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MACHINERY AND EQUIPMENT FOR LIVESTOCK BREEDING

*Approved by Ministry of education of Republic Belarus as a textbook
for foreign students in higher educational establishments for specialties
«Repair and maintenance production in agriculture»
«Logistics service in agroindustrial complex»
«Labour protection control in agriculture»*

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Рассматриваются схемы машин и оборудования для механизации технологических процессов: подготовки кормов к скармливанию и раздачи их животным, доения, первичной обработки и транспортировки молока в условиях животноводческих ферм, уборки и переработки навоза на фермах; приводится последовательность расчета их параметров. Изложены принципы формирования энергосберегающих поточных линий для механизации технологических процессов на животноводческих фермах.

Пособие предназначено для иностранных студентов учреждений высшего образования, магистрантов и аспирантов, конструкторов сельскохозяйственной техники, инженеров и научных работников.

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Schemes of machinery and equipment for mechanization of technological processes of preparation of feeds for feeding and distribution to animals, milking, primary processing of milk, harvesting and processing of manure on farms are considered; the sequence of calculation of their parameters is given. The principles of forming energy-saving production lines for the mechanization of technological processes on livestock farms are outlined.

This publication is intended for foreign students of institutions of higher education, undergraduates and graduate students, designers of agricultural machinery, engineers and scientists.

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INTRODUCTION

In market conditions products having the smaller cost and the best quality will be competitive. Such indicators can't be gained only for the account of resource-saving or energy saving. Complex savings are necessary, when all indicators characterizing technology are at optimum level. From this it follows that it is necessary to create the optimized low-cost mechanized operations and production processes which will allow create competitive products.

Today in literature questions of optimization of equipment configurations, performance of machines and the equipment, as well as effective techniques of flow-lines optimization and machines for mechanization of animal farms and complexes, are not provided with sufficient coverage. So, by optimization of machines for mechanization of animal farms the optimal solution shall be predetermined mathematically, by the correct choice of criterion function which would consider all complex of operational and ecological parameters (market requirements). Only in case of such strategy it is possible to achieve implementation of the potential reserves laid in the foundation of developing new generation technologies.

Increase in farm animal production, cost reduction for feed and work per unit of product are inconceivable without rational use of forages. It is important not simply to use all the feed supplies, but to use them with the maximum recoil, that is possible only when livestock is fed with balanced and nutritious feed mixtures taking into account productivity and physiological condition of each animal or small group of animals. As far as the preparation of balanced feed mixtures is the most power-intensive and costly process at animal farms and complexes, the considerable attention is paid to the questions of optimization of equipment configurations for flow lines for preparation and distribution of feed.

When speaking of complex mechanization, here the process of production in farms is perceived as a system of engineering and technical and related organizational and technological arrangements. In farm animal production participates a large number of machines of different brands, each of which, working as a part of engineering procedures, has as a direct, and indirect impact on operation of other machines and aggregates. To make an objective effectiveness evaluation of this or that machine it is reasonable to consider them in connection with the whole engineering procedure and the whole machine system. Often separate machine can have rather high technical and economic characteristics out of this system. However, in a processing line it sometimes can give even negative effect. Thus, it is necessary to take all operations of engineering procedure as a whole. When researching the set as a whole, it is reasonable to use system approach which consists in giving a complete view of a complex object. Changeability of working conditions is one of the most essential factors taken into account during the choice of the production technology of these or those products.

The research of interaction of engineering procedures for each work type and optimization of their parameters is constituted by subsystems or the second level – engineering procedure. Its elements are engineering procedures of feeding production and it distribution, milking of cows, milk preprocessing, removal of manure, etc.

When choosing the machines for livestock production it is necessary to recognize that the number of the machines entering each line should be minimum. The experience of use of flow lines has showed that with all other conditions being equal the machines are more reliable, when less number of machines is in the line, and when the line itself is shorter. Therefore in the textbook constructive schemes of machines and the equipment for feeding production and distribution, milking, preprocessing of milk, removal of manure on a livestock farm, and the problem of optimization of their characteristics are considered. Considerable attention is paid to techniques of building of processing flow lines.

Chapter 1

OVERVIEW OF STOCK-RAISING FARMS AND COMPLEXES, FEEDING-STUFF AND METHODS OF ITS PREPARATION BEFORE FEEDING

1.1 Basic concepts and definitions

The main commercial industries in agrarian sector are the cattle breeding, pig-breeding and poultry farming. Livestock production on an industrial basis is organized on farms and complexes.

The farm is a combination of the necessary main and auxiliary production constructions for a certain type of the cattle of various age, which is placed on the unified site plan and connected by convenient communications and highly effective service systems based on the electromechanical technology of livestock production. *The industrial livestock or poultry-farming complex* is a large agricultural enterprise specialized on production of high-quality products with the minimum labour and farm inputs, which is based on an energy-saving technology and a single production rhythm. Not only the large volume of the produced products and high level of mechanization of all processes, but also essentially new form of production management, which requires the single approach to the solution of all the technological, organizational, engineering, construction and economic problems are characteristic of industrial complexes. One of the main features of complexes is the high concentration of production, i.e. concentration on a farm of such quantity of animals or birds in case of which are provided the highest performance of work, the best use of means of mechanization and all fixed assets. *In stock raising*, there are the following types of livestock farms and complexes depending on specialization:

– dairy farm (milk production);

– meat and dairy farm (production of beef and milk);

– meat farm (production of veal and beef);

– breeding farm (raising of breeding bulls and heifers).

Depending on specific organizational and economic conditions agricultural enterprises apply year-round stall, stall and grazing or stall and camping housing of the cattle. Depending on the housing method there is differentiation between fastened and loose housing or combination of both (in the winter – fastened housing is used, and in the summer – loose housing). The important place in livestock production is taken by pig-breeding. The following farm types of this livestock production branch are known:

– breeding farms, which supply economic entities young breeders and sows;

– commercial reproductive farms, which raise pigs up to 4 months and sell them to other economic entities;

– farms with the integrated production cycle where there are permanent and once-only sows giving birth to young pigs, who are then raised and fed out to market standards;

– feeding farms, which purchase pigs at age of 4 months, raise them and feed them out to 100…120 kg of live weight and sell for meat.

Poultry breeding is one of the earliest and highly effective industries. Birds excel pigs and cattle considerably in precocity, fertility and efficiency of payment for forage. Products of poultry breeding are egg, meat, down and feather. Duck, goose and turkeys eggs are practically used only for incubation because of the low eating qualities. The enterprises engaged in breeding of hens are divided into the egg poultry, egg and meat poultry and only meat poultry types. The enterprises engaged in breeding ducks, geese, turkeys and are meat poultry enterprises. In poultry farming apply different methods of poultry rearing. They depend on a type of a bird, appointment and intensity of development of an industry. The main methods of poultry rearing are *cage-rearing*; *floor-housing*; *combined rearing*; *aviary housing*; *range rearing*; *range-pasture rearing*; *on a reservoir*.

Production processes in cattle breeding, pig-breeding and poultry farming are specific to each of these industries and depend on gender and age groups and animal species.

1.2 Feed characteristics

Based on the energy value feed is divided into bulky feed (0,6 feed units and less per 1 kilo) and concentrated feed (more than 0,6 feed units per 1 kilo). In feed processing industry feed is divided into several types: grass forage (grass, hay, straw, silage, haylage); root crops (beets, carrots, etc.); concentrates (grain, animal feed, oil cake, meal, etc.); food of animal origin (milk, milk products, waste dairy and meat industry); industry wastes (alcohol, sugar, canning, food, fat); microbiological synthesis feed (yeast, microbial protein); mineral and vitamin feed additives. Each type of feed used in cattle breeding is characterized by physical and mechanical properties. Knowledge of the properties of feed materials is essential for calculation of working elements, reduction energy - and metal intensity of food preparing machines, improves feed quality in preparation for feeding. *The physical properties of feed* include humidity, mesh size distribution (mesh sizes and their ratio), bulk density, density, porosity, water absorption, water loss, hygroscopicity, heat capacity, thermal conductivity, viscosity. *Mechanical feed properties* include coefficient of external and internal friction, side thrust, angle of repose, the characteristics of resistance to compression, cutting, destruction blow, etc. Table shows some of the physical and mechanical properties of the feed.

Table Physical and mechanical properties of the feed

Feed type	Humidity, %	Bulk density, t/m^3	Angle of repose, degree
Straw cutting	14…16	0,03…0,05	50…60
Grass cutting	12…14	0,10…0,12	–
Grass meal	12…14	0,18…0,20	34…65
Chopped fodder beet	86…88	0,67…0,74	35…40
Chop:			
– barley chop	14…15	0,46…0,65	32…36
– ear corn chop	14…15	0,68…0,78	33…36
– oatmeal chop	14…15	0,30…0,36	45
Crumbled feed staff	14…15	0,50…0,55	32…35
Pelleted feed staff	12…14	0,60…0,70	17…22
Feed mix for cattle:			
– straw + silage	60…64	0,15…0,17	54…58
– straw + silage + concentrated fodder + solution	65…67	0,20…0,35	55…60

The most important fodder properties for a given technological process are those that determine the response of the system to external mechanical impact. Such properties are referred to as *technological*, and may belong to physical or mechanical. For example, for filtration processes of the ingredients of animal feed shape and density of the particles are the most important properties, while for the grinding of feed grain it is strength properties that are important.

1.3 Methods of feed preparation and feeding

The aims of preparing feed are as follows:

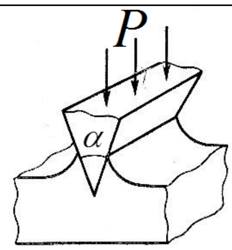
– increase digestibility and utilization of nutrients;

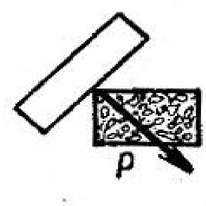
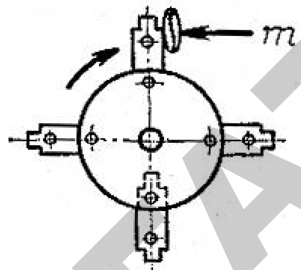
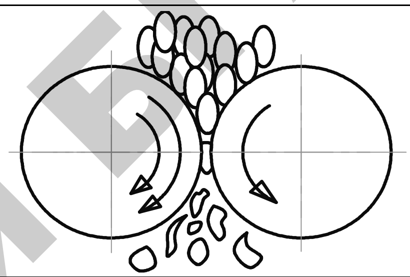
– improvement of technological properties;

– disinfection.

The following mechanical methods of preparation of fodder for feeding are the most common:

– *felling cut,* wherein the cutting angle is zero. In this case, the knife is directed vertically to the milled feed;



– *slidable cut,* wherein the knife is located in relation to the mass to be cut at an angle α of 25…30°. This process is the least energy-intensive;	
– *feed crush,* wherein the process is carried out by rapidly rotating hammers (from 40…100 m/s);	
– *flattening* is performed by rolls rotating at different speeds in opposite directions;	

Increase of animal productivity and reduction of feed costs per unit of production is inconceivable without the effective use of feeds. It is important not just to feed the fodder, but to make the most of their potential in the form of milk and weight gain. The following *methods of feeding* the animal fodder are known.

One of them involves separate and sequential distribution of gross, succulent and concentrated feed. This manufacturing process is very energy- and metal intensive because various feed distributive equipment is required - from motorized vehicles to handcarts. Concentrated feed delivery during this method is accompanied light fraction into environment, resulting in the loss of necessary elements for animals feed. A significant disadvantage of this method is the increase of feeding process period that worsens appetite of animals.

Another method of feeding consists in the simultaneous distribution of all kinds of feed in the form of feed mixtures. It allows to increase productivity of animals due to the complementary action of the mixture components and the increase of the feed eat ability in 5…9 % for dairy cows and in 10…15 % for fattening calves, as well as to reduce the loss of feed in 10...15 %. Feeding in the form of mixture can increase in the diet the share of low-value rough feed, which in pure form are eaten reluctantly. The composition of the feed mixture may be complemented with protein, mineral and vitamin supplements. Several studies have proved that productivity of animals receiving feed mixtures increases in 6,4...26,2 %, in comparison with separating feeding. Feed mixture greatly simplifies the organization of feeding process. In this case, different in physical and mechanical properties feeds are converted into a homogeneous mixture that allows to mechanize its distributing using the one type of feeders.

These advantages of animal feeding in the form of feed mixtures promoted widespread introduction of the following feeding types:

– haylage-concentrate;

– silage-concentrate;

– silage-, root crop-concentrate;

– hay-, haylage-concentrate and others.

However, mixture feeding does not provide individual feeding of animals, especially with high-energy feeds. This uneven distribution of feed reduces their energy efficiency, and therefore the profitability of the industry. In this case, haylage and silage are subjected to regrinding, that increases energy consumption and metal intensity of machinery and equipment. To solve the problem of the improvement of feed use efficiency, the following two areas in a normed animal feeding were highlighted:

– ensuring individual feeding based on complex automated systems;

– feeding of animal group.

Individual feeding system includes controlled by electric enabling device feed distributor. This system can identify the signal coming from the mini transmitter on each animal and organize individual feeding and regulation of feed, depending on the productivity and physiological state of the animal. The above considered method of animal feeding was implemented in mechanized *low-cost technology*. In this case, rough feed and ensilage are used as bulky components. In advance prepared and chopped tuberous roots, high energy bulk feed (mixed fodder), and various feed supplements are fed as a mixture forming a multi-component high-energy additive. This eliminates dust formation during the feed distribution process and the loss of nutritious juice of root crops, excluding repeated preparation of ensilage stalk feed.

With consideration for productivity of animals, bulky feed and higf-energy supplement are dosed, mixed and distributed to animals in the form of total mixed ratio. The use of this method of animal feeding provides reduction of energy- and metal intensity during preparation of feed for feeding: firstly, stalked feed is not exposed to regrinding; secondly, the feeding of expensive high-energy feed depends on animal productivity.

Considered method allows to reduce energy, materials, labor and financial costs of preparing the feed for feeding and feed mixtures compounding, and to increase energy efficiency of feed. The need for low-cost mechanized technology for feed production and distribution increases immeasurably when switching the herd on the same type feeding and collecting herds in homogeneous groups of animals with similar needs for nutrients.

Chapter 2

MECHANIZATION OF CONCENTRATED FOOD PREPARATION PROCESS

2.1 Concentrated food's zootechnical requirements and preparing machines

The destruction of grain feed is caused by the physiology of agricultural animals, since the rate of feed particles processing by gastric secretion is directly proportional to their surface area. Particles with a greater overall surface accelerate digestion and increase nutrient availability.

1. One of the well-known technologies of preparation of concentrated feed involves drying wet grain to 14 % and its storage in specialized premises, where it is necessary to maintain certain humidity conditions. The feed is fed in chopped form. Uniformity index of crushed grain content, which provides the same feeding value of feed, is to be at least 90...95 %. Grain fodder should not contain harmful add mixtures, earth, stones and straw impurities. The content of whole grains in the crushed product should not exceed 0,3...0,5 %, since the violation of these boundaries results in overspend of feed. However, excessive grinding of grain to dust reduces the efficiency of its use too. In addition, overgrinding of feed increases energy consumption for crushing process. As a criterion for the fineness of the product fineness modulus is used. In accordance with the zootechnical requirements, weighted average particle diameter of grain forage should be within the range of:

– 0,2...1,0 mm (fine grinding);
– 1,0...1,8 mm (medium grind);
– 1,8...2,6 mm (coarse grind).

Fineness modulus is determined by means of screen analysis. In this case weighing batch of groats (100 grams) is sifted through a set of screens with round holes with a diameter of 5, 3, 2 and 1 mm for coarse and medium grinding and 4, 3, 2, 1 and 0.2 mm for fine grinding. Weighted average particle diameter (i.e. fineness modulus) is calculated using the following formula:

$$M = \frac{d_1 m_1 + d_2 m_2 + ... + d_{n_\phi}\ m_{n_\phi}}{100} = \frac{\sum_{i=1}^{n} d_i m_i}{100}, \qquad (2.1)$$

where d_1 – the average size of the openings of two adjacent screens, mm; m_1 – mass of fraction yield, i.e the remnants on each screen, expressed as a percentage of the weight of the batch, g; n_f – number of factions into which the batch splitted, pcs.

The practical definition of the particle surface area is carried out by screen analysis method with the help of laboratory screening. As a result of screening of selected product sample, the particle classes different in grain size are obtained. The particles surface area (including their approximate cubic shape) of the entire sample is calculated with the formula

$$S = 6\sum_{i=1}^{n} \frac{m_i}{\rho\, d_i}, \tag{2.2}$$

where m_i – масса of the i particle class, kg; ρ – density of particle material, kg/m^3.

2. Grain conservation technology in the early stages of ripeness allows harvesting of crops of early waxy ripeness with its humidity of 14 to 40 %. Before storage the wet grain is destructed by flattening, which presupposes the destruction of grain by crushing method (the recommended thickness of flattened grain cereals should not exceed 1,1...1,8 mm). For grain viability during storage the mass is supplemented with preservative with observance of dosage and thorough mixing. When laying the grain for storage it is grain is firmed to remove air from the feed, and stored in air-proof trenches with walls and the top lined with plastic wrap or plastic bags.

The advantages of storage of feed grain harvested during its milk-wax ripeness are as follows:

– nutritional value of cereals during the milk-wax ripeness is at the highest, so a 1 hectare gives 10 % more of nutrients;

– the harvest is gathered 21...3 weeks earlier than usual, which is important for regions with unstable climate;

– grain drying and pretreatment is excluded.

There is a method of feeding fodder grain with the humidity of 14 to 40 %, in accordance with which the grain of wax ripeness must be crushed. In this case, the integrity of the grain is broken by cutting its cells into parts that minimizes the juice ooze, which is rich in nutrients.

Crushing grain increases its surface, which leads to the increased accessibility of feed. Crushing the grain reduces its geometrical dimensions that excludes the recovery of the same shape and reduces the energy demands for feed coefficient process before storage.

2.2 Classification, structure and operation of machines for preparation feeding grain

The choice of the method of preparation of concentrated feed is determined by a number of factors, including the physical and mechanical properties of the crushed grain. Depending on the nature and magnitude of the external forces applied to the grain, in the grain occurs *deformation* which may be ***elastic*** (reversible) and ***plastical*** (irreversible). Under the *elastic deformation*, after the removal of the external forces, the grain under the influence of interatomic forces returned to its original state.

Plastic deformations occur under loads exceeding the elastic limit. Unlike other deformations, plastic deformation develops with very low speed. The increase of plastic deformation depends not only on the magnitude of the applied stress (load), but also on the speed and duration of loading.

Development of elastic, and then plastic deformations in the grain under the exposure to external forces ends in its destruction. The destruction occurs when there is loading exceeding a certain limit, called the strength limit, or critical stress:

$$\sigma^* = \sqrt{\frac{2E\lambda}{\pi l_м}}, \qquad (2.3)$$

where E – elasticity modulus of material; $l_м$ – crack length of material, m; λ – specific fracture work per increment unit of product surface (material constants).

If the task is a *brittle grinding of grain*, then that effort should be made quickly, before the relaxation processes have developed in grains (stress relaxation). In this case the grain undergoes little deformation. If *ductile grinding* is needed, then the force should be applied slowly. In this case, not the brittle grinding, but grain flattening will prevail. To prepare grain fodder for feeding by applied shock the hammer crushers are used. In the machines of this type the grain grinding is caused not only by hammer blows, but also by concussions against ridged decks. *According to the layout* of the working chamber, the crushers can be horizontal and vertical. *Depending on the organization of work processes,* these machines are divided into:

– *in open-type crusher*, in which the material is not involved in a circular rotation. In such crushers the product is quickly evacuated from the working chamber, while the mechanical factor of grinding is a free hammer blow on incoming material. A process of open-type crusher is shown in fig. 2.1.

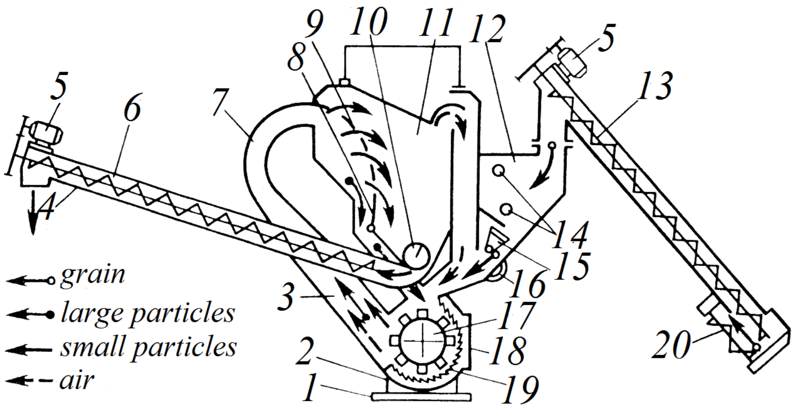


Fig. 2.1. Process flowsheet of an open-type crusher:
1 – framing; 2 –frame; 3 – grinding chamber; 4 – delivery screw conveyor; 5 –motor; 6 – screw conveyor; 7 – drop tubes; 8 – sleeve valve; 9 – separator; 10 – separating chamber's screw conveyor; 11 – separating chamber; 12 – grain tank; 13 – feed screw conveyor; 14 – level transmitter; 15 – tank sleeve valve; 16 – magnet; 17 – crushing drum; 18 – cover; 19 – deck; 20 – utility screw conveyor

Crusher rotor is positioned horizontally and fixed inside body. It consists of a shaft with a set of discs and freely oscillating on the axes hammers. On the inner cylindrical surface of the body decks are mounted, position of which is regulated relative to the hammers.

Outside, the body is mounted with the separation chamber, drop tubes and grain tank. Separation chamber serves as a vat in which the crushed product is separated into coarse and fine fractions.

Partitions in the chamber form channels - one for the air return to the grinding chamber, the other for the return of the coarse fraction for the regrinding. There is a screw conveyor in the bottom part of the chamber to offload the finished product of it. Grinding fineness is regulated by the turn of a sleeve valve in a separation chamber and a changeable separator.

Separator is mounted depending on the type of the crushed grain. Tank is equipped with magnets to trap metal impurities and low-level and high-level grain indicators by means of which a feed screw conveyor turns on and off.

– *in a closed-type crusher* the grain incoming in working chamber performs repeated circular motions. In closed-type crushers a screen and decks involve the whole rotor. A process of a closed-type crusher with a horizontal rotor is shown in fig. 2.2.

The crusher includes crushing device, fan, charge hopper, cyclone collector with rotary valve and two stub sockets, pressure and lateral pipes, filter and electric installation. All the units are mounted on the frame. Grain charge hopper has valve at the bottom which regulates the flow of bulk feed into the grinding chamber. The cyclone separator separates the air from the grain.

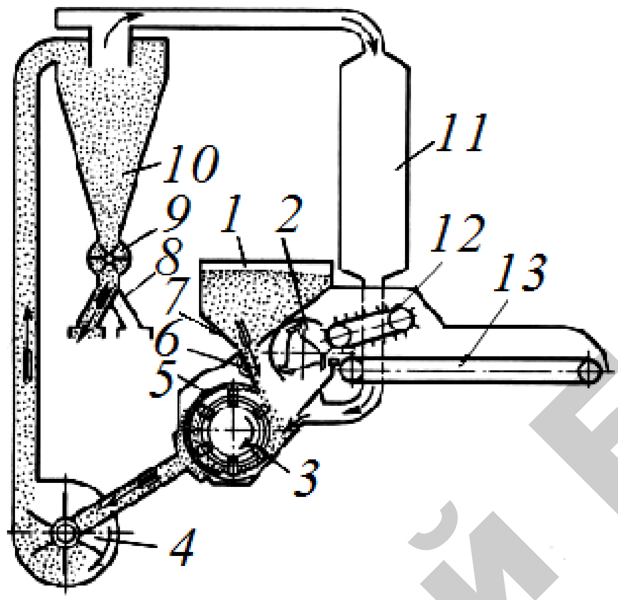


Fig. 2.2. Closed-type crusher process:
1 – grain tank; 2 – side frame; 3 – hammers, 4 – fan; 5 – screen;
6 – magnet separator; 7 – sleeve valve; 8 – bag holder socket,
9 – rotary valve; 10 – cyclone separator;
11 – filtration sleeve

At the bottom it is mounted with rotary valve. On the top there is cylindrical tube with twisted trailing enclosure. Rotary valve is designed for outputting the flour from the bottom of the cyclone separator and is composed of cast iron frame, the sidewalls and the rotor with rubber blades.

t prevents the exit of the airflow from the cyclone separator. The rotor is driven by V-belt drive, helical reducer and flexible coupling. At the bottom of the rotary valve there are two bag holders with butterfly gate.

Air tube connects the cyclone separator with the fan, and through a linen filter with a supply air pipe. Crushing apparatus consists of a cast iron frame, the sidewall, a hinged top, two ridged decks, removable screen and the crushing drum (Fig. 2.3). When grinding the grain, the crusher chamber is inserted with the sieve. The required degree of grain grinding is provided by changeable sieves with the holes of the target size, for example, 4; 6; 8 and 10 mm.

Crushing drum consists of disks mounted on a keyed shaft. The distance between the discs is fixed by spacers.

– The disk holes are mounted by axis with hammers. The distance between the hammers is adjusted by spacers. The upper window of the crushing apparatus chamber is connected to a charging hopper, the bottom window - with a supply air pipe, which has a longitudinal slot for directing the air flow into the grinding chamber. On the back wall of the upper window there is a magnet separator. When crushing grain feed, the chamber top is mounted with a changeable sieve. Bottom window of the top is connecting by a removable suction tube with a fan. Then the grinder is switched on and the degree of engagement is regulated with the grain tank valve.

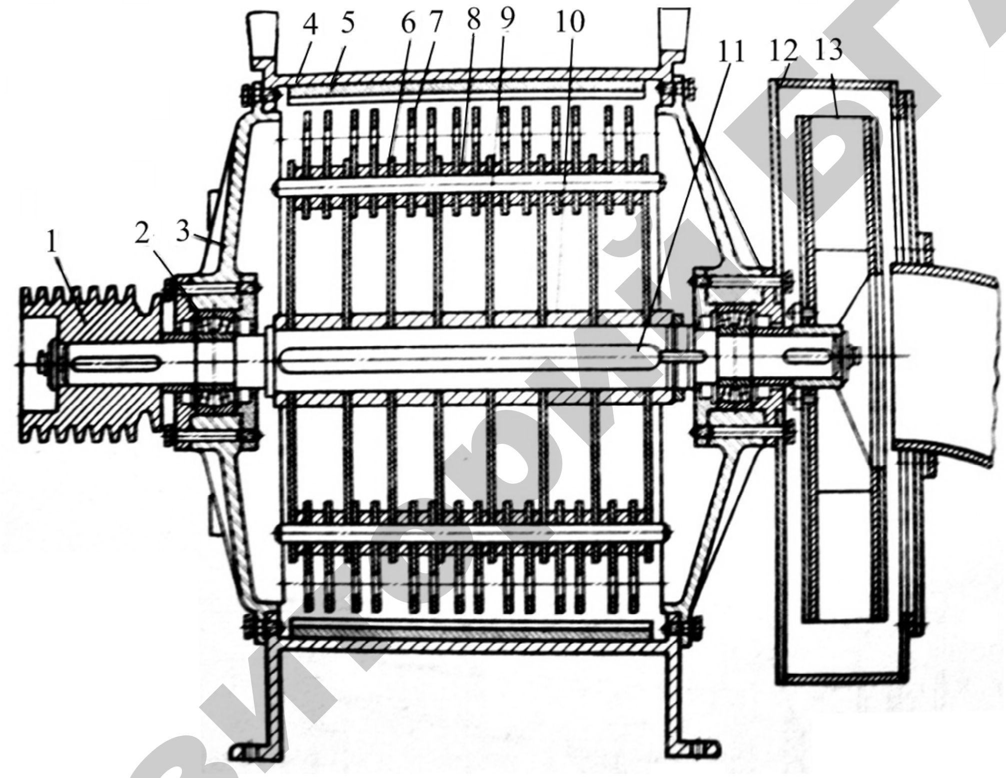


Fig. 2.3. Crushing apparatus with a fan:
1 – sheave; 2 – two-raw spherical bearing; 3 – sidewall; 4 – frame; 5 – deck;
6 – disk; 7 – hammer; 8, 10 – spacer; 9 – axe; 11 – drive key; 12 – fan cover;
13 – fan; 14 – suction tube

Loaded into the receiving hopper grain is cleared with a magnet separator of steel items and enters the crushing chamber. Under the influence of hammer blows it is partially destroyed. Unbroken particles are dropped to the decks and sieves, where it is finally crushed. The particles that are equal or smaller to the screen hole fall into the cavity of the grinding chamber, from which the air flow moves to the cyclone separator through the suction tube, the fan and air pipe. The cyclone separates the product from the air, then it goes through the rotary valve and the socket to the container, while the air goes through the discharge air pipe, filter and snorkel tube back into the grinding chamber. Part of the air comes out through the filter. This creates some unloading during the exit from the grinding chamber, thereby eliminating the atomization of crushed feed through the leaks of the chamber. The specialized crushing machines of a closed-type are those with vertical working chamber (fig. 2.4).

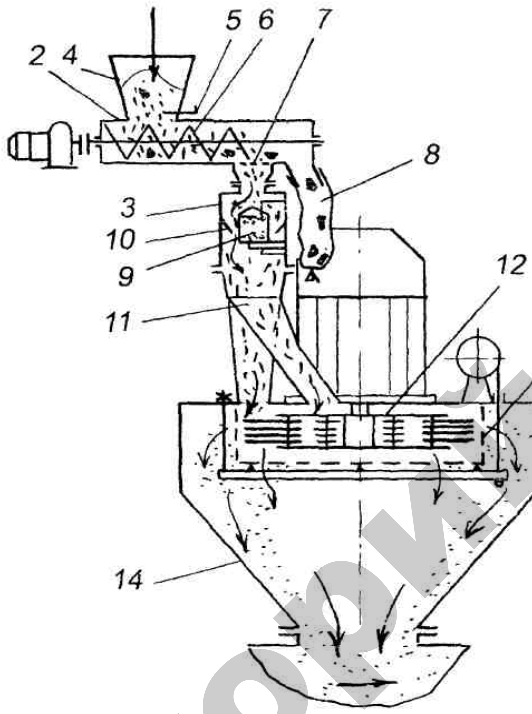


Fig. 2.4. Closed-type crusher with vertical working chamber process:
1 – crusher; 2 – feeder; 3 – magnet separator; 4 – funnel; 5 –check valve; 6 – screw; 7 – screen; 8 – sleeve; 9 – magnetic head;10 – reflector; 11 – distributor; 12 – rotor; 13 – sieve; 14 – funnel

Inside the working chamber of the crusher between two discs the rotor is mounted coaxially, which has fixed hinged hammers packs. On the outer perimeter of the rotor and bottom the sieves are fixed. The rotor drive and grain feeding tube is located on the upper base of the working chamber. A disposed vertically sieve is fixed inside the housing of the working chamber with a clearance which receives the milled grain and under the force of gravity is moved to the unloading chamber. The rotor is driven directly by the motor shaft, or through an intermediate V-belt transmission.

The active working element of a crushing machine is a hammer. According to the surface shape the hammers can be divided in the following types (fig. 2.5.):

– solid form. The advantage - the ease of manufacture;

– with projections. Advantage - increases productivity of machine;

– bulk (compound).

To fix hammers the holes are made. Hammers are made of steel St 25, St 65. The thickness of hammers can be between 2 and 5mm (the smallest value is selected by grinding grain).

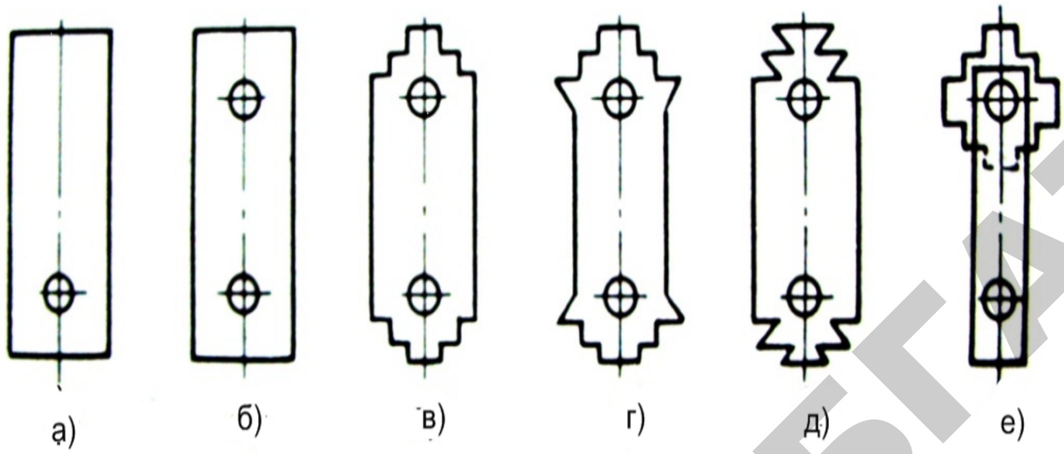


Fig. 2.5. Hammer shapes

Sieve of a grinder is designed to separate the crushed wheat from the uncrushed. On design (fig. 2.6.) the sieve can be with knockouts:

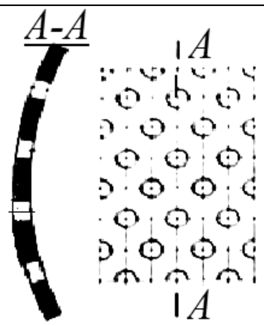
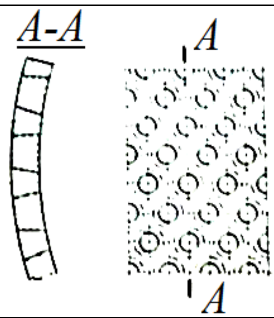
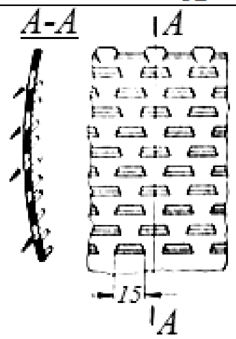
– with a cylindrical shape. The advantage with the cylindrical sieve apertures is easy to manufacture, but the disadvantage of relatively poor performance.	
– with conical holes. Advantage of the sieve with a conical shape of the holes, is to increase the productivity of crusher and lack of complexity in manufacturing.	
– Scale, with protrusions before the holes. The advantage is to increase the sieve mill productivity and lack of complexity in the manufacture and service life delimited.	

Fig. 2.6. Sieve shape

To activate grain crushing process, as passive working bodies, the internal perimeter of the processing chamber are set grooved deck - plates, which are made on the surface of the reefs, which can be of various forms: Rectangular; Semicircular; Triangular (most widely in agricultural production).

During the operation of the crusher feed moving along the surface of the deck, portable reduces its speed. In this case, due to the speed difference between the rotating working bodies and the food is its destruction. Almost all considered hammer mills grains have a significant *disadvantage* of heterogeneity of particle size distribution of the finished product. If grinding particles after sieving can be returned for re-grinding, then over grinding particles nothing can be done. They are a measure of the imperfections of the process, cause unnecessary energy consumption, harm the animal. This disadvantage can be eliminated by introducing into the process control scheme grinding sifting operation, i.e. by applying a multistage crushing with intermediate separation. Structurally grain grinder with *intermediate separation* of pro-product (fig. 2.7) is designed as a vertical working chamber with two tiers of successive sieves. Diameter of the holes in the top tier of the lattice is more than at the bottom. Above each tier of sieves mounted active working bodies.

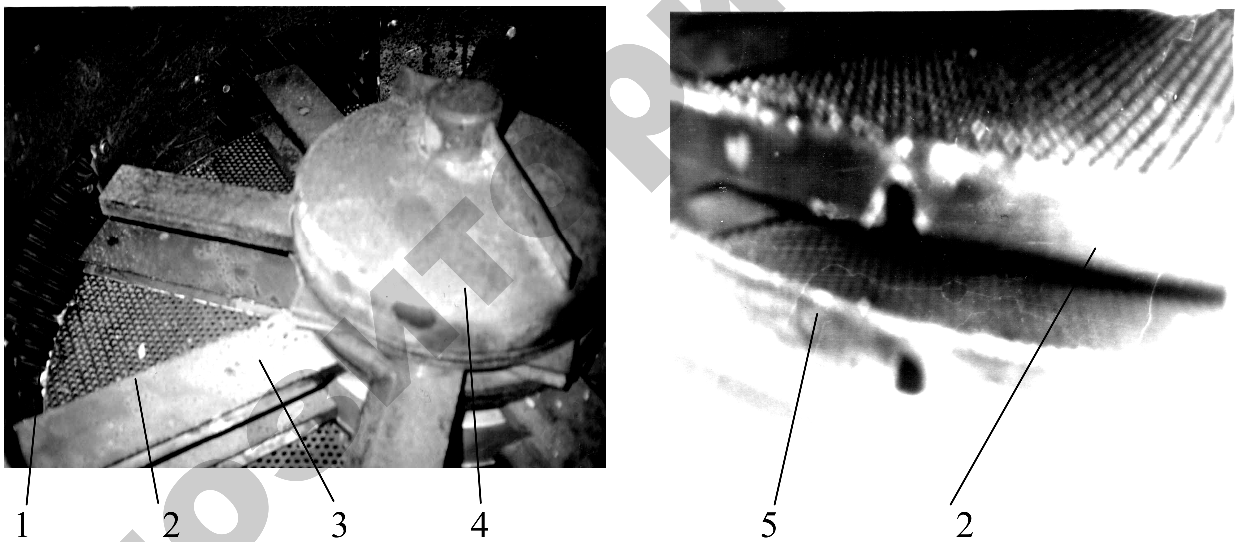


Fig. 2.7. Grain grinder with intermediate separation of product in working chamber: 1 – deck; 2 – sieve 3 mm; 3 – active working element; 4 – rotor; 5 – sieve 5 mm

Since the upper sieve is satisfied with a large diameter hole, provided pre-grinding grain alignment and its fractional composition, followed by evacuation of the working zone of the lower sieve, which provides the necessary grain size of grinding. Thus, a crushing machine arranged sequentially grain alignment facilitates its fractional composition, which prevents the formation of dust-like fraction. Corrugated deck reefs, used as passive working elements, mounted on the inner perimeter of the working chamber, which have a triangular shape.

The design of the shredder feed grain distributes grain forage along the perimeter of the working chamber during the entire grinding process allows more efficient use of working bodies and the surface of the perforated sieve, and thus reduces the energy density of feed milling process by increasing the productivity of the chopper. When installed in the processing chamber grinder sieves two - in the lower tier of holes with a diameter equal to 3 mm, and the upper - 5 mm, the specific energy consumption of the grain milling process was 8.19 kWh/t with productivity 2.68 t/h.

2.3 Determination of energy expenditure for grain destruction in crushers

The amount of energy for grain destruction by impact crushers can be determined by the formula

$$A_м = A_V + A_S + A_{раз}. \tag{2.4}$$

where $A_м$ – energy consumption for the movement under the hammer before the impact on grain, J.; A_V – the energy dissipated in the volume of the particles at their deformation (i.e spent on the internal friction and heating of the material to be ground); J; A_S – energy consumption for the formation of new cracks and surfaces in the material being ground, J.; $A_{раз}$. – energy on the destruction of grain, J.

The kinetic energy of moving hammer before blow:

$$A_м = \frac{m\,\upsilon_м^2}{2}, \tag{2.5}$$

where m – mass of the hammer, kg; $\upsilon_м$ – speed of a hammer before strike, m/s.

A_V work can be represented as part of the maximum potential energy of the elastic deformation of grains *Umax* stored during the impact:

$$A_V = \chi U_{max}, \tag{2.6}$$

where χ – energy scattering coefficient.

Maximum potential elastic energy stored in the body, as is known from the theory of elasticity, equals:

$$U_{max} = \frac{\sigma^2 V}{2\,E}, \tag{2.7}$$

where σ – stress in the caryopsis, N / m^2; V – volume of grains, m^3; E– modulus.

The kinetic energy of the moving hammer after hitting grains:

$$A_s = \frac{m\,\upsilon_ч^{\;2}}{2}, \tag{2.8}$$

where $\upsilon_ч$ – speed particles formed during the impact, m/s.

Substituting (2.4) value of components, energy balance equation at a single blow, enough to break the grains, is as follows:

$$\frac{m\upsilon_м^2}{2} = \chi\frac{\sigma^2 V}{2E} + \frac{m\upsilon_ч^2}{2} + A_{разр}, \tag{2.9}$$

Considering number of strokes, magnitude of energy used for the destruction of grain can be determined by the formula

$$A_{разр} = \left[\frac{m\upsilon_м^2}{2} - \left(\chi\frac{\sigma_n^2 V}{2E} + \frac{m\upsilon_ч^2}{2}\right)\right]n, \tag{2.10}$$

where n – number of blows to the caryopsis by hammer, blows.

For real conditions, grain grinders should be designed in such a way that the destruction process was held for 1...2 stroke or 30...50 strokes. Ignoring these ranges increase power consumption of grinding process. Energy costs for crushing 1 kg of barley (intermediate grinding) at $\upsilon_м$ = 130 m/s are equal to 2,3 kWh/t. However, currently the best examples of impact crushers have a specific energy of at least 5,0...5,5 kWh/t. This means that about half of the energy consumptions are unproductive in nature and associated with the generation of the accompanying air flow, friction material particles to each other and working bodies (which increases the heating of both), sub-optimal number of strokes to failure.

In assessing the strength of individual grains the size of the failure stress found that the cereal grain barley has greatest strength that being the main forage crop, is taken as a benchmark for comparative assessment. For various crops strength ratio of the grains include barley – 100 %; wheat – 91 %; rye – 83 %; peas – 65 %; oats – 54 %. From the data obtained, it follows that the guaranteed destruction of all without exception barley kernels observed at impact velocity of 130 m/s or higher, wheat - 115 m/s and higher, oats 105 m/s and above. This rate cause intense dynamic mode working on the principle of impact crushers.

Power demand for grinding feed is determined by the formula

$$N = N_{ИЗМ} + N_{Ц} + N_{X.X};\tag{2.11}$$

where $N_{ИЗМ}$ – power expended on the destruction of the material, kW; $N_{Ц}$ – power expended on food circulation in the working chamber, kW; $N_{X.X}$ – power idle crusher stroke, kW.

The power expended in the destruction of the material is given by:

$$N_{изм} = P_{изм}\, \upsilon_{м},\tag{2.12}$$

where $\upsilon_{м}$ – speed of hammers, m/s; $P_{изм}$ – the power expended on the destruction, N.

The power expended in the destruction:

$$P_{изм} = S \cdot \sigma_{разр},\tag{2.13}$$

where S – the working area of the hammer, m^2.

The working area of the hammer is determined by the formula:

$$S = l_{м.\,раб}\, h_{м},\tag{2.14}$$

where, $l_{м.\,раб}$ – working length of the hammer, m; $h_{м}$ – the thickness of the hammer, m; $\sigma_{разр}$ – destructive surface pressure N/m^2 (for cereals $\sigma_{разр}$= (70±20)10^5 N/m^2; for stalked $\sigma_{разр}$= (100,4±20)·10^5 N/m^2).

Power expended on food circulation in the working chamber and the idle mills turn, is defined by the formula:

$$N_{Ц} + N_{xx} = (0,15–0,20)\, N_{изм}\tag{2.15}$$

Crusher's parameters: definition. An important indicator of the mill feed is its performance. This figure depends on sieve surface. To determine this parameter, sieve analyzes workflow flowing into the grinding chamber. Grain fodder entering the shredder working chamber is exposed to the rotating working bodies, the action of which begins to move along the plane of the sieve. If the path meets particles crushed grain sieve opening, in which a particle fits into the solution, it moves toward the discharge channel. In this case, the capacity of the sieve can be determined by the formula

$$Q_{p1} = S_{сеп}\ \rho\ \upsilon_{пр}\ , \tag{2.16}$$

where $S_{сеп}$ – area of separating surface, m^2; ρ – density of feed, kg/m^3; $\upsilon_{пр}$ – the rate of passage of the ground product through a sieve aperture in m/s.

Separating surface area can be determined by the formula

$$S_{сеп} = S_{решета} - S_{отв}\ , \tag{2.17}$$

where $S_{решета}$ – area of sieve aperture, m^2; $S_{решета}$ – area of openings in a lattice, m^2.

Separating surface area can be determined by the formula

$$S_{решета} = DL\frac{\alpha}{360^0}\ , \tag{2.18}$$

where, D – diameter of the working chamber grinder, m; L – length of the working chamber grinder, m; α – angle coverage sieve crushing chamber; $S_{отв}$ – holes in the sieve area, m^2.

The area of openings in a lattice defined by the formula

$$S_{отв} = \frac{\pi\, d_{отв}{}^2}{4} Z_{отв}\ , \tag{2.19}$$

where $d_{отв}$ – diameter of the hole, mm; $Z_{отв}$ – the number of holes in the sieves.

Then surface separating area:

$$S_{сеп} = DL\frac{\alpha}{360^0} - \frac{\pi\, d_{отв}{}^2}{4} Z_{отв}\ . \tag{2.20}$$

Thus, the bandwidth depends on the speed of the sieve advancing ground product through the apertures in the sieve. From the sieved product circuit velocities through the holes in the sieve (fig. 2.8) that the particles move in space towards absolute velocity vector, which determines the rate of passage of the ground product through the apertures in the sieve, i.e. $\upsilon_{пр} = \upsilon_{абс}$.

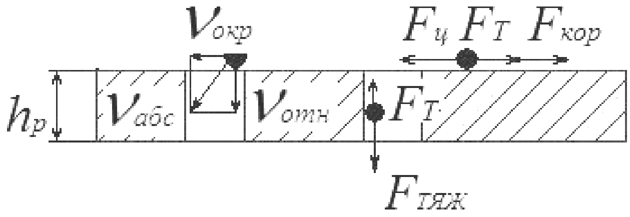


Fig. 2.8. Diagram for determining the capacity of the sieve

Absolute speed can be determine according to the formula

$$\upsilon_{абс} = \sqrt{\upsilon_{окр}{}^{2} + \upsilon_{отн}{}^{2}}. \tag{2.21}$$

To determine the peripheral speed of the feed particles $\upsilon_{окр}$, moving along the sieve surface, consider the force acting on it. On particles during grinding grain forage centrifugal force $F_{ц}$, coriolis force $F_{к}$ and friction force $F_{т}$, which can identify the following formulas:

The particle friction force on the surface of the sieve, which arises from the force of gravity:

$$F_{т} = f_{к}\, m_{к}\, g, \tag{2.22}$$

where $f_{к}$ – coefficient of friction against steel feed; $m_{к}$ – particle mass, kg;

Centrifugal force:

$$F_{ц} = m_{н}\, \omega_{р}{}^{2}\, r_{н}\,, \tag{2.23}$$

where $\omega_{р}$ – angular speed shredder rotor c^{-1}; $r_{н}$ – distance from the mounting axis to the hammer particle, m; $m_{н}$ – hammer weight, kg;

Cariolis force:

$$F_{кор} = 2\, m_{к}\, \omega_{р} \upsilon_{окр}, \tag{2.24}$$

where $\upsilon_{окр}$ – peripheral velocity of particles of grain forage, m/s.

Then we can write the equation of motion of a particle on the surface of the sieve:

$$m_{н}\, \omega_{р}{}^{2}\, r_{н} = f_{к}\, m_{к}\, g + 2\, m_{к}\, \omega_{р} \upsilon_{окр}.$$

From whence

$$\upsilon_{окр} = \frac{m_н\, \omega_p^{\,2}\, r_н - f_к\, m_к\, g}{2\, m_к\, \omega_p} . \qquad (2.25)$$

To determine the relative speed of the feed particles moving over the hole in the screen, consider the force acting on it. On grain forage particles the force of gravity and the force $F_{тяж}$ grain particles friction on the surface of the sieve F_{m1}, which can determine the formulas.

Gravity force:

$$F_{тяж} = \frac{m_к\, \upsilon^2_{отн}}{h_p}, \qquad (2.26)$$

where $\upsilon_{отн}$ – relative velocity of particles of grain forage, m/s; h_p – thick sieve, m.

The strength of the particles of the friction surface of the openings of the screen:

$$F_{m1} = f_к\, m_к\, g, \qquad (2.27)$$

where $f_к$ – coefficient of friction against steel feed; $m_к$ – particle mass, kg.

Then the equation of motion of a particle on the opening screen:

$$\frac{m_к \upsilon^2_{отн}}{h_p} - f_к\, m_к\, g = 0.$$

Where

$$\upsilon^{\mathbf{2}}_{отн} = f_к\, h_p\, g\, . \qquad (2.28)$$

The absolute velocity of the particles through the sieve aperture is defined by the formula

$$\upsilon_{абс} = \sqrt{\left(\frac{m_н\, \omega_p^{\,2}\, r_н - f_к\, m_к\, g}{2\, m_к\, \omega_p}\right)^2 + f_к\, h_p\, g} \, . \qquad (2.29)$$

Then the capacity of the sieve can be determined by the formula

$$Q_{p1} = (DL\frac{\alpha}{360^0} - \frac{\pi\, d_{отв}^{\ 2}}{4} Z_{отв})\, \rho \sqrt{\left(\frac{m_н\, \omega_p^{\ 2}\, r_н - f_к\, m_к\, g}{2\, m_к\, \omega_p}\right)^2 + f_к\, h_p\, g}\,. \qquad (2.30)$$

The capacity of the sieve in the chopper steady operation depends on several factors. So, with a positive increase in the ratio of geometrical dimensions of the working chamber and the shredder rotor performance will increase. The effective cross-section of sieve**'s** area (in case it**'s** maximum possible area) depends on the diameter of the holes and their number per unit surface. To increase the bandwidth should be selected largest diameter holes. Increasing the diameter of the holes and reduces the energy cost of running process. However, an increase in this parameter sieve entails growing size of the crushed material, under certain conditions, makes little sense for the chopper operation. The diameter of the holes in the sieve should be chosen depending on the requirements of the final product. The maximum number per unit area is determined based on constructive strength sieve.

The diameter of the working chamber is defined by the formula (fig. 2.9.):

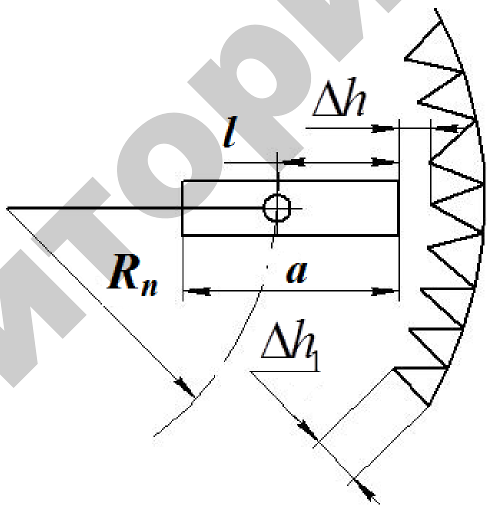


Fig. 2.9. Calculation of hammer mill diameter - sheme

$$Д = (l + R_n + \Delta h + \Delta h_1) \times 2; \qquad (2.31)$$

where l – the distance from the axis of the hammer attachment to its end face; R_n – the distance from the axis of the drum to the axis of suspension of the hammer, m; Δ_h – the gap between the ends of the hammer and the deck, m; Δh_1 – the thickness of the deck or screen, m.

The distance from the axis of the hammer attachment to its end face is determined from the relationship:

$$a = 1{,}5l, \rightarrow l = \frac{a}{1{,}5}, \tag{2.32}$$

where a – hammer length, m;

The distance from the axis of the drum to the axis of suspension of the hammer

$$l = \frac{4}{9} R_n \rightarrow R_n = 2{,}25 \times l, \tag{2.33}$$

The number of hammers in the grinding chamber can be determined by the formula

$$Z = \frac{LK_z}{\delta}, \tag{2.34}$$

where L – crushing drum length, m; K_Z – the number of the hammers going on the trail, $K_Z = (1...6)$; δ - thickness of the hammer, m.

To activate grain crushing process on the inner perimeter of the processing chamber are set grooved deck. In this case, due to the speed difference between the rotating working bodies and the food is its destruction. Parameters reef should ensure conditions for effective wasp-actions listed species destruction. From the diagram in fig. 2.10, a, and it is clear that for the direct impact of the central front face should be positioned at an angle *αr* to the working chamber radius. The most effective positioning the front facet at an angle αr from the condition:

$$arc\ cos\ (1 - l_в / R_н) \le \alpha_p \le \pi - 2(\varphi_1 + \varphi_2), \tag{2.35}$$

where $l_в$ – the distance from the particle to the departure point of the deck, m; $R_н$ – range described active working bodies, m; φ_1, φ_2 – feed angle of friction, respectively, on the surface of the deck, and an active working body, degree.

a) b)

Fig. 2.10. Determining schemes of the reef geometric parameters
a) fragment of the chopper working chamber: 1 – reef; 2 – knife;
3 – a rotor chopper; b) deck: 1 - reef

An important parameter that provides clean depressions formed adjacent reefs, is the wedge angle of the reef. The minimum value of taper angle facing towards the active working elements should be more than double the friction angle:

$$\gamma_{min} > 2\varphi. \tag{2.36}$$

For a rational number of reefs on the deck angle γ should be in the range of 80...100 degrees. Clogging troughs between adjacent reefs is due to constant displacement of feed particles along the active radius of the working member to the inner surface of the working chamber. The face of the reef ridge located over its front edge, limits the movement of the particles between the reefs and lead to their sticking. Eliminating this disadvantage is possible by replacing the static friction frictional movement. To this end, the reefs on the deck shall be located at an angle τ to the horizontal plane (Fig. 2.10b), the value of which should be in the range:

$$65^0...75^0 \geq \tau \geq 40^0...58^0. \tag{2.37}$$

Under certain angles reef unknown quantity is the length of the front face of the reef along which the ground material. To determine this parameter, we consider forces acting on the particle feed, while conveying it on the front plane of the reef. When grinding grain forage food particles entering the working chamber are exposed to the active working bodies. In this case, the feed stream changes its direction of motion. Feed particles begin to move at a speed relative to the inner surface of the working chamber. At the same time they acquire the drive speed, the influence of which the entrained in a circular motion. This feed particle moving along the frontal plane of the reef, the following powers:

– Force of particles friction on the surface of the reef:

$$F_{т} = f_{к} m_{к} g, \tag{2.38}$$

where $f_к$ – the coefficient of friction on metal; $m_к$ - particle mass, kg;

– centrifugal force:

$$F_{ц} = m_{к} \omega_{c}^{2} l_{ф.р}, \tag{2.39}$$

where ω_c – speed feed layer within the working chamber, s^{-1}; $l_{ф.р}$ – the length of the front plane of the reef, m;

– coriolis force

$$F_к = 2m_к\omega_c\frac{dl_{ф.р}}{dt}, \tag{2.40}$$

where $dl_{ф.р}/dt$ – velocity of particles of grain forage, m/s.

Then we can write the differential equation of motion of a particle with respect to the knife as follows:

$$m_к\frac{d^2 l_{ф.р}}{dt^2} - m_к\omega^2{}_c l_{ф.р} + 2\ f\ m_к\omega_c\frac{dl_{ф.р}}{dt} = -f_к\ m_к g\ . \tag{2.41}$$

This equation is a linear inhomogeneous 2nd order, its general solution is composed of the general solution of the homogeneous linear equation.

When the initial conditions t=*0*, $l_{ф.р} = 0$, $dl_{фр}/dt=0$, and the known values of the coefficients of the solution of equation (2.41) will take the form of:

$$l_{фр} = \frac{f_к g}{\omega_c^2}\left[\left(\frac{f_к + \sqrt{f_к^{\,2}+1}}{2\sqrt{f_к^{\,2}+1}}\right)e^{(f_к\omega_c+\omega_c\sqrt{f_к^{\,2}+1})t} - \left(\frac{f_к + \sqrt{f_к^{\,2}+1}}{2\omega_c^2\sqrt{f_к^{\,2}+1}}\right)e^{(f_к\omega_c-\omega_c\sqrt{f_к^{\,2}+1})t} + 1\right]. \tag{2.42}$$

To eliminate the clogging of milled grain fodder troughs between adjacent reefs, the front face of the reef on the deck at an angle τ to the horizontal plane (fig. 2.10, b). The equation for determining the length of the front face of the reef will be:

$$l_{фр} = tg\,\tau\frac{f_к g}{\omega_c^2}\left[\left(\frac{f_к + \sqrt{f_к^{\,2}+1}}{2\sqrt{f_к^{\,2}+1}}\right)e^{(f_к\omega_c+\omega_c\sqrt{f_к^{\,2}+1})t} - \left(\frac{f_к + \sqrt{f_к^{\,2}+1}}{2\omega_c^2\sqrt{f_к^{\,2}+1}}\right)e^{(f_к\omega_c-\omega_c\sqrt{f_к^{\,2}+1})t} + 1\right]. \tag{2.43}$$

From (2.43) it is clear that, in determining the length of the front surface of the reef should take into account the dynamic nature of the processes taking place by grinding grain forage harvester in the chamber.

2.4 Machinery for the preparation of grain for feeding milky wax ripeness and determination of their main parameters

For blanks for feeding of wet grain is used two ways: the destruction of the grain by flattening; the destruction of the grain by cutting. Flattening process involves the destruction of grains by crushing for the conversion of grain into flakes.

According to the principle of the conditioners are very similar and consist of a frame, cylindrical rollers with a smooth, grooved or micro-rough surface, rotating with different peripheral speeds in opposite directions, mechanisms of movement and adjustment of the gap between them. The process operates as follows. Grain is directed to the rolls conditioner (fig. 2.11). Rapidly rotating drum ahead of the particle in the grinding zone and treats her with riffles and particle overtaking slowly rotating drum, addictive product in the gap between the rollers, experiencing compression, shear and cut. This leads to the destruction of grain.

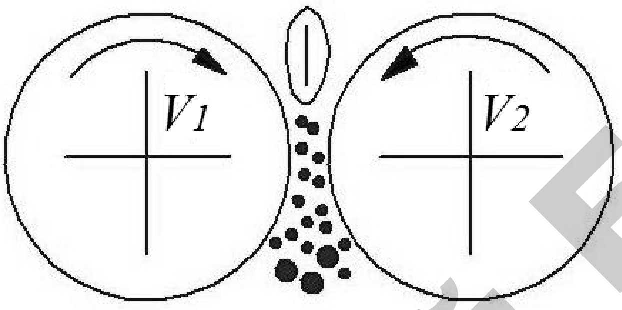


Fig. 2.11. Working process of flattening machine

Flattening process involves the destruction of grains by crushing for the conversion of grain into flakes. According to the principle of the conditioners are very similar and consist of a frame, cylindrical rollers with a smooth, grooved or micro-rough surface, rotating with different peripheral speeds in opposite directions, mechanisms of movement and adjustment of the gap between them. The process operates as follows. Grain is directed to the rolls conditioner (Fig. 2.10). Rapidly rotating drum ahead of the particle in the grinding zone and treats her with riffles and particle overtaking slowly rotating drum, addictive product in the gap between the rollers, experiencing compression, shear and cut. This leads to the destruction of grain.

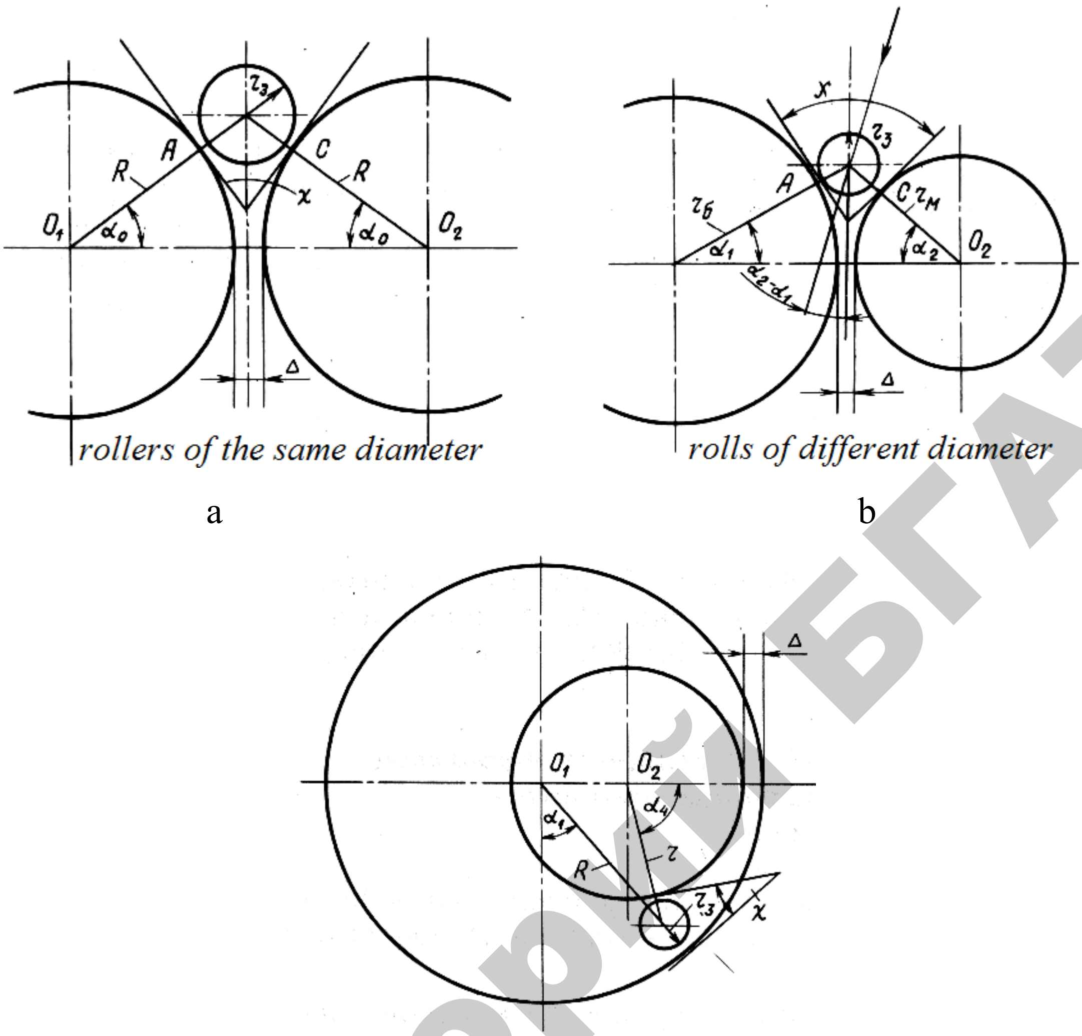
Milling quality is adjusted by changing the gap of a roller pair and the ratio of peripheral speeds of rollers. The gap between the rollers varies from 0,03 to 1,5 mm, a small change which leads to a significant change in the process of flattening.

By design, the execution of working rollers conditioner can be:

1) Contact with the outside work planes and the same size of the working bodies. Capture angle for crushers of this type is defined by the formula (fig. 2.12, a.)

$$\chi = \arccos\left[\frac{(2R+\Delta)^2}{2(R+r_3)} - 1\right], \tag{2.44}$$

where R – radius of the roll, m; Δ – gap between adjacent rollers, m; r_3 – radius of the grain, m.



a b

c

Fig. 2.12. Execution of working rollers conditioner – design

2) Contact with the outside working surfaces and different size of the working bodies (fig. 2.12, b). The angle for capturing rollers with different diameters:

$$\chi = \arccos \frac{\left(R_{\delta} + r_{M} + \Delta^{2}\right) + \left(R_{\delta} - r_{3}\right)^{2} - \left(R_{M} - r_{3}\right)^{2}}{2\left(R_{\delta} + r_{3}\right)\left(r_{M} + r_{3}\right)}, \qquad (2.45)$$

where $R_б$ – radius of the large drum, m; r_M – radius of the small drum, m.

3) Contact with the inner working surfaces of rollers. For conditioners with internal capture contact angle is determined by the formula

$$\chi = \arccos \frac{\left(R - r_{3}\right)^{2} + \left(r - r_{3}\right)^{2} - \left(R - r - \Delta\right)^{2}}{2\left(R + r_{3}\right)\left(r + r_{3}\right)}. \qquad (2.46)$$

An important indicator, providing workflow conditioners, a capture angle grain rolls. Conditions tightening product into the nip between the rolls is performed, if the capture angle less than or equal to $\chi \leq 2\varphi$, where φ - friction fig. 2.13 shows a schematic diagram of grain crushers. Grain crushers consist of one hopper for loading different kinds of grains. At the bottom of the hopper 2, the dispenser 1 has a frame 3 on which components are mounted on four smooth rollers bearing. One of the rollers mounted on the frame and is connected fixedly to the drive motor 5.

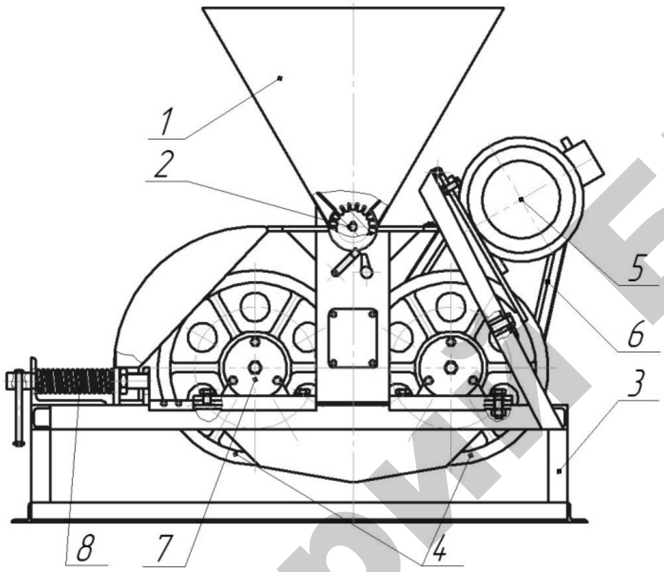


Fig. 2.13. Schematic diagram of grain crushers:
1 – dispenser, 2 – hopper, 3 – frame, 4, 5 – motor, 6 – belt, 7– roller,
8 – special mechanism

The second roller 7 can be moved horizontally along the frame, and it is spring-loaded by a special mechanism 8 for the passage between the rollers accidentally caught solid objects. When removing the belt from the motor 5 grain crushers drive can be powered from the take-of shaft of the tractor via the catdan shaft. The take-of shaft and cardan shaft of tractor transmission can be replaced by a hydraulic motor, running from the hydraulic system of the tractor or outside hydropower.

The design of the roller conditioner models «Murska 700S Russia" is different from other roller mills in that the surface has a point of rolling cylinders corrugation. A corrugation roller helps to capture wet and slippery grain that can not be done in mills with smooth rollers. Another feature is that both drums are leading.

Grain conditioner brand «Grinder Bagger» have a surface relief rolls in the form of grooves instead of teeth, allowing them full information (the gap between the rolls ≈ 0 mm) and, in turn, provides high-quality rolled grain of any size without destroying its structure. Conditioners grain from "Bio Mix" with three working rollers can simultaneously flatten the small and large grains, such as corn, beans, peas, along with oats, barley, and wheat. The width of the first slit is constant and the width of the second slit is adjusted to obtain the desired fraction.

2.4.1 Defining parameters of flattening machine

The main design parameters of calender unit are: the diameter and length of the roller, the circumferential speed of the rollers, the spring pressure on the roller. The diameter of the rollers is determined by the size of the particles in the product *B*, the size of the gap between the fixed rollers B_1 and the value friction angle *j* product on the working surface of the rollers (fig. 2.14, b). The value of the radius of the rollers can be defined as. Suppose we have two smooth roll radius *R* (fig. 2.14, a). At the time of entering into the slit the particle weighs on the rollers at points of contact with the power of n *P*. The force of friction *T* = *Pf* tangential (the figure shows the expansion of the Force for one half of the particles).

We expand the forces *F* and *T* on the horizontal and vertical components. Horizontal forces on the right and left sides of the particles, mutually-is mutually balanced. The vertical component of the friction force is directed downward. It tightens the product particles in the workspace and is equal to:

$$T\cos\alpha = fP\cos\alpha \qquad (2.47)$$

where, α – the angle of capture, compiled the direction of the reaction force *P* and line OO_1 centers, deg.

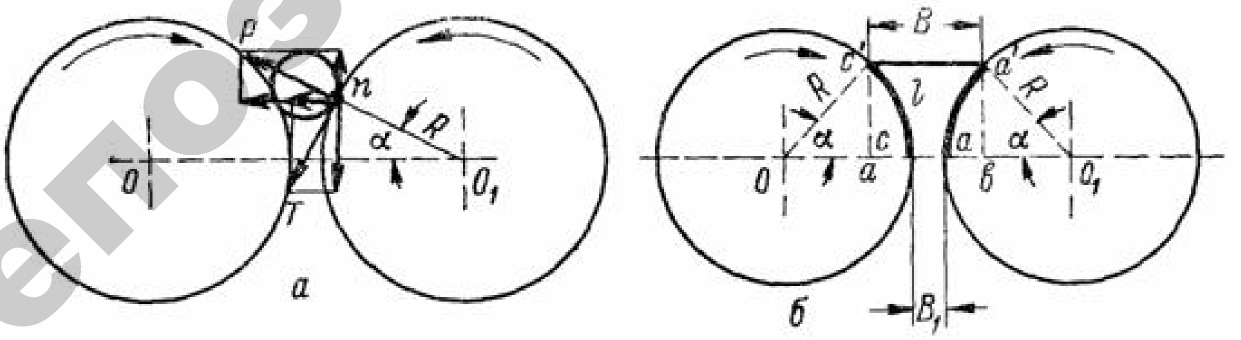


Fig. 2.14. Analyzing rollers workflow

The vertical component of the force *P* directed upwards and pushes the particle from the workspace. It is equal to *P sin α.* It is obvious that to capture particles rollers prerequisites:

$$2fP\cos\alpha > P\sin\alpha, \tag{2.48}$$

where $f\cos\alpha > \sin\alpha$ or $f > tg\,\alpha$.

Since $f = tg\,\varphi$, then

$$tg\,\varphi > tg\,\alpha. \tag{2.49}$$

Therefore, for a product gripping surfaces of rollers need to angle called gripping angle was less than the angle of friction between the particle and the roller. The relationship between the size of the product particles, the radius of the rollers and corner-scrap friction can be determined as follows (fig. 2.14, b). Let the initial size of the product particles be signed by B, and the gap between the rollers by B_1. Then we can write that

$$OO_1 = R\cos\alpha + R\cos\alpha + B = R + R + B. \tag{2.50}$$

From this equation, we find the radius of the rollers:

$$R = \frac{B - B_1}{2(1 - \cos\alpha)}. \tag{2.51}$$

However, since the angle a not exceed the angle of friction φ, i.e. the limit will be α = φ, we obtain the minimum required diameter of the rollers:

$$D_{min} = \frac{B - B_1}{1 - \cos\alpha} \tag{2.52}$$

The formula shows that the greater the particle size B, the larger should be D_{min} rollers. The effect of the gap between the rolls B_1, and the reverse friction angle φ of the product particles, i.e. greater than these values, the smaller the radius of the rollers may be.

The ratio of peripheral speeds of rollers - is the ratio between the shear and compressive forces in the roller pair. This ratio is usually denoted by the letter K:

$$K = \frac{\upsilon_\delta}{\upsilon_M}, \tag{2.53}$$

where $\upsilon_б$, $\upsilon_м$ – speed of fast- and slow- rotating rollers, m/s.

It is found that the optimum in terms of energy and quality of the grinding process, the speed of a rapidly rotating drum is equal to 5...6 m/s. For corrugated rolls optimum parity is R = 2,5; for micro rough K = 1.25. Efforts in roller conditioner determined by a load, which is created by springs of safety device. The area, which will operate this effort:

$$S_в = L_в l_в, \tag{2.54}$$

where Lв – length of rolls, m; $l_в$ – arc length on site shredding the material, m, $l_в = R\,\alpha = D\alpha/2$ (R – radius of the roll, m; α – angle of the arc, rad).

Average breaking force of grain:

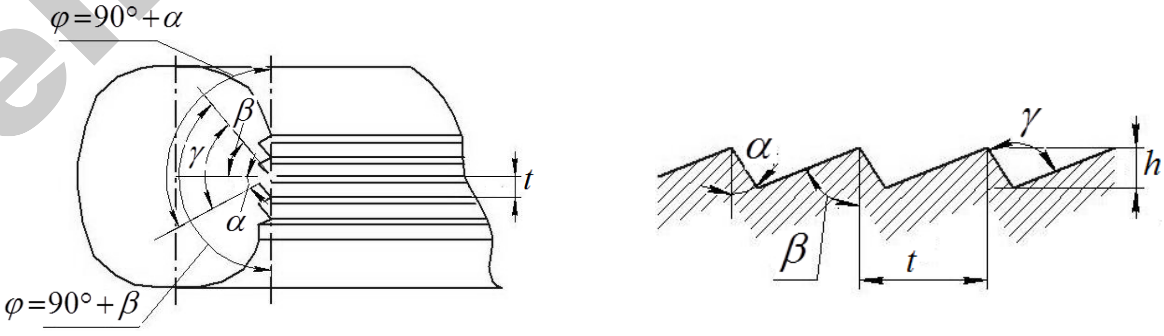
$$P_{ср} = \sigma_{сж} S_в \mu = \sigma_{сж} L_в l_в \mu, \tag{2.55}$$

where $\sigma_{сж}$ – tensile strength of the material under compression, N/m²; μ – Poisson's ratio.

The strength of the movable roller pressing springs should provide these values the efforts of the destruction of the grain.

The gap between the rollers varies from 0,03 to 1,5 mm and is adjustable grinding parameter. His change leads to a significant change in the grinding process. When operating the machine the gap has to be adjusted, because he wears the drum, and the physical and mechanical properties of the grains varies from batch to batch.

The main parameters of the working surface of the grooved rollers are: sectional shape of corrugation; relative position of the faces of corrugation; value of the slope to the corrugation forming drum; number per unit length of corrugation roll circle (1 cm). Each flute has two facets (fig. 2.15). Fringe smaller area called the face of the tip, the most - face back. Facets form between them a wedge angle γ. If the tops of riffles drop a perpendicular to the roll axis, the angle is divided into two unequal angles: α - the angle of the tip (30...40 °) and a β - the angle of the backrest (60...70 °).



a) b)

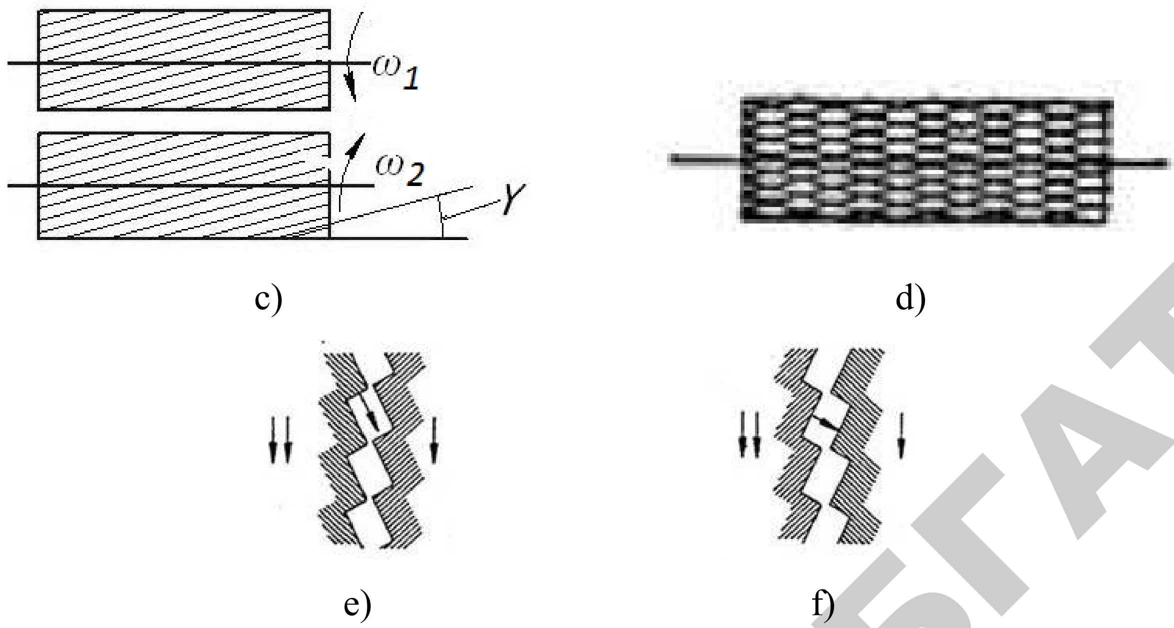


Fig. 2.15. The profile and the location of the corrugation rollers:
a, b – cross section of corrugation rollers; c – bias corrugation;
d – the intersection of corrugation pairwise rollers; e, f – interposition corrugation

The distance *h* between the circle and circle pits projections for-measurements along the radius of the drum, called a riffle height, the distance t in the circumferential direction between two vertices corrugation – step. Fluting at an angle y to the cylinder, which mitigates the work rolls. Otherwise we would have felt at clash of material on each riffle. Upon rotation of rollers towards each other, along the generatrix riffles cross in. The value of the line deflection riffles of the cylinder (slope) is generally measured in percentage terms. Recommended angle y = 6...10 % feed for grinding cereals.

Fluting on pairwise rollers usually has two options – the "edge on the tip" and "stupid on stupid." When placing the corrugation "edge on the island" is crushed product particle supported the cutting edge of the slowly rotating drum and the cutting edge of a rapidly crushed. In this case, the resulting particles are destroyed and cut shearing, which contributes to the formation of large grits fractions. When placing the corrugation "stupid on stupid" particles break down into smaller fractions. This or that the relative position of corrugation rollers pairwise reached by their respective laying in the roller coater. Obviously, first version of disposition corrugation should be used for preparation of feed. Slicing density corrugation is usually 4...12. The more it, the smaller the particle size distribution obtained by the product.

Performance of roller conditioners can be calculated, if you imagine the process of crushing the movement of the belt material width equal to the length of the roll of L, and the thickness of the gap Δ between 2 adjacent rollers. Then in one revolution of the shaft, the volume of material passing through the gap, counted:

$$V = \pi RL\Delta . \tag{2.56}$$

Then conditioner performance can be determined by the formula

$$Q = \pi DL\Delta\rho h , \tag{2.57}$$

where n – roller speed, min^{-1}; ρ – grain density, kg/m^3.

Since the rolls are moved apart by the width b, depending on the actual effort grinding and severity of safety springs, the actual performance is

$$Q = \pi DL\left(\Delta + b\right)n . \tag{2.58}$$

Performance conditioners with grooved rollers is determined by the formula

$$Q = (\Delta + h)L\upsilon_{cp}\rho , \tag{2.59}$$

where h – height of the reef, m; υ_{cp} – the average speed of rotation of roller in, m/s.

Engine power Ndv conditioners to overcome all resistance when operating the machine:

$$N_{\partial в} = \frac{N_1 + N_2}{\eta} , \tag{2.60}$$

where N_1 – the power consumed in the crushing taking into account the friction material on the drum, kW; N_2 – power consumed by friction bearings, kW; η – transfer efficiency, η = 0,90...0,95.

Engine power N_1 necessary for the destruction of the grain:

$$N_1 = 2\pi n\sigma_{сж} Ll\mu\Delta f \tag{2.61}$$

Power N_2, necessary to overcome the friction in the bearings of the two rolls, kW:

$$N_2 = 2\pi nd_{ш} f_1 G \tag{2.62}$$

where $d_ш$ – the diameter of the shaft journal, m; f_1 – the coefficient of rolling friction, is a-tion to the shaft, f_1 = 0,001; G – the load on the bearings, N.

2.4.2 Design and definition of the parameters of refiner of grain of milk wax ripeness

To grind grain moisture content of 14 to 40 % of the shredder is designed with a vertical working chamber, fig. 2.16. Chopper consists of a vertically arranged working chamber 1, within which, coaxially mounted to the rotor 2 blades 3 forming top and bottom grinding step, two blades in each tier. Each tier of the upper stage is set for 4 knives and lower knives 8. In order to evenly distribute the load on the rotor blades 2 each tier are displaced about a vertical axis relative to the adjacent blades.

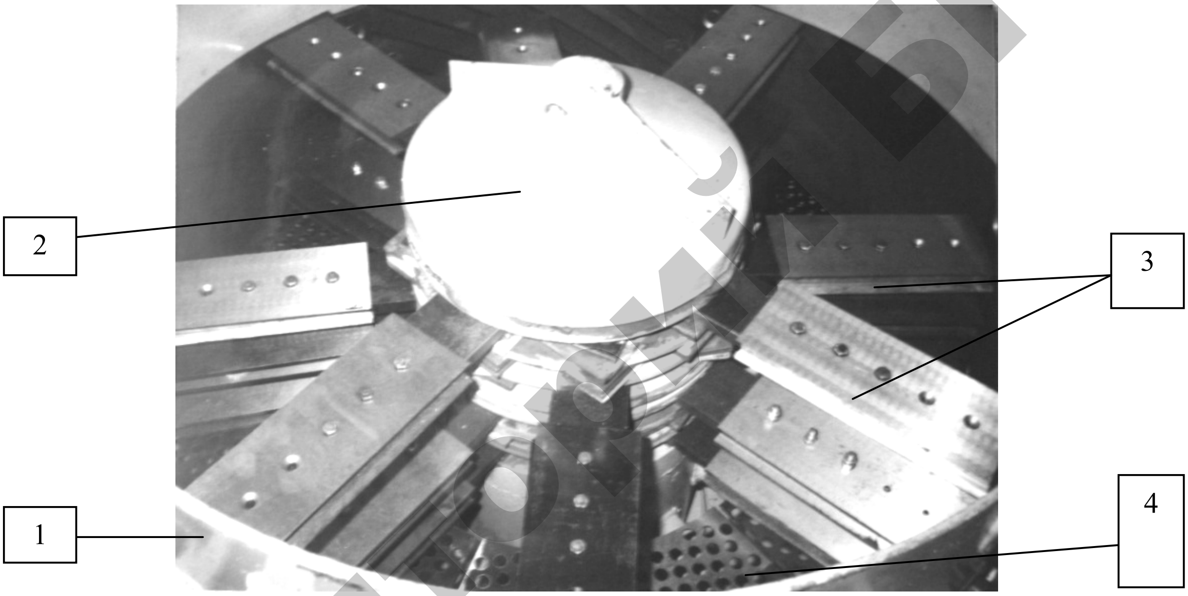


Fig. 2.16. Grain chopper with moisture content of more than 14 %:
1 – working chamber grinder; 2 – rotor; 3 – blades;
4 – sieve with a hole diameter of 15 mm

For wet milled grain separation in the processing chamber 1 at the last tier of blades, one set sieve 4, the diameter of cylindrical holes which equals 15 mm. The gap between the sieve 4 and 3 knives lower sieve is 5 mm.

Corrugated deck reefs which have a triangular shape used as passive working elements mounted on the inner perimeter of the working chamber. Rational wedge angle of the reef should be in the range of $80^0...100^0$, the slope faces the projection angle value must be within the range $65^0...75^0 \geq \tau \geq 40^0...58^0$. Grain chopper milk wax maturity is as follows. The wet grain is loaded into a vertically extending working chamber 1. Since the grain moves perpendicular horizontal rotating blades 3, the establishment of the counter-effect is provided on the entire perimeter of the working area of knives 3, destroying the structure of the grain.

Moving onto the surface of the sieve 4, grain regrinding and cha-particle comes into the working area of the holes. Repeated exposure grain knives cut it into small particles, each of which structure is not broken, and hence there is no loss of the most valuable part of the feed nutrient - cell sap.

The grain is delivered to the storage area, where stored. As the grain cut into individual particles, that is their displacement with respect to each other until a snug fit of the surfaces with less effort and reduces the consumption of preservative. Since the geometrical dimensions of the moist grain dropped without breaking the structure of cells and densely aligned adjacent surfaces of the particles, the violation occurs monolith compacted feed. This eliminates the formation of air voids in the compacted mass, and hence the oxidation processes, significantly reducing the quality of food.

Chapter 3
MECHANIZATION ROOT CROPS FEEDING PROCESS

3.1. The device and the machinery workflow for feeding root crops

Root crops are applied in the form of juicy lactiferous feed in dairy farming. They may be fed to animals in the form of a whole (excluding small) and a powdered - with a view to the mechanization of distribution and their inclusion in the process feed mixture composition.

Root crops usually contaminated soil, sand, and may contain impurities (stones, pieces of wood, metal, etc.). Before feeding to animals so they must be carefully cleaned, washed and milled. The actual pollution of root crops after harvesting may achieve up to 12...20 % by weight or more. Permissible pollution of the same after washing should not exceed 2...3 %. The degree of contamination is determined empirically by the formula

$$\delta = \frac{m_1 - m_2}{m_1} \cdot 100\%, \tag{3.1}$$

where m_1 – the total weight of the product portion to wash, kg; m_2 – mass portion of the clean product kg.

Product should be kept in the water for long, otherwise leached valuable nutrients (starch, sugar). Sink continuous of total time in water should be between 60...120 s. This time consists of the water absorption time (60...90 s), i.e., root crops stay in the bath loading and washing time (30...40 s), i.e. stay in the trough auger. According to the experimental data water consumption compose an average value of 250...300 kg per 1000 kg of root crops. Root crops are ground just before feeding, or no more than 1,5...2 hours before the feeding, as sliced they rapidly deteriorate. Thickness of cutting root vegetables when fed to cattle should be 10...15 mm, the calves - 5...10 mm. Physical and mechanical properties of root crops are presented in table.3.1

The specific resistance to cutting root crops varies between 1,48...1,96 kN / m.

For the separation of impurities from the root crops, two main methods are used:

1. Isolation weight impurities by using water.

Advantage:

– high-performance cleaning root crops;

– a fairly simple method of separating solid impurities;

– great flexibility of use for washing machines of various root crops (carrots, potatoes, beetroot);

–The relatively high productivity of machine used.

Table 3.1 Physical and mechanical properties of root crops

Indicators	Potatoes	Beet	Carrot
Diameter (width) mm	65...100	150...180	40...60
Length, mm.	75...100	150...200	150...200
Moisture,%	75	80	85
Bulk weight in kg / $cm3^3$	700	650	580
Modulus, MPa	34...54	54...83	69...83
Compressive strength, MPa	10...18	13...24	13...25

Disadvantages:

– relatively short shelf life of the final product (the presence of moisture reduces the shelf life);

– large amount of water used for the process;

– low temperature fluid freezing (0 ^{0}C).

2. Dry cleaning of root crops.

Disadvantages:

– large power consumption of the process;

– presence of dust in the performance process.

Advantage:

– no flow.

Root crops washing machines divided into the following types of machines.

Depending on the root crops cleaning technology they are distinguished:

– periodic;

– continuous.

Depending on *the design of the working bodies* root crops washing machines are divided into:

– cam washing machine (fig. 3.1, a) has a bath with a grid and a working body - the shaft with fortified on it with his fists on the helix. At the output end mounted on the shaft of the unloading blades which tubers are transferred outside the tub. Bath is a trench, partitioned into sections by length, which are arranged stone trap and hatches for removal of stones and mud. Water discharge processing is typically less than 0.6...0.8 l/kg.

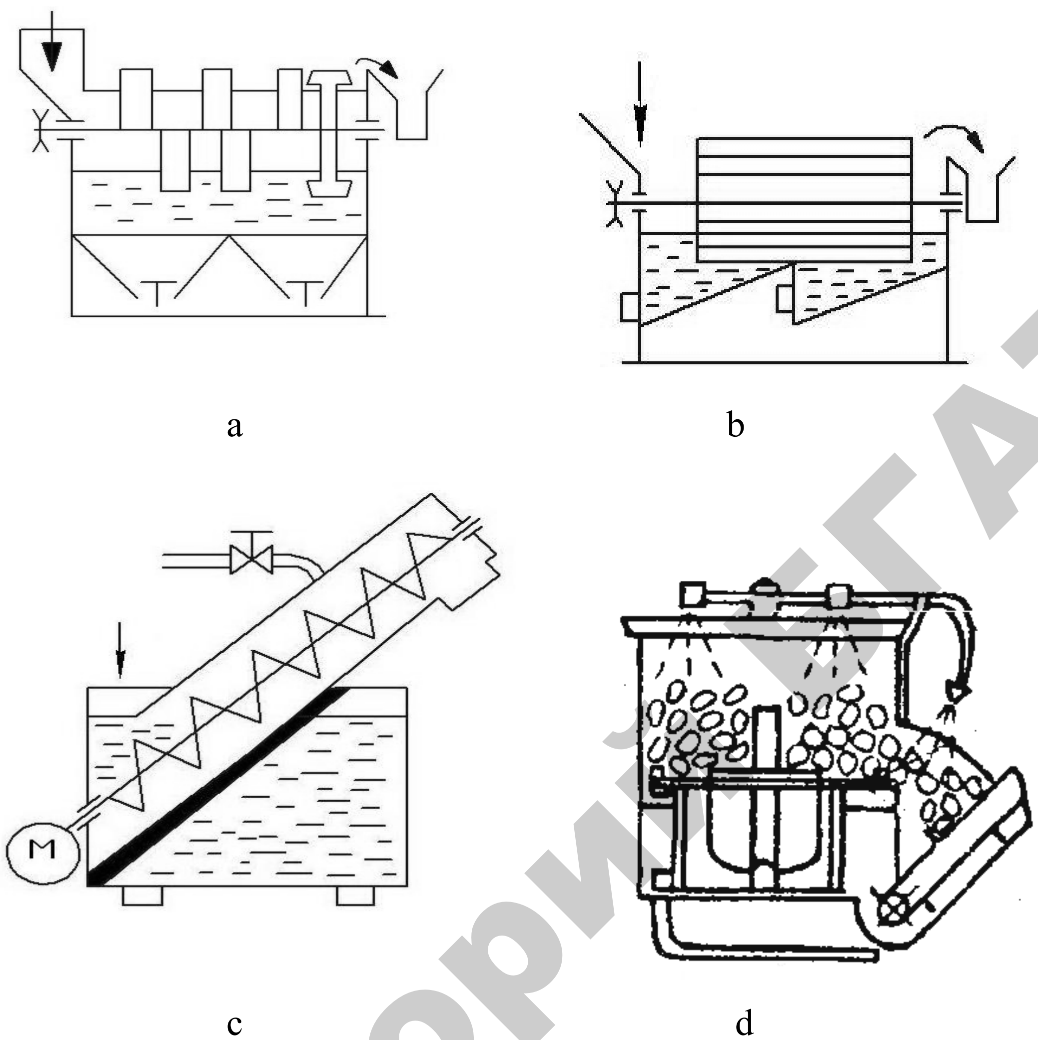


Fig. 3.1. Root crops washing machines:
a –cam; b – drum; c – auger; d – disk

– drum washing machine (fig. 3.1, b) actuator has a plate-drum rotating in a water bath. Root vegetables, doing a bath butt and moving along the drum, are the way in which is exempt from pollution. Recent settle to bottom of the bath, and washed roots are ejected from the blade drum, reinforced on its inner wall at the unloading end. Machines of this type are also used for cleaning (without water).

– auger washing machine (fig. 3.1, c) pre-constitutes a screw mounted in a bath. The screw is laid in the pipe having the slope to the horizontal 25...90 °. Pipe grid at the bottom of the supply part with the screw placed in the hopper with water, and root crops. By rotating the screw windings product capture and move it along the housing to the exit port. Towards the product in the auger housing washing water is supplied from the sump stream bath. Contaminants deposited on the bottom of the tub, which are periodically removed through a special hatch. An advantage of screw washers is their high capacity (from 3 to 6 kg/s) and the simplicity of the device.

– disk root cleaner (fig. 3.1, c) has washing working body in the form of a flat disk, which is welded to the surface of the projections that serve as shakers. Tubers, getting on a rotating disk, make with it a circular motion and relative to the disk surface, mixed under the influence of the projections. At the same time supplied from the ring sprinkler water washes dirt. Disc washer have not stone races and have high-water course in the processing of products. The main wash parameter is capacity, which can determine the formula

$$Q = V\rho\frac{\varphi}{\tau_M}, \tag{3.2}$$

where V – volume of the washing chamber, m^3; ρ – density of feed, kg/m^3; φ – bath fill factor (0,3...0,4); τ_M – stay tubers in the washing chamber (τ_M = 60...90 s)

Volume of the washing chamber:

$$V = \frac{\pi D^2}{4} H, \tag{3.3}$$

where H – height of washing chamber, m.

Diameter of washing chamber:

$$D = \sqrt{\frac{Q\tau_M}{\pi H \rho\varphi}}, \tag{3.4}$$

An important indicator of the centrifugal cleaning, is to determine the minimum *angular velocity* of the washing disc. To determine consider the scheme of forces acting on the tubers on shredder rotor (fig. 3,2).

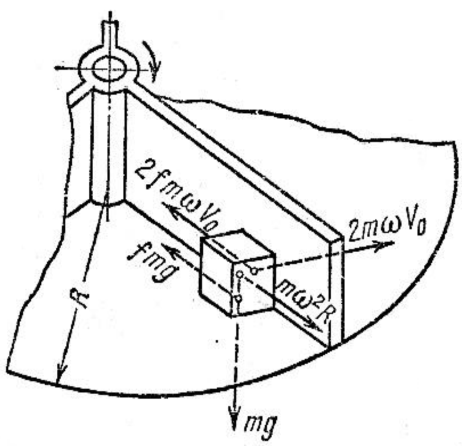


Fig. 3.2. Driving forces acting on the tubers on the wing

Moving along the plane of the tuber prevents $F_{тр}$ force of friction caused by the action of its gravity $F_{тяж}$:

$$F_{тр}=F_{тяж}\,f,$$

A condition in which the tuber will move by the wing has the form

$$mgf \leq F_{ц},$$

where $F_{ц}$ - centrifugal force N, determined by the formula

$$F_{ц}=m\,\omega^2 R. \tag{3.5}$$

Then

$$mgf = m\omega^2 R, \tag{3.6}$$

from whence

$$\omega = \sqrt{\frac{gf}{R}}.$$

Moving along the plane of the tuber of wing prevents Ftr force of friction caused by the action of its gravity $F_{тяж}$:

$$F_{тр}=F_{тяж}\,f,$$

A condition in which the tuber will move by the wing has the form:

$$m\,g\,f \leq F_{ц},$$

where $F_{ц}$ - centrifugal force, N, determined by the formula,

$$F_{ц}=m\,\omega^2 R.$$

Then

$$m\,g\,f = m\,\omega^2 R,$$

where the minimum angular velocity of the washing disk

$$\omega = \sqrt{\frac{g\,f}{R}}.$$

Crushers of root crops are distinguished (fig. 3.3) into disc, disc with a vertical shaft, drum and fixed knives. To include root crops Root cutting grinders, shredders, different from each other device working bodies and the degree of grinding material.

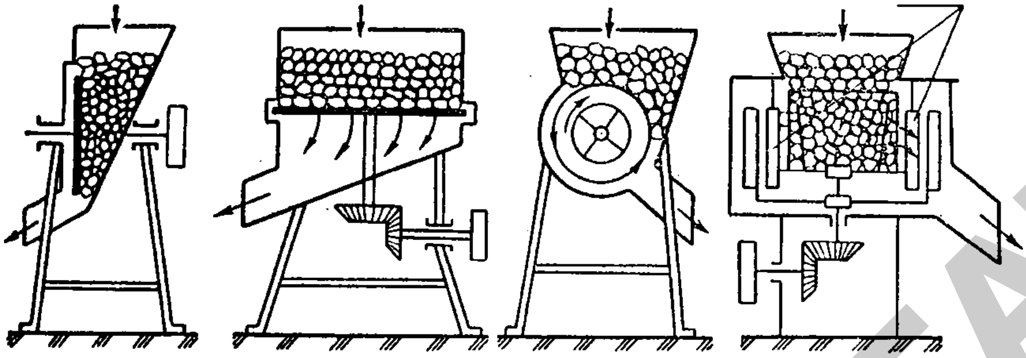


Fig. 3.3. Crushers of root crops

For grinding crops root washers combined with grinding machines and convered into root crops washing choppers. The workflow is based on all washes pollution department at friction root crops on the working bodies of the machine and each other. Dirt, diluted with water, is deposited in the appropriate containers machines.

Root crops washing chopper (fig. 3.4) is out of the bath 12, the vertical screw 11 with a disk-activator 13, the chopper 9, 2 conveyor for unloading stones, electrical equipment and drive. Bath 12 mounted thereon and units installed on the frame 1. The screw 11 is mounted in the bath itself, the upper end of the shaft which is situated in the bearing, which is located in the housing. The lower end of the screw shaft is supported on spherical bearings and nylon heel is activator 13.

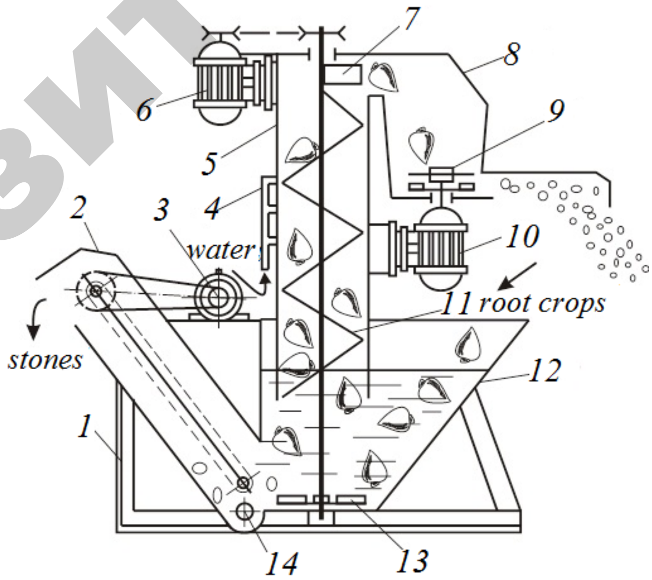


Fig. 3.4. Technological scheme of work of root crops washing shredder:
1 – frame; 2 – conveyor-stone-extrector; 3, 6 and 10 – motors;
4 – sprinkler water supply; 5 – casing; 7 – ejector; 8 – case of shredder;
9 – chopper; 11 – the screw; 12 – washing tank; 13 – disk-activator; 14 – hatch

Shredding unit 9 consists of a cast body and two discs, which are mounted directly on the shaft of the two-speed motor. The top drive is used for the initial shredding root crops. To him special bolt securing the two horizontal blade (fig. 3.5).

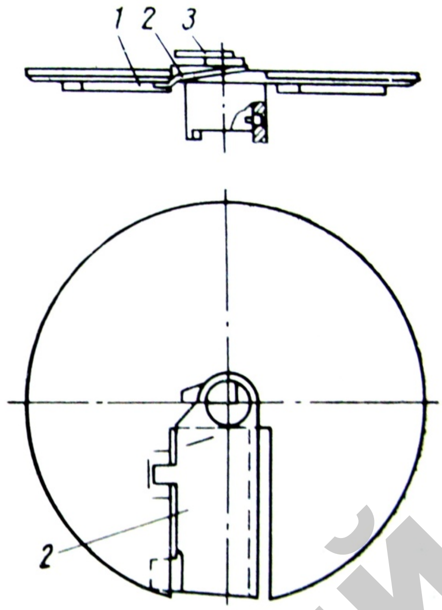


Fig. 3.5. Upper disc:
1– disc; 2 – knife; 3 – screw

Lower blade disc for fine grinding and root crops consists of upper and lower split disks of two internal and two external blades and knives with four vertical outer and inner grinding (fig. 3.6).

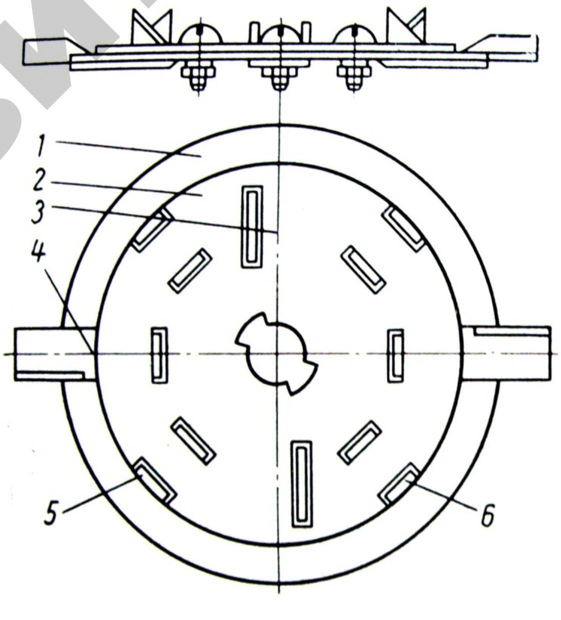


Fig. 3.6. Lower disc:
1 – lower disc; 2 – top drive; 3 – blade; 4 - outer blade; 5, 6 – knife

All working parts grinder successively planted on the motor shaft and fixed. Chopper also has a detachable deck (fig. 3.7).

Fig. 3.7. Deck:
1 – the case; 2 – deck; 3 – drive; 4 – knife

When preparing for cattle root crops deck removed, and the chopper motor speed reduced to 500 min^{-1}. To prepare root crops for pigs give deck and the motor speed is set 1000 min^{-1}. If necessary, processing frozen root crops set on the upper drive gear horizontal blades.

Considered root crops chopper also provides washing potatoes without crushing. In this case, removed from the shredding softener deck and upper disc, and in their place establish the disc ejector. In this case, the chopper motor speed should be 500 min^{-1}. Stone extractor transporter 2 is intended for unloading from the bath stones, sand and dirt. It consists of a main and a folding covers, swinging conveyor with scrapers and drive. The main casing is installed at the bottom of the door with a valve to clean and drain the water from the bath. The drive consists of a conveyor located on the bracket bath gear motor and chain drive. Each actuator chopper (auger shredder and conveyor) is individually driven by a motor. Before starting root crops washing -grinder to the work necessary to make sure that there are no foreign objects, and then open the faucet and fill the wash tub with water to the level of the overflow tube. After this series include a shredder and screw conveyor for unloading stones.

Auger motor include only when the motor chopper. It supplies the root crops to the rotating cutting blade, prevents clogging of the chopper at the time of launch. Turning conveyor is made independently of other mechanisms. The batch is made in a bath of root crops in the presence of water in the tub and at the working screw. Normal operation of the shredder is provided with a continuous supply of root crops. When all the arrangements are working, root crops with a loading conveyor receives the washing tub 12, where the action of water flow created krylachom 13, washed, washing captured screw 11 and transported up into the shredder chamber 9. As we move up the tubers are washed in the screw 11 counter flow water supplied through the water inlet 4. The amount of irrigation water supplied through the sprinkler, and is regulated by a valve depends on the degree of contamination of potatoes. As the clean water flow from sprinkler 4, merges into a dirty sewer through a manhole 14.

Peeled root crops are crushed by horizontal upper disc blades and under the influence of centrifugal forces come to the deck, which finally crushed the vertical blades and thrown out of the lower disk blade through the guide sleeve outside. Stones, lumps of earth and other foreign objects, having a higher density than root crops, sink to the bottom of the bath 12 and krylachom 13 are sent to the receiving neck stone extractor -transporter 2, which are ejected out of the machine. At the end of root washing shredder must be cleaned of dirt and crushed mass and mesh filter on the suction side of the pump. *In order to process* large root crops washing chopper (without shaft) screw can be fitted, which consists of a helix with a pitch of 380 mm (fig. 3.8). Application auger (screw conveyor) can process larger roots - with a diameter up to 350 mm.

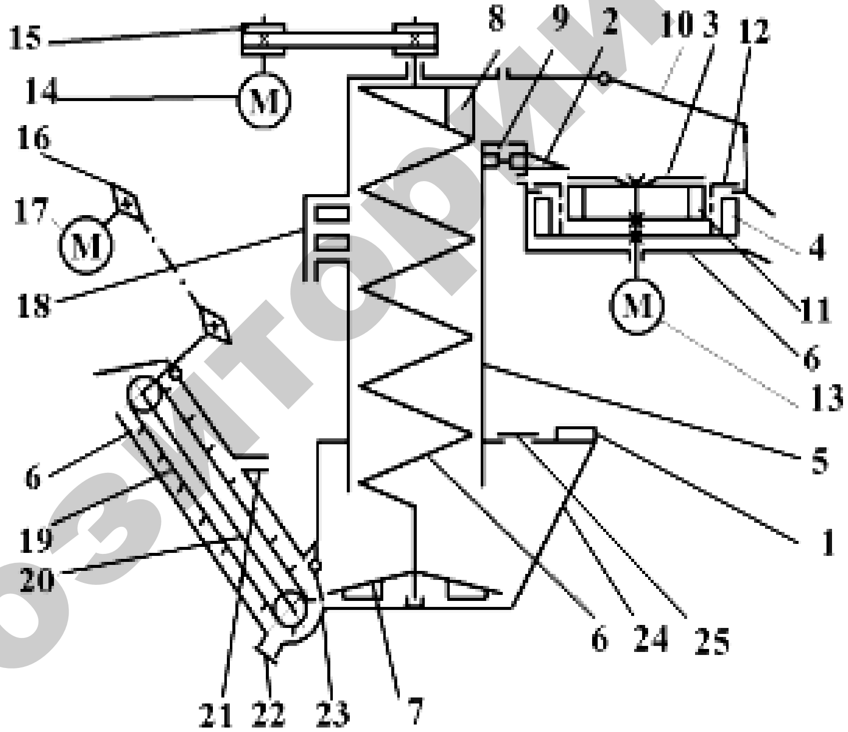


Fig. 3.8. Driving root crops washing chopper with without shaft screw*:*
1 – feeding tube; 2 – emphasis; 3 – removable blade horizontal; 4 – ejector;
5 – auger housing; 6 – screw; 7 – disc-activator; 8 – vane; 9 – limit switch;
10 – observation hatch; 11 – inner blade;12 – deck; 13, 14 – electric motor;
15 – transmission belt; 16 – transmission chain; 17 – gear motor;
18 – fitting for water supply; 19 – scraper conveyor; 20 – rod;
21 – outlet for removing excess water; 22 – door to remove dirt;
23 – valve; 24 – bath; 25 – inspection hatch

When the root crops are loaded into a rotating drum for dry cleaning, it separates the bulk of the earth, straw and plant residues. From the drum set with a gap with respect to the loading tray, root crops fall into the wash-tub stone-extractor, where the flow of water created by the impeller and screw threads, washed and fed to the grinding machine. The stones with a diameter exceeding 100 mm and the other heavy impurities are separated from the root crops still on the inclined wall of the wash tray, and getting on the wheel blade dropped to an inclined conveyor.

Chopper-stone trap can be equipped with universal (fig. 3.9) mechanism for dry cleaning from the ground plant residue, separating stones, washing and shredding root crops of all types and sizes. The use of dry cleaning reduces to 50 liters of water-course races on the 1 m roots and reduces the cost of the cleaning process. Dry pre-cleaning drum is made 660 mm in diameter and 950 mm in length. It is a shell with two grooves for V-belts connected to the rollers, which are welded on one side to the mantle, and the other left open. The drum is rotated by the motor, based on two pairs of support rollers mounted on the frame. The third pair of rollers arranged on the housing and creates a closed system.

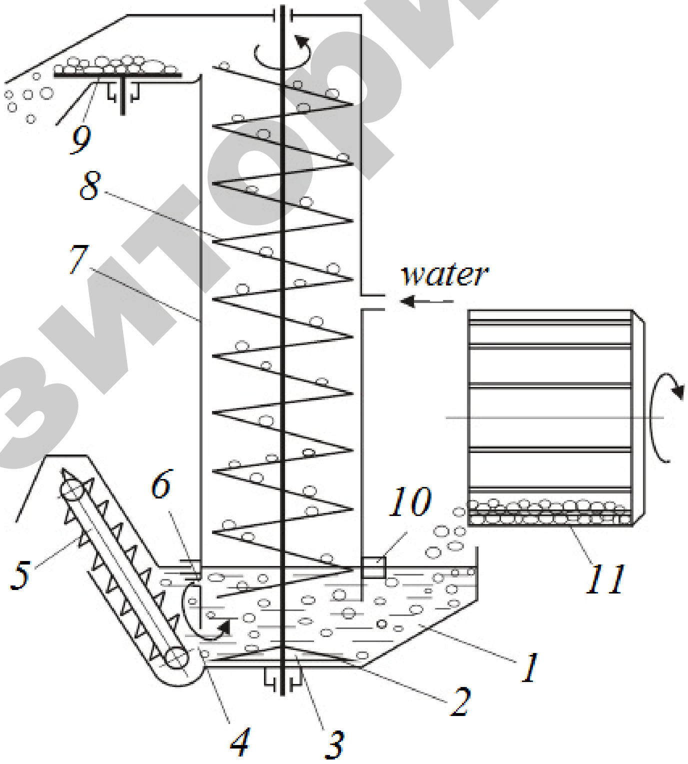


Fig. 3.9. Technological scheme of the unit:
1 – bath; 2 – cleaning disk; 3 – blade disk detergent; 4 – impurities exit window; 5 – conveyor for removal of impurities; 6 – ring slot; 7 – auger housing; 8 – screw; 9 – chopper; 10.11 – drum dry cleaning

Wash chopper root crops (fig. 3.10) can be equipped with a chopper drum equipped with hammers and equipped with shock-proof deck. If necessary, the chopper can be shut off and then washed roots generally come in the form of vehicles.

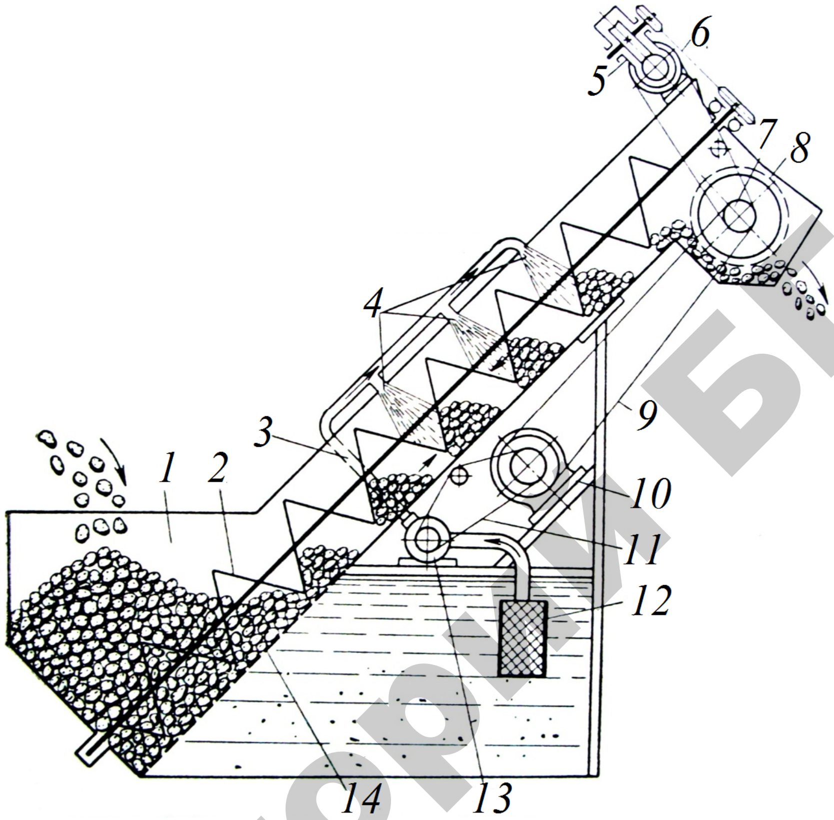


Fig. 3.10. Driving root crop harvester working process with a chopper drum type:
1 – receiving hopper; 2 – screw; 3 – pipeline; 4 – nozzle; 5, 10 – motors;
6 – chain transmission; 7 – shaft shredder drive; 8 – pin drum; 9, 11 – transfer;
12 – filter; 13 – centrifugal pump; 14 – socket with a lattice

3.2 Calculation of the chopper-wash parameters root crops

For transportation of root crops in the washes used auger. When determining the rotational speed of the screw washer disposed at an angle φ to a vertical plane, we consider forces acting on the root of the screw coil (fig. 3.11).

Moving tuber prevents friction force:

$$F_{mp}=f\,m\,g\,\sin\varphi, \tag{3.7}$$

where f – coefficient of friction; φ – the angle between the plane of the screw turns and twirl locally-acting force of gravity, degrees.

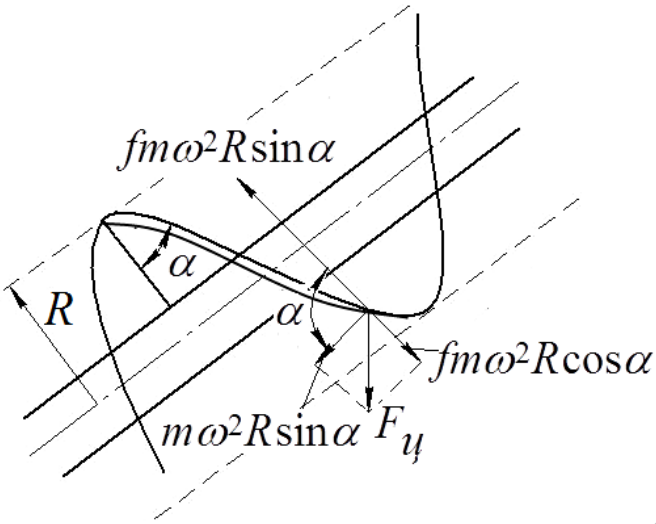


Fig. 3.11. The scheme to determine parameters of the washing screw

We expand on the strength of the gravitational force of normal pressure and mg sinφ tangent force *mgcosφ*. Centrifugal force to expand the power of the normal pressure on the flat coil:

$$F_ц = m\,\omega^2\,R\,sin\alpha, \tag{3.8}$$

and the force of a normal pressure at the end of the screw

$$F_ц = m\,\omega^2\,R\,cos\alpha\,. \tag{3.9}$$

Moving on tuber worm turns the plane lets you force-title of the centrifugal force:

$$F_ц = f\,m\,\omega^2\,R\,sin\alpha, \tag{3.10}$$

where R – screw revolution radius, meters; α - angle of the screw turns,degree; ω – angular speed of the screw turns, s^{-1}.

Then the maximum angular velocity can be determined from the expression:

$$fmgcos\varphi = fmgsin\varphi + fm\omega^2 Rsin\alpha, \tag{3.11}$$

from whence

$$\omega = \sqrt{\frac{2\,g\,\cos\left(1 - ftg\varphi\right)}{D\,\sin\alpha}}\,, \tag{3.12}$$

screw speed

$$n = \frac{30}{\pi}\sqrt{\frac{2g\cos\left(1 - f\,tg\varphi\right)}{D\sin\alpha}}\,. \tag{3.13}$$

where D – diameter of the screw turns, m.

Performance can be determined knowing the diameter of the screw washer and a predetermined mass of feed passing per unit of time:

$$Q = (D^2 - d^2)S\,\rho\,\omega\,K_1\,K_2, \tag{3.14}$$

where D – diameter of the screw, m; d – diameter of the screw shaft, m; S – pitch of the screw, m; ρ – density of feed, kg/m^3; K_1 – coefficient of filling auger working space, equals K_1 = 0,3...0,4; K_2 – filling screw is assumed to be from 0,25...0,35. For the case when the screw is set at an angle to the horizontal plane, 0.45...0.65.
Screw length:

$$L_ш = S\,t_м\,\omega\,/\,2\pi, \tag{3.15}$$

where $t_м$ – residence time in the sink, ($t_м$ = 60...120).

When the mass fraction of impurities root crops (6…7 %) of the screw length should be 2,5...3 m, with 20 % of pollution is required screw length up to 6 m, which is structurally difficult to perform. In such cases, the root successively passed through two washers.

Volume of washing chamber:

$$V = \frac{\pi D^2}{4} H\varphi\,, \tag{3.16}$$

where H – height of the wash chamber, m; φ – bath fill factor (0,3...0,4.)

The height of the wash chamber:

$$H_м = (0{,}85 \ldots 0{,}95)\,D. \tag{3.17}$$

The power required to drive the screw washer spent on:

– Product lift:

$$N_1 = Q\, \upsilon_к^2 = Q\, H^2\, \omega^2, \tag{3.18}$$

where Q – wash performance, kg/s; $\upsilon_к$ – speed lift tubers, m/s; H – height of lifting of the product, m; ω – angular velocity of the screw, c^{-1}.

– To overcome the friction of the inner surface of the casing of the screw:

$$N_2 = F_ц\, \upsilon_к f, \tag{3.19}$$

where $F_ц$ – centrifugal force moves the tubers on the inner surface of the housing, N.

– To overcome the frictional force of the helical surface of the screw:

$$N_3 = F_{тяж} R f \omega, \tag{3.20}$$

where $F_{тяж}$ – the force of gravity feed, H,

$$F_{тяж} = m\, g. \tag{3.21}$$

Adding components define the capacity of the drive washer screw.

3.3 Thermal processing of feed

Heat treatment of exposed potatoes, food waste, coarse and concentrated feed. The goal of treatment - improving the digestibility and disinfecting. Installations for the heat treatment of feed may be classified by the following features:

– by design – steaming chamber, herbal infusion container, steamers, mixers, potatoes steaming units;

– according to the process steps - continuous and batch;

– by type of use - landline and mobile;

– for treatment regimes - at atmospheric pressure and elevated (boilers, food processing waste, straw barotermochamber);

– on purpose - for potatoes, roughage and food waste.

By steamer the following requirements are applied:

– the possibility of mechanized loading and unloading the product;

– uniform heating of the total product;

– the safety and ease of maintenance;

– reliable operation;

– the product must not be contaminated with foreign matter.

As the coolants can be used gas and water vapor.

Using gas as coolant has some disadvantages: a low heat capacity and low heat transfer coefficient, a small efficiency when heated, and in case the flue gases and the treated material clogging the combustion products. Therefore, practically no gases are used for the thermal treatment of feed material.

The use of steam has following advantages over the water:

– enthalpy of saturated steam at atmospheric pressure to almost 6 times the heat content of water heated to a temperature of 100 ^{0}C;

– the ability to establish an ongoing process of steaming;

– no need to repeatedly heat the water, and the process necessary for a much smaller amount of water, which is usually drained after steaming cycle;

– the ability to organize the process of heat treatment at elevated steam parameters without compromising the security of work;

– infusion container device easier than digesters;

– heating feed mixtures with water can not be done if the zoo-technical requirements do not allow the wetlands feed mixtures.

Currently, the most widely used devices that use steam as a heat carrier. For its production industry produces boilers - steam generators operating on liquid and solid fuels. All boilers operate at low pressure not exceeding 0.07 MPa, and the steam temperature is 110...115 °C. Heat transfer can be carried out by direct contact of the heat carrier with the heated material and through the dividing wall.

Heat-exchange apparatus with a dividing wall in kormoprigotovlenii not apply because of the complexity of their design and to reduce the heat transfer efficiency.

Technological schemes that are subject to heat treatment of feed, can be very diverse and depend on the destination machine, and on the zootechnical requirements on the final form of the product. If the heat treatment must be subjected to a dry feed (straw, chaff, concentrates), its pre-soaked for increasing heat conductivity and accelerate the process of heating to a predetermined temperature.

Increased feed of moisture associated with increased thermal energy consumption for its handling, since heating to a predetermined temperature must be not only the product, but also the weight of water. Therefore, feed moisture should be minimized. Straw is best steamed loose state. Then the pairs free up every straw and quickly it heats. Soaking heated straw better in densified form, because it does not cool down quickly. Concentrated feed steams with continuous stirring better. This will be rapid and uniform heating of it. Potatoes have sufficient porosity to vapor passage, so it steamed in vats filled to the top. Resulting in potato of hardening condensation is removed as it contains unhealthy animal substance - solanine. Heat treatment of feed is used and the number of machine units, differing in design, size and performance.

Batch mixers are designed for the preparation of feed mixtures humidity 60 ... 80% of the crushed green and juicy fodder (root crops, silage, melons, etc.), as well as animal feed and concentrates (crushed coarse grains) on the pig and other farms. Feed can be formulated with or without steaming it.

The mixer of this type (fig. 3.12) consists of the following components and assemblies: housing 1, the agitator 3, installed inside the case, unloading auger 9, valve control system unloading the neck 7, steam distribution system 11, the drive frame, the motor 12, the gear 13, a V-belt transmission couplings sprinkler 15, temperature gauge, a gear motor 10 and the clutch.

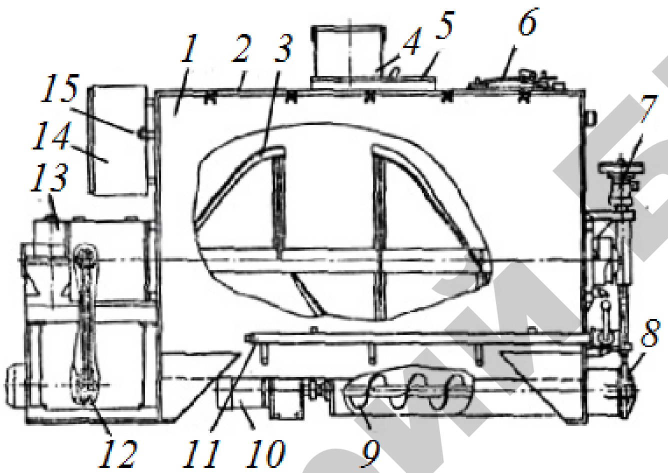


Fig. 3.12. Driving batch mixer:
1 – case; 2 – cover; 3 – mixer; 4 – feeding tube; 5 – damper; 6 – inspection hatch; 7 – drive unloading gate; 8 – unloading gate; 9 – unloading auger; 10 – unloading auger drive; 11 – steam manifold; 12 – electric motor; 13 – reducer; 14 – a control panel; 15 – sprinkler

The body of the mixer is a container for the preparation of feed mixtures. At the bottom of the housing 9 mounted unloading auger driven by a motor 10 through a gear coupling. The top of the housing cover 2 is attached to the manhole 6, and a boot neck 4. Sprinklers 15 in the housing end walls feed water to the reservoir through the flow mixer. The main working body of the mixer is a mixer 3, mixed feed and feed it to the unloading area. The agitator is driven by an electric motor 12 through a V-belt transmission and gearbox 13.

Steam distribution system include three-way valve, connecting flange, the main pipe, fittings and plugs for easy cleaning of the steam distribution system of feed mixtures residuals. Crane gives the steam and water into the mixer. After steaming pairs overlap and while water is supplied to the mixer, which prevents feed nozzles

To monitor the temperature of steamed food on the end wall of the body of the mixer mounted temperature gauge.

The control system (fig. 3.13) consists of an electric motor 3, the rotor 2, the rod 1, the upper and lower limit switches 8, 6 and 7 of the lever.

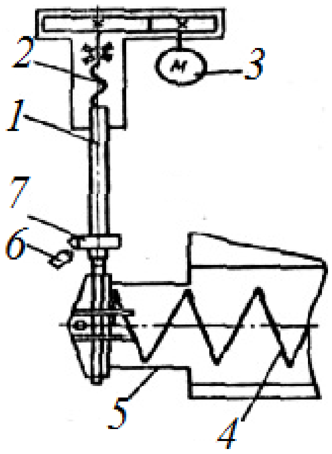


Fig. 3.13. Gate drive circuit unloading auger mixer:
1 – stock; 2 – screw; 3 – electricmotor; 4 – screw; 5 – case of the screw;
6, 8 – end turn off, whether; 7 – lever

When unloading of ready mix include motor drive for 3-engines. Stem 1, lifted the screw 2 at its rotation motor 3, lifts latch, opening unloading neck. In the uppermost position, the lever 7 by clicking on the limit switch 8 disables the motor drive 3 and will include unloading auger mixer. To prepare the feed mix without scalding include me-drive stirrer and charged with feed mixer components.

The enrichment feed by liquid fodder yeast, solution of molasses and other additives is performed after filling the main components of the mixer. After 10...15 minutes the finished ration is discharged. Preparation of wet mixtures with steaming made following manner. The mixer is poured the calculated amount of water is supplied to the steam that heats the water up to 90...95 °C. Turn on the mixer drive and load food to be steaming. After their steaming steam supply is stopped, and the food is kept 1...3 hours in a heated state. Then, cold water is poured into a mixer and the other components simultaneously charged. After 10...15 minutes stirring ready foodmixture discharged into vehicles.

The last valve position is adjusted in the following sequence: a fully closed or open neck of unloading, a limit switch is fixed so that the lever on the rod 7 has a power reserve of up or down 1,5...2 mm. To check the drive off valves when opening the unloading of the neck, by rotation of the handle shaft gear damper is removed up to the value of the stroke, and the piston rod presses the limit switch 8 disables the gate drive motor and includes the unloading auger drive. Limit switch installed near a manhole on the cover of the mixer body, at the opening of the hatch cover must disconnect electric mixer control circuit.

Potatoes steaming unit for washing, separation of light and heavy impurities, steaming, grinding and unloading potatoes is shown on fig. 3.14. He works in a cyclical mode.

Inside steaming tank is a disk valve, through which the potatoes evenly distributed throughout. The distributor is also a kind of level sensor - when filling the vat potatoes brake discs and filling auger motor is switched off. The continuation of the unloading auger 15 is crusher, in the middle of which is set crusher consisting of six grinding knives 14. The shaft of the screw passes through the inside of the hollow shaft of unloading and has a higher rotational speed.

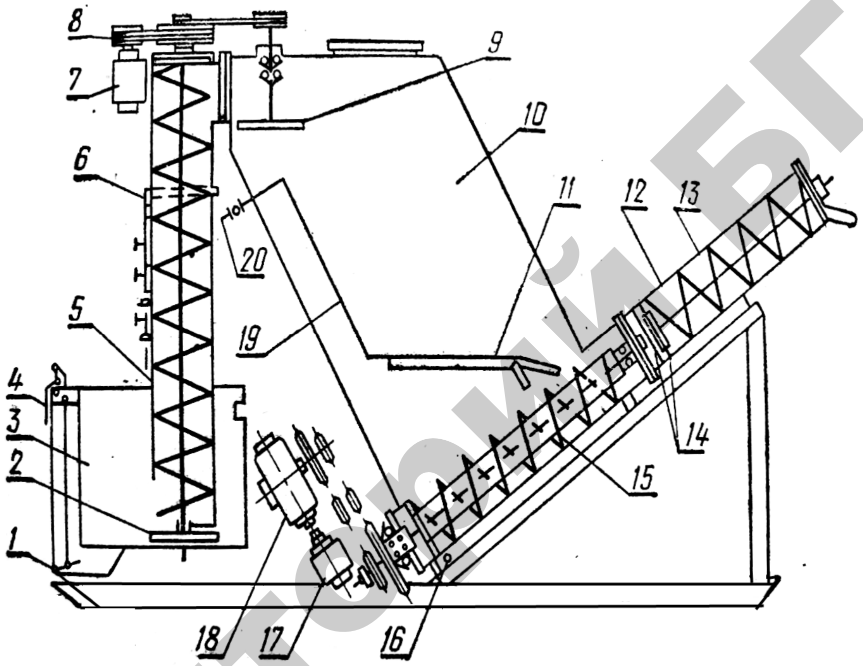


Fig. 3.14. Process flow diagram of potatoes steaming unit:
1 – stone collector; 2 – activator; 3 – washing; 4 – handle; 5 – filling auger;
6 – water distributor; 7, 17 – electric motors; 8 – drive; 9 – switchgear; 10 – digester;
11 – Steam collector; 12 – crushing auger housing; 13 – pug auger; 14 – knives;
15 – unloading auger; 16 – condensate pipe; 18 – reducer;
19 – reducer; 20 – steam line; 21 – valve

In potatoes steaming unit of continuous action in the end you are overweight, the screw can be installed crusher finger type. It has a conical shape, the screw turns within it also conical. Steamed potatoes under the pressure of the screw is pressed into the gaps between the fingers crushing and discharged into the mixer. At the end of crusher has a pocket to trap accidentally caught solid without passing through crusher impurities. A significant drawback potatoes steamer**'s** - uneven steaming food. To obtain high quality and homogeneous feed, it is necessary steam simultaneously with mixing. Therefore, mixing the crushed feed raw or steamed as a unified mixers used.

3.3.1 Calculation of heat consumption for steaming feed

Factors providing steaming process feed are steam parameters, heat capacity, the initial and final temperature of the product, steaming time, ambient temperature and steaming tank settings. The total amount of heat required for thermal processing of the product, the product is consumed for heating and heat loss Infusion container into the environment:

$$Q_{\partial} = Q_1 + Q_2 + Q_3 + Q_4, \tag{3.22}$$

where Q_1 – heat consumption for heating of the product; kJ; Q_2 – the heat consumption for heating Infusion container walls; kJ; Q_3 – heat losses to the environment, kJ; Q_4 – heat loss from condensate kJ.

On heating of the product, if the M1 mass and heat capacity c_1, heat is consumed:

$$Q_1 = M_1 c_1 (t_{кон} - t_{нач}), \tag{3.23}$$

where M_1 – the mass of slip–product, kg; c_1– Product specific heat kJ/kg. hail; $t_{кон}$, $t_{нач}$, – the final and initial product temperature, hail.

Specific expenditures related to product unit:

$$p_1 = \frac{Q_1}{M_1}, \tag{3.24}$$

For root crops p_1 = 0.14...0,16 kg/kg;
– Straw – 0,35...0,50 kg/kg;
– Water – 0,2 kg/kg.
The amount of heat for heating the infusion container walls

$$Q_2 = M_{\text{г}} c_r (t_r - t_{0r}) \text{ Дж}, \tag{3.25}$$

where M_r – the mass of the heated part of the tank, kg; c_r – specific heat of the material from which made a vat. For steel c_r = 460 J/kg. deg.; t_r – heating temperature of the tank, hail; t_{0r} – the temperature to which will have time to cool off a vat for the loading and unloading of the next portion of the product, hail. If there is a single steaming, then $t_r = t_{0r}$.

Specific consumption

$$p_2 = \frac{Q_2}{M_1} \tag{3.26}$$

For modern designs feed steamer specific steam consumption for heating the walls of p_2 = 0,01...0,025, kg/kg.

Consumption of heat loss to the environment determined by the formula

$$Q_3 = S\beta(t_{стен} - t_в)T, \tag{3.27}$$

where S – the surface area of the steamers, m^2; β – The total heat transfer coefficient, W/m^2; $t_{стен}$ – the temperature of the outer walls steamers, hail; $t_в$ – ambient temperature, degree; T – the process time per hour.

Specific consumption

$$p_3 = \frac{Q_3}{M_1} \tag{3.28}$$

For the most common structures p_3= 0,015...0,01 kg/kg. In the presence of p_3 insulation exceeds 0,005 kg/kg and saves pair 2...5 %.

The value of heat transfer coefficient β is:

$$\beta = 7,8 + 0,047\Delta t,$$

where Δt – the temperature difference between the walls and the ambient air steamers, deg, $\Delta t = t_{стен} - t_в$.

Table 3.3 The value of thermal conductivity and heat capacity

Product	Value	
	W/m^2	kJ/kg °
Feed	0,4	
Potatoes	0,85	3,6
Corn	0,55	2,3
Beet	0,90	3,8
Straw	0,55	2,3
Wood	0,57...0,65	2,5
Steal	0,115	0,48
Water	1,0	4,2
Brick	0,22	0,92

The amount of heat with the outgoing condensate:

$$Q_4 = V \rho \varphi_r U_{уд} c_в (t_к - t_в), \tag{3.29}$$

where V – volume steaming vat, m^3; $c_в$ – condensate heat (water), $c_в$ = 4,19 kJ/kg; $t_в$ – the water temperature is supplied to the steam generator, hail; $t_к$ – the average temperature of the condensate, at the beginning of steaming $t_к = t_0$, at the end of steaming $t_к = t_в$, hail; $U_{уд}$ – specific steam consumption for steaming, for existing units $U_{уд}$ = 0,17...0,22 kg/kg.

Calculation by the specific steam consumption is determined by the formula

$$\text{U}_{\text{уд}} = \frac{Q_4}{V \rho \varphi_{\text{г}} i}, \tag{3.30}$$

where i – steam enthalpy J/kg. The efficiency of the use of heat steamers

$$\eta = \frac{Q_1}{Q_4} 100, \% . \tag{3.31}$$

3.3.2 Calculation of steaming container batch

Basic parameters of steaming tank at periodic mode are determined by the performance of the equation:

$$Q = \frac{V \rho \varphi_3}{T}, \tag{3.32}$$

where V – volume of steamingvat, m^3; ρ – product density kg / m^3; φ_3 – fill factor; T – time steaming cycle, hour.

$$T = \tau_{загр} + \tau_{запар} + \tau_{выгр} \tag{3.33}$$

where $\tau_{загр}$, $\tau_{запар}$, $\tau_{выгр}$ – time respectively loading and unloading of steaming an hour.

The duration of load of potatoes in a vat depends on flow rate of vertical filling auger Q_2

$$\tau_{загр} = \frac{V \rho \varphi_з}{Q_2} = \frac{F H \rho \varphi_з}{Q_2}, \tag{3.34}$$

Performance of vertical auger:

$$Q_2 = \upsilon_{ocp}\pi\frac{D^2 - d^2}{4}\rho k_3\,, \tag{3.35}$$

where D – diameter of the screw, m; d – diameter of the shaft, m; k_3 – screw the fill factor (0,3...0,4); υ_{ocp} – the average speed of the material along the screw axis, for experimental data:

$$\upsilon_{\text{ocp}} = \frac{Sn}{60}\left[1 - \frac{2S^2}{\pi^2(D^2 - \mathrm{d}^2)} - \frac{4D\,d\,k_0}{(D+\mathrm{d})(\frac{S^2}{\pi^2} + dD)}\right], \tag{3.36}$$

where S – the screw step, m; n – screw speed, min^{-1}; k_0– factor experienced when S/D = 0,5...0,6 and n = 200...300 min^{-1}, k_0= 1,5...1,8. Big S and n values correspond to smaller values of k_0.

The duration of steaming tubers depends on the thermal characteristics of the initial and final temperatures of the tuber. potato steaming process feature consists in that the steam passing in the pores between tubers, condenses on the surface and the rate of advancement is reduced. Thus, the tuber is located near the steam distributor, begins to heat immediately after the start–up of steam into the vat, and located in a remote point of the tank – after a certain time. The velocity of propagation of a pair depends on the type of the steam nozzle (point, line or plane), and the initial vapor flow tubers temperature.

Therefore, the willingness of potato mass is determined by the availability of the remote start from the steam space of the tuber. Then the total length of the brew:

$$t_{запар} = \tau' + \tau''. \tag{3.37}$$

where τ'– the duration of the spread of steam per hour; τ'' – duration of tuber–heating hour.

The duration of the spread pair on the distance h at the start of the planar head (distribution front pair – plane):

$$\tau' = \frac{hF\rho}{G}. \tag{3.38}$$

The duration of the initial heating tubers t_0 to a predetermined temperature tr

$$\tau'' = 0,75\frac{R_к^2}{\pi^2 a}\ln(\frac{2(t_п - t_о)}{(t_п - t_г)}, \tag{3.39}$$

where F – cross–section of the vat, m^2; $h = H$ – height (H) steaming vat with steam nozzle arrangement in its bottom, m; ρ – bulk density of root crops, kg/m^3; t_n – steam temperature, ^{0}C; G – steam consumption, kg/s; $R_к$ – average equivalent in terms of tuber radius, m; α – coefficient of thermal conductivity of potato tuber, α = (0,10...0,12) 10^{-6}, m^2/s; t_o – the initial temperature of tubers, hail; t_r – readiness temperature of tubers, usually equal t_o = 94...98 ^{0}C.

The duration of the discharge steamed potatoes:

$$\tau_3 = \frac{V\rho\varphi_г}{Q_3}, \tag{3.40}$$

where Q_3– performance conveyor belt, kg/s

$$Q_3 = \frac{\pi(D^2 - d^2)}{4}Sn\varphi\rho, \tag{3.41}$$

where φ – the cross section of the filling ratio of the screw, with steamed potatoes unloading of tank φ = 1.

As can be seen from the performance of steaming process can be increased by reducing the time of filling the steam tank, which is possible by increasing the productivity of steam generator because it depends only on the properties of tubers and the saturated steam temperature. Performance Boot and unloading augers can be increased by the frequency of their rotation or change the design parameters. When developing new potatoes steamer units should strive to tank surface area per unit volume was minimal. At the same time there will be minimal heat loss through the tank wall. For a cylindrical tank that corresponds to the ratio:

$$\frac{D}{H} = 0,8...1,3$$

Then the volume of the steaming vat:

$$V = \frac{\pi D^2}{4}\cdot H = \frac{\pi D^2}{4}\cdot\frac{D}{0,8...1,3} = \frac{\pi D^3}{4(0,8...1,3)}. \tag{3.42}$$

On the other side of the performance equation volume of steaming tank should be:

$$V = \frac{Q\ T}{\rho\ \ \varphi_{3}}. \tag{3.43}$$

Solving these two equations define a tank diameter:

$$D = \sqrt[3]{\frac{4QT(0,8...1,3)}{\pi\rho\varphi_3}}. \tag{3.44}$$

3.3.3 Calculation of continuous infusion container

Continuous steaming process is characterized in that the finished cards, Fel is continuously discharged from the tank, and new portions of potatoes come in his place. Such a process can only be done provided that the mass is evenly potato layers without mixing them to move from input to output.

Promotion rate should be such that for the time of the tubers in the tank, they were brought to readiness. The speed of propagation of the front of steam coming out of the steam nozzle must be equal to the rate of movement of tubers. In this case, set a strict constant "hot" zone, constant in size, which will provide high–quality implementation of the process. Thus, the duration of the movement of potato tubers from the boot to the unloading hatch must be equal to the duration of the heating up of the tuber readiness:

$$\tau = 0,75\frac{R_k^2}{\pi^2 a}\ln(2\frac{t_n - t_o}{t_n - t_г}). \tag{3.45}$$

When working steaming vat in a continuous mode, performance is determined by the equation:

$$Q = \frac{\pi D^2}{4}\cdot\upsilon\rho\varphi_3, \tag{3.46}$$

where u – velocity of product movement in the steamers, m/s; D – diameter of steamers, m; ρ – the volume of the product weight, kg/m^3; φ_3 – filling factor steaming vat, φ_3 = 0,85...09.

The rate of movement depends on its height H or length L of the product and the time of motion equal to the time of its steaming:

$$\upsilon = \frac{H}{T}. \tag{3.47}$$

Accordingly,

$$Q = \frac{\pi D^2}{4} \cdot \rho \varphi_3 \frac{H}{T}. \tag{3.48}$$

Hence we define diameter of steaming vat

$$D = \sqrt{\frac{4 Q T}{\pi H \rho \varphi_3}}. \tag{3.49}$$

At steady state operation the amount of vapor steamers passing through Infusion container equals the amount of steam to heat the product:

$$P = Qc(t_{кон} - t_{нач}). \tag{3.50}$$

Substituting the value of the performance of the formula Q, we get a pair of consumption for heating of the product:

$$P = \frac{\pi D^2 \rho \varphi_3 H}{4T} \cdot c(t_{кон} - t_{нач}) = \frac{\pi D^2 \rho \varphi_3 Hc(t_{кон} - t_{нач})}{4T}. \tag{3.51}$$

But as the amount of steam passing through the Infusion container when weary hovering mode, equal to the amount of steam to heat the product, we can write:

$$\frac{\pi D^2}{4} \upsilon_n \rho_n k = \frac{\pi D^2 \rho \varphi_3 Hc(t_{кон} - t_{нач})}{4T}. \tag{3.52}$$

Solving this equation with respect to H, we get:

$$H = \frac{\upsilon_n \rho_n kT}{\rho \varphi_3 c(t_{кон} - t_{нач})}. \tag{3.53}$$

Performance unloading auger must be equal to performance steaming process. Since the latter depends on the size of tubers and its thermal properties, on the unloading auger drive put variable–speed drive.

3.3.4 Definition of the basic parameters of the steam distributors

Of great importance to the uniformity of the product in warm Infusion container design has a distribution steam line, which is a tube with holes formed on it. When the technical process steam is fed into the steam pipe (the pipe) and the pressure goes through the hole steaming forage.

Distribution steam pipe should provide an even distribution of steam in the feed steam stern. To fulfill this condition requires that the total cross–section of the openings on the steam manifold is equal to the internal diameter of the steam pipe. Consequently, a uniform distribution of steam condition can be written as follows:

$$\frac{\pi D_n^{\ 2}}{4} = \frac{\pi d^2 n}{4} z, \qquad (3.54)$$

where D_n – the internal diameter of the steam pipe, m; d – diameter of the holes, m; n – number of holes in a row; z – the number of rows.

The amount of steam flowing through the wire can be determined from the relationship:

$$P_0 = \frac{\pi D^2_{\ n}}{4} \cdot \upsilon_n \cdot \rho_{n,}, \qquad (3.55)$$

where υ_n – the rate of passage of steam through the steam line, $\upsilon_n > 25 \ldots 30$ m/s; ρ_n – vapor density, kg/m^3.

Substituting the value of steam flow, determine the diameter of the steam pipe:

$$D_n = \sqrt{\frac{4P_0}{\pi \upsilon_n \rho_{n,} T}}. \qquad (3.56)$$

Practice has shown that warming is the fastest, if the steam distribution around the steam pipe forming a continuous vapor layer thickness 0,03...0,04 m. To ensure this condition is countersink holes.

The number of holes in a row (fig. 3.15) depends on the length L and pitch distribution steam line holes S:

$$n = \frac{L}{S}.$$

Step openings is $S = 2l + d,$
where l – distance between adjacent channels, m; d – diameter of the hole, m.

The value l is determined by the height h of a continuous layer α, and a pair of exit angle $l = h / tg\alpha$.

Then $S = (2h / tg\alpha) + d$.

Substituting the values of n and S in equation conditions uniform distribution of vapor (V), and solving it with respect to d, we obtain

$$d = \frac{D_n}{2Lz}\left(D_n + \sqrt{D^2_n + \frac{8Lhz}{tg\alpha}} \right). \qquad (3.57)$$

The angle and depends on the angle countersink. If the latter is 120 °, then a = 30 °.

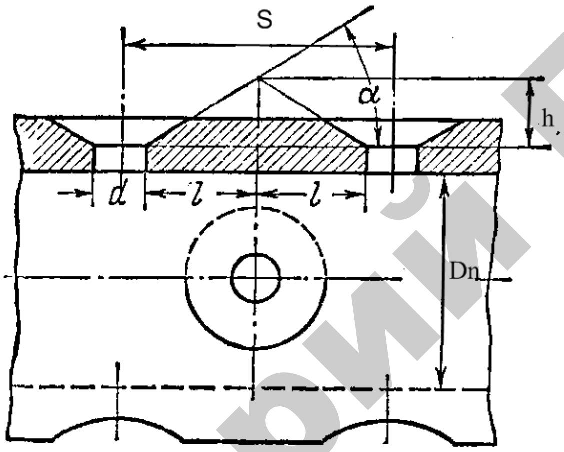


Fig. 3.15. Driving definition holes in the steam parameters

Number of rows of holes on the steam line is selected depending on the steam flow. When steam flows with a rate 0.055 kg/per second the number of rows z is taken 3...4, and at great expense z = 4...6.

The diameter of the pipe for the condensate output can be defined by the formula

$$d_к = 2\sqrt{\frac{qQ_0}{\pi \upsilon_к \rho_к}}, \qquad (3.58)$$

where q – the specific steam consumption; Q_o – performance steamers; υ_k – Average condensate output speed υ_k = 1,2...1,5 m/s; ρ_k – condensate density, kg/m^3.

Steam nozzle can be made planar, linear and point.

Chapter 4

MECHANIZATION OF CRUSHING PROCESS FOR STALK FEED AND ROOTS CROPS

4.1 Types of grinding machines feed stalk and root crops. The main cutting edge patterns

To stalk feeds include hay and straw. These feeds contain no more than 22 % moisture. Straw palatability ruminants provided above its splitting along the grain of not less than 85 %, and particles with a length not less than 10 ... 15 mm. Finer grinding straw harmful as its digestibility by ruminants does not increase, and reduced fat milk. Chopped straw can be fed to the animal as part of core–mosmesi and issuing general–purpose feeders.

The hay is ground to improve palatability and to improve its tech–nological properties. Chopped hay can be used for the preparation of complete feed mixtures. Average values of specific resistance to cutting hay varies between 5,7...12,0 kN / m. If hay resistance is taken as 100 %, the resistance is herbs 80...90 % straw 55...60 %. The rate of destruction of the stem feed stroke is 50...60 m/s. Silage feed is a source of digestible protein and carbohydrates. The quality of the final product is affected and the degree of comminution of green mass – cutting length must be at least 20 mm. Shredding of green mass allows you to give it to the general purpose trough feeders.

For the grinding of root crops and forages used stalked knife and hammer working bodies (fig. 4.1).

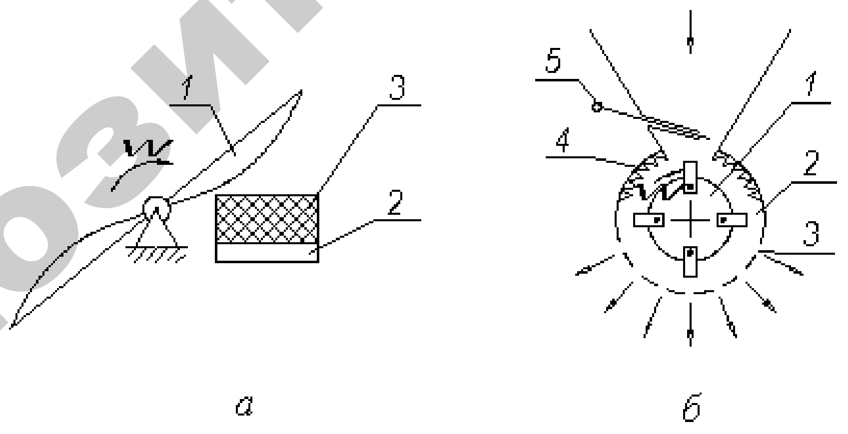


Fig. 4.1. Working bodies of shredders for root crops and stalked feed:
a) – cutlery: 1 – knife; 2 – shearbar; 3 – sample material;
b) – hammer: 1 – drive; 2 – hammer; 3 – sieves; 4 – deck; 5 – regulating valve

Hammer working bodies are universal, they can grow shallow because of all kinds of feed and are indispensable in the production of herbal, hay or straw flour that require particle size of not more than 1...3 mm. However, hammer crusher machines have a higher specific energy consumption to destruction. When the shock impact hammers on root crops is their destruction of a significant release of the cell sap, which does not meet the requirements of zootechnical.

The advantages of working elements of the knife are low specific energy consumption for a better grinding quality, the ability to grind feed any humidity. These advantages predetermined the most widely used for crushing of root crops and forages stalked stab working bodies. For any feed grinding humidity regrinding other components and mixing in the preparation of feed mixtures used forage chopper–mixer with a vertical working chamber containing a chopping blade unit (fig. 4.2).

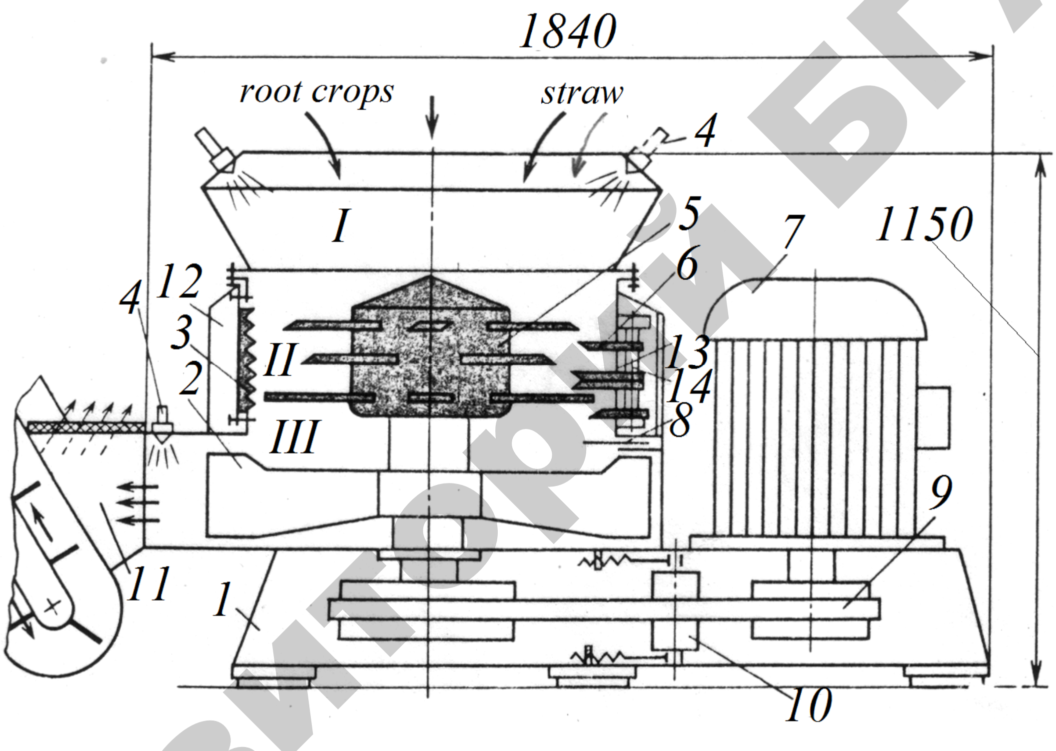


Fig. 4.2. Grinder–mixer with a vertical working chamber:
1 – frame; 2 – lobed spinner; 3 – deck gear; 4 – atomizer; 5 – rotor;
6 – counter–knife; 7 – electric motor; 8 – gate; 9 – V–belt drive; 10 – video;
11 – tank unloading conveyor; 12 – housing; 13 – counter–rotating; 14 – shaft

The machine consists of a reception I, II and working unloading III cameras, one above the other, the rotor blade, unloading conveyor, counter–rotating packets, the electric motor and V–belt transmission. For introduction to the mass of the treated liquid supplements provided by two nozzles – in the reception and unloading chambers.

There are six windows (fig. 4.3) in the walls of the working chamber in which to install packages, counter–rotating blades and deck gear. Window is closed to the outer side of the housing.

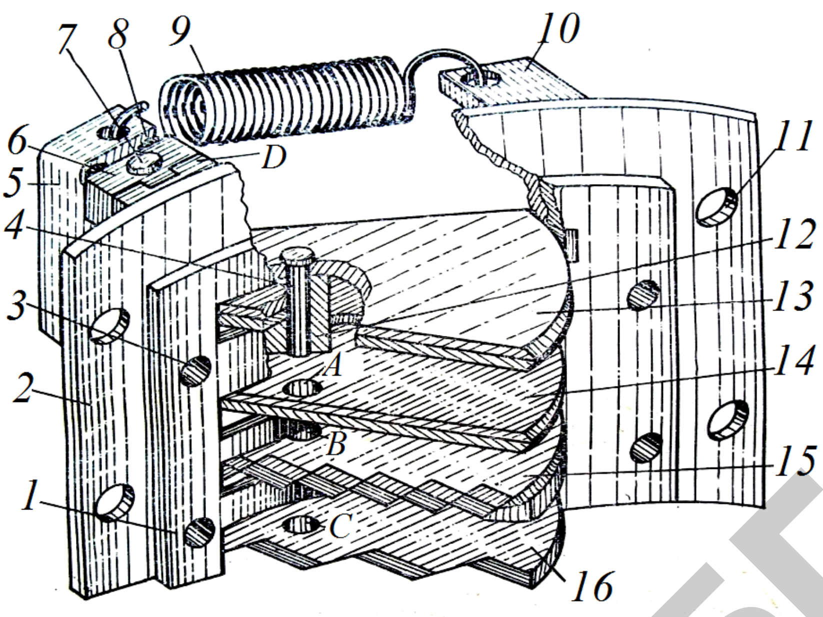


Fig. 4.3. Working chamber of grinder–mixer with a vertical working chamber:
1 – frame; 2 –plate; 3, 11 –holes; 4 – stop; 5, 10 – lever; 7 – shaft; 8, 9 –spring;
13,14 – counter–knifes; 15,16 –knifes

At the rotor blades arranged in tiers chopper performing the role of mixers. In the lower part of the rotor, located in the unloading chamber is throwing–fork raiser. Package counter–rotating blades, assembled on the shaft, is pivotally installed on the basis attached to the housing of the working chamber. When the mixer–chopper, loading the feed into the chamber and fall into the zone of interaction counter upper tier blades 6 of the rotor with cutting members, wherein the partially crushed and distributed evenly over the perimeter of the working chamber. Then feed particles are entrained on the smooth surface portion of the inner chamber and under the influence of gravity, several spirally moved downward. On the path of its movement of feed particles occur gear deck, and their speed is reduced.

Knives next tier, being longer, carry out grinding and further advancement of feed particles. This is one of the shredded fodder becomes greater speed than the other, which promotes the penetration of some particles in the feed and a lot of other effective mixing them. Moving under the influence of gravitational force down the food meets in its path verge knives and the counter–element lower tiers finally crushed. At the end of the process of unloading ration falls into the camera and shvyryalkoy thrown into the hopper unloading transport.

When grinding one kind or more feed components to be mixed and pulverized in the windows of all six packages mounted counter–rotating. Vertical loading of the grinder–mixer and the subsequent impact on the working bodies feed it as it moves in the chamber under the action of gravitational forces and allows combine it in a single machine three steps: mechanized loading, crushing and discharging. At the rotor blades arranged in tiers chopper performing the role of mixers. In the lower part of the rotor, located in the unloading chamber is spinner. The machine can be equipped with a paddle spinner or finger (fig. 4.4). The advantage of finger spinner is to reduce transportation energy costs feed into the working area of the window and a minimum unloading stern air flow, eliminating the separation of feed mixture.

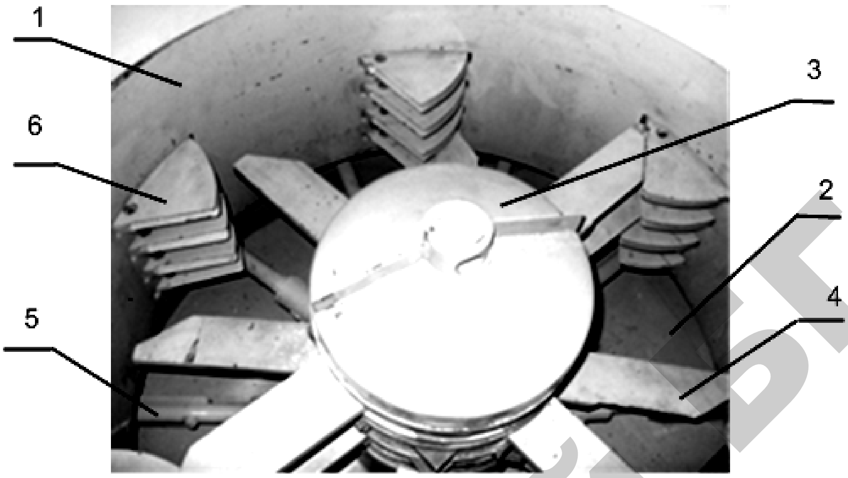


Fig. 4.4. Grinder–mixer with finger spinner:
1 – camera grinding and mixing; 2 – unloading chamber; 3– rotor; 4 – knife;
5 – finger spinner; 6 – counter–rotating

Performance machine running on a mix, can reach up to 25 t/h, to mix with the partial re–grinding – up to 15 t/h, in the grinding, for example, straw – up to 3...4 t/h with a length of cutting up to 30 mm and 4...8 t/h at a cutting length of up to 50 mm. The degree of crushing is controlled by changing the number of blades on the rotor, and a number of counter–rotating time of the product being in the working chamber (by means of an annular slide, mounted above spinner). The machine provides a mix of silage, straw, root crops and fodder to the degree of uniformity of 80 ... 90 %; Serves grinder–mixer one worker.

For all kinds of rough grinding and rich feeds may be used a drum type grinding apparatus (fig. 4.5), It consists of a sheaber and the knives of the drum, which may be straight or helical shape.

Spiral blades in the drum section are shaped on the drum are usually found 6 knives with an angle of 35°40' and the helix angle of ascent 70 °. Blade angles equal 75 °. The gap between the blade knives and shearbar set within 0,5...1 mm and regulate the movement of the drum with knives thrust screws. Figure 4.6 show cutting length can change the gear ratio of the drive transmission (replacing the stars on the shaft of the cutter drum) and the number of knives installed on the drum (cutting length increases with a decrease in their number). Wear limit edge cutting knife 10...12 mm, shearbar – 5 mm.

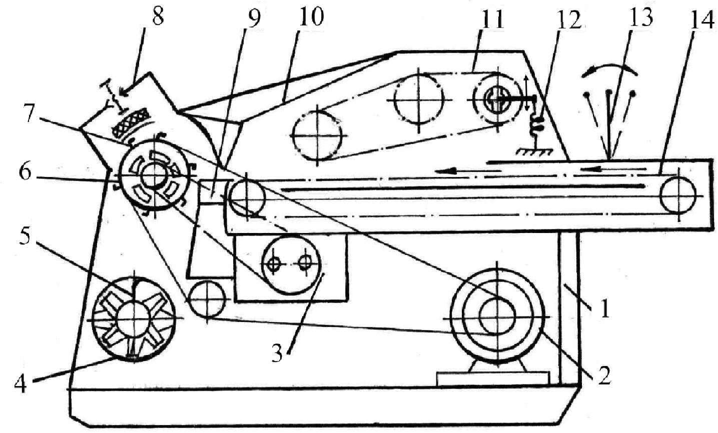


Fig. 4.5. Shredder feed grinding apparatus with a drum–type apparatus:
1 – frame; 2 – motor; 3 – gears; 4, 6 – apparatuses; 5 – screw; 7 – knifes;
8 – devise; 9 – plate; 10 – lid; 11 – tensioner supply conveyor; 12 – spring;
13 – lever; 14 – sealing conveyor

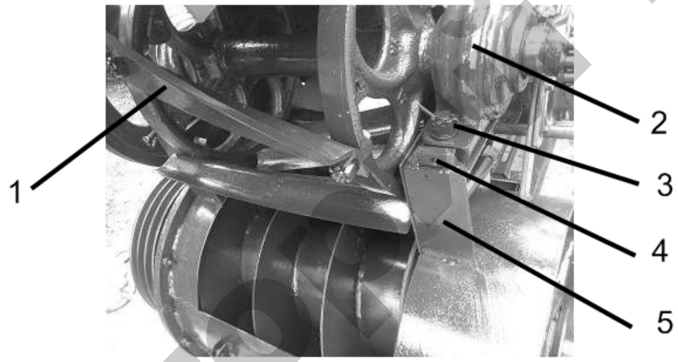


Fig. 4.6. Drum–type apparatus:
1 –knife; 2 –shaft; 3 –bolt; 4 – screw, 5 – plate

Pressure conveyor 2 is floating type in which the driven shaft with the tape moves in a vertical plane. This makes it possible to vary the height of the flow of feed to the feed conveyor and provides a seal them as you move up to the chopping–unit.

On the supply 1 and 2 pressure conveyors rotation is transmitted to the chain–governmental gears using the gear which allows their movement is "forward" and "back".

To protect both transporters from damage in case of overload on the drive shaft gear is mounted safety clutch, which must be adjusted to transfer torque. The rotational movement of the two conveyors obtained from electric engine. When grinding feed supplied from potatoes–feeder food is (or is placed manually) evenly on the conveyor belt 1, 2 sealed inclined conveyor, and is then sent to the chopping device. To avoid delaying the feed into the gap between the shearbar and ribbon supply conveyor, by moving the plate, the gap is set to a minimum.

4.2 Fundamentals of the theory of cutting blade

Cutting – a type of grinding associated with the blade. During cutting decreases the linear dimension of the material, increasing the number of new particles and their total surface area. For cutting flat surfaces characterized by the formation. The process of cutting blade is a special kind of grinding and therefore is subject to the general laws of grinding materials under the influence of external forces.

However, this process has its own characteristics. Important values belong to character–knife motion with respect to the product being cut. Depending on the nature of the knife cutting motion with respect to the product being cut is divided into slashing and sliding.

When chopping knife cutting vector material implementation rate per perpendicular–edge blade. Cutting the material into pieces under the influence of the knife is preceded by a preliminary compression of the material.

The amount of compression is determined Deplete the contact voltage by applying to, called critical power Fkr knife. In achieving its compression comes to an end and start the cutting process.

Thus, the critical force – this is the maximum force before the start–to–scrap cutting and for its determination the process of interaction of the knife and the counter with the material at the start of cutting.

The interaction of the knife and the cutting counter–rotating at the time of start cutting thickness h (fig. 4.7), during the deepening of the knife, the compression of the material begins at the value h compression channel as long as the cutting edge of the knife is not in will be the contact pressure greater than the tensile strength of the material. At the start of the cutting knife acting on the following forces:

$$F_{кр} = F_{рез} + T_1^{r} + T_2 , \qquad (4.1)$$

where $F_{рез}$ – cutting resistance, H; T^{r}_{1}, T_2 – the frictional force, respectively on the knife chamfer (due to the normal force N_1) and on the edge of the knife (due to the action of lateral pressure $F_{обж}$), N.

Resistance to cutting $F_{рез}$ consists of forces associated with overcoming the resistance of the layer of material compression $F_{1сж}$, cutting forces directly $F_{1рез}$,

$$F_{рез} = F_{1рез} + F_{1сж}.$$

Breaking strength recommends to determine by the formula

$$F_{1рез} = \delta\, \Delta l\, \sigma_p, \tag{4.2}$$

where δ – thickness (sharpness) of the knife blade, m; Δl – blade length involved in the cutting of the material, m; σ_p – destructive contact stress.

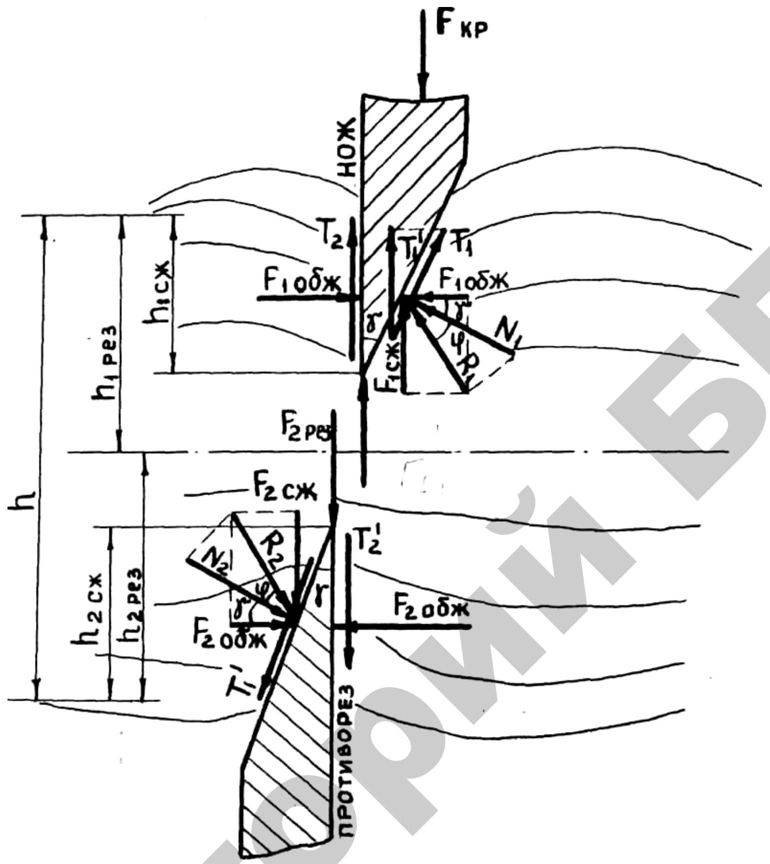


Fig. 4.7. Force knife and counter interaction with material during cutting

The friction force is on the verge of a knife:

$$T_2 = F_{1обж} f_н\,, \tag{4.3}$$

where $F_{1обж}$ – compression strength of the material, H; $f_н = tg\ \varphi$ – coefficient of friction of the knife feed material (φ – angle of friction).

The force of friction on the knife facet:

$$T'_1 = T_1 \cos\gamma\,, \tag{4.4}$$

considering the normal force N_1:

$$T'_1 = N_1 f_н \cos\gamma\,, \tag{4.5}$$

where γ – knife sharpening angle, deg.

Power N_1 can be expressed in terms of angle of friction:

$$N_1 = R_1 cos\gamma, . \tag{4.6}$$

where

$$R_1 = \sqrt{F_{1обж}^2 + F_{1сж}^2}\,. \tag{4.7}$$

To determine the resistance of the material compression $F_{1сж}$ consider the action of the unit attached to the blade force $dF_{1сж}$ (fig. 4.8), which will cause compression of the material by the amount $dh_{сж}$.

Since there is a force and moving work is done

$$dW_{1сж} = F_{1сж} dh_{сж}. \tag{4.8}$$

It is known that $F_{1сж} = \sigma A_H$ and $\sigma = E\varepsilon_{сж}$,
where $A_н$ – knife blade area, participating in compression, m^2.

Then, taking into account the above formula (4.8) takes the form

$$dW_{1сж} = E\varepsilon_{сж} A_н \, dh_{сж}. \tag{4.9}$$

Furthermore, the relative compression, as is known, is the ratio of the compressed $h_{сж}$ material to its total height h:

$$\varepsilon_{сж} = \frac{h_{сж}}{h/2}, \tag{4.10}$$

Substituting the value $\varepsilon_{сж}$ from (4.10) to (4.9), we obtain:

$$dW_{1сж} = \frac{2EA_н}{h} h_{сж} \, dh_{сж} \,. \tag{4.11}$$

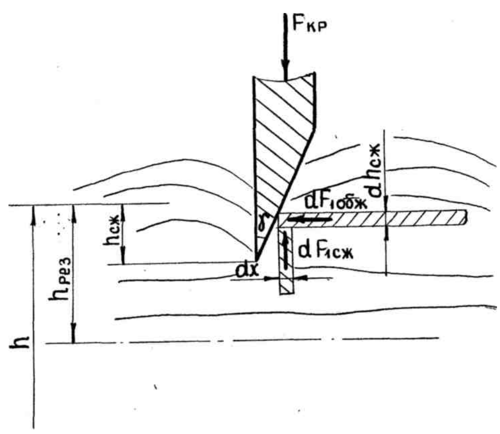


Fig. 4.8. Compression force definition – diagram

Full–size $W_{1сж}$ work can be obtained by integrating the formula (4.11):

$$W_{1сж} = \frac{E A_н h_{сж}{}^2}{h}, \tag{4.12}$$

where *compression force*, which carried out the work,

$$F_{1сж} = \frac{E A_н h_{сж}}{h}. \tag{4.13}$$

Equation (4.13) allows calculate analytically force is needed for compression–chained material before cutting – it depends on the area of the knife blade and the physical–mechanical properties of the material. The formula can lay specific physical and mechanical properties of a particular material (feed) with specific geometric blade**'s** parameters; you can also determine the power reduction.

The strength of the compression based on the theory of elastic deformations, equal

$$F_{1обж} = \varepsilon_1 E A_н, \tag{4.14}$$

where ε_1 – relative deformation of roughage in the horizontal direction, which can be expressed in terms of relative deformation in the vertical direction by the Poisson ratio μ:

$$\varepsilon_1 = \varepsilon_{сж}\,\mu. \tag{4.15}$$

Then the expression (4.14) takes the form:

$$F_{1обж} = F_{1сж}\,\mu, \tag{4.16}$$

or

$$F_{1обж} = \frac{A\sigma}{2}\mu. \tag{4.17}$$

Thus, the formula (4.1) can be represented in this form:

$$F_{кр} = (F_{рез} + F_{сж}) + (T_{г1} + T_2). \tag{4.18}$$

From equation (4.18) shows that the first two forces compress the material prior for cutting, and cutting is carried out, and according to Newton's third law, cause the same to counter–rotating forces. Two other forces of the equation (4.18) on the knife cause friction.

Substituting in equation (4.18) we obtain the value of:

$$F_{кр} = F_{рез} + F_{1сж} + F_{обж} f_н + f_н \sqrt{F_{1обж}^2 + F_{1сж}^2}\, cos\varphi\, cos\gamma. \qquad (4.19)$$

Express $F_{1обж}$ through $F_{1сж}$. Equation (4.19) takes the form:

$$F_{кр} = F_{1рез} + F_{1сж}\,(1 + f_н\,\mu + \sqrt{\mu^2 + 1}\, \cos\varphi\, \cos\gamma). \qquad (4.20)$$

Based on the well–known critical cutting forces can determine the work performed by the chopper:

$$W_{1рез} = \int_0^{h_{рез}} F_{кр}\, h_{рез} dh_{1рез} = F_{кр}\, h_{1рез}, \qquad (4.21)$$

or, according to the fig. 4.7,

$$W_{1рез} = F_{кр} \frac{h}{2}. \qquad (4.22)$$

The same work performs the cutting and counter–rotating blades. So, the energy intensity of the process of cutting with counter–rotating cutter should be theoretically two times less than those without such counter–rotating. In practice, the cutting process with the cutting counter–rotating somewhat delayed, especially when thick–cutting, and the work carried out by counter–rotating, slightly less than a knife made.

4.3 Determination of the angle of inclination of the knife during sliding cutting

Reduction of the energy costs for the cutting process, without reducing the reliability of cutting pairs, is possible by organizing cutting with slip, which can be carried out in various ways (fig. 4.9)

In the last two cases (fig. 4.9, b, c), the material will slide relative to the blade only if the angle of the solution of the cutting pair is greater than the friction angle of the material against the blade. Otherwise, the sliding cutting will degenerate into a chopping cut.

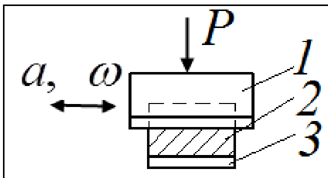
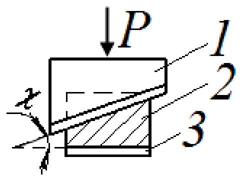
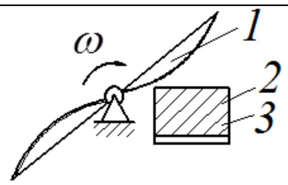
		
a) with the help of knife vibrations;	b) by setting the knife at an angle χ to the anti–rupture plate;	c) the cutting edge is executed by a certain curve, which is typical for rotational working elements.

Fig. 4.9. Types of cutting with sliding:
1 – knife; 2 – cut material; 3 – inconsistent plate

Fig. 4.10 shows the wedge, which is part of the knife with a one–sided sharpening on the surface of which the feed particle moves with sliding.

From fig. 4.10 it is seen that the angle of sharpening of the knife can be determined by the formula

$$tg\beta_{н} = \frac{tg\beta_{1}}{cos\tau}, \qquad (4.23)$$

where b1 is the value of the transformed knife sharpening angle, deg; T is the slip angle, deg.

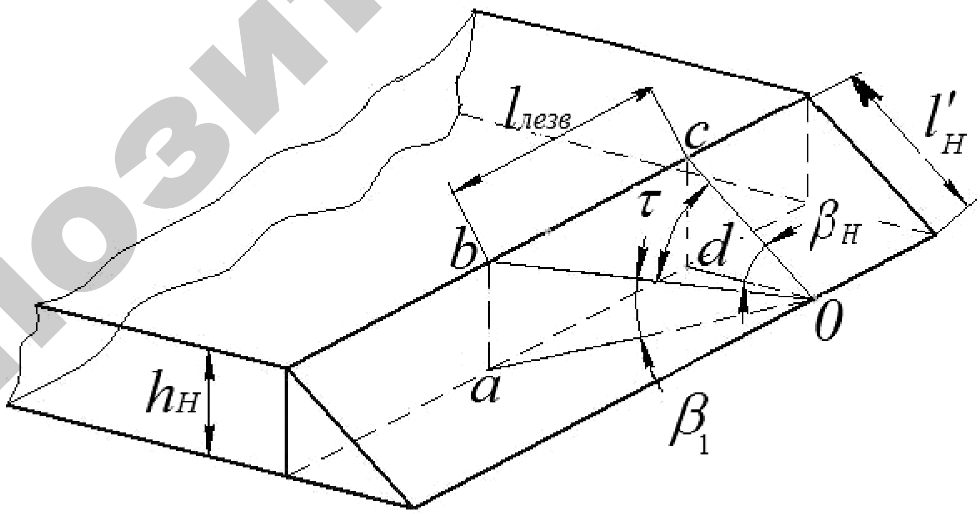


Fig. 4.10. Scheme for determining the angle of knife sharpening

Using trigonometric formulas, we can determine the value of the transformed angle of knife sharpening:

$$tg\ \beta_1 = \left(\frac{ab\ tg\ \tau}{bc\sqrt{1+tg^2\ \tau}} \right) [\frac{1}{\sqrt{1-\left(\frac{ab\ tg\ \tau}{bc\sqrt{1+tg^2\ \tau}} \right)^2}}]. \tag{4.24}$$

Then the expression (4.24) is substituted into equation (4.23), with replacing the symbols with the parameters of the knife:

$$tg\beta_н = \frac{h_н}{l^1_н \left(1 - \frac{h_н}{l^1_н + l_{лез}} \right) cos\tau} . \tag{4.25}$$

It can be seen from formula (4.25) that the angle of sharpening of the knife depends on its thickness hn, the length of the inclined face l1n and the amount of displacement of the particle of the lime forage along the inclined face.

With known knife parameters, it is necessary to determine at which distance the particle will move, moving along the inclined face of the knife.

The value of $l_{лезв}$ corresponds to the length of the knife blade which cuts feed particle. This process is accompanied by the interaction of the knife blade with the material. The work spent on the performed process can be determined by the formula

$$A_{рез} = P_{рез}\ 2r_ч , \tag{4.26}$$

where $P_{ре}$ – is the cutting force, Н; $r_ч$ – is the radius of the section of the feed particle, m.

The cutting force can be defined as the multiplication of the edge area of the blade of the knife and the breaking contact stress σ_p:

$$P_{рез} = \delta_{лезв}\ l_{лезв}\ \sigma_p , \tag{4.27}$$

where $\delta_{лезв}$ – is the thickness of the knife blade, m; $l_{лезв}$ – the length of the knife blade involved in the cutting, m.

Then the work spent on separating the particle by the cutting force can be determined by the formula

$$A_{рез} = \delta_{лезв}\ l_{лезв}\ \sigma_p\ 2\, r_ч . \tag{4.28}$$

In the process of cutting, the feed particles move along the blade of the knife, overcoming the frictional force. The work spent on overcoming this force is determined by the formula

$$A_{тр} = F_{тр}\ l_{лезв}\ , \tag{4.29}$$

where $F_{тр}$ – frictional force, H.

Then the work required to cut the feed particle and overcome the friction force when moving the feed particle along the blade of the knife is determined by the formula

$$A = A_{рез} + A_{тр}\ . \tag{4.30}$$

Where

$$l_{лезв} = \frac{A}{\delta_{лезв}\ \sigma_p\ \ 2r_ч + F_{тр}}\ . \tag{4.31}$$

The work on overcoming these forces can also be determined from the expression:

$$A = \frac{N_{рез}}{\omega_p}, \tag{4.32}$$

where $N_{рез}$ – the power spent by the knife on the cutting process, kW; ω_p – is the angular velocity of the rotor of the shredder, s^{-1}.

By substituting in the formula (4.31) the values of the considered forces, we obtain an expression that allows us to determine the length of the particle cutting path:

$$l_{лезв} = \frac{N_{рез}}{(\delta_{лезв}\ \ \sigma_p\ 2r_ч + F_{тр}\ )\omega_p}\ . \tag{4.33}$$

Formula (4.33) makes it possible to determine the angle of sharpening of the knife taking into account the length of the blade of the knife participating in cutting of feed particles:

$$tg\beta_н = \frac{h_н}{l^1_н\left(1 - \dfrac{h_н}{l^1_н + \dfrac{N_{рез}}{\left(\delta_{лезв}\,\sigma_p 2\,r_ч + F_{тр}\right)\omega_p}}\right)cos\tau}. \tag{4.34}$$

It can be seen from formula (4.34) that the angle of sharpening of the knife is one of the complex parameters of the working element. It would seem that the cutting machines need to be designed with slip angles $\tau \rightarrow 90$ °. However, when designing cutting machines, it is necessary to take into account other factors. First, maintaining the large angles is associated with maintaining high ratios of the tangential and normal velocity components of the blade $\tau = \mathrm{arctg}\ (V_\tau/V_n)$, which can often be accomplished only by lowering Vn, that leads to a loss of cutting machine productivity. Secondly, the friction force path increases in F $(\cos \tau)^{-1}$ times and consequently, the specific energy expenditure increases by the same factor. According to various experimental data, the optimal value of the angle of slip lies in the interval 20...60°.

In accordance with equation (4.34), we obtain a graphical dependence of the knife sharpening angle on its thickness and the power used for cutting of the feed (fig. 4.11).

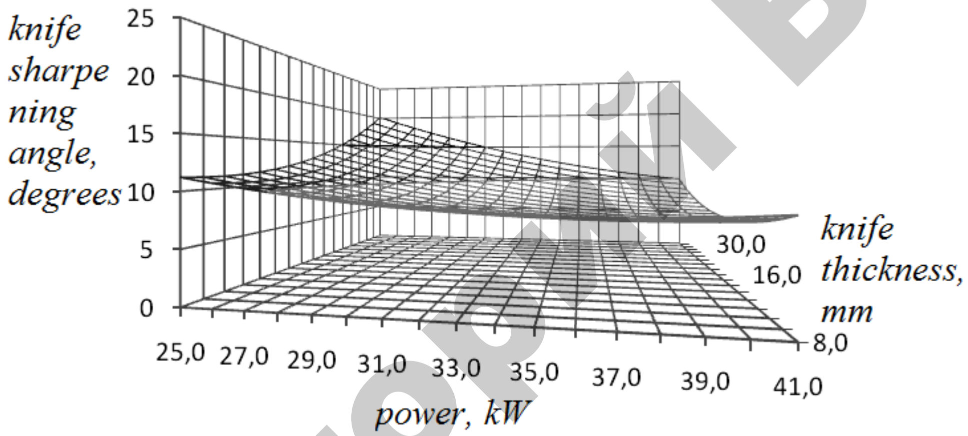


Fig. 4.11. Dependence of the angle of sharpening of the knife on its thickness and power on the cutting of the feed

An analysis of the obtained graphical dependence shows that with an increase in the thickness of the knife, the angle of its sharpening increases. With increase of power for feed cutting, the angle of sharpening of the knife decreases. On the basis of the graphical dependence, it is established that for a knife 10 mm thick with an electric motor power of 39 kW, the knife sharpening angle is 8 degrees.

4.4 Justification of the choice of knife thickness

It has been experimentally established that the energy consumption for cutting work increases with an increase in the thickness of the knife (29, 30). Obviously, the thickness of the cutting element must be the smallest. However, in this case, it will bend in the smallest section, that can be the reason for the breakage of the knife when meeting with the opposing element.

A technical solution that reduces the likelihood that knife will encount a plate is to increase the gap between them. However, solving the reliability problem raises the problem of increasing energy costs for the performed process. With a significant gap, the feed particles will simply move between the planes of the cutting pair. To justify the thickness of the knife, let's consider the forces which arise in the process of the food movement in the solution of the cutting pair. During the working process the knives move over the opposing elements (fig. 4.12). Since there is a gap between them, the food enters this space.

The value of the compressible feed layer depends on the size of the clearance between the knife and the opposing plate. That is why the equality is valid. Then:

$$F_{обж} = \mu \frac{E}{2} \frac{h_{сж}^2}{\Delta_1}, \tag{4.35}$$

where Δ_1 – is the clearance between the knife and the opposing plate, m; μ – is the Poisson ratio; E – modulus of feed deformation, N/m^2; $h_{сж}$ – the value of the compressed feed layer, m.

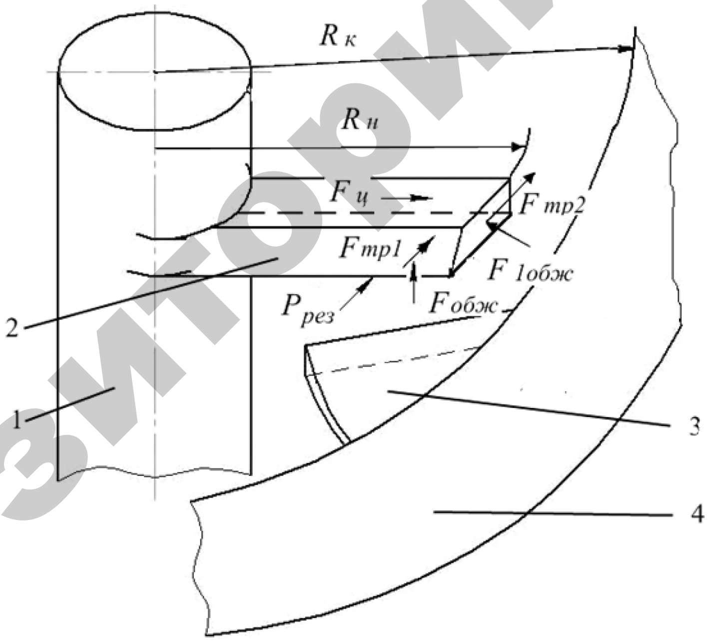


Fig. 4.12. Sheme of knife thickness definition:
1 – rotor; 2 – knife; 3 – contradictory element; 4 – working chamber chopper

The force of compression creates a friction force, which can be determined from the formula

$$F_{тр1} = F_{обж}\ f_к, \tag{4.36}$$

where $f_к$ – coefficient of friction of food for metal.

The work spent on overcoming the frictional force Fmp1 is determined by the formula

$$A_{тр_1} = \mu \frac{E}{2} \frac{h_{сж}^2}{\Delta_1} f_к \, l_{н.пр} , \tag{4.37}$$

where $l_{н.\ пр}$ – is the length of the knife moving above the plane of the opposing plate, m.

During operation of the chopper, the feed enters the gap between the inner surface of the working chamber and the knife.

As a result, a horizontally directed crimping force arises, the value of which can be determined from formula

$$F_{1\ обж} = \mu \frac{E}{2} \frac{h_{сж}^2}{\Delta_2} , \tag{4.38}$$

where Δ_2 – is the gap between inner surface of the working chamber and the knife, m.

There is a friction force From the force of compression, which can be determined from the formula

$$F_{тр\ 2} = F_{1\ обж} \; f_к . \tag{4.39}$$

The work spent on overcoming the frictional force $F_{тр\ 2}$can be determined by the formula

$$A_{тр_2} = \mu \frac{E}{2} \frac{h_{сж}^2}{\Delta_2} f_к \, b_н . \tag{4.40}$$

Critical cutting force arises during cutting the feed with a blade of a knife on its active part, which can be determined by formula

$$A_{кр.рез} = P_{рез} \; l_{лезв} , \tag{4.41}$$

where $P_{рез}$ – is the cutting force, H; $l_{лезв}$ – the length of active part of the knife blade, m.

Rotational motion of the knife is carried out by the action of the circumferential force, the work on overcoming of which can be determined from the expression:

$$A_{окр} = F_{окр} \; R_н = \frac{N_{дв}}{\omega_p} , \tag{4.42}$$

where $N_{дв}$ – is the power transmitted from the electric motor to the rotor of the shredder, kW; ω_p – is the angular velocity of the rotor of the shredder, s^{-1}.

The obtained equations allow determine the bending load acting on the knife during the reference cutting:

$$A_{тр_1} + A_{тр_2} + A_{окр} + A_{кр.рез} = A_{изг}\,. \tag{4.43}$$

Fastening of the cutting element in the grinder excludes its movement in the vertical plane, which allows, with some assumptions, to treat the knife as a cantilever beam with a fixed end. For such fastening, the maximum bending load acting on the knife can be determined by the formula

$$P_н = \frac{b_н h_н^{\,2}}{l_{кр}} \frac{\sigma_в}{6}, \tag{4.44}$$

where $b_н$ – is the width of the knife, m; $h_н$ – knife thickness, m; $l_{кр}$ – length of the knife from the axis of attachment to the end, m; $\sigma_в$ – permissible ultimate strength, MPa.

Load acting on the knife

$$A_{изг} = P_н R_н, \tag{4.45}$$

where $P_н$ – load acting on the knife, H; RH is the radius described by knives, m.

Substituting the corresponding components into equation (4.43), we obtain:

$$\mu\frac{E}{2}\frac{h_{сж}^2}{\Delta_1} f_к\, l_{н.пр} + \mu\frac{E}{2}\frac{h_{сж}^2}{\Delta_2} f_к\, b_н + \frac{N_{дв}}{\omega_{рот}} + P_{рез}\; l_{лезв} = \frac{b_н h_н^{\,2}}{l_{кр}}\frac{\sigma_в}{6} R_н. \tag{4.46}$$

From the equation (4.46) we determine the thickness of the cutting element:

$$h_н = \sqrt{\frac{\left[\mu\,\frac{E}{2} h_{сж}^2\, f_к \left(\frac{l_{н.пр}}{\Delta_1} + \frac{b_н}{\Delta_2}\right) + \frac{N_{дв}}{\omega_{рот}} + P_{рез}\; l_{лезв}\right] 6\, l_{кр}}{b_н\, \sigma_в\, R_н}}. \tag{4.47}$$

Analysis of formula (4.47) shows that with the increase of power transferred to the knife, its thickness also increases.

When calculating it is necessary to take into account the physico–mechanical properties of the feed and the design features of the shredder.

4.5 Determination of energy costs for the shredder drive

The power used to cut the feed is one of the main parameters determining the choice of the electric motor for driving the feed shredder. When designing machines for grinding feed, it is necessary to take into account the parameters that determine the amount of energy spent on the working process of the machine. The magnitude of this power can be determined from formula

$$N_{пр} = \frac{W_{рез} \;\; z_{м}}{t_{рез}\omega_{p1}}, \tag{4.48}$$

where $W_{рез}$ – the moment of the amount of motion of the knife during the cutting process, kg m^2/s; Z_m – number of knives fixed on the rotor, pcs; $T_{рез}$ – time spent on the cutting process, s; ω_{p1} – is the angular velocity of the knife after the cutting process, s^{-1}.

In the general form, the angular momentum of the knife during the cutting process is determined from formula

$$W_{рез} = \left(m_{н} + m_{л}\right) r^{2}_{ин} \;\; \omega_{p}, \tag{4.49}$$

where $m_н$ – is the mass of the knife, kg; $m_л$ – weight of food on the knife blade, kg; $r_{ин}$ – is the distance from the rotor axis to the center of gravity of the knife, m.

Since the cutting process takes place over a certain period of time t, it is possible to determine the moment of the amount of motion of the knife:

$$M_{н} = M_{кр.рез.} \;\; t_{рез}. \tag{4.50}$$

The moment of the amount of motion when cutting with a knife rotating around the axis can be expressed by the dependence:

$$M_{н} = I_{p}\left(\omega_{p} - \omega_{p1}\right), \tag{4.51}$$

where I_p – the moment of inertia of the rotor, $I_p = m_н\ r_{ин}/2$, kg m^2; Rn is the radius described by the knives, m; ω_p– is the angular velocity of the rotor, s^{-1}.

Since in the equations (4.50) and (4.51) the left–hand sides are equal, the following equality holds:

$$I_p\left(\omega_p - \omega_{p1}\right) = M_{кр.рез.}\ t_{рез}. \tag{4.52}$$

The phenomenon of cutting materials occurs as a result of the critical cutting force Pcr. The action of this force during the cutting period gives rise to a moment, the magnitude of which can be determined from the expression:

$$M_{кр.рез.} = P_{кр.рез}\ l_{лезв}, \tag{4.53}$$

where $P_{кр.рез}$ – is the critical cutting force, H; $l_{лезв}$ – the length of the knife blade involved in cutting the feed, m.

From equation (5.52) we express the angular velocity of the knife after the cutting process:

$$\omega_{p1} = \frac{I_p\omega_p - P_{кр.рез}\ l_{лезв}\ t_{рез}}{I_p}. \tag{4.54}$$

The amount of power consumed for cutting can be determined from formula

$$N_{пр} = \frac{\left(m_н + m_л\right) r^2_{ин}\ \omega_p z_м}{t_{рез}\ \omega_{p1}}. \tag{4.55}$$

If the knives on the rotor are hinged, the moment of the amount of motion of the knife during the cutting process is determined from the formula

$$W_к' = \left(m_н + m_л\right)\left(r^1_{ин}\right)^2 \omega_p, \tag{4.56}$$

where $r^1_{\grave{e}\psi}$ – is the distance from the axis of the rotor to the center of gravity of the knife when it deviates from the axis of symmetry, m.

Since the knife is fixed articulated, after the cut it deviates from the radial axis by an angle α1. For a hinged knife, the distance from the rotor axis to the center of gravity of the knife when it deviates from the symmetry axis is:

$$r^{I}_{ин} = R_{под} + C - \psi \cos \alpha^{I},\tag{4.57}$$

where α^{I} – is the maximum deviation angle of the hinged knife from the radial position, deg; $R_{под}$ – the distance from the axis of the rotor to the axis of fastening the knife, m; C – is the distance from the axis of attachment of the knife to its center of gravity, m.

Substituting into (4.56) the value of (4.57), we obtain:

$$W'_{к} = \left(m_{н} + m_{л}\right)\left(R_{под} + \psi - \psi \cos \alpha^{1}\right)^{2} \omega_{I} \ .\tag{4.58}$$

Analysis of formulas (4.57) and (4.58) shows that the magnitude of the moment of the number of motion of the knife when cutting the feed depends on the dynamic interaction of the knife with the material and the design features of the grinder. This parameter increases with the hinged attachment of the knife, since in this case the stiffness of the connection between the knife and the rotor is broken. As a result, the blades in operation deviate from the radial position, which transforms the cutting angle.

Thus, the least energy–intensive is a grinding machine with a rigid knife attachment on the rotor of the machine. In this case, during the execution of the technological process, there is no disturbance in the dynamics of the movement of active working elements and the reliability of the grinder is increased.

4.6 Determining the parameters of the drum–type cutting device

The main parameters of the tympanic cutting device are: the height h of the location of the throat relative to the axis of the drum shaft and the diameter D of the grinding drum. The location of the neck relative to the axis of the drum shaft along the vertical (fig. 4.13, a) is due to the kinematic operating mode and in this case depends on the ratio of the translational velocity vs the stalk layer and the horizontal component vgor of the peripheral speed vb of the drum.

From the diagram (fig. 4.13, b), it can be seen that the horizontal component of the speed of the blade when the drum rotates by 90° changes its value from –vb to 0 and by rotating another 90° from 0 to + vb. At the same time, the translational velocity vs of the feed of the layer remain constant in magnitude and direction. When the blade meets the layer in the quadrant I, the particles of the layer will be repelled by a knife, which prevents the mass from entering the drum and breaks the cutting process.

When meeting in the quadrant II, the blade will assist in drawing the layer into the gap of the cutting pair, thus ensuring a more favorable working environment for the apparatus. Consequently, the supply of material to the knife should be carried out in the drum quadrant II and precisely about the horizontal diameter of the drum, since then its resulting velocity will be directed almost along the layer, and the working process conditions will be violated. In connection with this, the neck in these devices is made of a small height, and the material in the drum is fed in a thin layer.

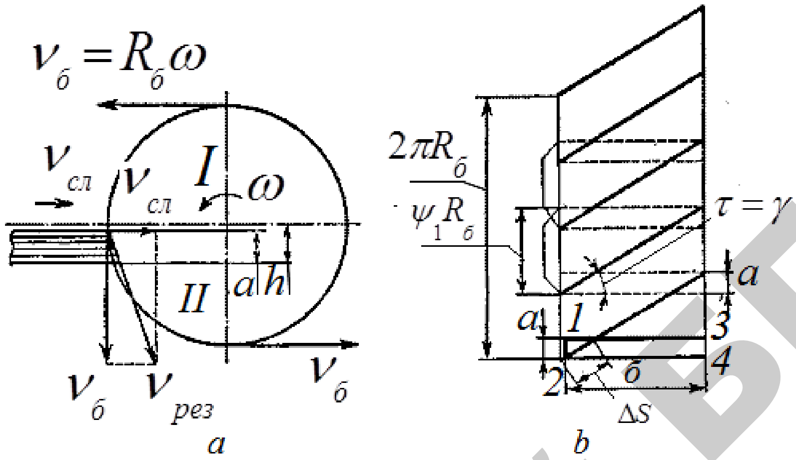


Fig. 4.13. Scheme to justify the location of the neck with respect to the drum axis (a) and scanning of the knife drum (b)

Reznik recommends determining the height h of the shaft of the drum above the counter plate by formula

$$h = a + \frac{Dv_{сл}}{2v_б}; \tag{4.59}$$

where D – is the diameter of the drum, m; a – thickness of the layer equal to the height of the neck, m.

The working edge of the opposing plate is located on the circumference described by the blade of the knife and is removed from the vertical diameter of the drum by a distance:

$$u = \frac{h}{tg\psi_n}; \tag{4.60}$$

where ψ_n – is the angle determining the position of the working edge relative to the axis of rotation of the drum ($sin\ \psi_n = 2h/D$), deg.

The cutting device of the drum type is characterized by simple relations between the main parameters of its operating mode. If we consider the sweep of the drum (fig. 4.13, b), then the spiral blade of the knife will be represented by a straight line inclined to the generatrix of the cylinder described by this blade, at the angle χ of the solution. In this apparatus, the slip angle is equal to the angle of the solution, and they have constant values (within 24...30 °). With a constant radius of the drum, the normal component of cutting speed also has a constant value.

The working process per pass of one knife is characterized by the fact that the loaded section *S* of the blade changes its value: from point 1 to point 2 it increases, then to point 3 it retains its maximum value and then decreases to zero. The nature of the change in the length of the loaded section can be represented graphically as an equilateral trapezoid. The change in the length of the loaded section causes a proportional change in the total resistance to cutting. To align the load on the shaft, the knives on the reamer of the drum are placed with overlapping. Overlapping should be equal to the thickness of the layer, i.e. the height of the neck. In this case, the length of the arc of coverage per blade of one knife is determined by the formula

$$R_б\psi_1 = \frac{2\pi R_б - za}{z} = l_б tg\tau\ ; \tag{4.61}$$

where $R_б$ – is the radius of the drum, m; Ψ_1 – angle of incidence of the knife, degrees; z – number of knives, pcs; $l_б$ – drum length, m; a – throat height, m.

The main parameter of the drum–type cutting device is the radius R_b of the drum, which can be found by formula

$$2\pi R_б = z\left(l_б tg\tau + a\right). \tag{4.62}$$

With a considerable length of the drum $l_б$, it can be equated to the width b of the neck, and then

$$R_б = \frac{z\left(l_б tg\tau + a\right)}{2\pi}. \tag{4.63}$$

The number of blades is from 2 to 8 and is necessarily even for balancing reasons.

In the presence of great dynamic advantages due to the uniformity of the load on the shaft and the absence of the need to have a balancing flywheel, the drum machines are not without flaws. The need to feed the material with a thin layer limits the productivity of the machine. In addition, the presence of spiral knives causes a noticeable axial force on the shaft, and the manufacture of knives and sharpening them during operation are quite complex.

4.7. Calculation of the feeders of grinders

In the grinders of coarse forages with the grinding apparatus of the drum type, the cutting process is preceded by the operation of feeding the feed to the working zone of the cutting pair of a knife–countermeasure performed by the feeder. In order to reduce energy costs for the cutting process, the feed mass is subjected to preliminary compaction.

As a rule, the clamping elements of the feeder are loaded with a spring, which allows them to move in a vertical plane. When performing the work process, the coarse feed of even layers is laid on the feeding conveyor and moved to the working area of the pressure roller (conveyor). Under the action of the clamping element, the mass layer is compacted and fed to the working zone of the cutting device that performs cutting of the feed. In accordance with the foregoing, the degree of compaction of the layer and the thickness of the feed layer depend on the parameters of the working organs of the feeder. The feeding apparatuses of the following schemes were most widely used of the known designs of feeding apparatuses. The feeder, consisting of the feeding conveyor and the upper and lower feeding rollers (fig. 4.14), is characterized by a prosthetic structure.

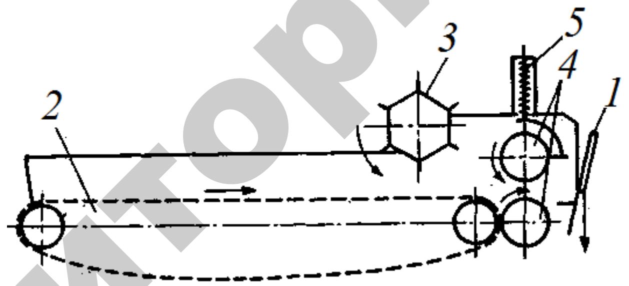


Fig. 4.14. Driving feeding device roughage with rollers:
1 – knife; 2 – feed conveyor; 3 – roller presser; 4 – roll feed; 5 – spring

To determine the parameters of the feeder with the rollers, which ensure the compaction and feeding of the feed layer, consider the design scheme of the working process (fig. 4.15).

So the upper roller exerts pressure on the mass, then this force P can be divided into two components

– radial $$N = \frac{P}{\cos\alpha};$$

– horizontal $$H = Ptg\alpha,$$

where P – is the pressure force of the upper roll on the feed layer, H; α – is the angle of capture of the mass layer, deg.

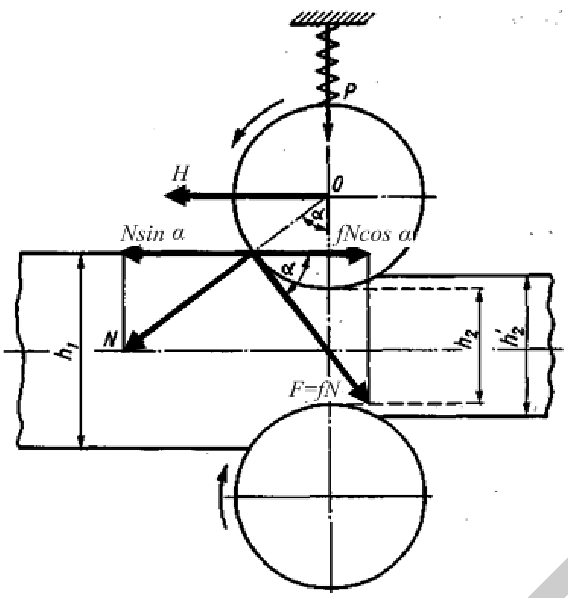


Fig. 4.15. Scheme for the calculation of feed roll parameters

The radial force N induces frictional force on the plane of the roller $F = fN$. Having decomposed these forces into vertical and horizontal it follows that the vertical force compresses the feed layer from the initial thickness h_1 to the value h_2. In the calculations it is recommended to take the ratio $h_1/h_2 = 0{,}4...0{,}6$.

To fulfill the condition of tightening the feed between the rolls, the condition is required

$$fN\cos\alpha > N\sin\alpha, \tag{4.64}$$

where f - coefficient of friction, $f = tg\alpha$.

Therefore, in order to ensure the capture of the feed layer, it is necessary that the friction angle φ is greater than the angle of capture of the layer of mass α.

An important parameter of the feeding device with rolls is the diameter of the rollers. This parameter depends on the ratio of the thickness of the inlet layer and the packed layer of the feed. From the triangle aOc it is clear that

$$\frac{h_1 - h_2}{2} = \frac{D}{2} - \frac{D}{2}\cos\alpha\,. \tag{4.65}$$

Whence $$D = \frac{(h_1 - h_2)}{1 - \cos\alpha},$$

where h_1 – is the thickness of the incoming feed layer, m; h_2 – thickness of the packed layer of food, m.

Since

$$\cos\alpha = 1\Big/\sqrt{1+tg^2\alpha}\,,\quad tg\alpha \le f = tg\varphi,$$

Then

$$D \ge \frac{h_1 - h_2}{1 - \dfrac{1}{\sqrt{1+f^2}}}\,. \tag{4.66}$$

To fulfill the last condition, the values of the diameter of the rolls are large, which is constructively inconvenient (4.66). Therefore, the diameter of the rolls is less than the calculated value. Rollers carry toothed or serrated (fig. 4.16) .In this case, the friction force is replaced by the force of adhesion between the material and the roller. In order to ensure an uninterrupted tightening of the mass between the rollers rotating at a circumferential speed $\upsilon_{\text{в}}$ coming from the feeding conveyor at a speed of v, it is necessary to observe the conditions $\upsilon_{\text{в}} > I_{\text{тр}}$. In the calculations take $\upsilon_{\text{в}} = (1{,}25...1{,}31)\ B_{\text{тр}}$.

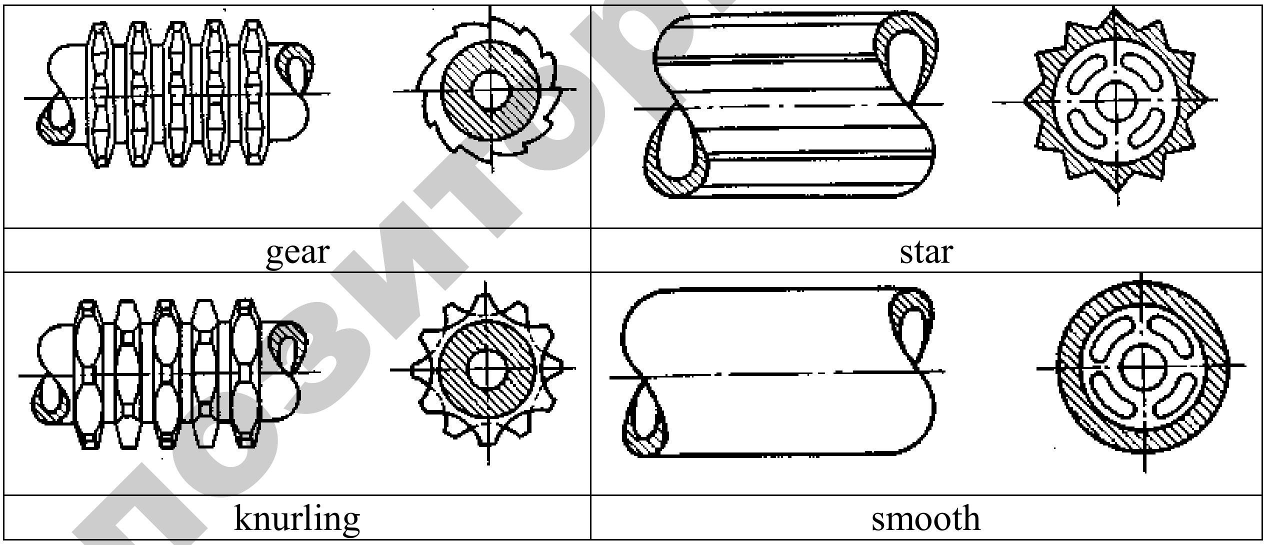


Fig. 4.16. Types of rollers

The speed of the upper υ_1 and the lower υ_2 rolls play an important role in the process of mass motion from the feeding rollers. In order to ensure the uniformity of feed supply to the grinding machine, it is recommended that $\upsilon_{\text{м}} = 0{,}88...0{,}93$ $(\upsilon_1 = \upsilon_2)$.

When the work process seal layer stalked feed undergoes several phases. The supply of feeder food fit evenly on the conveyor belt 2 and the pinch roller compacted 3. Due to the displacement of stems of plants, pinch roll 3 press mass to seal the coefficient 0,7...0,8.

However, when leaving the work area, with its mass of expanded again by acquiring a state in which the compression ratio is equal to 0,4...0,5. This status feed back layer requires its seal, so the food goes to the feed rollers 4 of which goes to the chopping device. The design of the feeding apparatus of roughage with prostate rollers different designs, however, reseal feed requires additional energy costs. A feeder consisting of feeding and pushing conveyors allows reduce unproductive energy costs (fig. 4.17).

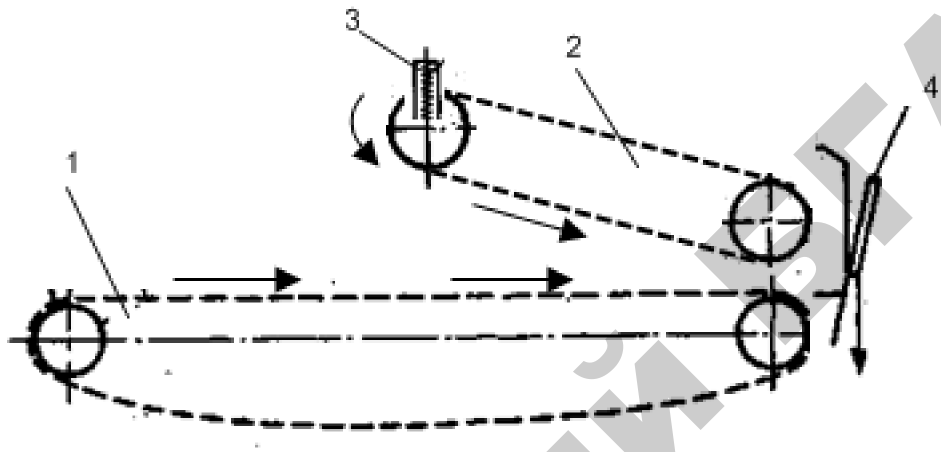


Fig. 4.17. Scheme of feeder for coarse forages with a clamping conveyor:
1 – feeding conveyor; 2 – clamping conveyor; 3 – a spring; 4 – knife

Feeding conveyors ensure a uniform compaction of the stalks, which reduces the phenomenon of expansion of the packed feed layer at the outlet from the feeder. However, the feeding apparatus considered is the most complicated in its structure and less reliable in operation.

Chapter 5
MECHANIZATION OF FOOD DOSAGE

5.1 Zootechnical requirements and classification of batchers

Dosing is the process of measuring a material with a specified accuracy that does not exceed the established requirements. Dispensers that ensure the operation of the mixers must produce the components of the future feed mixture in accordance with the recipe.

The following requirements are imposed on batchers:

– accuracy and stability of power supply, necessary to ensure the constancy of the specified flow within the limits of permissible deviations. Inaccurate dosing of components reduces the feed and biological nutritional value of feed mixtures;

– regulation of the rate of feed delivery within the specified limits;

– the ability to work with different materials;

– simplicity of the device, low metal and energy consumption;

– high performance;

– ability to create automatic lines.

Quality of dosing is carried out on such parameters as technological tolerance, absolute error of dosing, coefficient of variation.

The accuracy of dosing by any type of dispenser is determined by zootechnical requirements and is limited to the technological tolerance:

$$\varDelta_T = Q_{max} - Q_{min} / Q_{cp}, \tag{5.1}$$

where Q_{max}, Q_{min}, Q_{cp} – the maximum, minimum and average consumption (productivity) of the dispenser, kg/s or m^3/s (when operating at the same set dose).

When dosing feed, it is necessary that the maximum relative error of dosing does not exceed the technological tolerance:

$$\mathrm{Q_{max}} - \mathrm{Q_{min}} / \mathrm{Q_{cp}}, \leq \Delta_{\max} \leq \Delta_{\mathrm{T}}. \tag{5.2}$$

The process tolerance is different for different materials. For example, when dosing of stalk feeds, it can be taken to $\Delta_{\mathrm{T}} = 0{,}1$.

The average absolute error of dosing is determined by the formula

$$\delta = \frac{\sum_{i=1}^{n} \left(Q_i - Q_\delta \right)}{n} \tag{5.3}$$

where Q_i – is the actual flow rate; Q_p – calculated or set flow rate; n – is the number of measurements.

The estimated index of the relative error is the coefficient of variation:

$$v = \frac{1000}{Q_{cp}} \sqrt{\frac{\sum_{i=1}^{n} \left(Q_i - Q_{cp} \right)}{n - 1}} \tag{5.4}$$

The main factors that adversely affect the dosing process are:

– clogging of working organs with large–sized inclusions;

– a different degree of compaction of feed in feed storage bins, depending on the filling height;

– increased moisture content of the components, contributing to caking, clumping and arborization;

– the presence of protuberances and other obstacles in the bottoms and walls of the over–hopper bunkers that inhibit the free yield of products.

Depending on the method of dosing (by volume or weight), the dispensers are divided into:

– according to the method of issuing a predetermined amount of a substance, dispensing and continuous dose dispensers are distinguished;

– volumetric, for which a periodic repetition of the cycle of output of a material is characteristic. The following types of volumetric dosing are known:

1. The drum type dispenser is designed for dispensing bulk products (mixed feed, flour, cereals).

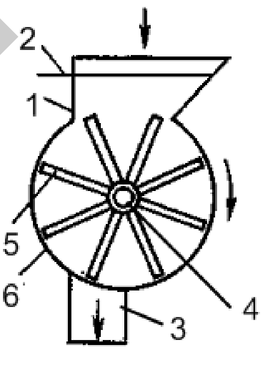


Fig. 5.1. Scheme of volumetric drum type dosage of feeds:
1 – bunker; 2 – damper with control mechanism; 3 – discharge nozzle;
4 – drive shaft; 5 – drum; 6 – housing

A special feature of the design of this dispenser (fig. 5.2) is the drum 1, fixed in a steel casing, on the shaft, which is made up of individual sprockets 3, between which are disks 2 dividing it into four sections. Depending on the physical properties of the components, drums of different shapes are used: A – for cereals; B – for mealy; B – for non–friable; D – for components included in recipes in small quantities.

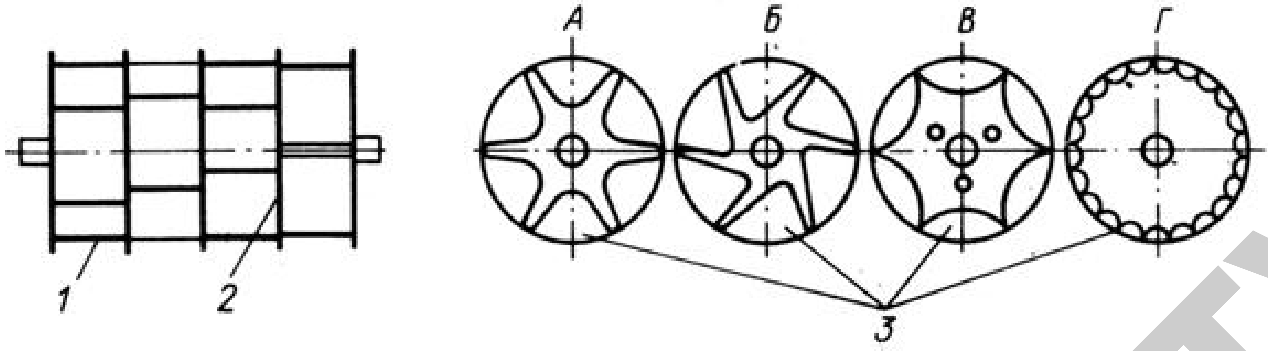


Fig. 5.2. Diagram of the working part of the drum metering device:
1 – drum; 2 – dosing disk; 3– drum rollers

The technological scheme of supplying the product to the drum type dispenser is as follows. The components enter the receiving part of the dispenser, where the drum cells are uniformly filled. Rotating, the product empties out of them and is removed from the dispenser. The dispenser is considered simple and reliable. Its disadvantage is the relatively low productivity. The feed enters the receiving branch pipe, loosens and enters the drum pockets. The drum, rotating, throws the food into the outlet. The speed of rotation should not exceed 30...40 min^{-1}.

The capacity of the dispenser is changed due to changes in the rotational speed and geometrical parameters of the drum and is determined by the formula

$$Q = \frac{\pi}{30} \omega z S_{ж} l_{ж} \rho \varphi, \tag{5.5}$$

where ω – is the angular velocity of the drum, s^{-1}; z – number of blades, pcs; $S_{ж}$ – is the cross–sectional area of one trough, m^2; $l_{ж}$ – length of the trough, m; ρ – density of feed, kg/m^3; φ – coefficient of filling cells (φ = 0,8...0,9).

The power to drive the dispenser depends on the frictional force of the food captured by the drum, on the overlying layers of the feed. The frictional force at sliding of forage is defined by the formula

$$P_{тр} = P_{н} S_{ж} f = \frac{F_{тяж}}{S_{к}} S_{ж} f \,, \tag{5.6}$$

where $P_{н}$ – feed pressure on the drum surface, N/m^2; $S_{ж}$ – is the cross–sectional area of one trough, m^2; $S_{к}$ – cross–sectional area of the hopper throat above the drum, m^2; f is the feed friction coefficient of feed; $F_{тяж}$ – gravity of the feed, N.

The power for driving the dispenser drum can be determined from formula

$$N = \frac{P_{тр}\ \upsilon K_1 K_2}{1000\eta} = \frac{P_н S_{жс} f\upsilon\ K_1 K_2}{1000\eta}\ , \tag{5.7}$$

where υ – is the peripheral speed of the drum, m/s; K_1 – coefficient, taking into account the resistance of the food to crushing, for powdery materials K_1 = 1.0, for lumpy K_1 = 2.0; K_2 – factor, taking into account friction losses of the dispenser working elements (K_2 = 1,1...1,2); Η is the transmission efficiency.

Drum peripheral speed:

$$\upsilon = \omega R_б\ , \tag{5.8}$$

where $R_б$ – is the radius of the drum, m.

2. The tray dispenser is designed for dosing salt with a moisture content of 3...4 %, Chalk with a moisture content of not more than 6...8 % and other components of mixed fodders. The disk dispensers are characterized by high accuracy and wide range of capacity control.

In operation, the material to be dispensed comes from the hopper 5, the rotating horizontal disc 4, from which the components are dropped by the fixed scraper 2 (fig. 5.3).

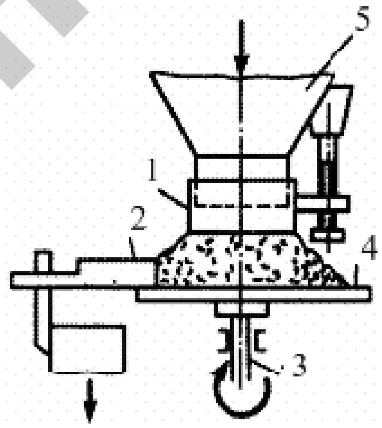


Fig. 5.3. Scheme of the disk dispenser:
1 – cuff; 2 – scraper; 3 – shaft; 4 – disk; 5 – receiving hopper

The components on disc 4 come from receiving hopper 5 and are distributed along the disc in the form of a truncated cone. The dimensions of the cone are regulated by a cuff 1.

The principal and calculation schemes are shown in fig. 5.4. For one turn of the disk, a portion of material is taken from it, having a volume of a ring with a triangular cross section.

The feed of the dispenser is determined by formula

$$Q = V \rho n / 60, \tag{5.9}$$

where V – is the volume of material discarded from the disk per its rotation, m^3; ρ – bulk density of feed, kg/m^3; n – is the rotational speed of the disk, min^{-1}.

We define the volume V as follows:

$$V = F\, 2\pi\, R_o, \tag{5.10}$$

where F – is the cross–sectional area of the annular layer, m^2; R_0 – is the distance from the axis of rotation of the plate to the center of gravity of the cross section, m.

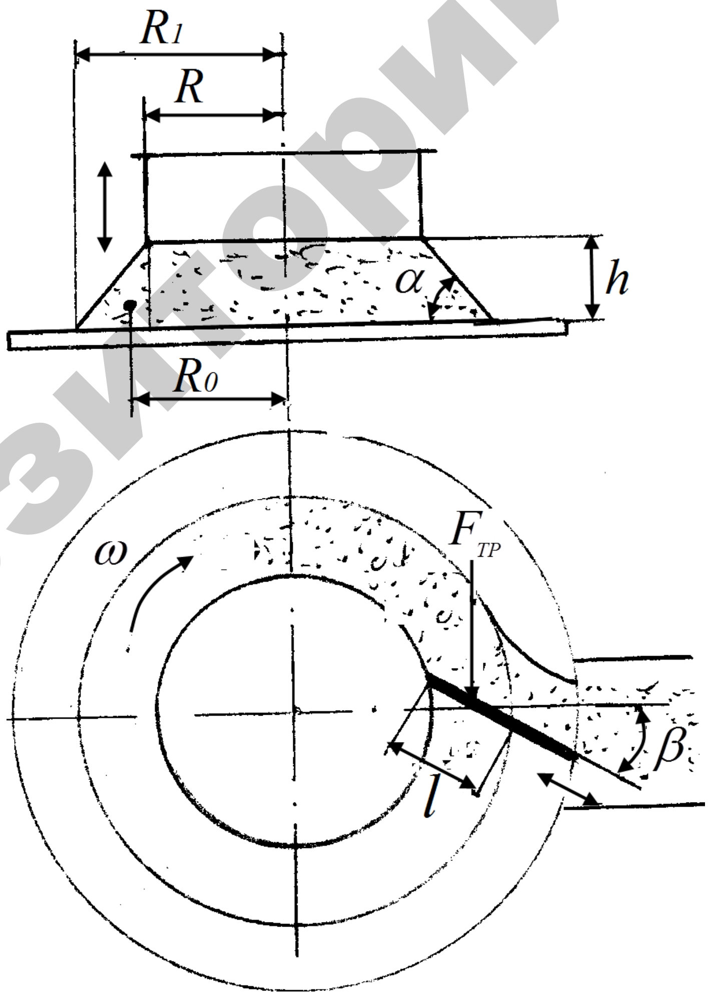


Fig. 5.4. Scheme for calculation of the disk dispenser

The value of R_0 is:

$$R_0 = r + a' = r + \frac{h}{3tg\alpha}. \qquad (5.11)$$

Since

$$a' = \frac{1}{3}a = \frac{h}{3tg\alpha}, \qquad (5.12)$$

Then

$$F = \frac{1}{2}a\,h = \frac{1}{2}\frac{h}{tg\alpha}h = \frac{1}{2}\frac{h^2}{tg\alpha}. \qquad (5.13)$$

Thus

$$V = \frac{1}{2}\frac{h^2}{tg\alpha}2\pi\left(\mathrm{r} + \frac{h}{3tg\alpha}\right). \qquad (5.14)$$

The feeding of the tray dispenser is finally equal to:

$$Q = \frac{60\pi\ h^2\ \rho}{tg\alpha}\left(\mathrm{r} + \frac{h}{3\ tg\alpha}\right), \qquad (5.15)$$

where h – is the height of the casing over the disk, m; r – shell radius, m; α – is the angle of the natural slope of the material during motion, deg.

The limiting angular velocity of a disk is determined from the condition that the centrifugal force of inertia must be less than the frictional force of the product about the disk:

$$m\,\omega^2\ R < m\ g\ f_1, \qquad (5.16)$$

$$\omega\ \leq\ \sqrt{\frac{g\ \ f_1}{r}}, \qquad (5.17)$$

where R – is the radius of the lower base of the cone of the material, m; f_1 – is the coefficient of friction of the material against the disk.

Power to drive the disk dispenser is:

$$N = N_1 + N_2 + Nxx, \tag{5.18}$$

where N_1 – the power to overcome the resistance of the feed from friction it against the plate, kW; N_2 – the power to overcome the resistance of the feed from its friction against the scraper, kW; N_{xx} – power consumed by the dispenser in idling mode, kW.

Power to overcome the resistance of food from friction to the plate:

$$N_1 = F_{TP}\, V, \tag{5.19}$$

where F_{TP} – is the frictional force that occurs when the food moves along the plate, H,

$$F_{TP}=mg, \tag{5.20}$$

Feeding speed on the plate, m/s.

$$V = \omega\, R_0 = \frac{n\,\pi\, R_0}{30} = \omega\left(r + \frac{h}{3tg\alpha}\right). \tag{5.21}$$

The limiting speed of the plate:

$$n_{np} = \frac{1}{2\,\pi}\sqrt{\frac{g\ \ f}{R + \dfrac{h}{tg\,\varphi}}}\ , \tag{5.22}$$

where f – is the friction coefficient of the plate; φ – is the angle of the slope of the metering material, deg.

The power used to overcome the friction of the feed on the scraper:

$$N_2 = N_1 \cos\beta\ , \tag{5.23}$$

where β – is the angle of installation of the scraper, deg (for $\beta = 0$, $N_1 = N_2$).

Then:

$$N = F_{TP}\, V(1 + \cos\beta)\, N_{xx}\ , \tag{5.24}$$

3. Screw feeder is used for dosing and feeding of cereals, small–lump and mealy components. It consists of a hopper 1, under which there is a screw conveyor 3. Due to the speed variator, which is installed in the drive device, regulates the productivity of the dispenser, changing the speed of rotation of screw 2 (fig. 5.5).

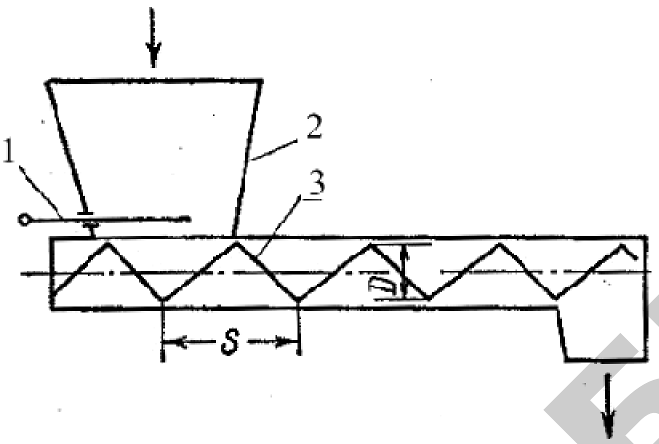


Fig. 5.5. Scheme of volumetric screwt ype dosage of feeds:
1 – damper; 2 – bunker; 3 –screw

The productivity of the dispenser can be determined by the formula

$$Q=\pi\,(D^2-d^2)\,S\,\rho\,\varphi\,\omega, \tag{5.25}$$

where D – is the diameter of the screw casing, m; d – is the shaft diameter, m; S – pitch of screw turns, m; φ – is the duty cycle; ρ – density of feed, kg/m^3; ω – is the angular velocity of the screw, s^{-1}.

The power expended on the drive of the screw feeder can be determined from formula

$$N=Q(L\,K\pm H)K_1, \tag{5.26}$$

where K – is a coefficient that takes into account the displacement resistance; K_1 – is a coefficient that takes into account the loss of power to bearing friction.

4. Tape dispenser (fig. 5.6). The belt dispensers have a greater productivity, they consist of a belt conveyor, above which a feed hopper is installed. To adjust the feed rate, a flap is mounted in the hopper. The disadvantage is the relatively low accuracy of dosing feed.

The capacity of the volumetric belt dispenser can be adjusted within wide limits by changing the speed of the belt or the position of the slide gate valve and determined by the formula

$$Q = b\, h\, \upsilon\, \rho\, \psi, \tag{5.27}$$

where b – is the width of the feed layer on the belt, m; h – is the thickness of the feed layer on the belt, m; υ – belt speed, m/s; ρ – bulk density of feed, kg/m^3; ψ – is the duty cycle.

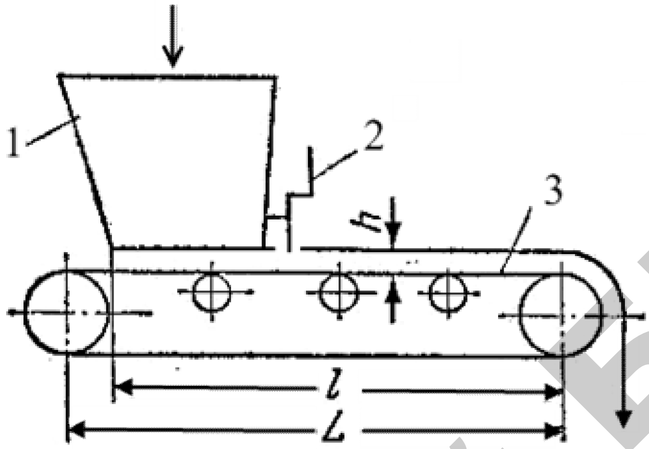


Fig. 5.6. Volumetric belt dispenser:
1 – bunker; 2 – damper; 3 – belt conveyor

The power expended on the drive of the belt feeder

$$N_1 = K_1 \left(m_o \upsilon + QL \right) g \pm H\, Q\, g, \tag{5.28}$$

where m_o – is the mass of the tape, m; L – length of the conveyor, m; H – the height of the lift of the feed (taken into account for the case when the conveyor is installed at an angle to the horizontal plane), m; Q – productivity of the dispenser, t/h; K_1 – is a coefficient that takes into account the power loss to bend the conveyor belt on the drum.

The power spent on friction of food about the trough:

$$N_2 = P_n f \upsilon, \tag{5.29}$$

where f – is the coefficient of friction; P_n – is the force of normal feed pressure on the trough, N.

$$P_n = mg \cos\theta, \tag{5.30}$$

where m – is the mass of the feed, kg; θ – is the angle of the natural slope of the feed.

If the enclosing walls of the trough are inclined to the horizon at the angle of the natural slope, then the mass of the material bounded by the volume $h^2l/tg\theta$ (l – is the length of the trough, θ – is the angle of the natural slope) will be applied to them.

Then the power expended on friction of the feed about the trough:

$$N_2 = h^2 l \rho g f \upsilon \frac{\cos\theta}{tg\theta}, \tag{5.31}$$

where h – is the height of the sides, m;

The power expended on the drive of the belt conveyor can be determined by the formula

$$N = \frac{N_1 + N_2}{\eta}, \tag{5.32}$$

where η – is the transmission efficiency.

5. Bulk dosing unit for loose feeds (fig. 5.7). Such batchers are usually installed under the hopper. When the damper 2 is opened, the food from the hopper 1 enters the cavity of the dispenser 3. When the feed fills the cavity 3, the damper 2 is closed. When the damper 4 is opened, a predetermined portion of the feed enters the mixer.

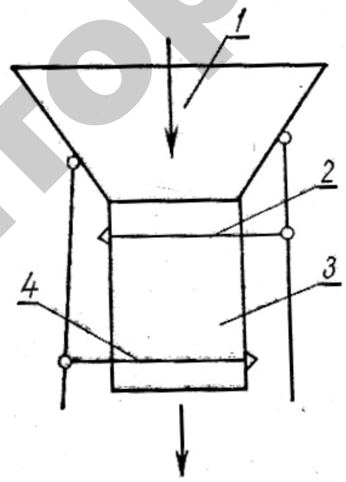


Fig. 5.7. Volumetric Portion Dispenser:
1 – bunker; 2 – damper; 3 – cavity of dispenser; 4 – outlet flap

The capacity of the dispenser is determined by the formula:

$$Q = \frac{V\rho}{t_{выд}}, \tag{5.33}$$

where V – is the volume of the cavity of the dispenser, m^3; $t_{выд}$ – the time of issue of one portion ($t_{выд}$ = 30...120 s).

A similar kind of dispensers (screw, tray) can also dispense crushed root crops.

6. It is most difficult to dose silage, hay, crushed coarse fodder (not shredded at all, not dosing). As feeder–dosers for the accumulation and dosed supply of stalk materials in feed processing lines, chain–slat feeders with bitter devices of various design are used.

The following constructions are distinguished:

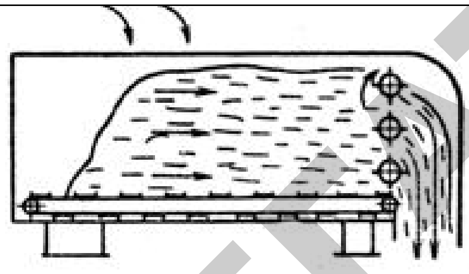
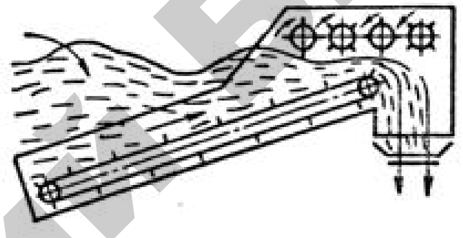
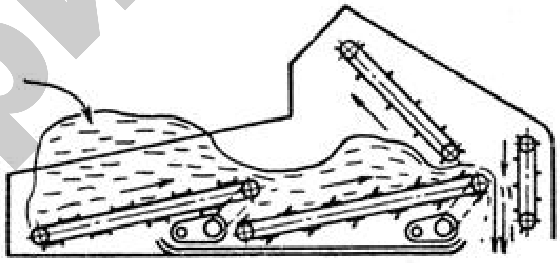
– horizontal with vertical arrangement of biter	
– Oblique with horizontal position of biter	
– Stepped with combing conveyor	

Fig. 5.8. Chain–slat feeders

The technological process of dosed supply of stalk and other cohesive feeds is as follows. A monolith of feed material located in the hopper of the feeder–dispenser is fed by a chain–slat conveyor to the combing device consisting of biters. The pins of the rotating biters scrape the material in contact with them and unload them onto a transverse conveyor or are fed directly to the processing line for collecting and mixing the components of the prepared feed. In any case, the throughput of the biter system should somewhat exceed the capacity of the feeder, which supplies feed to beaters. Under this condition, the system provides a dosed delivery of connected feed materials with permissible deviations. To increase the accuracy of dosing of stalk feeds, a two–stage system consisting of feeders–dispenser of biter type and leveling device that provides smoothing of the feed stream coming from the feeder and automatic control of its operation is often used.

As the volume of fodder mass in the bunker decreases, a shift begins, and then the upper layers of the mass collapse. The amount of feed given out here decreases sharply and amounts to 60...70 % of the norm of the set issuance.

Mass (weight) dispensers allow to make recipes of mixtures with accuracy more than volumetric (in conditions of constant microclimate) with an error within ± 0,1...1 %, and therefore their application is mandatory in the lines of preparation of premixes, protein–vitamin additives And high–quality mixed fodders with the introduction of components that make up less than 3 % of the mixture.

The use of mass dispensers in the feed industry is usually combined with batch mixing of components and automation of line management. By design, weighing batchers are similar, differ only in the number of feeders and bucket load capacity. Each individual component in accordance with a preset recipe is fed to the scales by feeders that are individually driven by speed motors. Feeders are switched to sequentially feed components from bins automatically, after receiving a portion of the specified mass. At the end of each serving, the screw conveyor switches to a reduced speed for a more careful pouring. The weighted portion is also automatically discharged from the scoop to the mixer or to the collecting hopper. After this, a new weighing cycle begins. Mass continuous dosing is hampered by the lack of reliable and simple in design weight continuous dosing devices. The existing systems are inferior to the batch systems in terms of accuracy and economic indicators.

Chapter 6
MECHANIZATION OF FOOD MIXING

6.1 Assessment of mixing quality

The need for preparing feed mixtures is determined by the fact that in none of the feeds there is a complete set of nutrients. Feeding of full–flow mixtures increases the productivity of animals by 25...30 % with the reduction of fattening terms by 15...20 %. The feed consumption also decreases. The use of mixtures that are inhomogeneous in their composition for feeding animals significantly reduces their productive effect. Mixing refers to the technological process of moving particles of material, as a result of which in any volume of the mixture will contain a predetermined number of its components. In addition, mixing feeds are used to intensify the processes of heat and mass transfer. The machines in which the mixing process takes place are called mixers, and their working elements are stirrers.

In dynamics – mixing is a set of processes of directed formation of systems of a homogeneous composition, density, and physico–mechanical properties of a set of required components. Sometimes the mixing is combined with the grinding of the components. Depending on the type and method of keeping animals or birds, the adopted type of feeding, and the availability of feed in the farm, the feed mixtures are prepared in different consistencies:

– dry (mixed fodder and feed mixtures) – humidity W = 13...15 %;

– wet loose W = 45...70 %;

– liquid (flowing) W = 75...85 %.

With periodic mixing in the mixer comes a set of components and during the mixing occurs:

1. Moving a group of adjacent particles from one place of the mixture to another by the introduction or sliding of layers and the gradual redistribution of the particles of various components through the newly formed boundaries of their separation. In this case, the particles are evenly distributed in the mixture when mixed. At the beginning of the process, the quality of the mixture is improved mainly as a result of the movement of particles from one place of the mixture to another by the introduction or sliding of the layers. The speed of the mixing process is practically independent of the physico–mechanical properties of the components, since the process is at the level of large volumes. The most important role at this time is played by the design of the mixer, which gives the mixture a certain character of motion;

2. Concentration of particles having similar dimensions, shape, mass in different places of the mixer under the action of gravity forces (gravitational forces).

In this phase, the density, shape and nature of the particle surface, the particle size distribution, the moisture content of the component, its flowability begin to influence the mixing efficiency. The closer the components are to their properties, the more effective the process of mixing them. The greater the difference in the physico–mechanical properties of the blended components the longer this process goes.

With a large number of components, the share of each of them decreases, and the duration of the process increases. The latter process prevents the uniform distribution of particles.

With continuous mixing, the flow of components, their mixing and the delivery of the finished mixture take place continuously. The quality of the finished mixture obtained in these mixers depends not only on their design, but also on the uniformity of the dosing of the components. Therefore, the mixer should not only mix well the components, but also smooth the pulsation of their supply. In such cases, in addition to the transverse, there must necessarily be a longitudinal movement, created usually by blades, which are capable of moving components, both in the direction of movement and partly back. The effectiveness of mixing is estimated by the degree of heterogeneity of the mixture:

$$Q = \frac{100}{B_t}\sqrt{\frac{\sum_{i=1}^{n}(B_t - B_0)^2}{n-1}}, \tag{6.1}$$

where B_t – is the fraction of the smaller component in the sample; B_0 – is the fraction of the smaller component in the ideal (calculated) mixture; n –is the number of samples.

Homogeneous is a mixture in which, in any small volume, the ratio of components corresponds to the ratio of the components as a whole for the mixture.

If Q> 30 % – the mixer works poorly. With an ideal mixing of $C_i = C_0$, therefore $Q = 0$. A qualitative characteristic of the mixing process is the unevenness (heterogeneity) of the mixture, estimated by the coefficient of variation of the Cx of the controlled or control component introduced in an amount of 1 % to the mass of the whole mixture. To calculate the variation coefficient of the monitored or control component, 15...20 samples are taken at regular intervals when the finished mixture is unloaded by a continuous mixer or from the entire volume of the mixture in a batch mixer. The sample weight for mixed feed should be:

– 5 g, wet and liquid mixtures for pigs;

–100 g, dry mixtures and for cattle;

– 300 g, wet mixtures for cattle.

For continuous mixers, the unevenness (inhomogeneity) of the mixture is determined by the formulas:

– arithmetic mean concentration of the control component

$$X = \frac{\sum x_i}{n}, \tag{6.2}$$

where x_i – is the concentration of the controlled or control component in the samples (weight, relative, number of units); n – is the number of taken samples.

– indicator of process variability

$$\sigma_x = \sqrt{\frac{\sum (x_i - x)^2}{n-1}}, \tag{6.3}$$

– coefficient of variation

$$C_X = \frac{\sigma_x}{x} 100\%. \tag{6.4}$$

For batch mixers, a more objective estimate is obtained by calculating the exponents and C_X by the formulas, where instead of the mean value of the control component in all samples, the calculated (theoretically expected) amount of this component in each sample x_p is used:

– indicator of process variability

$$\sigma_x = \sqrt{\frac{\sum (x_i - x_p)^2}{n-1}}, \tag{6.5}$$

– coefficient of variation

$$C_X = \frac{\sigma_x}{x_p} 100\%. \tag{6.6}$$

The homogeneity of the mixture θ is related to the inhomogeneity by the relation

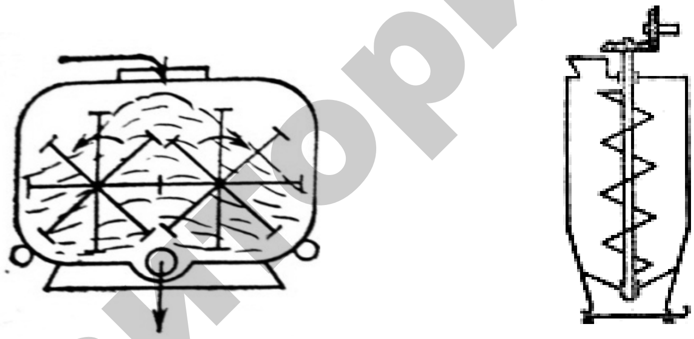
$$\theta = 100 - C_x, \tag{6.7}$$

Smaller C_x and greater θ call more uniform of the mixture which characterizes efficiency of mixers. The calculated value of the homogeneity of the mixture should not exceed the zootechnical norms. Regardless of the types of mixers and their design features, some common properties of the $\sigma(t)$ and $C_x(t)$ variability indices are found in the mixing process, namely, as the duration of mixing in batch mixers and the operating length of the continuous mixers increases, the decrease in these indices is identical and Approaches a certain limit. Increasing the mixing time or the length of the mixer does not improve the uniformity of the distribution of components. In accordance with zootechnical requirements, the uneven mixing in the preparation of feed mixes for cattle should be no more than 20 %, and when feeding additives – no more than 10 %. The currently used mixers for the preparation of feed mixtures can be classified as follows.

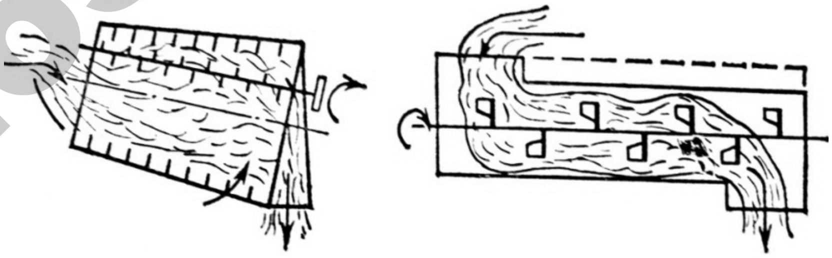
6.2 Types of feed mixers

By the nature of the process, the following mixers are distinguished:

– portioned (periodic) action;



– continuous operation.

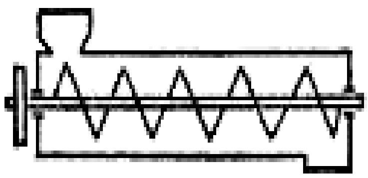
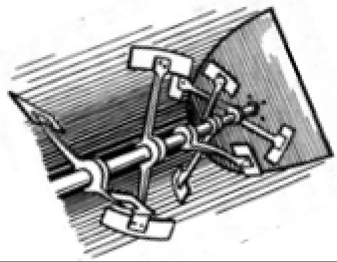
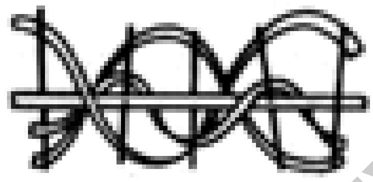


According to the main production purpose, depending on the type of miscellaneous feed mixers can be designed for:

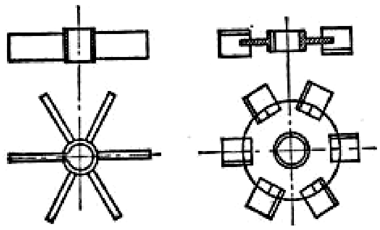
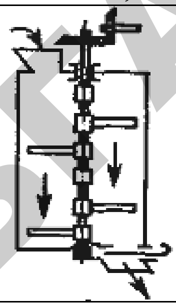
– preparation of dry loose (mixed fodders),

– loose wet

– liquid (consistent) feed.

screw	blade	belt
		

– for liquid

turbine	propeller	blade;
		

Mixers are used for the construction of the working bodies (stirrers):

– for loose feeds

– for loose wet (stalk) forage – screw and lobed.

Depending on the speed of the mixers, the mixers are divided into:

– low–speed ones, in which the kinematic regime $K = (\omega^2 R/ g) < 30$ (here ω is the angular velocity, s^{-1}, R is the rotational radius of the mixer, m),

– and fleet, for which $K > 30$.

According to the number of mixers, the mixers are divided into single– and double–shaft mixers. By the location of the working chamber:

– vertical;

– horizontal;

– Inclined or planetary.

For the preparation of moist feed mixtures from the stalk forage and root crops, predominantly slow, horizontal single– or double–shaft bladed mixers of batch action are used. Liquid components are mixed, as a rule, mechanically in apparatus with agitators. But in a number of cases, pump circulation or pneumatic mixing is used.

For the mechanical method, slow–speed bladed impellers or high–speed blades are used – turbine and propeller. Bladed stirrers are used for mixing small volumes of high viscosity liquids, propeller mixers for small viscosity liquids. Turbine agitators allow a wide range of viscosities.

To the design and modes of the working bodies of the mixers are required:

– elimination of stagnant zones and separation of the mixture by granulometric composition,

– ensuring fast loading of components and unloading the feed mix.

6.3 Calculation of mixer parameters

Screw feed mixers are intended for the preparation of a mixture of all types of feed, except for liquid, although in small quantities liquid additives are allowed, and consist of a loading chamber, casing, auger, discharge funnel and drive mechanism. The working process is reduced to the fact that the components fed continuously into the loading chamber are subjected to intense action of the rotating screw inside the circular casing. As a result, the food is stratified, individual layers receive different circumferential speeds, pour one against the other and are unloaded, gradually moving to the unloading window.

Technological calculation of the mixers provides for the determination of the feed and power required for its drive, as well as the design parameters: the dimensions of the container and the operating elements and the speed of rotation of the auger. Theoretical submission

$$Q_T = \upsilon_0 F \rho \varphi_H, \tag{6.8}$$

where υ_o – is the axial velocity of the mass of the feed, m/s; F is the cross–sectional area of the screw, m^2; ρ – volumetric mass, kg/m^3; Φн is the filling factor of the screw section by the mass being transported (for horizontal screws φ_n = 0,3...0,4, for vertical screws – φ_n = 0,7...0,8).

For a horizontal continuous auger, this formula will have the form

$$Q_T = \frac{\pi (D^2 - d^2) S n_c \rho \varphi_H}{4}, \tag{6.9}$$

where D – and d are the diameters of the screw and its shaft, m; S – screw pitch, m; n_c – is the rotational speed, min^{-1}.

Replacing n with its value, we get:

$$Q_T = \frac{(D^2 - d^2) S \omega \rho \varphi_H}{8}. \tag{6.10}$$

When moving granular or pasty materials, the axial velocity will be much lower than the theoretical, since the material partially rotates together with the screw. This causes the stratification of part of the flow and the lag of some particles from the mass moving with the maximum possible speed:

$$\upsilon_{omax} = r\omega tg\ \alpha, \tag{6.11}$$

where α – is the angle of the helix, hail.

Minimum axial speed:

$$\upsilon_{0\min} = r\omega \sin\alpha (\cos\alpha - f \sin\alpha), \tag{6.12}$$

Where f is the coefficient of friction.
In this case, the minimum feed of the horizontal screw:

$$Q_{\min} = 0,25\pi (D^2 - d^2)\omega r_c \sin\alpha_c (\cos\alpha_c - f\sin\alpha_c). \tag{6.13}$$

The actual supply of single–screw mixers will lie within Q_{max} and Q_{min}. Feeding of twin and multi–screw mixers:

$$Q_Д = z\varphi_1 Q_{расч}, \tag{6.14}$$

where z – is the number of screws, pcs; φ_1 – is a coefficient that takes into account the overlapping of the free section of the screw by working parts; $Q_{расч}$ – the feed, found by the formulas (6.9) or (6.10).

Power required to drive screw mixers:
– horizontal

$$N_{шт} = 0,01kQL, \tag{6.15}$$

– vertical

$$N_{шт} = 0,01QL, \tag{6.16}$$

where k – is the reduced coefficient of resistance to movement of food along the auger casing (for grain and compound feed k = 1,2, salt k = 2.5, root crops k = 8…10).

When calculating a screw mixer of periodic action, the capacity of the mixer is determined by the formula:

$$Q = M\left(\frac{60}{T_ц}\right), \tag{6.17}$$

where M – is the mass of the portion loaded into the mixer, kg; $T_ц$ – cycle time, h:

$$T_ц = T_{загр} + T_{см} + T_{разгр}, \tag{6.18}$$

Reducing $T_ц$ increases the productivity. Usually $T_{см}$ = 5...8 min, $T_ц$ = 15...12 min.
The total volume of the mixing chamber is:

$$V = \frac{M}{\varphi \rho}, \tag{6.19}$$

where φ – is the volume utilization factor, φ = 0,8...0,85; ρ – is the density of the feed, kg/m^3.

Given the diameter D, we define H from

$$\frac{H}{Д} = 2 \ldots 2,5. \tag{6.20}$$

Diameter of the screw shaft d = (0,25...0, 35) D.
Screw in the process of work must repeatedly throw the mass up. On the basis of the multiplicity of the kn material and its mass transfer, we determine the required hourly productivity of the screw M:

$$Q = M k_n \left(\frac{60}{T_ц} \right), \tag{6.21}$$

where k_n – is the folding frequency of the material, k_n = 6...10.

Paddle mixers are periodic and continuous and are suitable both for the preparation of liquid, and thick mushy mixtures. In mixers, components are mixed with horizontal, vertical or inclined blades rotating around a vertical or horizontal shaft (fig. 6.1). Blades with their frontal surfaces are installed perpendicular or inclined to the direction of movement. The continuous mixer is filled with food by 30 %. To ensure a high quality mixing of ingredients, the speed of the blades should be chosen in such a way that individual layers of feed moving at different circumferential speeds are not thrown, but gradually poured one against the other.

The speed of rotation of the blades is determined from the condition that the centrifugal force imparted to the material by the rotating blade should be less than or equal to the force of gravity of the material itself,

$$m\omega^2 R \le mg\,, \tag{6.22}$$

where m – is the mass of the material displaced by the blade, kg; ω – angular velocity of blade rotation, s^{-1}; R – is the greatest radius of the blade, m.

The condition at which $m\omega^2 R = mg$, will correspond to the maximum permissible frequency of rotation of the mixer shaft.

Expressing the angular velocity through the shaft rotation frequency $\omega = \pi n/30$ and solving the equation with respect to n, we get:

$$n_{\max} = \frac{30}{\pi}\sqrt{\frac{g}{R}} = \frac{30}{\sqrt{R}}\,. \tag{6.23}$$

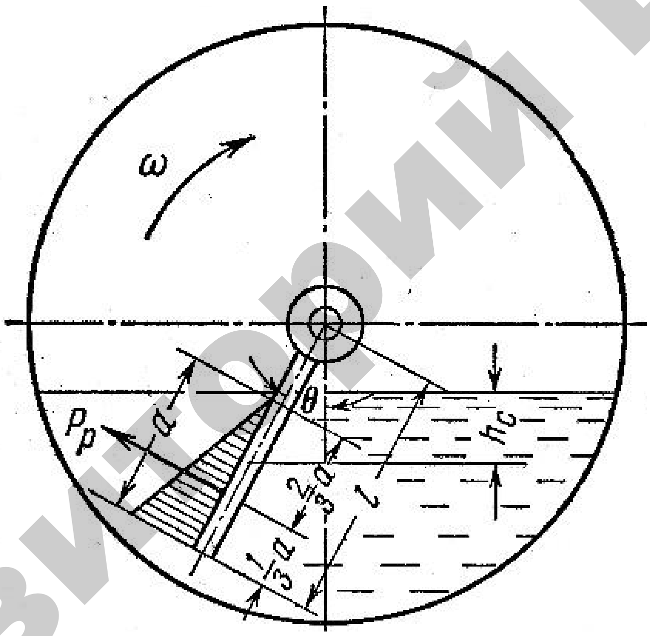


Fig. 6.1. Paddle mixer – digram

Productivity of a continuous mixer blade:

$$Q = 60\frac{\pi D^2}{4} S n \rho \varphi = 15\pi D^2 S n \rho \varphi. \tag{6.24}$$

where D – is the external diameter of the blades, m; S – blade pitch, m; n – frequency of rotation of the mixer shaft, s^{-1}; ρ is the volumetric mass of the product, kg/m^3; φ – feed rate, depending on the design of the blades and their location on the cotton wool (φ = 0,6...0,8).

The power required to drive a blade mixer

$$N_л = \frac{\left(P_p\, v_p\ + P_0 v_o\right) z}{1000},\qquad (6.25)$$

where z – is the number of working blades, pcs; P_p – is the radial component of the product resistance force acting on the blade, H; v_p – is the circumferential velocity of the point of application of the resultant feed resistance forces acting on the blade, m/s; P_o – is the axial component of the product resistance force acting on the blade, H; v_o – is the axial velocity of the point of application of the resultant, m/s.

The velocities can be determined from formulas

$$v_p = (2l\cos\theta + b)\omega\ ,\qquad (6.26)$$

$$v_0 = v_p \cos\alpha\ \sin\alpha\ ,\qquad (6.27)$$

where l – is the length of the blade, m; θ – is the angle of rotation of the blade, deg; b – blade width, m; ω – is the angular velocity of the blade, s^{-1}; α – is the angle of inclination of the blade to the plane of rotation, deg.

Forces are determined by formulas

$$P_p = 9{,}81\rho h_c F tg^2(45^\circ + \frac{\varphi}{2})(\cos\alpha + f\sin\alpha),\qquad (6.28)$$

$$P_0 = 9{,}81\rho h_c F tg^2(45^\circ - \frac{\varphi}{2})(\sin\alpha + f\cos\alpha),\qquad (6.29)$$

where h_c – is the average depth of immersion of this blade, equal to half of the greatest depth of immersion of the blade, m; F – is the area of the blade, m^2; φ – is the product friction angle along the blade; f – is the friction coefficient of the product along the blade.

Mass of feeds loaded into the tank of a batch mixer:

$$m_n = V\varphi\rho_{см}\ ,\qquad (6.30)$$

where V – is the geometric volume of the mixer, m^3; $\rho_{см}$ – bulk density of the mixture, kg/m^3; φ – is the filling factor (determined experimentally). For batch mixers, the filling ratio φ, depending on the type of mixes being prepared, ranges from 0,6 to 0,8.

For continuous mixers, the actual mixing capacity is a variable, numerically always smaller than the capacity of the mixer and the capacity occupied by the feed in the mixer.

6.4 Mixer feed designs

To prepare wet feed mixtures from the stalk forage and root crops, one–or two–shaft mixers are used for portioning (fig. 6.2).

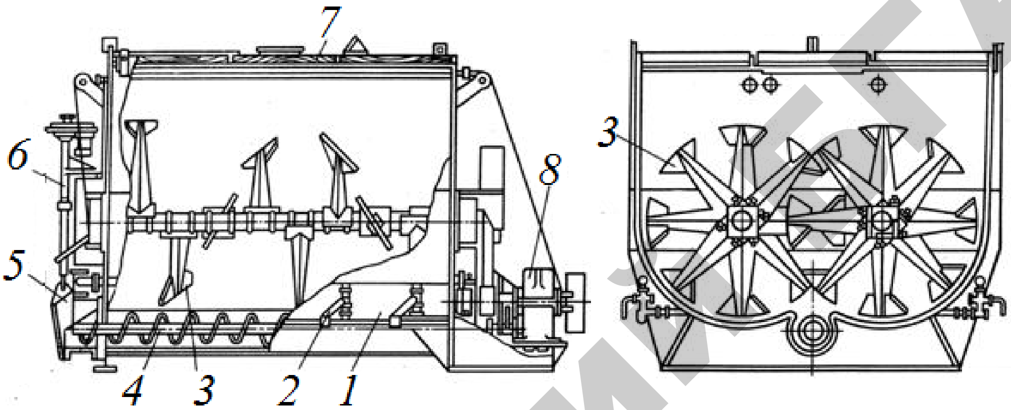


Fig. 6.2. Mixer feed, scheme:
1 – housing; 2 – steam distributor; 3 – paddle stirrer; 4 – discharge screw;
5 – discharge throat with wedge gate; 6 – control system; 7 – cover; 8 – drive

The mixer is intended for preparation of raw and steamed feed mixtures. The mixer consists of a casing 1, a steam distributor 2 with taps, two paddle mixers 3, an unloading screw 4, an unloading neck 5, lids 7 and a control system 6 with a gate valve and screw connection. Agitators and auger are driven by drive 8. Between the end walls of the body are welded three pipes, serving to supply water and solutions.

Two laminar mixers are installed inside the casing. Each consists of a shaft with 8 blades and bearing blocks fixed on the end walls of the shell. The blades are mounted on the shaft along a screw line at an angle of 45 ° and are fastened by ladder. The blades of the right mixer are mixed and the feed is directed towards the drive station, and the blades of the left mixer – towards the discharge neck, which ensures a good mixing of the feed. In the lower part of the mixer there is an auger with a diameter of 320 mm and a foot of 250 mm, feeding the mixed mass to the discharge nozzle. On top, the housing is hermetically sealed. In one of them there is a hatch with a slide gate and traction, and in the other – a manhole. On the side of the cover on the bracket there is a limit switch that disconnects the mixer mechanism when the lid is opened. The system for supplying steam to the mixer consists of a manifold with a manometer and two distribution pipes, which are connected to the steam pipes by five couplings. The steam supply is controlled by a switch. To ensure that the feed does not enter the distribution pipes, the cranes must be closed after the end of the steaming.

The first is the feed, which must be steamed. The crushed coarse feed is loaded with simultaneous moistening. Me shacks are included not later than when filling 1/3 of the technological volume and continue to load. At the same time, the filling factor of the mixer tank should not exceed 0,6...0,7 for thick mixtures with straw included and 0,8 for fodder with humidity over 70 %. Then tightly close the manhole covers, open the valve on the steam line and the coupling valves on the distribution pipes. The pressure of the supplied steam and the temperature of the mixture are monitored by a manometer and a thermometer. On average, the time of steaming in the mixer is 1 to 3 hours. At the end of steaming, it is necessary to close the couplings and valves on the steam pipe and for 40...60 minutes to withstand the digestion food. After that, add water to cool the feed and load the other components. When preparing feed mixtures without fuming, all components entering the mixture can be fed simultaneously. Stir food for 10 minutes, and when enriched with carbamide and other chemical solutions – 15 minutes.

Chapter 7
MECHANIZATION OF FOOD SEALING

7.1 Classification of feed compaction methods

Two methods of compaction of feeds are known for their storage for storage – granulation and briquetting. Granules are compacted to a density of 800...1300 kg/m3 in cylindrical or shaped pieces with a thickness or diameter of up to 25 mm feed components or mixtures crushed into flour. The diameter of granules for chickens aged 1...7 days should be 1...2 mm, 7...30 days – 2,2 mm, more than 30 days – 3 mm, for adult poultry – 4...6 mm; for pigs–weaners – 8 mm, young pigs more than 4 months – 10 mm; For sheep, calves – 5...7 mm; For cattle – 14...20 mm.

Briquettes are compacted coarse forages (grass or straw cutting) and feed mixtures, including coarse feeds, the particle size of which is 20...70 mm, formed into cylindrical, up to 65 mm diameter, or another shape with the largest dimensions of 80 mm, density 500...900 kg/m3. Diameter of briquettes for cattle 30...65 mm, the size of briquettes of rectangular shape 60x50 mm.

The feed mixture for compaction must be homogeneous by at least 90 %. The amount of non–pressed mass should not exceed 6 %, the allowable heating during the pressing process should not exceed 90 °C for granules and 70 °C for briquettes. The coefficient of grinding of cutting into flour for briquetting should not exceed 20 %; Loss of carotene in the process of pressing 5 %, loss of product by weight 1 %. The required quality of pellets and briquettes is also determined by their density, strength and crumbliness, which depend on the moisture content of the material, the granulometrical or fractional composition, the temperature and pressure of pressing

Crumbling characterizes the degree of cohesion of particles that make up granules or briquettes and is determined from expression

$$K = \frac{m}{M}100\ \% \qquad (7.1)$$

where m – is the mass of crumb that has passed away after testing granules or briquettes for crumbling, t; M – is the initial mass of food samples.

For the normal course of the process of granulation of herbal flour or mixed fodder mixtures, the optimum moisture content is 15...16 %, temperatures 60...70 °C. At a moisture content of more than 16 %, the particles become elastic, they are less compressed. Surface moisture contributes to a better approximation of particles and their compaction, especially with steam conditioning or with the presence of devices for active redistribution of the humidifier.

The process of granulation will be more effective for fine grinding, since the friction coefficients are smaller than for coarse grinding. The most favorable conditions for granulation of mixed fodders are created by processing them with steam with a pressure of 0,25...0,4 MPa (flow rate 0,4…0,5 kg per 1 kg of feed). The strength of briquettes depends on the duration of the stress in the load zone. When the briquetted mass is heated to 70...80 °C, the relaxation period is almost halved.

The pellets can be prepared by pelletizing mealy forages in beads at a moisture content of 30 to 35 %. However, since the components must be finely grinded, and the resulting granules must be dried to reduce their moisture content to 12...14 %, this method is not widely used.

7.2 Types of working elements for compaction of feed materials

To seal the feeds, stove–and–ring and ring presses are most widely used. The stamping tools work on a batch feed basis. They can be closed (fig. 7.1, a) or open (fig. 7.1, b, c) by pressing chambers.

In the open chamber, the back pressure is created by friction of the pressed material against the walls, while the compaction and ejection are performed in one stroke of the stamp. In a closed chamber, these operations are performed separately, and the counter pressure is created by a fixed stop. Working bodies with a closed chamber are less energy–intensive than with an open chamber, since work on pushing a row of briquettes at the maximum stamp effort is excluded here. Stamping tools act on the principle of batch feed. However, in enclosed chambers, an equal supply of material is required for each stroke of the stamp, which is an extremely difficult task. The second disadvantage of closed chambers is the small holding of the briquette under pressure.

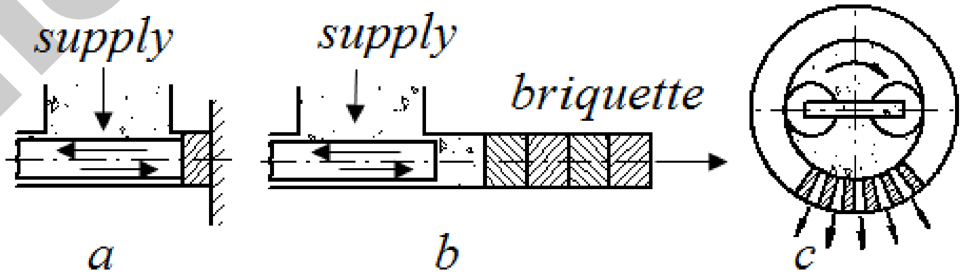


Fig. 7.1. Types of working elements for compaction of feed materials:
а – a stamp with a closed chamber; б – stamp with an open chamber;
в – ring with an open chamber

The advantage of stamp presses is low energy consumption, obtaining large diameter briquettes, which is important for reducing the surface of the feed where the oxidation process develops. In addition, the device of the matrix channel allows you to adjust the pressing pressure, and, consequently, the density and strength of the briquettes. Stamp presses are more versatile, they can be briquetted with a variety of raw materials. Ring presses (fig. 6.1, c) have annular matrices with perimeter press channels, through which the feed is pushed. The advantage of the circular working bodies is the continuity of the technological process. However, they are relatively energy–intensive (up to 100 kJ/kg) and require careful preparation of the material before granulation and briquetting – homogeneous grinding and uniform humidity. Most often, presses with a vertically rotating die are used for compaction in pellets, for compacting into briquettes, with rotating rollers. Other working organs: screw, roll, roll – for some reason or other have not found application in the practice of feed preparation.

The process of forming a briquette from loose fodder in an open chamber includes several stages. When pressure is placed on the stamp (fig. 7.2), the feed particles approach each other, the voids between them decrease, and the air is forced out through the gap between the stamp and the wall of the chamber. The approach of particles and the compaction of material increase with increasing force on the stamp.

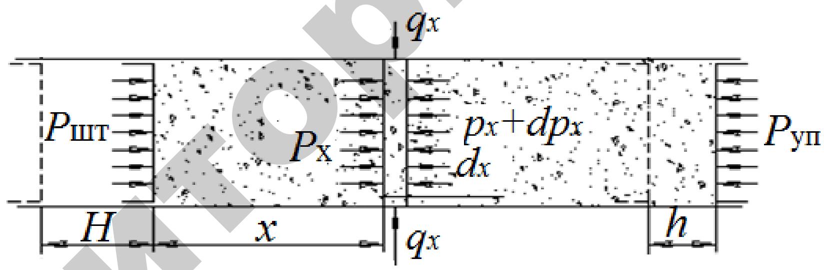


Fig. 7.2. Scheme of pressing the food in an enclosed chamber

As the particles approach each other, their intermolecular forces are produced. The manifestation of cohesion forces is the stronger, the larger the contact surface of the particles, i.e. the more the particles are brought together. As a result of these forces, consolidation of the pressed portion and the formation of a monolithic start–up of the briquette are taking place. The actual process of pressing the feed material in the chamber is accompanied by friction between it and the walls of the press chamber. In connection with this, the stamp must overcome additional efforts. Consequently, the total pressure on the stamp $P_{шт}$ is

$$P_{шт}=P_x+P_F, \tag{7.2}$$

where P_F is the pressure due to the external friction of the deformable material; P_x – is the axial pressure on the material.

The pressure P_F is determined by the relation:

$$P_F = \frac{F}{S}, \tag{7.3}$$

where F – is the frictional force at pressing, H; S – is the cross–sectional area of the press chamber, m^2.

Friction, in addition to additional energy costs, leads to an uneven briquette density. This is due to a drop in pressure in the material being pressed as it moves away from the stamp. The relationship between lateral and axial pressure is adopted in the form of a simplified linear relationship:

$$q_x = \xi P_x \tag{7.4}$$

where q_x – is the lateral pressure on the feed; ξ – is the lateral expansion coefficient.

The axial pressure is determined from the expression:

$$P_x = P_{шт}\ exp(-f\xi \frac{l}{S} x), \tag{7.5}$$

where f – is the coefficient of friction; l – perimeter of the chamber cross–section; x – is the thickness of the material to be compressed.

Accordingly, the pressure on the stop will be:

$$P_x = P_{шт}\ exp(-f\xi \frac{l}{S} x_{en}). \tag{7.6}$$

Thus, the greatest density of the briquette is reached at the stamp, the smallest at the stop, which leads to great crumbling of the briquettes or their destruction after removal from the chamber. To reduce this phenomenon, it is possible to tamper with the stamp at the end of its working stroke. In this case, the stresses in the briquette would be relaxed and it would expand less after the load was removed. But this is due to the loss of productivity by the briquette press. The specific costs of the working process in the open chamber can be written in the form of the following balance:

$$A = A_{сж} - A_{ур} \tag{7.7}$$

where $A_{сж}$ – is the energy expended on the compression of the material; $A_{ур}$ – is the work of elastic expansion of a compressed material (returning some of the energy to the press mechanism at the beginning of the back stroke of the stamp).

The energy costs in the open chamber, compared to the closed one, are relatively small. However, other shortcomings of the closed chamber are constructively not overcome, and presses with such a camera have not received distribution.

The patterns of pressing in a closed chamber. The working process in an open chamber consists of two stages (Figure 7.3):

1. Compress another portion of the source material. This stage is no different from the compression in a closed chamber; only here the emphasis is served earlier with pressed briquettes;

2. Pushing the entire series of previously compacted samples and pushing one of them out of the chamber.

In the open chamber, the counter pressure is created at the pushing stage – only due to frictional forces.

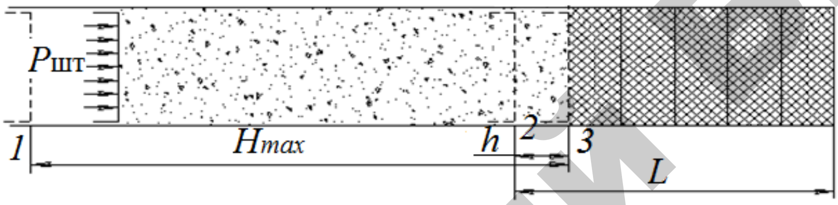


Fig. 7.3. The scheme of pressing the food in an open chamber

A feature of the open press chamber operation is a large expenditure of energy on the process of pushing the briquette tape, which exceed the cost of briquette formation by 1.5...2.0 times. The total energy balance for the working process of a stamp machine with an open chamber can be written in the form:

$$A = A_{сж} + A_{пр} - A_{у.р} \tag{7.8}$$

where $A_{пр}$ – is the energy expended on pushing the briquette tape.

Since in the open pressing chamber the emphasis in forming the next briquette is previously compacted briquettes held in the chamber by frictional forces, the task of determining the length L of the pushing chamber is actual.

The average lateral pressure can be determined by the formula:

$$q = \frac{P_{шт}^{max}\xi}{2} \tag{7.9}$$

The frictional force that occurs between the briquette tape and the walls of the chamber during the pushing can be determined by the expression:

$$F = fqLl = \frac{1}{2} f \xi P_{шт}^{max} Ll . \tag{7.10}$$

At the stage of pushing the feed with a stamp, the frictional force is overcome:

$$F = P_{шт}^{max} S \tag{7.11}$$

Equating the values of the frictional force, we have:

$$S = \frac{1}{2} f \xi L l \tag{7.12}$$

Whence the length of the push chamber:

$$L = \frac{2S}{f \xi l} \tag{7.13}$$

For a chamber of circular cross–section, we have $l = \pi d$, $S = \frac{\pi d^2}{4}$. Then, the formula (7.13) becomes simpler:

$$L = \frac{d}{2 f \xi} \tag{7.14}$$

The length of the chamber should be increased (0.3 m or more) for large diameters of briquettes. To form granules with a diameter of about 1 cm, the length of the spinner is 5...6 cm. In the open pressing chamber, the samples are placed under a load for some time, which promotes relaxation of stresses and the production of quality briquettes and granules. The average speed of pushing compressed samples should not exceed the value:

$$\upsilon = \frac{L}{T_p} \tag{7.15}$$

where T_p – is the relaxation time (according to SV Melnikov: for coarse–grained feed T_p = 23...25 s, for grass meal T_p = 12...17 s.)

The working bodies of the ring press are a matrix–roll pair, the design parameters of which can be different depending on the purpose (granulation and briquetting) and the kind of material to be compacted.

Regardless of whether the matrix or the carrier of the rolls rotates, the pressed material is tightened into the wedge gap between the matrix and the roll and compacted as the gap is reduced (fig. 7.4).

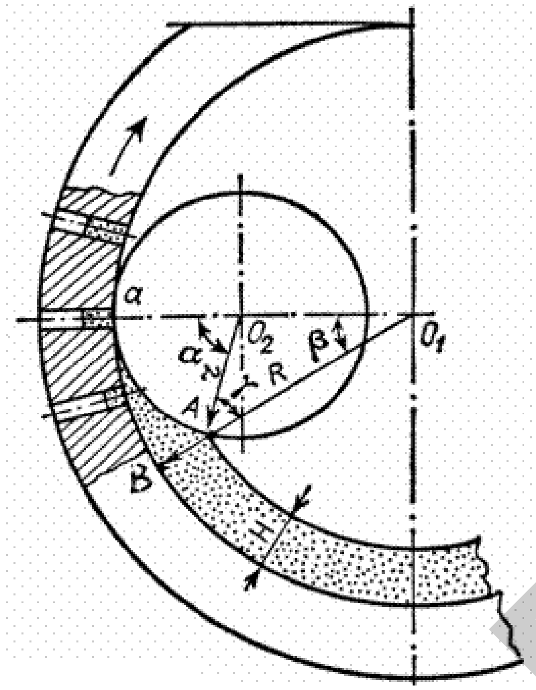


Fig. 7.4. Scheme for the calculation of a ring press with a rotating matrix

At the moment when the density of the material in the wedge gap becomes approximately equal to the density of the pellets or briquettes in the die channel spinnerets, the material begins to be pressed into the spinnerets. In this case, the granules or briquettes move in the spinnerets and are squeezed out of them.

After passing through the spinneret, the minimum section of the wedge gap is elastically expanded in the densified layer in the spinnerets. The briquette or pellet is formed by repeatedly pressing into the spinneret individual portions of the dense food. Let's define theoretically parameters of the ring press. From the circuit in fig. 6.3 that the side A_1O of the triangle AO_1O_2 can be expressed by the cosine theorem:

$$AO_1 = \sqrt{r^2 + (R-r)^2 - 2r(R-r)cos(180-\alpha)}, \quad (7.16)$$

where r – is the radius of the carrier, m; R – is the radius of the matrix, m; α – is the angle of pressing, deg.

From here

$$R - H = \sqrt{r^2 + (R-r)^2 - 2r(R-r)cos\alpha}, \quad (7.17)$$

where H – is the thickness of the layer of the initial material on the matrix, which is tightened into the annular gap (the remaining portions of the feed are shifted by the roller).

The height of the layer can be found from formula

$$H = R - \sqrt{r^2 + (R-r)^2 - 2r(R-r)\cos\alpha}\,. \tag{7.18}$$

In this formula, the pressing angle α is unknown. To find it, consider the triangle AO_1O_2, from which it follows that $(\pi - \alpha) + \beta + \gamma = \pi$, where β – is the angle of the capture arc in the roll rotation zone, deg; γ – is the angle of capture of the material, deg. $\gamma = \alpha\left[1-(r/R)\right]$.

In order that the roll can capture a layer of bulk material and then compress, it is necessary that the angle γ does not exceed the angle φ of friction of the material with the surface of the roll. Consequently, the following condition must be satisfied:

$$\gamma \le \varphi\,. \tag{7.19}$$

Then define the angle of pressing:

$$\alpha \le \frac{\varphi}{\left[1-(r/R)\right]}\,. \tag{7.20}$$

The relationship between the radius of the roller and the matrix is strictly limited: – for two rolls, r/R = 0,42…0,45; – at three r/R=0,40…0,42.

The productivity of the press can be calculated by the formula:

$$Q = 2\pi\left(R - \frac{H}{2}\right)Hz\omega\delta\rho\,, \tag{7.21}$$

where z – is the number of rollers; ω – is the angular velocity of the matrix or the driver of the rolls, s^{-1}; δ – coefficient, taking into account slipping of the working elements and perforation of the matrix.

The ratio of the total area of the spinnerets to the total internal surface of the die is:

– for granulating matrices – 0,4...0,5;

– for briquetting – 0,70...0,75.

The minimum frequency of rotation of the matrix is determined taking into account the best conditions for the distribution of material on the inner surface of the matrix, for example, that it is retained on the surface of the vertical matrix. Podkolzin writes this condition in the form:

$$\frac{\omega^2 R}{g} \geq \frac{1}{sin\varphi} \tag{7.22}$$

From where

$$\omega_{min} = \sqrt{\frac{g}{Rsin\varphi}} \tag{7.23}$$

The maximum angular velocity is limited in connection with the possible destruction of hot granules or briquettes from tensile stresses arising from the action of centrifugal forces. Granules and briquettes can be peeled off when they reach the required length in relation to their transverse dimension d. For pellets according to zootechnical norms, we have a length

$$l = \theta d \tag{7.24}$$

where θ – is the ratio of the length of the granules to its diameter (θ =1,5...2,0).

Hence we have the formula for calculating the maximum rotation speed of the matrix:

$$\omega_{max} = \sqrt{\frac{\sigma}{(R+L)\theta d \rho}}\,, \tag{7.25}$$

where σ – allowable stresses of rupture (for granules from grass meal σ =13...17 kPa); L – is the length of the spinneret, m; ρ – is the density of the granule, kg/m^3.

The energy costs for the pressing process in the ring press can be represented in the form of the following components:

$$A=A_{сж}+A_{пр}+A_{ст}-A_{ур}, \tag{6.26}$$

where $A_{ст}$ – is the work of pushing the material from the bridges between the die dies.

In connection with the availability of energy consumption, the energy consumption of annular briquette presses is 1,5 times higher than that of stamps with an open chamber. Another disadvantage of the ring presses is the grinding (chopping) of the stalk feed, which is harmful to ruminant animals.

At the same time, ring presses are indispensable in the production of pellets of herbal flour, mixed fodder and finely ground feed mixtures.

7.3 Equipment for pelletizing and briquetting of feeds

To obtain granules by a dry method, the greatest distribution was obtained by roller presses with a ring matrix.

In the equipment, the herb flour from the drying units comes through the intake 1 (fig. 7.5) to the cyclone 2 and out of it into the feed hopper 4.

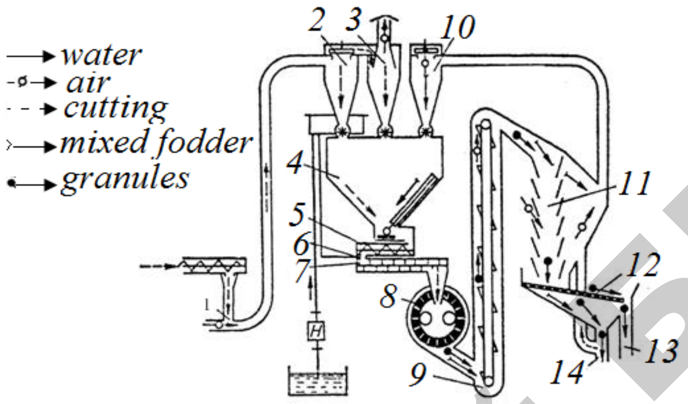


Fig. 7.5. Scheme of the technological process of operation of the roller press with a ring matrix

The cyclone 3 serves to trap the dust carried away with the airflow from the cyclone 2. In order for the flour to be uniformly fed to the granulation, And no vaults were formed in the bunker, a planetary stirrer was installed in it, driven by the shaft of the screw feeder 5, which regulates the amount of flour fed to the granulation by changing the speed of rotation, since it is driven by an autonomous electrode Through the V–belt variator. When exiting the dispenser, the flour is moistened with water coming through the water injection system into the nebulizer 6. Antioxidants and binders can be introduced together with water.

Uniformity of humidification and homogeneity of mixing is provided by a high–speed bladed air–mixer 7, which also has an automatic drive from an electric motor.

From the mixer, the conditioned grass meal is gravity fed to the press receptacle 8, from which the guide vanes are fed to the inner surface of the matrix. The monoliths of the pressed material pressed out by the pressing rolls from the working holes of the matrix meet with fixed knives and break off, forming granules.

The granules exiting the press have a high temperature (75…85 °C) and are unstable. They flow through the tray to the noria 9, which lifts them and sends them to the cooling column 11.

The air being sucked through the column by the cyclone fan 10 cools the granules that enter the screening screen 12 and from there to the sampler 13, from where they are sent for bagging or transported To the place of storage in bulk. Some of the flour may not be granulated (up to 5...7 %). Some of the hot granules can crumble into crumbs. This trifle passes under a sieve, through the sampler 14 air is transported to the cyclone 10 and sent for repeated granulation.

The main part in the set of equipment is a granulator, which consists of a screw feeder 5, a blade mixer 7, a press 8, a reducer of a dispenser drive. The basis of the granulator is a press (fig. 7.6), consisting of a reducer and a pressing unit with a vertical annular matrix and two passive rolls.

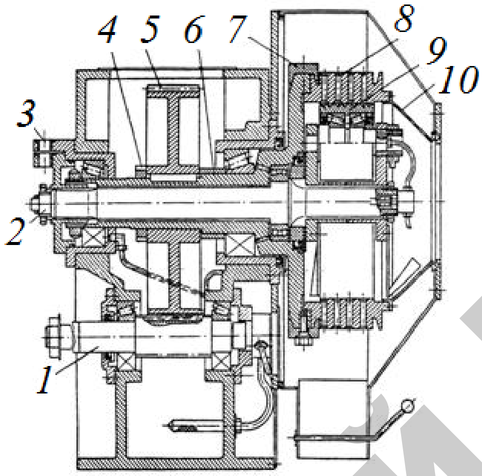


Fig. 7.6. Scheme of the press–granulator:
1 – pinion shaft; 2 – axis; 3 – shear pin; 4 – the nut; 5 – a cogwheel; 6 – main shaft; 7 – fastening segment; 8 – matrix; 9 – pressing rollers; 10 – flour receiver

The electric motor through the elastic coupling is connected to the pinion shaft 1. The gear wheel 5 is fixed in constant engagement with the pinion shaft fixed rigidly with a key and nut 4 on the hollow main shaft 6. To the flange of the main shaft, the segments 7 are fixed with a die 8 which is fixed by the tongues from turning .

A conical receiver 10 is attached to the outer end of the rotating matrix, forming, together with its internal cavity, a compression chamber. Inside the main shaft there is an axle 2, at one end of it there are two plates, between which on the eccentric axes the pressing rollers 9 are mounted.

The clearance between the working surfaces of rollers and the matrix is 0.3–0.5 mm, it is regulated by means of special levers and bolts On the front plate of the rollers. At the other end of the axis 2, a flange is placed on the splines, which is connected through a shear pin 3 to the rear roller.

Under normal load, the shear pin keeps the axle from rotating, and the press rollers rotate only around their fixed axes. If the press is excessively overloaded with the granulated mass, or if a foreign object gets into the gap between the rollers and the matrix, the shaft will jam and the torque from the die will be transferred to the axis 2 and through it to the shear pin 3. After cutting the pin, the flange will begin to turn and the pressure on the pushbutton of the final switch that will detach all the electric motors from the mains and stop the press, preventing it from breaking.

Chapter 8
MECHANIZATION OF FEEDING

8.1 Zootechnical requirements for feeder devices

Proper organization of distribution of feed to animals is important. In terms of labor intensity, it is 30…40 % of the total labor costs for animal care. The following zoo–technical requirements are imposed on feed–distributing devices:

– deviation of the dose by weight per head for cattle is: stalked forages 10 %; Root crops of 15 %; Mixed fodder and concentrated feed 5 %; Mineral additives 5 %;

– the duration of the operation of distributing feed in one room should not exceed 30 minutes when using mobile means and 20 minutes – when distributed by stationary means;

– feed distributors must be universal with regard to the possibility of issuing all types of feed;

– to have high productivity and the ability to regulate the rate of issuance for 1 head from the minimum to the maximum, depending on the diet taken;

– do not create excessive noise in the room;

– mechanically cleaned of residues of feed;

– to be reliable in work.

By the nature of use, feeder–driving machines can be stationary and mobile.

8.2 The device, process of operation and calculation of the parameters of a stationary belt feeder

Stationary distributors are installations mounted in one or more interlocked premises and distributing feed to the animals along the front of the feeding. The belt conveyor–distributor of feeds (fig. 8.1) provides distribution of all types of feeds (except for liquid ones) when servicing cattle and sheep. The feed dispenser includes feeders 6, a drive and tension station, an operating element (traction chain 8 and belt 7), a loading hopper, electrical equipment. Stretching station with a loading hopper is located outside the end wall of the barn, in a tambour with a through passage for a mobile feed dispenser.

The working organ of the feed distributor moves the feed along the groove. It is a closed loop consisting of a tape and a chain that are connected by a safety device. The limit switches stop the operating element in the extreme travel positions with the help of abutments with skis. The feed chute, along with the guide for the working organ, simultaneously serves as animal feeders, to which the carbide holder brackets are attached. The drive station drives the feeder feeder and consists of a frame, a reducer 12, an electric motor 13, drive sprockets 9, 10, 11.

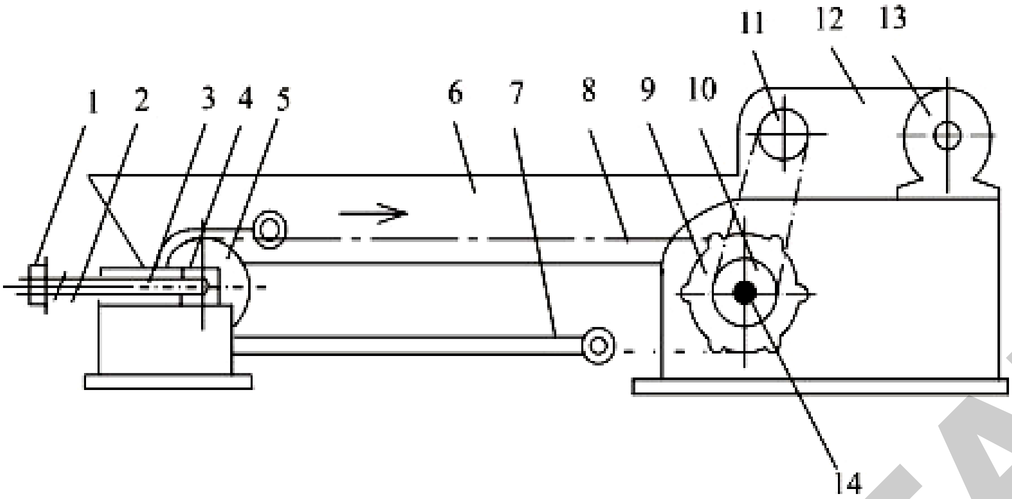


Fig. 8.1. Technological scheme of belt feeder:
1 – nut; 2 – tension station; 3 – tension screw; 4 – slider; 5 – driven drum;
6 – feeder; 7 – tape; 8 – chain; 9, 10, 11 – gears; 12 – reducer;
13 – electric motor; 14 – drive shaft

The productivity of a belt conveyor can be determined by the formula

$$Q_л = F_л V_л \rho , \qquad (8.1)$$

where $F_л$– is the cross–sectional area of the feed on the belt during its movement, m^2; $V_л$– speed of the belt, m/s; ρ – density of feed, kg/m^3.

In fig. 8.2 it can be seen that the total cross–sectional area of the feed on the belt can be expressed as the sum:

$$F_л = F_1 + F_2. \qquad (8.2)$$

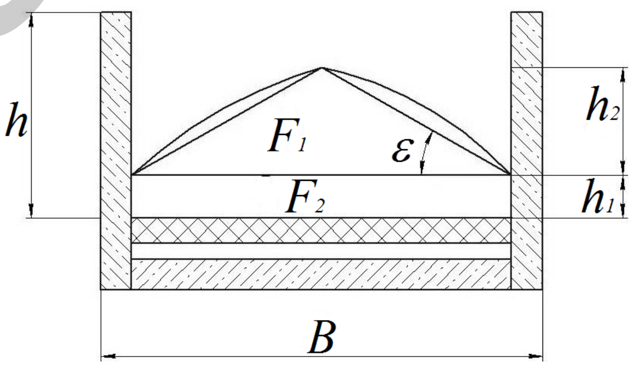


Fig. 8.2. Calculation of the parameters of belt feeder

The area F_1 can, with some assumption, be represented as the area of an isosceles triangle whose lateral walls are inclined to the base at an angle of the natural slope of the transported food in motion:

$$F_1 = \frac{h_2 B}{2}. \tag{8.3}$$

Wherein:
$$h_2 = \frac{B}{2} tg\varepsilon. \tag{8.4}$$

The area F_2 with some assumption can be expressed as the area of the rectangle:

$$F_2 = h_1 B. \tag{8.5}$$

Taking into account the above equations, the total cross–sectional area of the feed on the belt is:

$$F_л = B\left(0,25 B tg\varepsilon + h_1\right) \tag{8.6}$$

Substituting expression (8.6) into formula (8.1), we obtain an equation for determining the productivity (supply) of a flat ribbon located in the feeder chute:

$$Q_л = B V_л \rho \left(0,25 B tg\varepsilon + h_1\right). \tag{8.7}$$

8.3 The device and calculation of the main indicators of stationary chain–scraping feeder devices

Chain–scraper conveyors are distributed on farms for the distribution of dry concentrated, coarse and juicy forages, wet sticks. Design simplicity, their ability to feed food in any of two directions, and (if necessary) – simultaneously in both directions are advantages of these conveyors. The main disadvantages of chain–scraper conveyors are abrasion of feed during transportation, great resistance from friction of feed slip along the walls of the trough, rapid wear of the chain and trough. On farms, a stationary scraper feeder (fig. 8.3) is used for transportation, group dosing and distribution of feed products along the feeding front in the premises for keeping young cattle. It is a horizontal chain–scraper conveyor 2 of an open type, mounted on the bottom of the trough 1, consisting of two parallel and circular channels. The feed is loaded near the drive and moved by scrapers along the feeder trough until it is evenly filled all along the length. The plant capacity is 15 t/h.

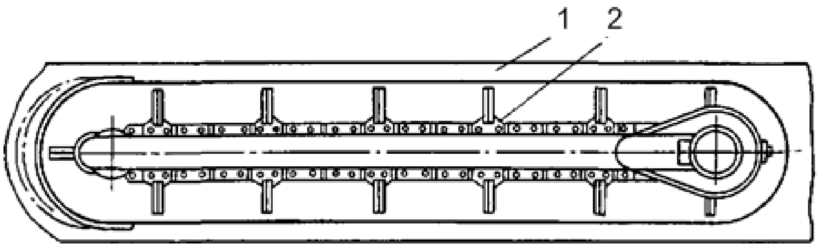


Fig. 8.3. Stationary scraper feed distributor**:**
1 – feeder; 2 – chain–scraper conveyor

When a chain–scraper feed–forwarding conveyor runs, each scraper conveys a portion of the feed, which, with some approximation, can be taken as a prism. The latter has in the longitudinal section a shape close to an unequal trapezoid (fig. 8.4). Then the volume of food portion:

$$V_л = \frac{l + l_1}{2} h_c b_c \, , \tag{8.8}$$

where h_c and b_c – height and width of the scraper, m.

Then

$$l = l_1 + h_c tg\, \varepsilon \, , \tag{8.9}$$

where ε– angle of collapse (shedding) of the transported norm, deg.

Substituting the value of l in equation (8.8), we obtain:

$$V_л = \left(l_1 + \frac{h_c ctg\, \varepsilon}{2} \right) h_c b_c \, . \tag{8.10}$$

In calculations, the value of the angle ε is taken to be 0,7...0,8 of the angle of the natural slope of the feed at rest.

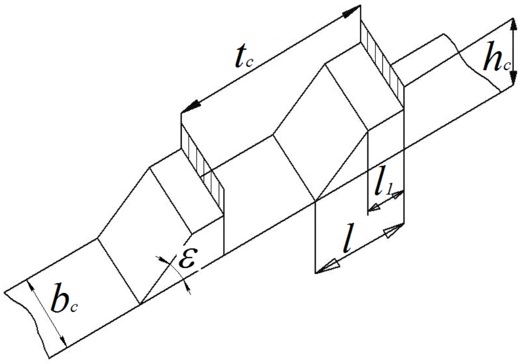


Fig. 8.4. Calculation of the parameters of a chain–scraping feeder

Productivity of chain–scraper conveyors:

$$Q_л = \frac{V_л v_ц \rho}{l_c}, \tag{8.11}$$

where $v_ц$ – is the speed of the chain with scrapers, m/s ($v_ц$ = 0,25...0,5 m/s); ρ – density of feed, kg/m^3; l_c – distance between scrapers, m.

The value of lc is recommended to choose more than the length l of the portion of the transported food. Accept l_c= (6…8)/h_c. Substituting in (8.11) the expression (8.10) and replacing $l_1 = k_1 h_c$ and $b_c = k_2 h_c$ (k_1 and k_2 – with the proportionality coefficients), we obtain:

$$Q_л = \left(k_1 + \frac{ctg\,\varepsilon}{2}\right) k_2 \rho \frac{v_ц}{l_c} h_c^3, \tag{8.12}$$

From where the calculated height of the scraper:

$$h_c = \sqrt[3]{\frac{Q_л}{\left(k_1 + \frac{ctg\,\varepsilon}{2}\right) k_2 \rho \frac{v_ц}{l_c}}}. \tag{8.13}$$

In some cases, when calculations are sufficient to calculate the productivity and parameters of the chain–scraper conveyor, one can use the simplified formula

$$Q_л = bhv_ц \rho \psi k_n. \tag{8.14}$$

Taking into account that $b = k_2 h$, we can write:

$$h = \sqrt{\frac{Q_л}{k_2 k_n v_ц \rho \psi}},$$

$$b = \sqrt{\frac{k_2 Q_л}{k_n v_ц \rho \psi}}, \tag{8.15}$$

where h and b – height and width of the trough (according to internal measurement), m; k_n – is a coefficient that takes into account the effect of the elevation angle β of the conveyor, k_n = 1– (0,01...0,02) β; ψ – is the feed rate of the trough (when the trough is open ψ <0.5, with ψ = 0.9 closed).

Required power for driving a chain–scraper conveyor:

$$N = \frac{g}{\eta}\left(f_1 M L v_ц \pm M H v_ц + 2\chi_1 M_Ц L v_ц \right), \tag{8.16}$$

where g – is the acceleration due to gravity, m/c^2; η – efficiency Transfer; f_1 – coefficient of feed friction against the trough; M – load per 1 m of conveyor length, kg/m, $M = Q/v_n$; L – length of the conveyor, m; H – height of the forage lift, m; χ_1 – coefficient of resistance of the moving parts of the conveyor (for roller chains χ_1 = 0,15...0,2; for rollers χ_1 = 0,1...0,12); $M_ц$ – weight of one meter of chain with scrapers, kg/m.

In equation (8.16), the first term expresses the required power to move the feed horizontally, the second – the power for transporting the feed along the vertical (upwards – the plus sign, downwards – the minus sign), the third – the idle power of the conveyor.

Often, chain–scraper conveyors work under conditions such that their scrapers are completely immersed in the transported feed product. If the height of the feed exceeds the height of the scrapers, the feed dispenser acts as a conveyor with submerged scrapers. Its performance:

$$Q_л = k_0 k_y k_r k_n h b \rho \psi, \tag{8.17}$$

where k_0 – is the speed coefficient taking into account the lag of the transported feed from the chain with scrapers (for grain crushed forages k_0 = 0,45...0,8, for unmilled k_0 =(0,6...0,9); k_y – the coefficient of compaction of the material being transported in the trough under the action of scrapers (for grain crushed feed k_y = 1,05...1,1); k_r – geometric coefficient of productivity, taking into account the loss of the useful volume of the trough occupied by the chain and scrapers (k_r = 0,95); k_n – factor, taking into account the angle of the conveyor lift (k_n = 1– (0,01...0,02) M).

8.4 The device and calculation of the basic indicators of stationary pipeline fodder devices

On the farms for the delivery and distribution of feed pipeline devices are used. There are several ways to feed the pipes:

– with the help of a fan;

– the pump;

– compressed air or gravity dilution.

Pipeline devices in which feeds are supplied by the head of air created by the fan are used to transport and distribute coarse and dry concentrated feeds. The disadvantage of this type of delivery and distribution of feeds is limited distribution due to the energy intensity of the process and the undesirable separation of feed resulting from its feeding through the pipe.

Economical is the method of feeding feeds through the pipes from the feeding station to the point of delivery using compressed air, pumps or vacuum. Such methods of transportation are used only for the supply of fodder by liquefied water through pipes. The mass ratio of dry food products and water is usually from 1:1 to 1:2.5, which corresponds to the moisture content of the semi–liquid feed of 65...75 % and higher.

The installation for transporting and distributing semi–liquid feeds through compressed air pipes contains the following main assembly units: compressors, receivers, blowing boilers, main feeder with taps and a device for automatically changing the feed direction, feed hoppers, feeding lines with a set of bilateral feeders.

The calculation of the installation is carried out in this order. First, you select the diameter of the feed line, which can be calculated by a joint analysis of the hydraulic resistance when moving the feed mix and the capital costs for the installation of the feed line. For a relatively small length of feeder pipe (150...200 m), when choosing its diameter, the head loss reduction factor should be considered the main factor, as the cost of the feed line is low, and the maintenance costs do not depend on its diameter.

Total hydraulic losses in the feed line:

$$\sum \Delta h = \Delta h_л + \Delta h_м + \Delta h_Г , \tag{8.18}$$

where $\Delta h_л = \Delta h_Г \sum l_п$ – linear head losses on straight sections of the feed–water along the entire length, kN/m^2; $\Delta h_м = \Delta h_Г \sum l_м$ – head loss in local resistance, kN/m^2; $\Delta h_Г = H \dfrac{\rho}{\rho_в}$ – geodetic head losses, ie, head loss to overcome the height difference between the beginning and end of the main feed, here H is the geometric height of the feed, m; ρ and $\rho_в$ – are the density of the feed mix and water, kg/m^3), kN/m^2.

Linear pressure losses are determined using the well–known Darcy–Weisbach formula:

$$\Delta h_л = 9{,}81\lambda \frac{l_к v_{ср}^2}{2 g d_к}, \tag{8.19}$$

where λ – is the coefficient of hydraulic resistance; $l_к$ – is the length of the feed channel, m; v_{cp} – is the feed flow rate in the pipeline, m/s; g – acceleration due to gravity, m/s^2; $d_к$ – diameter of the feed duct, m.

The coefficient of hydraulic resistance for the structural regime of motion:

$$\lambda = \frac{64}{Re}, \tag{8.20}$$

where Re – is the generalized Reynolds criterion.

Loss of pressure in local resistances is within 10 % linear, ie:

$$\Delta h_м = 0{,}1\Delta h_л. \tag{8.21}$$

In order to ensure the operation of a pneumatic feed handling machine in accordance with the time required by the order of the working day on the farm, it is necessary that the actual productivity (supply) of the installation be equal to or greater than its required production capacity, i.e., $Q_ф \geq Q_п$.

Required installation capacity:

$$Q_п = \frac{q_H m_i}{\rho\, t_{ус}}, \tag{8.22}$$

where m_i – is the number of animals served, pcs; q_H – average daily amount of required feed for one animal, kg; $t_{ус}$ – the duration of the installation per day according to the daily routine, h.

The actual capacity of the installation with a single blowing boiler:

$$Q_ф = \frac{60W_{пк}}{t_3 + t_П + t_0}, \tag{8.23}$$

where $W_{пк}$ – useful capacity of a blowing boiler (take 0,7...0,8 from the full capacity of the boiler Кк), m^3; t_3 – loading time of the boiler, hour; $t_П$ – the time of feeding a portion of feed in the boiler to the feed hopper, hour; t_0 – loss of time for preparatory–final operations (opening and closing of cranes, valves, creating pressure in the receiver and boiler), hour.

The capacity of the blowing boiler must be equal to the capacity of the feed hopper:

$$W_{кб} = \frac{m_i q_H}{a_к \rho \psi}, \tag{8.24}$$

where $a_к$ – number of animal feedings per day, pcs; ψ – is the hopper filling factor.

The time for feeding a portion of food into the feed hopper depends on the length of the feed line, the speed of the feed in the pipe, the useful capacity of the boiler, the capacity (mass flow) of the feed line:

$$t_{П} = \frac{60W_{пк}}{Q_{к}} + \frac{l_{ср}}{60v_{ср}}, \tag{8.25}$$

where Q_K – is the productivity of the feed channel, m^3/h; $l_{ср}$ – average length of a path of moving of a forage on кормопропровод from a blowing boiler to кормоприемного bunker, m; $v_{ср}$ – average speed of feed, m/h.

Productivity of the fodder:

$$Q_{к} = 900\pi d^2 v_{ср} \tag{8.26}$$

When using two blowdown boilers in the installation, its actual capacity:

$$Q_{ср} = \frac{60m_i q_H}{t_ц \rho}, \tag{8.27}$$

where $t_ц$ – is the time of the feed distribution cycle per day, hour.

Then

$$t_ц = t_з n + t_П n + t_0 n, \tag{8.28}$$

where n – is the number of portions to be distributed within 24 hours, pcs; t_0 – time spent during the day for preparatory and final operations for maintenance of the installation, hour.

Number of servings to be distributed within 24 hours:

$$n = \frac{m_i q_H}{W_{пк} \rho}. \tag{8.29}$$

Receiver capacity:

$$W_{рес} = W_{пк} + \frac{\pi d^2 l_{\max}}{4}, \tag{8.30}$$

where $l_{\max}$ – length of receiver ресивера, m.

The consumption of compressed air, or the capacity of the compressor station:

$$Q_{к.ст} = \frac{Q_ф \varepsilon}{k_у}, \tag{8.31}$$

where $Q_ф$ – actual supply of the installation, m^3/h; ε – is the degree of air compression in the feed duct; $k_у$– coefficient of leakage of compressed air ($k_у$ = 0,85).

Degree of air compression:

$$\varepsilon = \frac{p_{раб}}{p_{ат}}, \tag{8.32}$$

where $p_{раб}$ – is the working pressure required to feed the blowing boiler to the feed hopper, kN/m^2; $p_{ат}$ – atmospheric pressure ($p_{ат}$ = 98,1 kN/m^2 or $p_{ат}$ = 1 at).

The design working pressure in the pneumatic system should be equal to the total head loss in the feed line:

$$p_{раб} = \sum \Delta h. \tag{8.33}$$

When leaving the feed from the feed, it is necessary that it pours out not freely, but with a certain high–speed head (30...50 kN/m^2), which prevents the presence of feed masses in the pipes. That is why in order to ensure a supply of compressed air to maintain pressure at the end of the feed, the pressure in the system of the pp receivers must exceed the pressure of the rudders in the pneumatic system. In this way:

$$p_р = p_{раб} + (30...50). \tag{8.34}$$

To distribute the semi–liquid feeds serve feeder–dispensers. The cycle time of a single feed for $t_ц$ is composed of the time necessary to fill the feeder–dispensers $t_з$, the time taken to discharge the forage from the feeding lines to the feeders, $t_в$ and the time for the preparatory–final operations t_0:

$$t_ц = t_з + t_в + t_0. \tag{8.35}$$

Knowing the total cycle time $t_ц$ of a single feed distribution and based on the zootechnical conditions and design features of the distributor, the values of the components of this time $t_з$, $t_в$ and t_0 take.

Taking into account the time $t_з$ and $t_в$, the required productivity when unloading the feed from the feed hopper and filling the feeder pipelines:

$$Q_з^/ = \frac{G}{t_з d_к}, \tag{8.36}$$

where G – is the daily amount of feed (according to the ration) required for animals fed by the fodder distributor, kg; d_k – diameter of feeder pipelines, m.

The required productivity of the distributor at the unloading of feed from feeder–dispensers in feeders:

$$Q_в^/ = \frac{G}{t_з d_к} \tag{8.37}$$

Productivity at the end of feed from the feed hopper to the feeder–dispenser:

$$Q_з = \frac{\pi d_0^2}{4} v\rho, \tag{8.38}$$

where d_0 – is the diameter of the hose feeding feed from the feed hopper into the feeder–dispenser, m; v – is the rate of feed expiration, m/s.

Taking into account that $Q_з^/ \le Q_з$ equations (8.36) and (8.38) can be written:

$$d_0 = \sqrt{\frac{4G}{\pi v\rho t_з d_к}}.$$

Diameter of feeding lines–batchers:

$$d_к = \sqrt{\frac{4 q_H n_i}{\pi L \Psi \rho d}}, \tag{8.39}$$

where n_i – is the number of animals served by the distributor, pcs; $q_н$ – average daily amount of feed required for one animal, kg; L – total working length of feeder–dispensers, m; Ψ – filling factor of feeding lines–metering devices; d – is the diameter of the dispenser, m.

Due to the small diameter of the pipe, large head losses are possible, but small costs are required for the construction of the feed line. With a larger pipe diameter, the losses in the feed line will be lower, and the cost of its installation will increase. The most favorable from the point of view of the least hydraulic resistance is a diameter equal to 100...150 mm. If the diameter is more than 150 mm, this does not give a noticeable energy effect, since the head loss in pipes 150 mm in diameter and, for example, 200 mm differ slightly. However, the use of pipes with a diameter of 200 mm leads to a sharp increase in the metal capacity of the feed network.

A significant disadvantage of stationary feed dispensers is insufficient redundancy of necessary equipment reliability. The process of distributing feed throughout the feeding front is stopped when any of the technical elements of this feed dispenser fail.

8.5 The structure and process of operation of mobile feed dispensers

On the farms of cattle, mobile bunker feeders are operated with the drive of working bodies from the tractor's VOM. Advantage of mobile feed distributors of machines is the lower unit cost of these machines in comparison with stationary machines. In this case, mobile feed dispensers work longer and can distribute feed in several rooms. If the feed distributor fails at any stage of the operation, the distribution of feed will increase only in time and is produced by other machines participating in this process. In addition, only mobile feed dispensers can organize mechanized feeding of fodder in the walking areas and in summer camps. Increase their efficiency of use can be achieved by reducing idling and expanding functions.

The most common are trailed mixers–distributors of batch action. Most of them consist of a uniaxial chassis on which a feed hopper is mounted with a transverse discharge conveyor and a discharge door flap. The drive of the working bodies of these machines is carried out from the tractor's power take–off shaft. Screws are installed in the hopper for transportation and mixing of feeds. They can be located inside the hopper depending on the machine layout – horizontally or vertically (fig. 8.5). The number of them can vary from one to four, and the diameter from 254 to 600 mm. With the increase in the volume of the hopper, the diameters increase, and the radial size of the screws located at the bottom is less than the upper ones.

The replaceable capacity of the mixer–distributors is 3,2...6,4 t/h, and the specific energy consumption is 5,4...9,04 kWh/t. The volume of the mixer–distributor bunker is in the range of 4...36 m^3.

The energy consumption of the mixing process is higher with the vertical arrangement of the screws in the center of the hopper. This is due to significant loads on the vertical auger. Mixers–distributors in this design version are aggregated with tractors of higher power. The overall dimensions of machines with a vertically positioned auger are larger in height and width than those with a horizontal auger of the same capacity.

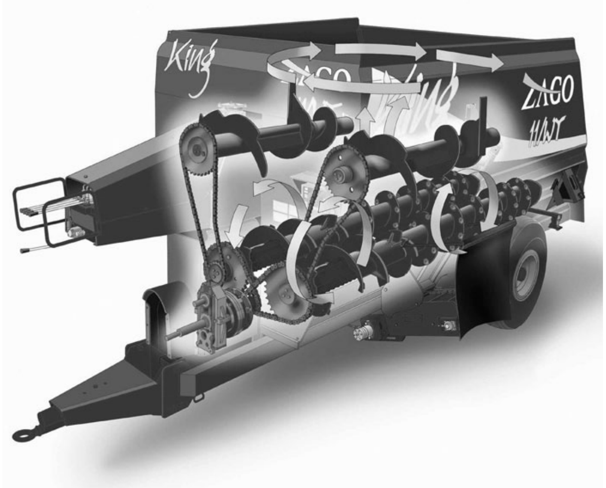
	
– with horizontal arrangement of augers	–with vertical arrangement of augers

Fig. 8.5. Choppers–mixers–distributors of feeds

The advantages of these machines are also:

– simplicity of structures;

– possibility of bunker loading from all sides;

– great adaptability to processing of stalked forages.

However, in comparison with mixers–distributors with horizontal augers, they consume 30 to 40 % more energy, and require a gate height of at least 2,3...2,7 m and a width of the feed aperture of at least 2,4 m. The grinding time And mixing the feed mixture is 10...15 min.

It is established that about 25 % of all costs associated with feeding animals are for loading fodder in vehicles. Moreover, in small and medium–sized farms, the use of forklift trucks does not bring the economic benefit due to low workload – the time lost by loaders reaches 40 %. Increase the effectiveness of this technological operation is proposed by equipping feeders with forage feeders.

The use of distributors with a self – loading system allows you to exclude the tractor on loading operations. This helps to reduce the energy intensity and metal consumption of the operations performed, allows the tractor driver to be freed. Also, the loss of time associated with the mismatch in performance of various machines employed in the performance of this process is eliminated.

A number of mechanisms are used to perform the technological operation of loading feeds: fork and clamshell grippers, milling cutters, combing combs and various knife structures for cutting feed blocks. Since the feed mix includes silage (haylage) and stalked forage, it is more technologically to organize the loading of grab grips or milling drums. Each of these technical systems has its advantages and disadvantages. Grapple forklifts are widely used due to their simple construction and reliability (fig. 8.6).

a b

Fig. 8.6. Grinder–mixer–distributor with clamshell gripper (a).
Chopper–mixer–distributor with milling drums (b)

They are more versatile in terms of types of feed materials and can be used when loading stalk feeds. At the same time, they have a significant drawback when loading silage (senage). Performing a technological operation, the fodder monolith is destroyed, which causes loss of nutrients due to secondary fermentation.

This disadvantage will be eliminated when loading the stalk feed with milling drums, the cutting elements of which consistently cut off the stalked feed from the monolith and feed them to the conveyor or to the loading ladle of the machine (fig. 8.7).

a) b)

Fig. 8.7. Mixer–chopper feed:
a) with a vertical cone–shaped screw
b) with horizontal augers

The recommended circumferential speed of the cutting drums at the selection of silage forages is set at 8...12 m/s, on roughage this indicator is recommended to be equal to 30...35 m/s, the productivity of the milling drum loading device depends on the number of animals and should be within 14,4...28,8 t/h. The upper value is desirable when the unit is operating on a farm with a capacity of up to 5000 heads. The drive of the milling drum is carried out, as a rule, by a hydromotor.

Sometimes the mixing is combined with the grinding of the components. When preparing wet feed mixes, grinders and mixers with knife work tools are used – the auger is equipped with knives installed on its turns (fig. 8.7, a, b). This machine design allows you to combine two operations, grinding feed and forming a feed mix.

Self–propelled mixers–feeders Siloking (fig. 8.8) contain a hopper with a capacity of 10...30 m³ for 60...300 cows, one or two augers. High maneuverability and productivity ensure the use of self–propelled mixer–feed mixers both on modern livestock complexes and in narrow and low buildings. The feature of the self–propelled mixer–feeder SILOKING is a three–point chassis. In this case, the pivoting device at the rear of the machine is a twin–wheel rotating by 150 degrees. As a result, a high maneuverability is ensured with an extremely small turn diameter. In addition, this avoids the loss of feed during loading and ensures that they are dispensed to feeders.

Fig. 8.8. Siloking self–propelled mixer–feed mixer

The cutter, equipped with knives, keeps the structure of feeds as much as possible and optimizes the quality of the feed mix. The conveyor feeds the components of the feed into the mixing bin. In the hopper the mixing of feeds ensures the auger.

The feed dispenser–mixer approaches the place of storage of feed, which is fed into it by the loader. During transportation, the components are thoroughly mixed and re–ground before the farm is placed, and then evenly distributed along the feeders. However, preparing a feed mix with these machines does not ensure individual feeding of animals, since high–energy feeds are fed to animals without taking into account their productivity. Such uneven distribution of feed reduces their energy output – feed mix for a certain group of animals' leads to a decrease in the productivity of other groups. Inadequate nutrition of fed feed to the needs of animals is also a reason for reducing the overall level of milk yield.

Reduce the energy consumption and metal consumption of the process of distribution and formation of the feed mix can be a mobile modular mixer–distributor (fig. 8.9). The machine contains two bins – for voluminous stalk feeds and a multicomponent high–energy additive. Mixing of these feeds to animals is carried out in a continuous flow – as the feeder moves along the feeders.

Fig. 8.9. General view of the mobile mixer – feed distributor:
1 – bunker for stalked forages; 2 – module for a multicomponent high–energy additive; 3– dosing flap of high–energy additive; 4–unloading chain–slat conveyor

The machine consists of a wheelbase on which a hopper for stalked forages is fastened, on the bottom of which an unloading chain–slat conveyor moves, which moves the stalked forage to the unloading window. Dosage of stalked forages is carried out before the unloading window installed from the discharge hut. Transportation, mixing and delivery of multicomponent high–energy additives by animals are carried out in the module located on the opposite side of the hopper hopper. Dosage of feed flow through this window is carried out by an adjusting damper.

The module (fig. 8.10) for a multicomponent high–energy additive is a hopper 1, inside which, in one horizontal plane, there are two screws 2, 3.

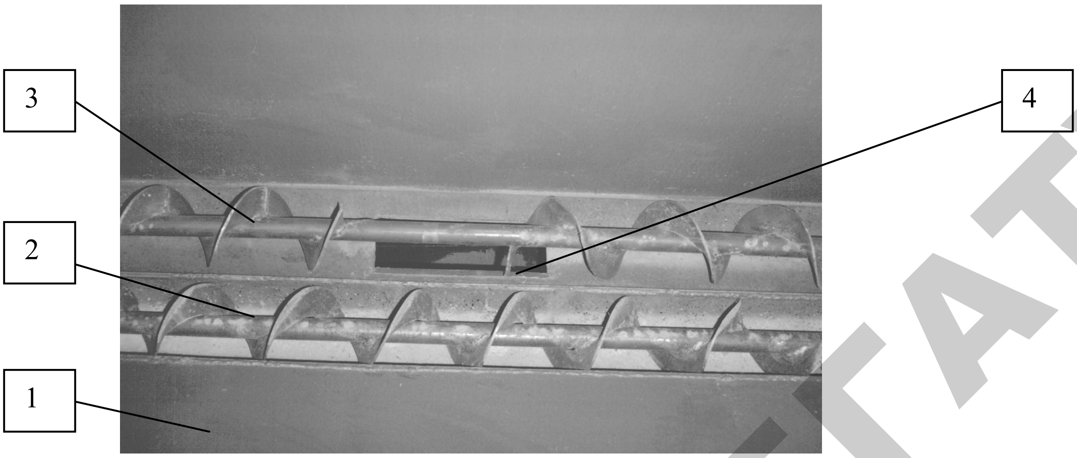


Fig.8.10. Module for multicomponent additive:
1 – bunker; 2, 3 – augers; 4 – discharge channel

These technical elements of the mixer perform two technological operations – mixing of high–energy feed components and feeding the mixture through the discharge channel 4 to a meeting of suspended in the stalk forage. In accordance with the scheme of the mobile modular mixer–distributor of feeds, an operational flow chart of the mobile workflow is compiled (fig. 8.11).

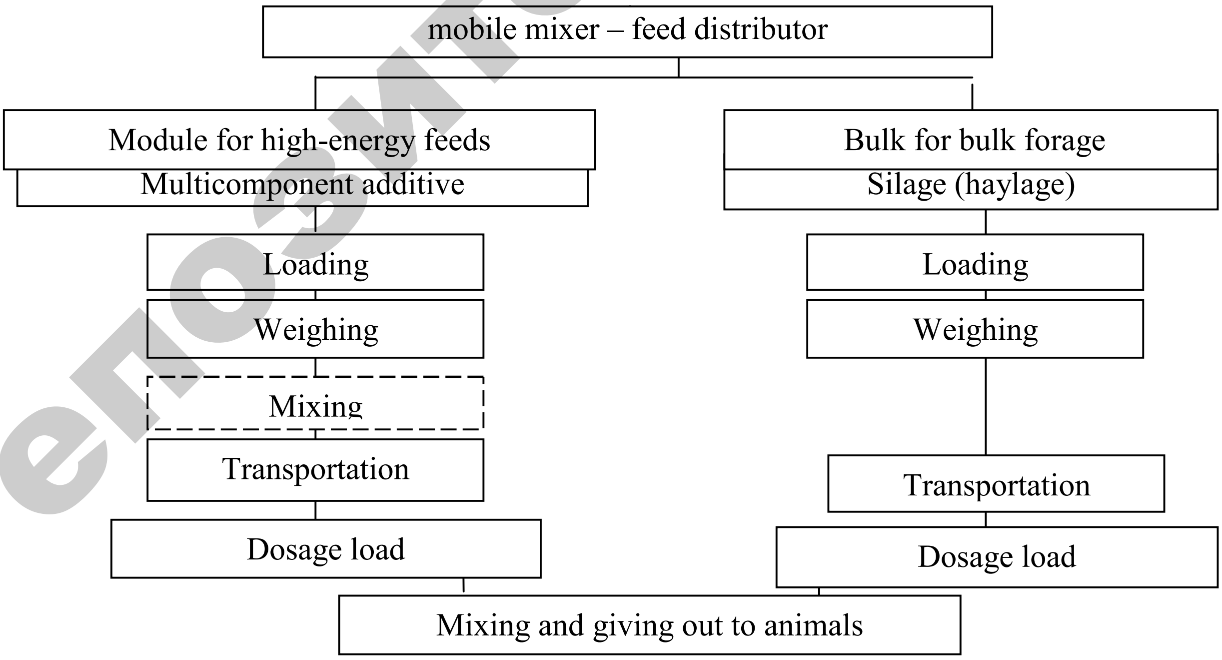


Fig. 8.11. The operational scheme of the working process
of the mobile modular feed mixer–distributor

When feeding food to animals, the stalk feeds are dosed out from the hopper to the transverse discharge conveyor. High–energy feeds through the unloading window, made in the working area of the screw turns, are fed to the stream of stalk feeds. The fodder mix is formed from the intersecting in the air streams of ensilared stalked and high–energy fodder. Implemented in the car technology of distribution of animal feed reduces the energy intensity of the process by 10,1 %.

8.6 Determination of the basic design parameters of mobile feed distributors

8.6.1. Determination of the unloading window squire of dosage feed distributors

An important element of the technological process of distributing feed to animals is the observance of a given rate of delivery to animals. Ensure unhindered, dosed, dispensing of feed can only be determined by the area of the discharge window dispenser. In general, the area of the unloading window can be determined by the formula

$$S_{о\,к} = \frac{Q_к}{v_к \;\; \rho}, \tag{8.40}$$

where $Q_к$ – is the given maximum rate of feed distribution to animals per unit time, kg/s; $v_к$ – speed of feed in the working area of the unloading window, m/s; ρ – is the density of the food, kg/m^3.

The specified maximum rate of feed for animals per unit time can be determined from formula

$$Q = \frac{v_{раз}\, m_к}{l_{разд}}, \tag{8.41}$$

where $v_{раз}$ – the speed of movement of the aggregate during the distribution of feed, m/s; $m_к$ – is the mass of fed feed, kg; $l_{разд}$ – the length of the feeding front of one animal, m.

The speed of movement of feed in the working area of the unloading window can be determined by formula

$$v_к = l_ч\, \omega_{тр}, \tag{8.42}$$

where $l_ч$ – is the distance from the feed particle to the edge of the unloading conveyor in the working area of the unloading window, m; $\omega_{тр}$ – angular velocity of the unloading conveyor, c^{-1}.

In formula (8.42), the unknown quantity is the distance lh. To determine it, let us consider the forces acting on the particle of the feed at the moment of descent from the unloading conveyor:

– gravity:

$$F_т = m_к\, g\,, \tag{8.43}$$

where $m_к$– is the mass of the feed particle, kg; g – acceleration of gravity, m/s^2;

– friction force of feed about discharge conveyor:

$$F_{тр} = f\, m_к\, g,, \tag{8.44}$$

where f – is the friction coefficient of the feed particle;

– centrifugal force:

$$F_ц = m_к\, \omega_{тр}^{\;2}\, l_ч\,, \tag{8.45}$$

– Coriolis force:

$$F_{кор} = 2\, m_к\, \omega_ш \frac{dl_ч}{dt_ч}\,, \tag{8.46}$$

where $dl_ч / dt_ч$ – is the velocity of the particle moving in the radial direction, m/s; $t_ч$ – is the time of moving the feed particle in the radial direction, c.

Force of friction of a particle:

$$F_{тр\,л} = f\,(F_{кор} + F_{тр})\,. \tag{8.47}$$

Then, taking these forces into account, we obtain a differential equation for the relative motion of the feed particle:

$$-m_к \frac{d^2 l_ч}{dt_ч^{\;2}} + m_к \omega_ш^{\;2} l_ч - f\, m_к\, g - f\,(2 m_к \omega_ш \frac{dl_ч}{dt_ч} + f\, m_к\, g) = 0. \tag{8.48}$$

After the transformation, equation (8.48) has the form:

$$l_ч = \frac{fg}{\omega_ш^2}\left[\left(1-\frac{f+\sqrt{f^2+1}}{2\sqrt{f^2+1}}\right)e^{t_ч\omega_ш(f+\sqrt{f^2+1})}+\left(\frac{f+\sqrt{f^2+1}}{2\sqrt{f^2+1}}\right)e^{t_ч(f-\sqrt{f^2+1})}-1\right]. \quad (8.49)$$

Substituting the values (8.49) into equation (8.42), we determine the required area of the unloading window:

$$S_{ок} = \frac{v_{раз}\, m_к}{\omega_ш\, \rho\, l_{разд}\left[R_ш-\left(C_1 e^{(f\,\omega_ш+\omega_ш\sqrt{f^2+1})t_ч}+C_2 e^{(f\,\omega_ш-\omega_ш\sqrt{f^2+1})t_ч}-\frac{fg}{\omega_ш^2}\right)\right]}. \quad (8.50)$$

Thus, when determining the size of the unloading window intended for unloading feed from the hopper, it is necessary to take into account the physical and mechanical properties of the feed, the parameters of the unloading device, the rate of feed distribution to the animals. To change the rate of feed feeding, the unloading window is overlapped by a movable flap.

8.6.2 Determination of energy costs for screw drive when mixing feeds

For the preparation of animal feed mixtures are used augers. These working bodies, moving the feed components in the hopper, provide due to the mutual intersection of feed streams, the receipt of feed mixtures.

The nature of the movement of feed in the hopper is different, so the energy costs for the process will depend on the individual working areas of the screw.

Since the screw is located at the bottom of the hopper, part of the energy will be spent to overcome the friction force of the feed along this surface (fig. 8.12). In general, this power can be determined from the formula

$$N_{тр} = F_{тр} v_{см1}\,, \quad (8.51)$$

where $F_{тр}$ – is the friction force of the feed about the bottom of the hopper, H; $v_{см1}$ – speed of feed displacement along the axis of the bottom of the hopper, m/s.

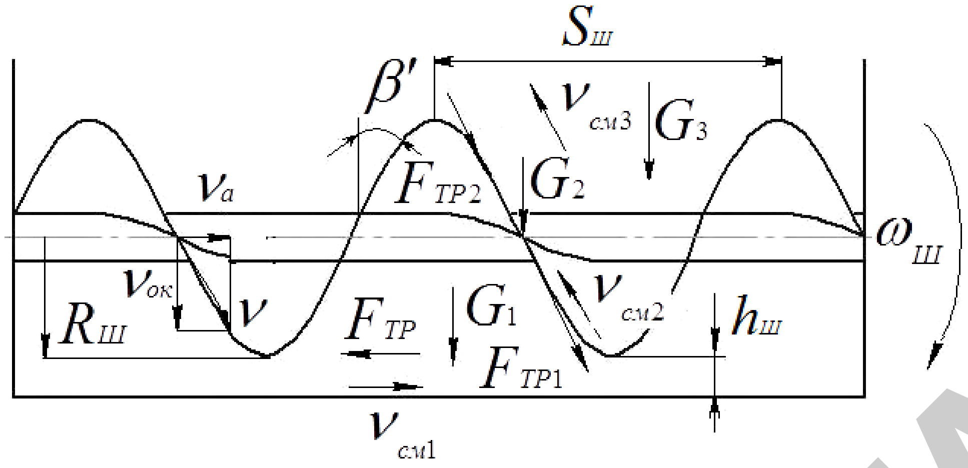


Fig. 8.12. Scheme for determining the power of a screw drive

The frictional force $F_{тр}$ is produced by gravity of the feed located in the working part of the screw. Then:

$$F_{тр} = m_{кор} \ g \ f, \tag{8.52}$$

where $m_{кор}$ – is the mass of food transported by the augers along the bottom of the hopper, kg,

$$m_{кор} = W_к \, \rho,$$

$W_к$ – is the volume of feed, m^3; ρ – density of feed, kg/m^3.

Since the auger is covered only by the bottom wall of the hopper in a limited area, the volume of transported food can be determined by formula

$$W_к = \frac{\pi \ R_ш \ h_ш \ \alpha_{ш1} \ L_ш}{360^o}, \tag{8.53}$$

where $R_ш$ – is the screw radius, m; $h_ш$ – the gap between the screw turns and the bottom wall of the hopper, m; $\alpha_{ш1}$ – the angle of coverage of the screw by the bottom wall of the hopper, degrees; $L_ш$ – screw length, m.

The speed of feed displacement along the axis of the bottom of the hopper due to the small gap between the screw turns and the wall will be equal to the axial speed determined by the formula

$$v_{см1} = v_a = S_ш \omega_ш . \tag{8.54}$$

Then the power to overcome the friction force of the feed at the bottom of the hopper:

$$N_{mp} = \frac{\pi\ R_{ш} h_{ш} \alpha_{ш\,1} L_{ш} \rho}{360^{0}} g f S_{ш} \omega_{ш}\ . \qquad (8.55)$$

where $S_{ш}$ – is the pitch of the screw turns, m; $\omega_{ш}$ – screw speed, s^{-1}.

When auger rotates, the forage mass moves along its planes. For this part of the screw, the amount of energy spent on overcoming friction on the winding can be determined from formula

$$N_{mp1} = F_{mp1}\, \mathbf{v}_{см2}\ , \qquad (8.56)$$

where F_{mp1} – is the friction force of the feed on the surface of the screw turns covered by the bottom of the hopper, N.

The frictional force F_{mp1} can be determined by the formula

$$F_{mp1} = m_{к1}\, g\ f\ , \qquad (8.57)$$

where $m_{к1}$ – weight of feed in the interturn space, kg,

$$m_{к1} = W_1\ \rho\ \ \frac{\alpha_{ш\,2}}{360^{o}}\, ,$$

where W_1 – volume of feed in the interturn space, m^3; $\alpha_{ш2}$ – the angle of the auger is not covered by the rear wall of the hopper, hail;

The volume of feed in the interturn space on the length, the current step, is determined by the formula

$$W_1 = \left(D_{ш}^2 - d_{в}^2\right)\ S_{ш}\ \ K_V\, , \qquad (8.58)$$

where $D_{ш}$ – is the screw diameter, m; $d_{в}$ – diameter of auger shaft, m; K_v – is a coefficient that takes into account the use of the interturn space.

Then the frictional force Fmp1 can be determined by the formula

$$F_{mp\,1} = \left(D_{ш}^2 - d_{в}^2\right) S_{ш} K_V \rho \frac{\alpha_{ш\,2}}{360^{0}} g\ \ f. \qquad (8.59)$$

Since the screw turns are limited by the bottom of the hopper, the forages are moved predominantly in the axial direction.

Then we can assume that $V_{см2} = V_a.cos\beta$ 'and the energy consumption for moving the feeds along the screw turns is determined by the formula

$$N_{тр\,1} = \frac{\alpha_{ш\,2}}{360^0}\left(D_ш^2 - d_в^2\right)2S_ш K_V \rho g f \omega_ш \cos\beta' . \qquad (8.60)$$

For a part of the screw, not limited to the bottom of the hopper, the energy costs for moving the feed through the turns can be determined from the formula

$$N_{тр2} = F_{тр2} \; \mathbf{v}_{см3}, \qquad (8.61)$$

where $F_{тр2}$ – is the frictional force arising when the feed is moved along the screw turns not bounded by the bottom of the hopper, N.

The frictional force caused by gravity can be determined from the formula

$$F_{тр\,2} = \left(W_\delta - W_ш\right)\rho g f , \qquad (8.62)$$

where W_δ – the volume of the bunker, m^3; $W_ш$ – is the volume of the screw, m^3.

Since the upper part of the auger is not limited to auxiliary planes, the rate of movement of the feed through the turns will be equal to the sum of the axial and circumferential velocities:

$$v_{см3} = \sqrt{v_a^2 + v_{ок}^2} . \qquad (8.63)$$

The value of the circumferential velocity can be determined from the expression:

$$v_{ок} = v_a tg\left(\beta' + \varphi_{тр}\right) = S_ш \, \omega_ш tg\left(\beta' + \varphi_{тр}\right) . \qquad (8.64)$$

where β – the angle of the screw screw line elevation, degrees; $\varphi_{тр}$ – is the angle of friction, deg.

Then

$$v_{см3} = \sqrt{\left(S_ш \omega_ш\right)^2 \left[1 + tg\left(\beta' + \varphi_{тр}\right)^2\right]} . \qquad (8.65)$$

Substituting the values (8.62) and (8.65) into equation (8.61), we obtain:

$$N_{тр2} = \left(W_{\delta} - W_{ш}\right)\rho g f \sqrt{\left(S_{ш}\omega_{ш}\right)^{2}\left[1 + tg\left(\beta' + \varphi_{тр}\right)^{2}\right]}. \qquad (8.66)$$

Summing up the energy costs for overcoming frictional forces, we determine the resulting power on the screw drive, which depends on the physical and mechanical properties of the feed and the geometric parameters of the hopper and the screw.

8.6.3 Formation of feed mix in mutually intersecting flows of feed components

The process of continuous formation of the mix from the intersected flows of ensilared stalked and high–energy feeds, which are loaded onto the conveyor of the feed distributor from the corresponding hoppers, is shown in fig. 8.13. An important parameter characterizing the quality of the feed mix is the determination of the range of flight of particles of a multicomponent high–energy additive. This parameter depends on the velocity of the particles, which can be determined by decomposing it into two independent ones – a uniform rectilinear motion V_n and a free fall with a relative velocity $V_{от}$.

Then the absolute speed of movement of the particles of stalked feed can be determined by the formula

$$v_{ч}^{\,2} = v_{n}^{2} + v_{от}^{2}. \qquad (8.67)$$

In formula (8.67), $v_n = h_{гор}\,\varphi'$, where $\varphi' = d\varphi/dt$ – is the angular velocity of the moving of the particle of the multicomponent additive during the time interval; s^{-1}; $h_{гор}$ – the distance of the moving of a particle of a multicomponent additive, m.

The relative velocity of moving a particle of a multicomponent high–energy additive is determined by the formula

$$v_{от} = dh_{вер} / dt, \qquad (8.68)$$

where $h_{вер}$ – is the fall height of a particle of a multicomponent high–energy additive, m.

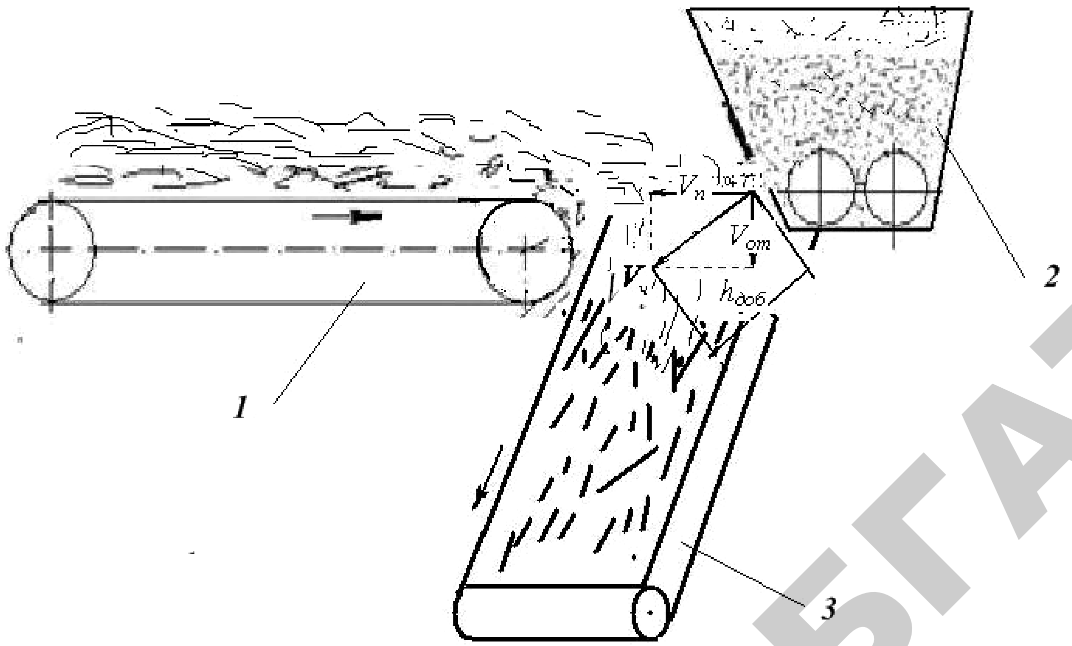


Fig. 8.13. Scheme for calculating the range of flight intermittent in the suspended state of feed streams:
1 – transporter of stalked forages; 2 – mixer–metering device of a multicomponent high–energy additive; 3 – unloading conveyor

Then the absolute velocity of the particle motion of the multicomponent high–energy additive can be determined by the formula

$$v_ч{}^2 = h^2_{гор}\,\varphi'^{\,2} + h'^{\,2}_{вер}. \tag{8.69}$$

To solve equation (8.69), we use the Lagrange equations of the second kind:

$$\frac{d}{dt}\left(\frac{\partial E_{доб}}{\partial \varphi}\right) - \frac{\partial E_{доб}}{\partial \varphi} = Q_{\varphi}, \tag{8.70}$$

where Q_h, Q_φ – are the generalized forces acting on the particle of the multicomponent high–energy additive, H; $E_{доб}$ – is the work expended when moving a particle of a multicomponent high–energy additive, kg m^2/s^2.

By transforming equations (8.70), we define the work spent when moving the feed particle on the elementary path of possible displacements, can also be determined from formulas

$$Q_h = m_ч\, g - f\,N,$$

$$Q_{\varphi} = (N - m_{ч}\, g)\, h_{гор}. \qquad (8.71)$$

where f – is the friction coefficient of a multicomponent high–energy metal additive; $m_{ч}$ – is the mass of a particle of a multicomponent high–energy additive, kg.

Since in the equations (8.70) and (8.71) the left parts are equal, then after aligning them and after the transformation, we obtain a linear inhomogeneous second–order differential equation with constant coefficients.

$$h_{вер}'' + 2\,f\,h_{вер}'\,\omega_{ш} - h_{гор}\,\omega_{ш}^{2} = g(1 - f). \qquad (8.72)$$

Representing equation (8.72) as the sum of a particular and general solution, we determine the flight range of a multicomponent additive particle into the layer of stalked feeds by the formula

$$h_{доб} = \frac{g(1-f)}{\omega_{ш}^{2}} \left(\left(1 - \frac{f + \sqrt{1+f^{2}}}{2\sqrt{1+f^{2}}}\right) e^{\omega_{ш}\, t \left(f + \sqrt{f^{2}+1}\right)} + \frac{\left(f + \sqrt{1+f^{2}}\right)}{2\sqrt{1+f^{2}}}\; e^{\omega_{ш}\, t \left(f - \sqrt{f^{2}+1}\right)} - 1 \right). \qquad (8.73)$$

On the basis of the obtained equation, it is established that at the time of flight of high energy feed particles, equal to 2...3 sec. And the angular velocity of the unloading auger $\omega = 4.4\ s^{-1}$, the flight range in the layer of stalked feeds of the particle of the multicomponent high–energy additive is 8 mm. The process of continuous formation of the feed mix takes place on the overload conveyor by randomly absorbing the particles of a multicomponent high energy additive by the flow of a silage stalk feed. Particles of a multicomponent high–energy additive under the influence of gravity $F_{тяж.ч}$ move to the lower layers of a siloed stalk with a height l_{δ} to a depth $l'_{ч}$:

$$F_{тяж.ч.} = \frac{f_{доб}\ m_{к}\ g\ l'_{ч}}{l_{\delta}}, \qquad (8.74)$$

where $m_{к}$ – is the mass of a particle of a multicomponent high–energy additive, kg; $l'_{ч}$ – depth of moving of a particle of a multicomponent high–energy additive, m; l_{δ} – height of the layer of stalked forages on the conveyor belt, m; $f_{доб}$ – coefficient of internal friction of particles of a multicomponent high–energy additive.

The displacement of a particle of a multicomponent high–energy additive into the layer of stalked forages is counteracted by the frictional force, the magnitude of which depends on the vertical pressure of the particles of the layer of stalked forages:

$$F_{тр} = P_{д}\, S\,, \tag{8.75}$$

where $P_{д}$ – vertical pressure of stalked forages on a conveyor belt, Pa; S – area of the stalk feeds of the conveyor belt located on the belt, m^2.

The vertical pressure can be determined by the formula:

$$P_{д} = \rho_{стеб}\, l_{\delta}\, g, \tag{8.76}$$

where $\rho_{стеб}$ – is the density of the stalked feed, kg/m^3; l_{δ} – height of the layer of stalked forages, m.

Then the frictional force $F_{тр} = \rho_{стеб}\ l_{\delta}\ g\ S$. Since $F_{тяж} - F_{тр} = 0$, equations (8.74) and (8.73) allow us to determine the depth of penetration of a particle of a multicomponent high–energy additive into the layer of stalked fodder:

$$l'_{ч} = \frac{f_{стеб}\,\rho_{стеб}\ l^{2}_{\delta}\ k\ S}{f_{доб}\ m_{к}}\,. \tag{8.77}$$

8.6.4. Determination of technological parameters of mobile feed distributors

The capacity of the feeder bunker is chosen in such a way that for one–time loading the machine can serve one or several cattle–breeding premises. In addition, the amount of food in the dispenser should be equal to or multiples of the amount of feed consumed for a single feeding of livestock, located in one row of the room, i.e.,

$$G = q_{п} n_{i} n_{ж} \tag{8.78}$$

where q_{n} – is the amount of feed required per head, kg; n_i – number of animals in the same row of rooms, pcs; $n_{ж}$ – the number of rows of animals served by a single load of the hopper of the feeder, pcs.

Since on livestock farms, especially for the dairy herd, the distribution of feed is carried out two or three times a day, the one–time weight of feed that must be given to animals can be calculated by the formula:

$$G_p = \frac{\sum q_i n_j}{k}, \tag{8.79}$$

where k – is the multiplicity of animal feeding on the farm.

The required amount of food, placed in the bunker of the machine,

$$G_p = V_p \rho \, \psi, \tag{8.80}$$

where Vp – hopper capacity, m^3; ρ – density of feed in the dispenser's hopper, kg/m^3; ψ – filling factor of the hopper (ψ = 0,8...0,9).

After that, the dimensions of the feeder bunker are determined: the width B, the height H and the length L, starting from the condition that $V = BHL$, and the constituent elements of the cattle–breeding room (the width and height of the gate, the distance between the troughs, etc.) in which the machine will operate.

To ensure a one–time distribution of feed, it is necessary to have a capacity of the mixer–distributor hopper equal to

$$V_p = \frac{M_p}{\rho} = \frac{\sum q_i n_j}{k\rho}, \tag{8.81}$$

where ρ – is the density of feed, kg/m^3.

The capacity of the feed mixer hoppers, providing in one distribution of all the animals' forages, shown in fig. 8.14, shows that it is not advisable to make bunkers of such large dimensions, since the machines will in this case operate no more than 1 ... 3 h / day, and the cost of their production will be very large.

For efficient use of the mixer dispenser, especially for self–propelled mixers, the capacity of the hopper must be such that the animals are provided with feed for several hours (cycles) of dispensing during the time set by the zootechnical requirements.

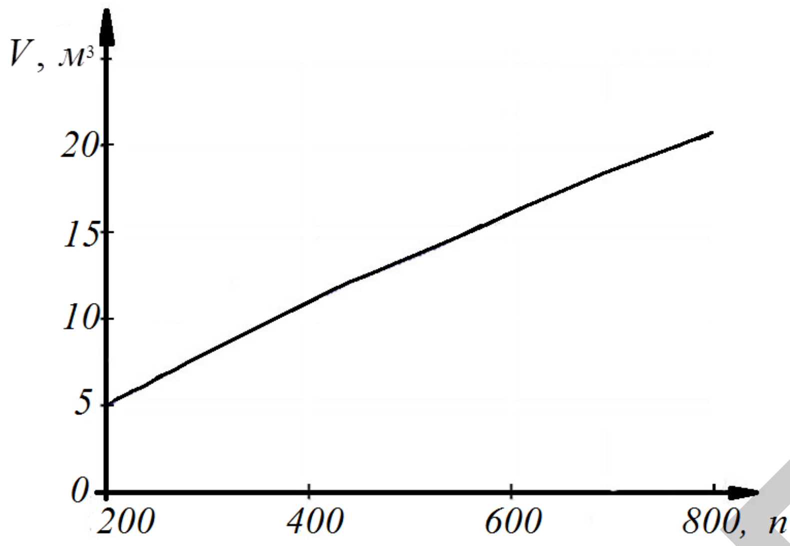


Fig. 8.14. Dependence of the capacity of the mixer–dispenser hopper on the number of served livestock

In this case, the hopper capacity of the mixer dispenser will be:

$$V_m = \frac{V_p}{n_ц}, \tag{8.82}$$

where $n_ц$ – is the number of feed distribution cycles for animals, pcs:

$$n_ц = \frac{t_c}{\sum_{i=1}^{n} t_i},$$

where t_c – changeable working time of the mixer–distributor; $\sum_{i=1}^{n} t_i$ –the time needed to perform all the technological operations in one cycle of the machine.

The given recommendations on the choice of the volume of the mixer–distributor hopper are applicable for the operation of single–hopper machines. When the animals are given one machine of two different in physical and mechanical properties of feed, the volume of each bunker can be determined by the formulas:

– for stalk feeds:

$$V_к = \frac{m_{об}}{K^{об}{}_з \, \rho_к \, k}, \tag{8.83}$$

– for a multicomponent high–energy additive:

$$V_{\partial} = \frac{m_{\partial}}{K^{\partial}{}_{з}\,\rho_{\partial}\,k}, \tag{8.84}$$

where $m_{об}$ – is the mass of stalked forages, kg; m_{∂} – is the mass of a multicomponent high–energy additive, kg; $K^{об}{}_{з}$, $K^{\partial}{}_{з}$ – filling factors of the hopper, respectively, for stalk feeds and a multicomponent high–energy additive; k – multiplicity of feeding of animals.

When choosing the volume of the bunker, the most numerous group of animals should be taken into account. The disadvantage in this case is the incomplete loading of the mixer–distributor when distributing feed mixtures of a small group. Installing the extender bead in the distributor–mixer it is possible to vary the coefficient of machine utilization.

The capacity (supply) of the distributor's unloading devices must be coordinated with the amount of feed distributed per 1 m of the feeder's length (feeding front) and the speed of the dispenser's movement along these feeders.

For a mobile distributor, the amount of food distributed per 1 m of the feeding front to one side is determined as follows. The feeding conveyor (longitudinal conveyor) at a certain speed moves to the bi–ters during the time t part of the feed per 1 m of the feeder, i.e.,

$$q = q_0\, v_n t \tag{8.85}$$

where q – is the amount of feed distributed per 1 m of the feeder length, kg/m; q_o – the amount of feed per 1 m of the hopper's length, kg/m; v_n – speed of the feeding conveyor, m/s; t – is the time when the feeder feeds 1 m of the feeder's length (the time it takes the unit to travel 1 m), p.

Then

$$t = 1/v_a, \tag{8.86}$$

where v_a – is the speed of the feed unit (the speed of the distributor along the feeders), m/s.

Substituting the value of t into formula (8.85), we obtain

$$q = q_0 v_n / v_a \tag{8.87}$$

Then

$$v_n = qv_a/q_0 \qquad (8.88)$$

The amount of feed distributed per unit length of the feeder can also be determined from expression

$$q = BH_0 v_n \rho k_0/(v_a K_б), \qquad (8.89)$$

where H_0 – is the height of the feed product in the hopper, m; k_0 – coefficient of feed lag in the hopper from the longitudinal conveyor (k_0 = 0,94...0,96); $K_б$ – factor of decrease in speed of a tractor due to slipping of wheels ($K_б$ = 0,95...1,0).

If the feed distributor distributes the feed to the right and left sides simultaneously, then the feed conveyor speed should be increased 2 times.

If the feed is continuously distributed along the feeding front (one side) of the given amount of feed, the productivity of the unloading conveyor should be equal to the productivity of the feeding conveyor and be coordinated with the forward speed of the feed assembly. This condition for the belt conveyor can be written as follows:

$$Bhv_л\, \rho_1\, K_n = BH_0 v_n\, \rho\, k_0 = qv_a\, K_б, \qquad (8.90)$$

where B – is the internal width of the trough of the unloading conveyor, m; H – height of the layer of transported mass, m; $v_л$ – conveyor belt speed, m/s; ρ_1 and ρ – density of the mass of the feed on the unloading conveyor and in the hopper of the time–sensor, kg/m^3; K_n – coefficient, taking into account the decrease in productivity due to the movement of food on the belt with some slippage (K_n = 0,94...0,98).

The amount of feed distributed per 1 m of the feeder length by the main working organ – the scraper conveyor, can be represented by the following formula.

$$q = bhv_c\, \rho\varphi_c k_a/(V_a\, n_к), \qquad (8.91)$$

where b and h are the length and height of the conveyor scraper, m; v_c – speed of the scraper conveyor, m/s; φ_c – is a coefficient that takes into account the filling of the space between the scrapers with feed; k_a – is a coefficient that takes into account the reduction in the feed rate of the conveyor due to the angle of lift of the feed; n_K – is the number of rows of feeders into which feed is fed individually.

The productivity of feed distributors with screw dispensing–discharging organs is expedient to calculate according to the clarified formula of Professor VV Kransnikov:

$$Q_в = \frac{\pi}{4}\left(D^2 - d^2\right) S n_ш \rho K_n, \tag{8.92}$$

where K_n – is the differential coefficient of productivity; D and d are the diameters of the screw and shaft, m; S – screw pitch, m; $n_ш$ – screw speed, s^{-1}; $\varphi_ш$ – is the filling factor of the screw.

Then

$$K_n = K_з K_\beta K_v K_y \tag{8.93}$$

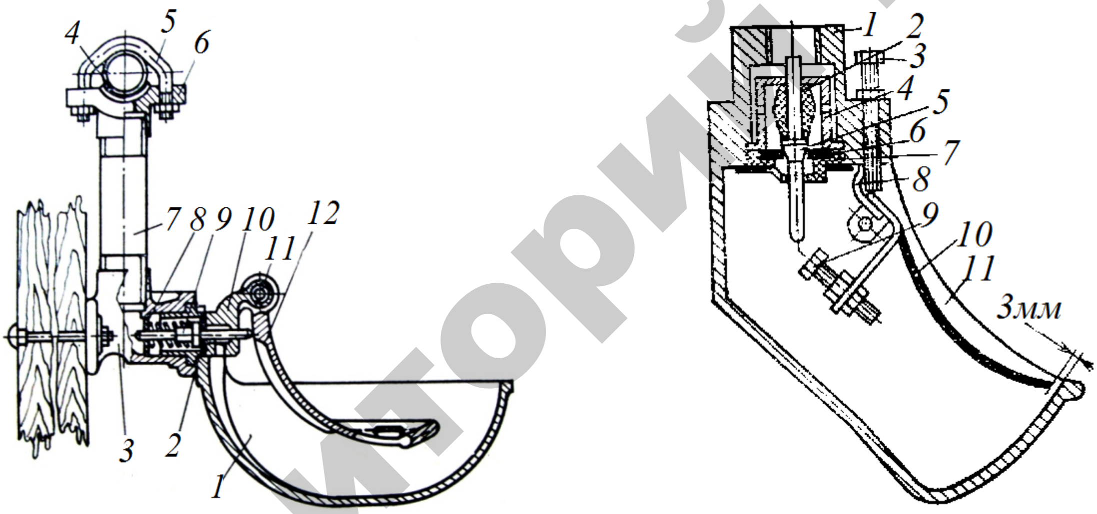
where K_3 – coefficient, taking into account the effect of the design of the loading device on productivity (K_3 = 0,5...1,0); K_β – is a coefficient that takes into account the effect of the slope of the screw on productivity (K_β = 0,3...1,0); K_v – coefficient that takes into account the lag of the average axial velocity of the product from the circumferential speed of the screw (K_v = 0,9...0,6, larger values are chosen for high–speed augers); K_y – is the coefficient of using the interturn volume (K_y = 0.2...0.9).

Chapter 9
WATER EQUIPMENT FOR ANIMALS AND POULTRY

9.1 Classification of drinkers, their arrangement and operating principle

Group drinkers are used for drinking horn cattle with loose housing system, pigs and poultry. Group drinkers can be stationary and mobile. They are equipped with troughs or several individual drinkers for drinking animals.

The water level is regulated by a float type valve mechanism. These include the group drinker VUK–3, AGK–12, AGK–4 with electric heating for horn cattle, AGS–24 for pigs, AP–2 groove and P–4A cup drinkers for poultry. In individual cup drinkers the amount of water entering the drinking cup is regulated by a special pedal, which is pressed by the animal itself. These include drinking bowls PA–1A, AP–1 for horn cattle (fig. 9.1).



a b

Fig. 9.1. Automatic cup waterbowls:
a) PA–1A: 1 – drinking bowl; 2, 4 – rubber gaskets; 3 – the case; 5 – collar; 6 – saddle; 7 – riser; 8 – grating; 9 – valve; 10 – valve box; 11 – axis; 12 – lever
b) – PSS–1: 1 – branch pipe for connection to the water supply; 2 – shock–absorber; 3, 9 – adjusting bolts; 4 – glass; 5 – valve; 6 – valve saddle; 7 – valve cover; 8 – spring; 10– bowl cover; 11 – cup

The drinking bowl is connected to the water supply network at a height of 500 – 600 mm from the floor of the stall. For the pig–poultry in recent years began to be widely used nipple drinkers type PBS–1 of various sizes or combined. The drinking bowl has the form of a cylinder, inside of which there is a nipple, pressing on which the animal displaces the stop valve from the hole in the water pipe and drinks water (fig. 9.2).

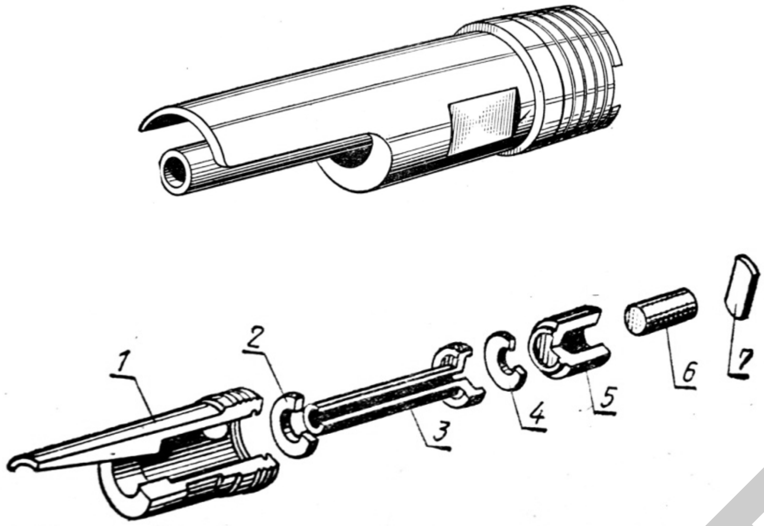


Fig. 9.2. PbS–1 Nipple automatic water bowls:
1 – housing; 2,4 – rubber gaskets; 3 – the nipple; 5 – the valve;
6 – the shock–absorber; 7 – stop

For the drinking of chickens, vacuum drinkers are used; for drinking, nipple drinkers are used in cellular batteries, water enters the bird in the form of a droplet along the cylinder of the nipple that it nests. The use of drinkers especially pacifier and nipple in pigs and poultry, allow the introduction of special devices in the water supply system by medics. These are the containers in which water–soluble drugs, vitamins, and other substances dissolve. That allows solving problems of drinking water with the prevention and treatment of animals. The mobile drinker PAP–10A is designed for mechanization of drinking of cattle in summer camps, remote from water sources. Aggregate the drinking bowl with tractors of classes 9 and 14 kN. The main components of the auto–drinking machine (fig. 10.3) are the water distributor VR–3.0 and the KAW–10 machine.

The water dispenser consists of a cistern, a frame and a centrifugal vane pump with a drive, running gear. The tank is a reservoir, in the upper part of which there is a neck closed by a lid. The pump provides a suction height of 4,5 m at the maximum number of revolutions (2900 min^{-1}) the pressure developed by it is 0,3–0,5 MPa. Pump drive from tractor PTO through cardan gear and V–belt drive.

The KAK–10 set includes 10 drinking bowls PA–1, left, right and front lines. The cistern is filled with water from open water sources by means of a pump or by gravity from a water pipe. At the drinking site, the drinking bowl is disconnected from the tractor and set in the working position. The number of serviced animals is 110 heads. The group drinking machine with electric heating AGK–4A is used for drinking animals on open areas in winter. It can also be installed indoors. In the body of an auto–drinker with thermal insulation there is a drinking bowl, a valve–float mechanism, an electric heating element and a temperature regulator. The electric heater is located under the bottom of the bowl. In the warm months of the year, it is turned off.

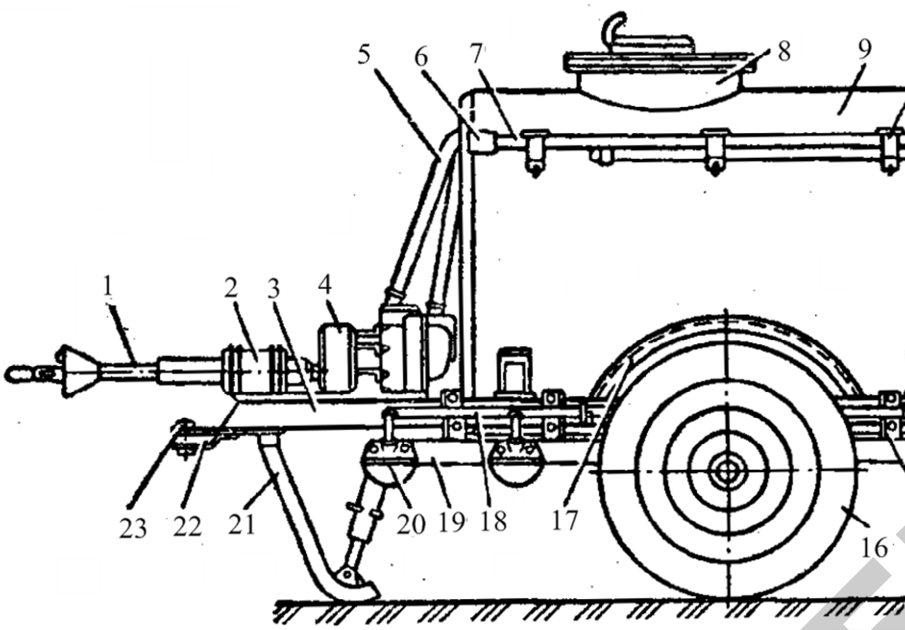


Fig. 9.3. Schematic diagram of the mobile auto–drinking machine PA–10A:
1 – cardan shaft; 2 – housing; 3, 14 – frame; 4 – pump; 5 – drain hose; 6 – filter; 7 – suction sleeve; 8 – neck; 9 – tank; 10 – collar; 11.19 – bracket; 12, 13 – valves; 15 – corner of the frame; 16 – running gear; 17 – a mud flap; 18 – left main line; 20 – drinking bowl; 21 – step; 22 – trailing bracket; 23 – cooker

Animals get access to water by pressing one of the four valve caps located at the top of the drinker. As the level of water in the bowl decreases, the float drops, the valve opens, and water from the water pipe enters the drinking bowl. The water temperature in the bowl is regulated within 4...180 C, changing the gap between the membrane and the microswitch. It is supported automatically by the thermostat. The drinking bowl is designed for 80...100 animals.

9.2. Calculation of water demand

The amount of water (drinking, technical) that the projected water supply system must supply is determined by the design norms of its consumption by the consumer of each species and their number, taking into account the long–term plan for increasing water consumption. Average daily water discharge on the farm is determined by the formula:

$$Q_{ср.\ сут.} = q_1 n_1 + q_2 n_2 + \ldots + q_m n_m,$$

where q_m – is the average daily water consumption rate, l; n_m is the number of consumers.

The magnitude of this expense is not sufficient for the calculation of the water supply network, therefore, determine the maximum daily water flow by the formula:

$$Q_{\text{мах. сут.}} = Q_{\text{ср.сут}} \cdot \alpha_{\text{сут}}, \tag{9.1}$$

where $\alpha_{\text{сут}}$ –is the daily non-uniformity coefficient, $\alpha = 1.3$.

The maximum hourly flow is determined by the formula:

$$Q_{\text{мах.ч.}} = Q_{\text{мах.сут.}}\ \alpha_{\text{ч}} / 24, \tag{9.2}$$

where $\alpha_{\text{ч}}$ – coefficient of hourly unevenness, $\alpha_{\text{ч}}$ =2,3.

The second water flow is equal to:

$$Q_c = Q_{\text{мах.ч.}} / 3600,\ \text{l/s}, \tag{9.3}$$

The daily flow rate of the pumping station must be equal to the maximum daily water flow in the room or farm, and the hourly consumption of the station (pump) is determined by the formula:

$$Q_{\text{нас.}} = Q_{\text{мах.сут}} / \tau, \tag{9.4}$$

where τ – the duration of the pump or station per day, h. Based on comparative technical and economic calculations, the operating time of the pumping station is assumed to be 7 or 14 hours. According to the value of $Q_{\text{нас}}$ and the required head is selected according to the operating characteristics of the type and grade of the pump. The required power of the electric motor for driving the pump is determined by the formula:

$$N = (Q_{\text{нас}}\ \rho\ H\ K_з\ g) / (\eta_п\ \eta_н), \tag{9.5}$$

where $Q_{\text{нас.}}$ – volume flow of water (pump feed), m^3/s; ρ – density of water, kg/m^3; H – full pump head, m (taken from the technical specification); $K_з$ – power factor that takes into account overload capabilities during pump operation; g – acceleration of free fall, m/s^2; $\eta_н$ – efficiency pump according to the technical specification; $\eta_п$ – efficiency transfer from engine to pump.

Chapter 10
MECHANIZATION OF MANURE REMOVAL AND STORAGE

10.1 Physical and mechanical properties and methods of manure management

In livestock raising two methods of keeping animals are used– on litter and without litter – are of greatest use. Large–scale farms are spreading the cotton–free method of keeping animals. It is less labor–intensive, since it allows the use of complex mechanization and automation of works related to manure removal from production facilities. With this content of animals, liquid (semi–liquid) manure is obtained.

Unmodified (pure) manure is very homogeneous in composition. The average particle size of pure cattle manure is 2,6 mm, particles greater than 10 mm long contain no more than 1 %.

On small farms of cattle, the maintenance of animals on the litter is common. In this case, solid (dense) manure is obtained. The litter absorbs liquid excretions of animals and formed nitrogen, improves the physical–chemical and biological properties of manure, which becomes less moist, more friable, easier to decompose during storage. In the presence of litter, the floor of the stall is more even, warm and clean, the transport of manure is facilitated, and its introduction into the soil is made easier.

Different types of litter absorb an unequal amount of liquid. So, straw, sawdust and shredded shavings absorb water in an amount 2...3 times their mass (at a humidity of 10…14 %), and dry up peat – 5...7 times.

The approximate norms of the daily expenditure of litter per animal are given in Table. 10.1. When calculating machines for manure cleaning, it is necessary to know the values of the viscosity and the ultimate shear stress.

Table 10.1 Norms of daily bedding consumption per animal

Method of content	Litter, kg	
	Dry straw	peat
Binding:		
Dairy cows	2,2...3	2...3
Young growths	1,5...2	1,2...2
Calves one year old	1,2...1,8	1...1,5
Tethered and free–walking:		
Cows	4...4	3...5
Young	3...5	2...4

The viscosity of liquid manure, like the ultimate shear stress, increases with decreasing humidity. Thus, when the moisture content of cattle manure is reduced from 94 to 82 % viscosity increases from 0,13 to 2,6 Pa, and shear stress – from 1,5 to 100 Pa.

For the calculation of machinery, a characteristic of the physical and mechanical properties of manure is required (table 10.2).

Table 10.2 Physical and mechanical properties of manure

Indicators	Значения
Density of straw manure, kg / m^3: At a humidity of 75 to 85 % From 0 to 20 %	 530...890 1010...4700
Density of liquid manure, kg/m^3	1010...1020
Coefficient of sliding friction of non–pigmented manure at critical humidity: 64,4 %, according to steel 67,6 %, for concrete 0,4 % by the board of pine	 0,9 1,04 1,02;
Coefficient of sliding friction during the movement of manure with straw litter at a critical humidity: 71,4 %, according to steel 73,4 %, for concrete 72.8 % on the board of pine	 0,67 0,68 0,77.

The main influence on the properties of manure is moisture, the value of which is due to the adopted system for its removal. Thus, the moisture content of cattle manure during mechanical removal and litter content is 75...90 %, in case of a non–woven cloth – 88...95 %, for a gravity system – 94...96 % and for a flushing system – 96...98 %.

Cleaning of stalls and replacement of bedding is carried out in the morning and in the evening (before milking). Within a day the yield of manure is uneven. More than 30 % of the daily yield in cows is observed at the time of feeding. In the stall period excreta allocation in them occurs up to 10...15 times a day.

During stall–pasture maintenance of animals the yield of excrement in the pasture period should be taken in the amount of 50 %, and when walking – 85 % of the calculated value. In approximate calculations, the weight of excrement can be considered equal in dairy cows on average 8 % of the animal's weight.

The daily yield of manure can be determined by formula

$$Q_{cym} = \sum_{i=1}^{n} g_i \cdot m_i, \tag{10.1}$$

where g_i – the rate of manure from 1 head. With approximate calculations, the weight of excrement can be considered equal in milk cows on average 8 % of the animal's weight; – the number of animals in the room.

When the animals are kept in an animal bedding mode, the manure from the stalls is removed 2 to 3 times a day. When keeping livestock on deep bedding, 2...3 times a year.

By designation, manure harvesting is divided into:

– cleaning means for premises;

– means of manure accumulation and disposal;

– means of transporting it and processing for the purpose of subsequent disposal.

Cleaning manure from livestock housing can be carried out in two ways – mechanical and hydraulic.

10.2 Mechanical means of manure removal

With the mechanical method of manure cleaning, scraper conveyors and mobile means are used.

Scraper conveyors include:

1 Chain–scraper conveyor for removal manure by circular action (it is designed to remove manure from livestock premises with caged cows and the simultaneous loading of manure into vehicles (fig. 10.1)). It consists of horizontal 1 and inclined 2 conveyors, which are activated by means of individual electric drives 4 and 3. The horizontal conveyor is installed in manure ditches located along rows of stalls in cattle rooms or inside. Manure in manure ditches is discharged by operators manually with special scrapers.

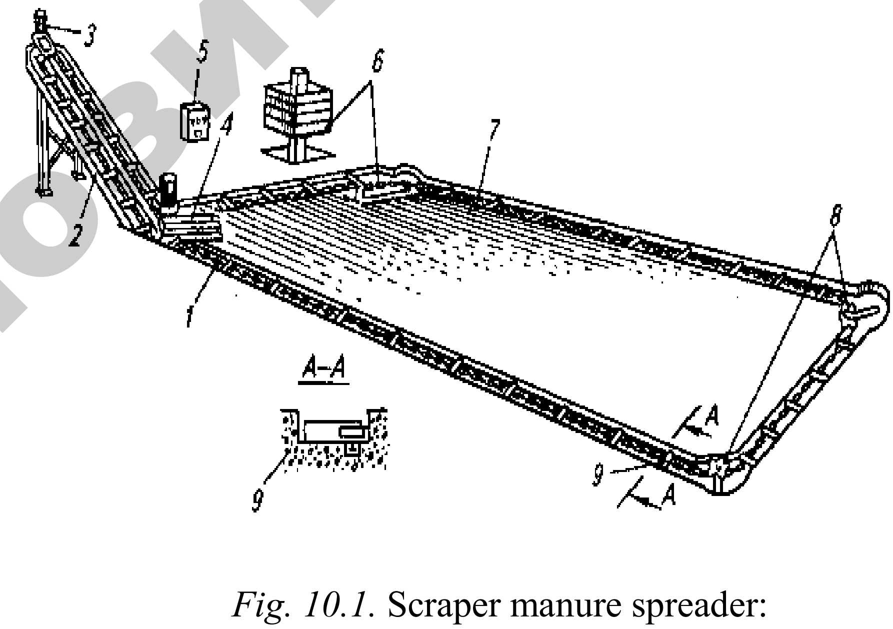


Fig. 10.1. Scraper manure spreader:
1 – horizontal conveyor; 2 – inclined conveyor; 3 – drive of inclined conveyor; 4 – drive of horizontal conveyor; 5 – control cabinet; 6 – tension device; 7 – chain; 8 – rotary devices; 9 – scraper

The horizontal conveyor consists of:

– a drive station 4 including an electric motor, a two–stage reduction gear, a V–belt drive and a drive sprocket,

– closed circular chain of chain 7 anchor type, with fixed on it with a step of 1,12 m scrapers 9;

– the self–tensioning device of the chain 6, which provides automatic tension of the chain and in a timely manner compensates for its stretching and out–noses;

– two rotary devices 8.

The inclined conveyor 2 is a metal chute that is supported by one end on the stand, and the other is buried in the pit. In the center of the gutter is an anchor type chain with scrapers placed in 0,46 m increments. A turn and tension device is provided at the ends of the chute. The movement of the chain of the inclined conveyor is ensured by an individual electric drive 3, consisting of an electric motor and a two–stage reducer. Install the inclined conveyor in a separate room at an angle of no more than 30° to the horizon, which makes it possible to supply manure to a height of 2,7 m from the zero mark of the floor of the barn.

When removing manure, the first include an inclined conveyor and then a horizontal conveyor. The scrapers of the horizontal conveyor catch manure and move along the bottom of the manure channel to the place where manure is dumped onto the inclined conveyor.

By means of a chain with scrapers of the inclined conveyor, the manure moves up its trough and is discharged into the vehicle. The speed of the chain of the inclined conveyor is much higher than the horizontal, which is necessary to ensure the discharge of liquid manure.

Feed or capacity of the horizontal conveyor:

$$W_{тр} = b_с \cdot h_с \cdot V_{тр} \cdot \rho_{экс} \cdot \phi_н , \tag{10.2}$$

where $\varphi_н$ is the coefficient of filling of inter–scraper space, $\varphi_н$ = 0,75...0,9; b_c, h_c – respectively, the length and height of the scraper, m; $V_{тр}$ – mean speed of the scraper, $V_{тр}$ = 0,72 m/s; $\rho_{экс}$ – density of manure, kg/m^3.

Duration of the transporter per day:

$$\tau_р = \frac{Q_{сут}}{W_{тр}} , \tag{10.3}$$

where $Q_{сут}$ is the daily yield of manure from a particular livestock house, kg.

Number of transporter starts per day:

$$K_{вкл} = \frac{Q_{сут}}{b_к h_к l \rho_{экс} \varphi_н}, \qquad (10.4)$$

where $b_к, h_к$ – the width and height of the canal, respectively, m; l – length of the canal, m.

Scraping machines with reciprocating movement of working bodies, called deltacrepers, ensure the mechanical transportation of manure from livestock premises and its supply by means of special transverse manure conveyors to manure collectors or a vehicle.

Depending on the type of traction body, reciprocating conveyors are divided into two types: (rod and rope) and – according to the location of the suspension axis of the scrapers – into two groups:

1) with a vertically located axis, when the scrapers unfold in a horizontal plane parallel to the bottom of the trough;

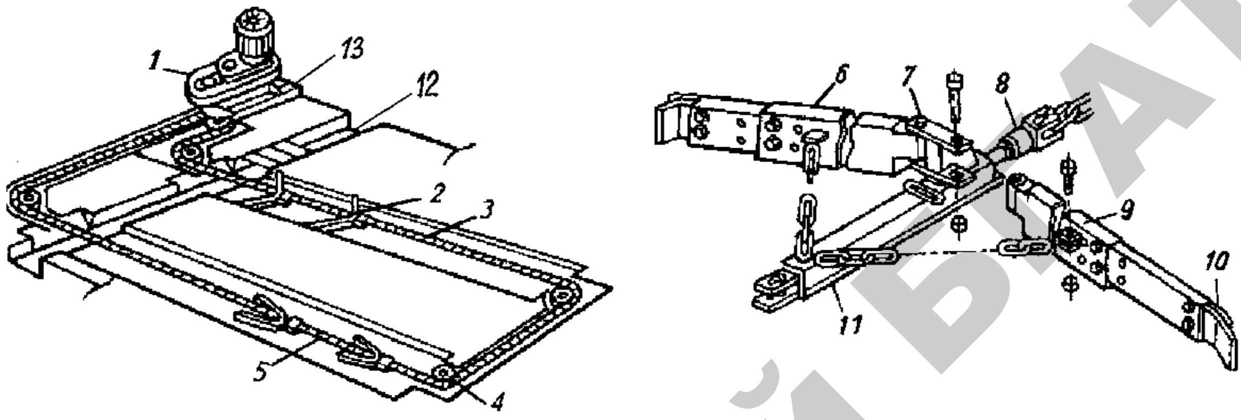
2) with a horizontally located axis, when the scrapers unfold in the longitudinal–vertical plane.

The most common conveyors belong to the first group.

A scraping unit with a vertical axis (fig. 10.2) is designed to remove the pig–laying manure from open longitudinal passages in cowsheds up to 80 m long with box and combi–box animals. Installation can work in two modes – manual and automatic. Device: an electric drive 1 is installed on the frame 13, which includes an electric motor with a power of 2,2 kW, a reducer, on the drive sprocket of which the circuit 3 of the working circuit is fixed (fig. 10.2, a). Four delta scrapers are fastened to the chain. There are rotary devices 4 at the corners of the chain circuit. There is a reversing mechanism designed to automatically reverse the drive motor in order to ensure the reciprocating motion of the chain circuit. The reversing mechanism is operated by a fixed stop on the chain.

The scraper is designed to move manure through the channel. It consists (fig. 10.2, b) of the slide 11, the hinge device 7, the scrapers 6, 9 and the tensioning device 8. To clean the passage walls, rubber cleaners are installed at the ends of the scrapers 10. The wipers also provide a silent run of the scrapers. The unit operates in automatic mode with the reciprocal motion of delta scrapers. When turning on the scraper moves at a speed of 0,063 m/s, so do not bother animals, that is, manure cleaning is possible in their presence. If, for one pass, the first pair of scrapers moves toward the transverse channel 12, then their scrapers, due to frictional forces on the floor, unfold and move the manure.

On the other pass, the second pair of scrapers idle in the folded state and in the direction opposite from the transverse channel. After dropping the manure into the transverse channel, the movement reverses, and the work cycle is repeated with the scrapers of the other pair of scrapers opened. Manure from the barn is fed into the receiving funnel of the pump, which moves the manure through a pipeline to the manure storage.



a – general view

b – Scraper

Fig. 10.2. Scraper installation:
1 – electric drive; 2 – scraper; 3 – working contour; 4, 8 – respectively rotary and tension devices; 5 – intermediate rod; 6, 9 – scrapers; 7 – hinge; 10 – rubber cleaner; 11 – slider; 12 – transverse channel; 13 – frame

Installation serves cows located in two group machines. For two– and four–row arrangement of cow stalls, a manure–laying installation is used (fig. 10,3), which consists of two horizontal rod conveyors of reciprocating motion. Such conveyors have advantages over scraper conveyors of circular motion. Thanks to the reciprocating movement of the rod, manure is delivered to the place of discharge in the shortest possible way. In the absence of guiding blocks and sprockets on the way of movement of manure, the operational reliability increases. By means of the guides and the rigid rod, lifting of the scrapers is prevented and the conveyor is stable.

The calculation of rod scraper conveyors is reduced to determining the stroke of the rod, the pitch of the scrapers and their dimensions. The values of the feed and the required power are calculated by analogy with chain–scraper conveyors. The rod run *S* should allow the scraper to free turn into the working position after it has passed by the manure portion left by the previous scraper.

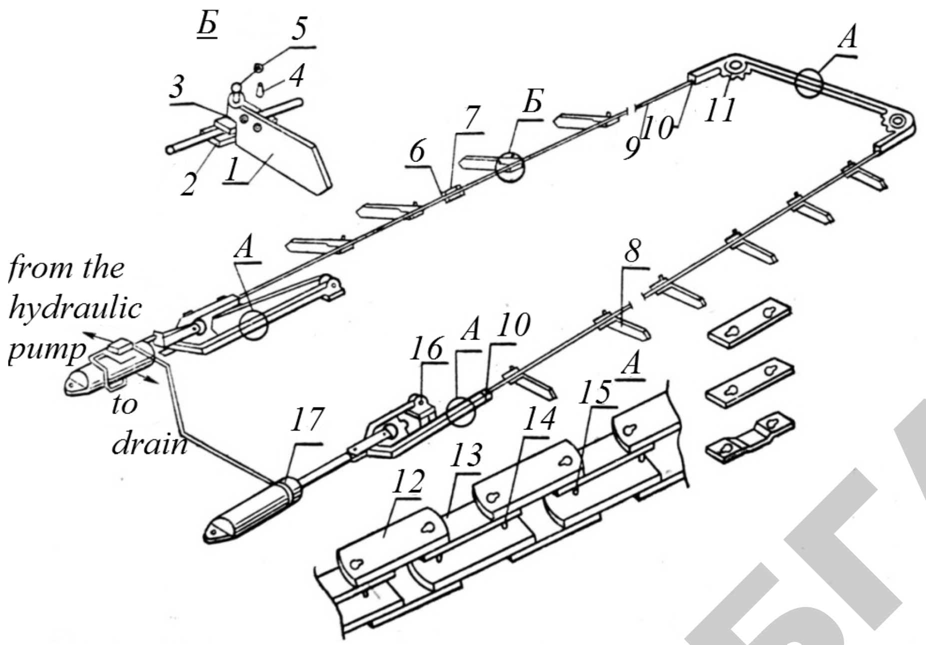


Fig. 10.3. Transporter of reciprocating motion:
1 – scraper blade; 2 – thrust bearing; 3, 6 – lock nuts; 4 – support; 5 – lock ring; 7 – coupling; 8 – right scraper; 9 – rod; 10 – adapter; 11 – rotary device; 12 – bar; 13 – support bar; 14.15 – fingers; 16 – bracket; 17 – hydraulic cylinders

Consequently, the condition

$$S \geq \Delta l, \tag{10.5}$$

where Δl – is the path length of the rod, on which the scraper passes from idle position to working position, m.

The following types of conveyor drives are known: chain–link, chain–crank–conrod, hydraulic and with the reversal of the electric motor. According to the results of the research the most reliable is the chain drive. For its operation, the following condition is necessary:

$$S = A + 2r_H \geq t_c + B \tag{10.6}$$

where A – is the distance between the center of the guide and the guide sprocket, m; r_H – is the radius of the initial circle of the asterisks, m; t_c – is the distance between the scrapers (step), m; B – is the length of the scraper, m.

For a crank–crank drive, we write this condition as follows:

$$S = 2r_к \tag{10.7}$$

where $r_к$ – is the radius of the crank, m.

The main factor determining the length of the rod's path on which the scraper passes from idle to working position is the angle of inclination of the scraper to the bar when idling. To determine the optimum value of the angle of the scraper relative to the rod, it is necessary to determine its influence on the feed of the conveyor (fig. 10.4).

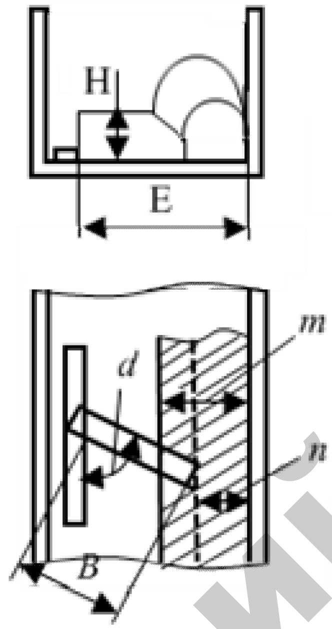


Fig. 10.4. Angle effect between scraper and rod on the formation process of the dragging body (calculation scheme)

To do this, it should be borne in mind that the cross–sectional area of the body formed by the scraper during idling depends mainly on the average value of the base of the drawing body, that is, $m_{cp} = (m + n)/2$. The working width of the conveyor chute is $E = B sin\alpha + m_{cp}$, hence:

$$m_{cp} = E – B\ sin\alpha, \qquad (10.8)$$

where α – is the angle of inclination of the scraper to the rod at idle, deg; B – length of the scraper, m.

It can be seen from expression (10.8) that to increase mcp, and ultimately the load–carrying capacity of the scraper can be achieved by decreasing the angle α, the optimum value of which lies in the range of 17…20 °. The step of the scrapers should be longer than the length of the lower base of the dragging body, i.e. $t_c \geq l2$ (where l2 is the length of the lower base of the drawing body in front of the scraper).

To select the step of the scrapers, refer to fig. 10.5, which makes it possible to establish the dependence of this quantity on the length 12 of the lower base of the drawing body before the scraper. As a result, we get:

$$l_2 = l_1 + h/tg\varphi, \tag{10.9}$$

where l_1 – length of upper base of dragging body, m; h – is the height of dragging body**'s** layer, m; φ – is the angle of natural slope of moving loads, deg.

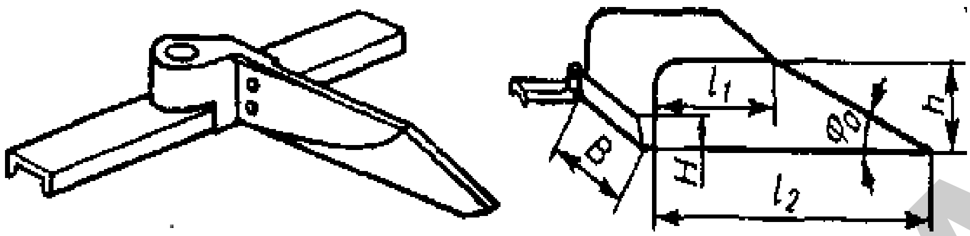


Fig. 10.5. The design scheme of the wire drawing of bonded cargo (before scraper)

Besides scraper–rod**'s** fitting angle α and pitch tc, shape and clearance between scraper and side walls also exert great influence on scraper**'s** conveying ability. The greatest transporting ability is possessed by scrapers with the ratio of height to its length *N/V*, i.e *H/B* = 0.2...0.25, where *H* – is the height of the scraper, m. Wide, but shallow ditch is preferable, since in this case the resistance of movingto manure will be more less.

As the *H/B* ratio increases, the friction force of the manure against the side wall of the ditch increases. To eliminate the pinching of bonded weights between the scraper and the wall at the time of the scraper's turn, the end of the scraper must be cut at an angle to its base. The amount of pinching angle between the side wall of the ditch and the scraper tip should be greater than the sum of the friction angles of the manure against the ditch wall and the scraper. The speed of movement of the conveyor scrapers varies within the limits of 0,2...0,4 m/s.

The disadvantage of the conveyors with direct drive to one boom is that, during the working stroke, it is subject to stretching, and when idling, it is compressed. More reliable work is provided in those cases when the bar works only on stretching. This requirement is fulfilled with a cable drive from the drum, chain or plunger of the hydraulic cylinder.

Mobile devices for harvesting manure on farms include hinged and trailed devices for tractors. The mobile unit is a tractor with a bulldozer attachment. Such an aggregate is used to remove manure from open manure passages of cattle–breeding premises for cattle and feed it into a transverse canal located inside the room or to push manure into a storage facility located near the farm.

When manure is cleaned with a bulldozer from premises for boxing or combi–box maintenance of animals, the manure pass must be in the form of a rectangular tray with a width of at least 2200 mm and a depth of 200 mm. If it is used in premises for tethered cows, the passage is made in the form of two grooves with a depth of 150...200 mm and a width of 550 mm with a distance between them of 1100 mm. The total width of the passage must be 2200 mm. The bulldozer shovel must match the shape of the canal. In the middle of it there is a pivotally mounted scraper 1100 mm wide. When using a bulldozer, the floor of the passages must be monolithic of concrete not lower than grade 200 and a thickness of at least 180 mm with a slope of 0,5 % in the direction of transport of manure.

The tractor with a mounted scraper performance is determined (with some approximation) by the amount of machine time on removing 1000 kg of manure, according to formula

$$t_б=1000\, l_б / q_б \cdot \upsilon_б, \qquad (10.10)$$

where $t_б$ – is time spent on removing 1000 kg of manure from a barn by bulldozer's mounting, c; $l_б$ – average length of manure transport path, m; $q_б$– amount of manure, harvested in bulldozer working stroke, kg; $\upsilon_б$ – average working speed of the tractor with bulldozer's mounting, m/s.

It should be noted that the ratio $1000/q_б$ determines the number of flights to be completed for harvesting 1000 kg of manure. Resistance to the movement of manure, moved by a tractor with mounted scraper on a walking platform with a hard surface or in a manure pass of a barn, can be calculated from the dependence

$$P=9{,}81\, k_б f_{cm}\, M, \qquad (10.11)$$

where $k_б$ – is a coefficient that takes into account the angle of the scraper setting (table 10.3); f_{cm} – coefficient of static friction; M – is the mass of the dragging body, kg.

Table 10.3 Values of the coefficient $k_б$

Dung	Position angle of working element, degree			
	0	15	30	45
Strawing	1	0,85	0,75	0,65
Peating	1	0,95	0,85	0,70
Excrement	1	0,95	0,90	0,80

The work of the bulldozer is in many respects similar to that of a forklift truck. Values of the rated carrying capacity of the latter with a bucket for bulk materials, depending on the traction class of the tractor are given below. The productivity of the bulldozer weigher can be calculated from the formula

$$Q = P\,n, \tag{10.12}$$

where P – is the load capacity of the bulldozer, kN; n – number of working cycles of the loader for 1 hour.

It can also be represented as:

$$Q = \frac{V\rho\psi_{зал}360}{t_ц} \tag{10.13}$$

where V – bucket capacity, m^3; ρ – bulk density of manure, t/m^3; $\Psi_{зал}$ – coefficient of filling the bucket (usually $\Psi_{зал}$ = 0,5...0,9); $t_ц$ – is the cycle time, including the time taken to scoop up the load, turn the tractor, switch gears and unload the load from the bucket (as a rule, determined experimentally), p.

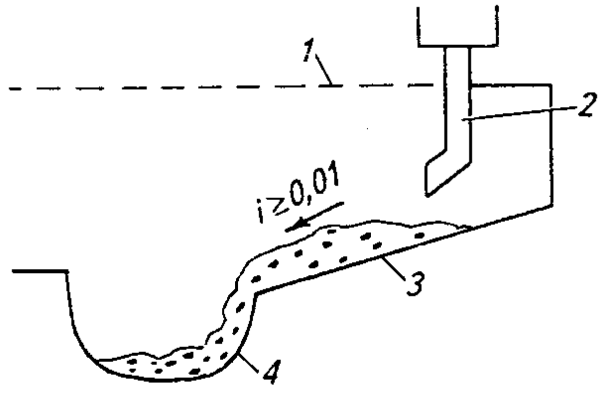
The weight of the drawing body in front of the scraper will depend on the length of the track, the width of the aggregate gripping and the frequency of manure removal from the sites. Mobile means for collecting litter manure are used both for tied and loose–fitting maintenance. In order to avoid cooling rooms, the wagon type entry gate creates protective air curtains with air intake from the middle part of the room.

10.3 Hydraulic manure removal systems

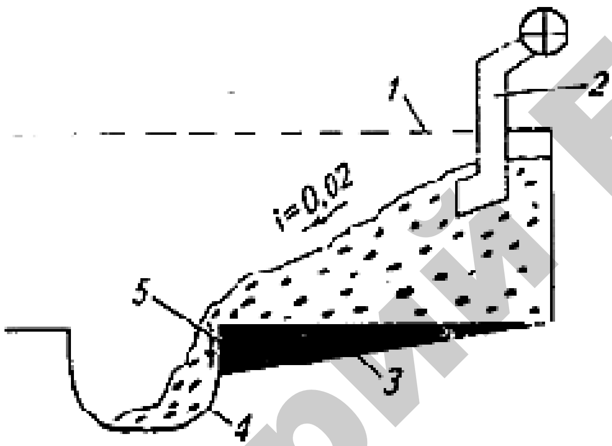
Hydraulic manure removal systems are used for indoors manure cleaning at livestock farms. There are the following types of hydraulic manure removal systems:

Gravity continuously operating system is designed for removal of manure from the barns at liquid animal maintenance and is based on the principle of self–moving mixture of excrement. The system operates continuously, upon receipt of manure mass through cracks of channel gratings and its run–off through the open end of the channel (fig. 10.6). Manure movement occurs through the channels under the influence of gravitational forces (manure flows through the channel under the effect of the slope).

Manure flows along the channel under the influence of external (coercive) power (manure runoff in the channel by water flow) (fig. 10.6, a).



a



b

Fig. 10.6. – Hydraulic manure systems:

a) plan of the flushing continuously operating system;

b) gravity flowing continuously operating system for remove dung:

1 – slatted floors; 2 – flushing system; 3, 4 – longitudinal and transverse channels; 5 – sill; 6 – sliding

Manure layer thickness increases along the channel in the direction opposite to its motion. Manure mixture moves at a certain angle to the channel bottom. With overpressure generated by the difference of layer thickness, there is a force that moves the manure through the channel. Manure mixture continuously flows out of the channel. Mixture flow rate is small (1...2 m/h), and its movement is barely noticeable.

Gravity continuously operating manure system (fig. 10.7) consists of longitudinal channels taking manure 3, which are covered by slatted floors 7, intra–farm pumping station (not shown), the flush water pipe for recycled water 1 and the main reservoir 6. The nut 4 is made at the junction of longitudinal channels to transverse of height 100...150 mm. When starting the system longitudinal channel is pre–filled with water from pipe–line 1 to the height of the nut.

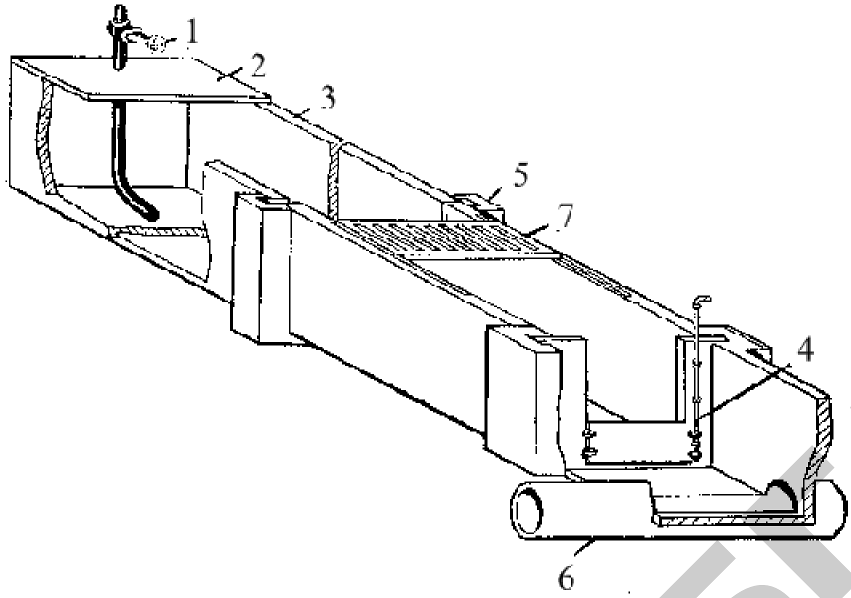


Fig. 10.7. The device of longitudinal manurereceivering channel for remove dung within gravity flowing continuously operating system: 1 – flushing water; 2 – a plate; 3 – manurereceivering channel; 4 – nut; 5 – coupling; 6 – main collector; 7 – grill

The workflow of the gravity continuously operating system based on self–movement of excrement mixture, i.e. using the viscoplastic properties of liquid manure. The system operates continuously as the mass comes through the cracks of channel gratings and runs off through the open end of the channel in the overall cross–collector 6. The thickness of the manure layer increases during the length of the channel in the direction opposite to its motion.

Under the influence of overpressure generated by the difference in layer thickness, is made a force moving the manure down the channel. Manure mixture continuously flows out of the channel at a very low speed and its movement is barely noticeable.

Gravity periodic operating system (fig. 10.8) is arranged similarly to a continuous system, but there are differences, including:

– manure taking channel 3 is done with bottom slope of 0,005...0,007. This slope is done in order to ensure cleaning (washing) of the channel. Having the steep slope bottom, the liquid excrement (urine) would quickly ran down, and feces remain in the channel;

– at the end of the channel is set manure taking throttle gate 5;

– at the end of the channel is set manure taking reinforced concrete bulkhead 6 in order to prevent contact between adjacent manure taking channels and eliminate drafts and prevent harmful gases in animal houses.

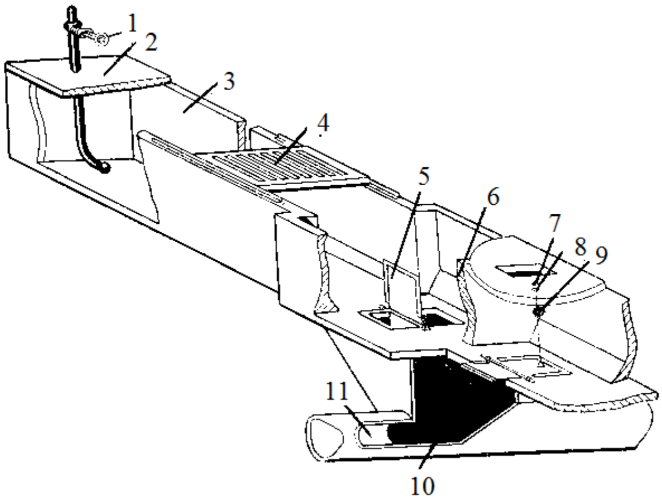


Fig. 10.8. The device of manurereceivering channel
for remove dung within gravity flowing periodical operating system:
1 – flushing water; 2 – plate; 3 – manure taking channel; 4 – model;
5 – throttle damper; 6 – bulkhead; 7 – bracket; 8 – handle; 9 – ring;
10 – downshaft; 11 – main collector

Flap of the throttle gate in a vertical position is raised with help of a cable or rod, and is lowered by gravity. In the working process animal excrement, falling through the gratings 4, are accumulated in manure taking channels 3 according to the sanitary requirements till the level at the head part not less 0,3 m till the lower surface of the grating floor. The storage period is 7...14 days depending on the species of animals, feed ration and season. When the channel is full, the gate is opened and the accumulated manure is let out, thus the Gravity system is launched. Remaining at the nut level in the channel layer of manure is displaced by entering to the channel fresh manure mass.

For periodic cleaning manurereceivering channels from dung and its sludge as and within gravity continuously operating system recycled water is fed into the flushing water supply. This system is effective for liquid manure and requires minimal labor. The main disadvantages of this system – increased water consumption and a significant release of hydrogen sulphide during the dumping of the manure mass, which worsens the microclimate. Manure channel depth depends on the height of a layer of manure at which it starts to flow. The minimum (initial) manure flow depth at which possible the movement of visco–plastic mass through the channel of unit width can be determined by the formula (10.9)

$$h_0 = \sqrt{2\tau_0 L_K / (\rho g)}$$

where τ_0 – limiting shear stress, Pa; L_K – channel length, m; ρ – density of the manure mass, kg/м^3; g – acceleration of gravity, m/c^2.

The initial and final gravity depth of the channel (fig 10.10.):

$$H_{Н.К.} = \Delta h + h_0 + h_{сл} + h_{рез} \quad (10.14)$$

$$H_{К.К.} = h_{пор} + h_{сл} + h_{рез} + h_0$$

where Δh – the exceeding height of the nut over the bottom of the channel in the initial part, m, $\Delta h = h_{nut} - z$, here h_{nut} – the height of the nut, m. It is usually taken $\Delta h = 0{,}05 \ldots 0{,}1$ m; z– difference between the marks of the beginning and end of the channel, m; h_{layer} – the thickness of the liquid layer over the nut, $h_{\text{layer}} = 0{,}05...0{,}1$ m; h_{res} – reserve channel depth, m, ie, the minimum distance from the highest level of the mass at the beginning of the channel till the grid floor ($h_{\text{res}} = 0{,}3...0.35$ m).

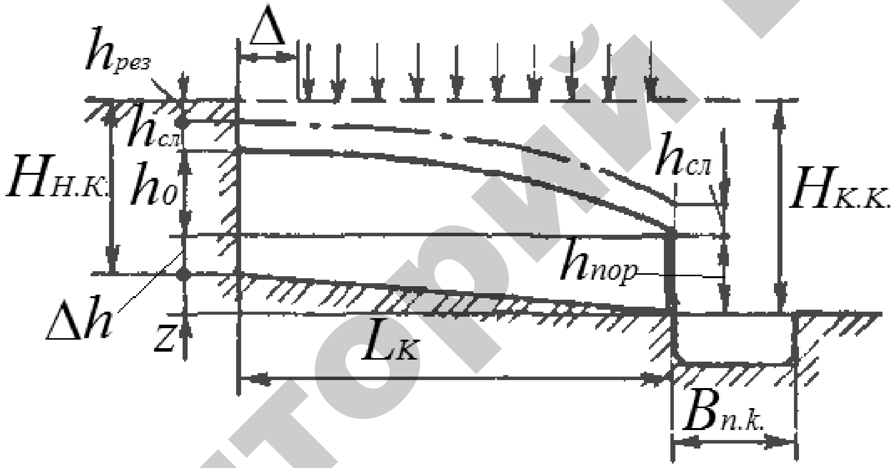


Fig. 10.9. Estimated manure removal channel scheme

Direct water wash manure system includes longitudinal channels with the slope of $0{,}007^{o}...0{,}01^{o}$ and cross channels – with a slope of $0{,}02^{o}...0{,}03^{o}$. Outside the livestock buildings and on the site to the receiving tank cross channels are replaced by the pipes. To remove and transport the manure mass industrial water is supplied under pressure 0,2...0,3 MPa. For one volume of excrement 6...10 volumes of water is spent.

Water wash method has some disadvantages:

– extensive water usage;

– the need to have very large storage tanks for liquefied manure;

– high cost of wastewater treatment plants.However, this method can be fast enough to remove manure from livestock buildings, what almost fully meets the requirements of the veterinarian.

10.4 The technological schemes of transportation and storage of manure

To deliver manure from livestock buildings to manure storage the following devices are used:

– tractor truck, which is installed in the manure vestibule (manure get into truck through an inclined line of the dung remover or scraper equipment) and with filling transported to a storage tank, where it is unloaded;

– bucket conveyors. Special insulated manure pit cleaners are constructed with a capacity of up to daily manure mass output, which is overloaded into vehicles with help of the the slatted apron or bucket manure loader. Stationary manure loader consists of a frame, a leading and tensioning rollers, chains with buckets, suspension and electric drive, which is located at the upper end of the loader frame. While the loaded strand moving, the buckets capture manure mass bottom–up and move. When passing through the upper drive sprockets they overturn and dump the contents into the vehicle. The conveyor is made operational by an electric motor through a gearbox and chain drive. Maximum angle of the conveyor inclination is 63 °, flow up is to 30 m^3/h;

– pneumatic installation. Liquid manure is discharged from the manure pit with the help of the pneumatic installation, evacuated tanks of screw pump and others. When using conveyor it is possible to mechanize the process of manure applying (85 % moisture and higher) to a storage tank at a distance till 500 m with a full exception of mobile machinery and manual labor. Manure coming from internal dung removers enters the manure pit, which then is tightly closed. By using compressed air it is fed through the distributor to manure pipe–line and then to dung–yard;

– scraper equipment. Scraper equipment can be used for the manure transportation of coming with dung remover, if the dung–yard is located at a distance of no more than 50 m from the barn;

– pumping stations. *The main assembly* units: piston pump, hydraulic station, manure pipe–line and control cabinet. When equipment works the manure is supplied under the influence of its own weight into the working chamber because of vacuum created by the pump. After its filling a valve closes hopper window and opens the discharge valve of manure pipe–line. The piston pump, making a stroke, pushes the manure from the working cylinder through the manure pipe–line into the storage. Manure pipe–line is laid underground below the frost penetration. The air intake chamber is brought into action by the water cylinder. Herewith the straw materials are easily cut, thereby ensuring safe transportation of solid manure along the pipeline. For reliable operation of the equipment, the manure humidity should be at least 76 % and the cutting length of the dunnage – no more than 10 cm.

– gravity manure taking system directly into the store, located at the distance more than 100 m from the livestock buildings, is useful only in case of good feature, providing the necessary bias of manure pipe–line.

10.5. Processing and manure storage

Farms must have manure storages. Randomly stored manure can be a source of pollution of the environment. Manure stores must be not less than 50 meters from the farm. They can be underground and surface. In areas with long cold winters manure stores are made closed.

To choose the manure storage one must have an annual output of livestock excrements on the object Q. The required volume of manure storage:

$$V_{н-х} = \frac{Q_{год}}{H_б \cdot \rho_{экс}},$$

where $H_б$ – height; $H_б$ = 1,5...2,5 м.

On farms *with litter animal welfare* and mechanical manure cleaning system its moisture does not exceed 75 %. Solid manure is decontaminated by self–heating in clamps. Prepared at spring and fall manure is taken to the field, and with the help of spreaders of organic fertilizers is applied to the soil. Such technology of recycling is simple, does not require any additional processing of manure mass and does not pose any danger of pollution and contamination of the environment.

In practice, there are two main directions of liquid manure handling with 90...98 % moisture when used as an organic fertilizer:

– processing non–fractionated manure;

– Separation of liquid and solid fractions.

Indecomposable liquid (semi–liquid) manure is treated in two ways – by homogenizing and composting. In case of detection in manure, which is in quarantine tanks, agents of dangerous diseases it is recommended to be disinfected:

– *chemical method.* The chemicals with help of which the manure can be decontaminated include formaldehyde, ammonia, chlorine, sodium hypochlorite;

– *biological method.* Of the biological methods of liquid manure decontamination most effective are intensive oxidation and thermophilic fermentation in digesters;

– *thermal method.* Along with thermophilic fermentation and manure mass disinfection a valuable organic fertilizer is received.

Technological process of homogenization of liquid manure is as follows: from livestock buildings the manure is sent to a separator of mechanical impurities. Passed through the separator the manure is removed into the quarantine tanks, where it stands for 6...7 days to detect infection. Quarantine tanks should be adapted for the disinfectional, chemical or heat treatment of manure. The number of slurry tanks should be not less than two. The capacity of each slurry tank is equal to the ten–day flow of manure with two slurry tanks, five–day flow with four slurry tanks. Under such conditions, a seven–day quarantine period of manure exposure and additional reserve for carrying out, if necessary, of disinfecting manure processing is achieved.

Disinfected manure is supplied by pumps to storage–blenders, where it is kept for 6...7 months for dehelmintization and is periodically homogenized for the purpose of deodorization and prevention of the dense sediment formation at the bottom. After storage at the storage–homogenizers the manure is discharged from them and used as organic fertilizer.

One of the most promising and cost–effective methods of processing, storage and decontamination of manure – is its composting. By fertilizing properties composts are as good as manure, and some of them (eg, peat–manure with ground phosphate) are even better. As a result of composting manure mass becomes loose, that makes it possible to completely mechanize all processes associated with the loading, transport and introduction of compost on the fields with help of commercially available tools.

For composting the solid manure is used (with bedding content of cattle) with humidity about 65 %, undivided liquid manure with humidity 90...92 %, and a solid fraction after manure separation with humidity up to 75 %. The starting materials for the preparation of composts are peat, manure, chopped straw, slurry, wood and other foliage. Composts are prepared as follows. From storage places compostable material is delivered to the crowding pen. Before applying of manure the compostable material is uniformly distributed over the area of the section in the quantity necessary to obtain a mixture of desired humidity and is maintained for 6 days. in order to detect infections. Uninfected or decontaminated manure is thoroughly mixed with compostable material by repeated puddling and moving with help of bulldozer. Mixture Humidity should not exceed 70...75 %, since reliable biothermal decontamination is impossible with greater value.

In the process of manure composting with peat and straw there is temperature up to 65 °C in the organic mass, which provides decontamination of most species of pathogenic microflora, helminth eggs destruction and loss of seed germination of weeds. The content of available nutrition elements for plant rises in organic mass (nitrogen, phosphorus, potassium and others.). Another method of processing, storage and decontamination of liquid manure is to split it into liquid and solid fractions. This reduces storage costs, as solid fraction is looaded at sites with a hard surface and after 2...3 months is used as a fertilizer. Storage can be used for the liquid fraction stores, which aren't necessarily equipped with stirring devices. The liquid manure is separated into fractions by flotation and filtering installations.

Separation by filtration units – forced filtration through a porous membrane capable of retaining the suspended particles and let liquid to pass. The process of filtration occurs under the action of forces:

– mechanical (gravity, inertial force and surface pressure);

– gravity (in drum sieves), inertia (in vibrating screens, vibrofiltrs, centrifuges);

– surface (in filter presses and vacuum filters).

To separate the manure into fractions inertial inclined screens are used. Advantages of such screens – simple device and operation, high reliability performance. Constructions of the screens are fundamentally the same. They consist of a box with the filter partitions, the inertial vibrator, the spring cans and drive arrangement. The screen works as follows. The liquid manure goes by the tray on the top sieve. This ensures uniform distribution of the manure mass along the width of the filter septum. Here gross mechanical inclusions are delivered, which are sent to the dump. Manure purified from coarse inclusions falls onto the lower sieve, where it is filtered, the liquid fraction is drained into the sump and is discharged for further processing.

Sedimentation and flotation separation by units based on the bundle by the deposition of suspended solids under the force field, or separating them from the liquid in the form of sludge. The deposition takes place in the gravitational and inertial fields of mechanical forces.

The gravity separation method on the liquid manure fraction (settling) is based on the settlement of the particulate precipitate under gravity force. To do this, a variety of septic tanks are use: vertical, horizontal, radial. In the process of deposition under the action of inertial forces, in particular centrifugal, centrifuges are used and other settings.

Vertical sumps of continuous action are designed for separation of fluid (not less than 96,5 %) manure in the flow. They serve to deliver the fine particles from the filtrate obtained by machine fractionation of the manure using shakers, vibrating screens, arc sieves, filtering centrifuges and others. Advantages of vertical settling tanks – simple device and easy operation; require less accommodate place; provide a high effect separation (lightning).

Horizontal settlers of batch operation are rectangular with dimensions of the bottom 100 x 25 m and a depth of 2 m. At the bottom of septic tanks in the longitudinal direction there is laid drainage of perforated iron (steel) pipes with holes with a diameter of 16 mm, staggered by 150 mm. Pipes are poured by large gravel. Each drain line at the output ends with a damper located in the well and opened after filling (accumulation) the settler with solid manure fraction. At the end of the settlers the dam gates are placed for clarified liquid discharging.

The sump is filled with manure of humidity 90...92 % for 30...45 days. Predrying (dehydrated to a moisture content of 75%), during which drainage operates, finishes after 45...60 days. The dried solid fraction of manure is unloaded for 30...40 days.

Gravel is discharged from the trench with an excavator with a special bucket and then it is washed in a special unit. For the supply of liquid manure in the settlers the pumps are used. for the sediment discharged from the radial and horizontal tanks sewage pumps are used.

Centrifugal sumps in technological lines of liquid manure processing are used as secondary settling tanks for the separation of the sludge mixture obtained in the process of biological treatment of the liquid fraction in aerotanks and for clarifying (lightening) the liquid fraction. The circular tanks advantages are small depth, providing of high quality lightening. The disadvantage – formed in the radial sump sediment is characterized by high humidity. The disadvantages of circular tanks are bulkiness and capital intensity. Therefore machine methods of deposition of suspended particles are preferred in practice.

10.6. Calculation of the tray gravity manure removal system

When having the tray gravity manure removal system, every row of machines or stalls is provided with a longitudinal channel with a semicircular bottom R = 15 cm and a width of the top 60...70 cm (for pigs) and 70... 80 cm (for cattle).

Manure channel length:

$$L_к = n_i m \cdot b + \Delta L,$$

where n_i – the number of animals staying along the manure channel; b – width of the stall, m; ΔL – length of the channel that goes beyond the stall, m.

The minimum depth h_{min} (m) of the channel at the head part which is required for normal self–floating of mass:

$$h_{min} = (h - z) + h_1 + h_2 + h_3,$$

where h – height of the nut, h = 0,10...0,12 m; z – difference between the beginning and the end of the channel markers, z = (0,005...0,006) $L_к$, m; h_1 – minimum initial flow depth at which possible the movement of visco–plastic manure mass along the channel, h_1 = 0,015 $L_к$, m; h_2 – thickness of the liquid layer over the nut (at manure humidity 86...92 % the value is h_2 = 0,05...0,1 m); h_3 – the minimum distance from the highest level of the mass at the start of the channel to the slotted floor over the canal, h_3 = 0,25...0,35 m.Volume flow of manure channel:

$$Q = 3600 \cdot F \cdot V_{av},$$

where F – cross–sectional area of the mass layer over the nut, m^2; V_{av} – average speed of manure mass in the channel, V_{av} = (8,3...30) 10^{-3} m/s.

The cross–section of the channel is determined by:

$$F = b\ h_2,$$

where b = 0,8...2 m – channel width of gravity system.

Needed volume flow of the channels:

$$Q_n = q\ n_i/(\rho\ \tau),$$

where q – the daily output of liquid manure from one animal (solid, liquid excrement and water for flushing), kg; n_i – number of animals in the livestock premises; ρ – density of the liquid manure; τ – duration of manure hydroremoval lines.

The daily output of manure on the farm is counted using the formula:

$$G_{сут} = n_i\ (q_n+q_м+q_в+q_п),$$

where q_n – the average daily release of solid excrement by one animal, kg; $q_м$ – the average daily release of liquid excrement by one animal, kg; $q_в$ – the average daily consumption of water per flush manure per animal, kg; $q_п$ – the average rate of litter per animal, kg; n_i – the number of animals on the farm.

The annual manure output (tons):

$$G_{год} = 1/1000\ (G_{сут} \cdot \tau_{cm}),$$

where τ_{cm} – the duration of the stall period.

Knowing the daily manure output on the farm of the total population and the duration of its storage, determine the manure storage area (м^2):

$$F_x = (G_{сут}\ Д_{хр}/\rho)/h,$$

where h – the height of manure stacking, h = 1,5...2,5 m; $G_{сут}$ – the daily output of manure on the farm of the total population, kg; $Д_{хр}$ – duration of manure storage at the storage tank, days; ρ– Manure density t/m^3.

Chapter 11
MECHANIZATION OF FARM ANIMALS MILKING

11.1. Physiological basis of machine milking of cows and milking device

Milking process – is the process of the milk removal from the teat in various ways: by calf, by way of manual squeezing or milking. There are manual and machine milking of cows. Manual milking is similar to the natural impact of milk while calf sucking, it doesn't almost injure teats of the udder and there is resting phase, and mastitis does not exceed 2...3,5 %. Manual milking does not require any special tools and is gentler as compared with the machine milking. It is used in private farms, as well as in regions where there are a lot of cows that do not meet the requirements of machine milking. The disadvantage of hand milking is low productivity and large labor intensity.

Machine milking makes work easier, considerably increases its productivity and contributes to the improvement of the sanitary quality of milk. In terms of machine milking it is needed to address the causes hindering the milk ejection reflex, as the response to the stimulus of milking can be interrupted by a nervous state of the animal caused by violation order in the barnyard or sudden external stimulation. The disadvantage of machine milking of cows is mastitis incidence (30 %), the cause of which is primarily the bad cow selection for the technology of machine milking because of the mismatch of the udder parameters.

To minimize the harmful effects of milking machines on the breast they are constantly improved. It should also be taken into account the individual characteristics of cows and their habits. Selection of cows into groups, similar in many ways and, in particular, the type of nervous activity is a must event and provides a more efficient and high–performance use of milking areas.

Milking unit must be:

– simple to manufacture and maintain;

– reliable in work;

– should not require manual adjustment during milking;

– must allow easy visual inspection of the expiration of the udder milk.

There are various types of milking machines, but every milking machine (fig. 11.1.) includes:

– vacuum installation, including vacuum pump 1 with an electric motor, silencer and fuse 2;

– interceptor trap 8;

– vacuum line 3 with a vacuum gauge and vacuum regulator 7;

– milking units 5 connected to the vacuum line via vacuum valves 4.

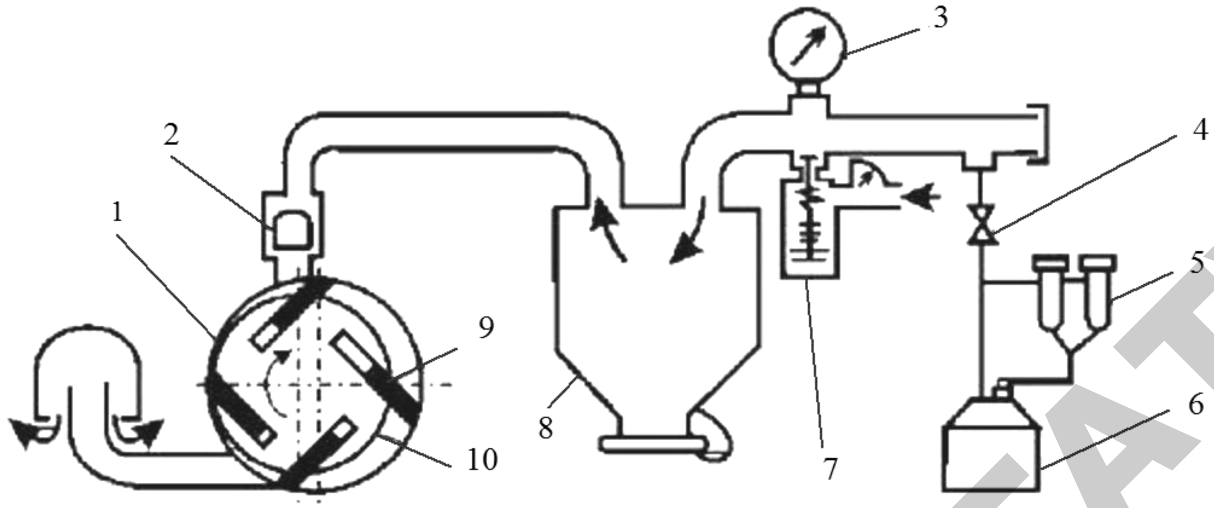


Fig. 11.1. Milking machine scheme:
1 – vacuum pump; 2 – vacuum line; 3 – flow gauge; 4 – vacuum cock;
5 – milking apparatus; 6 – milk pail (bucket); 7 – ragulator; 8 – interceptor trap;
9 – blade; 10 – rotor

The technological process of milking machine operation is as follows. Deficiency generated by a vacuum pump applies through interceptor trap and vacuum–pipe to the milking apparatuses, making the process of milking and collection of milk in the milking bucket 6. When the milking machines and milk line works milk is sucked from the milking apparatus into the milk line through which it is transported into milk storage.

The main controlled parameters during operation of milking machines are:

1. Performance of vacuum units or stations. For individual vacuum units the performance should at least be 30...45 m^3/h.

2. The level and stability of the operating vacuum. The vacuum level must correspond to the nominal working vacuum of used milking machines – 45, 46, 48, 50 kPa.

3. The ratio of the vacuum in the milk line and vacuum line for milking machines with the milk line. The best is the excess of vacuum in the milk line by 1...2 kPa over corresponding values vacuum line. The equality is acceptable. At higher relative vacuum in the vacuum line the glasses fall off, teatcup liner is quickly worn out; milking process becomes dangerous to animal health. The duration of milking one cow should be not more than 7 minutes.

The main working element of milking installation serving for milk extract from the udder of cow and its collecting in a container or milk line is a milking apparatus. It consists of teat cups, collector, pulsator, hoses and connecting nozzles working due to the discharge created by vacuum pump in vacuum line (main pipe) (fig. 11.2).

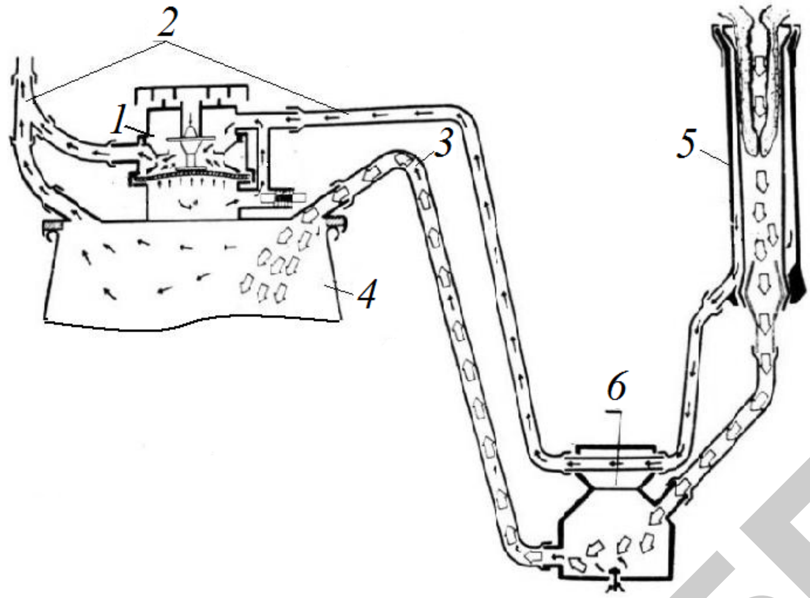


Fig. 11.2. Scheme of the milking machine:
1– pulsator; 2 – vacuum line; 3 – milk hose; 4 – milking pail or milk line;
5– teat cup; 6 – collector

The milking machine must meet the following requirements:

– milking machine should not subject the nipples of the animal to excessive compression;

– does not cause irritation of the animal;

– create a vacuum and ensure that the duration of vacuum phase in accordance with udder pressure and velocity of milk flow;

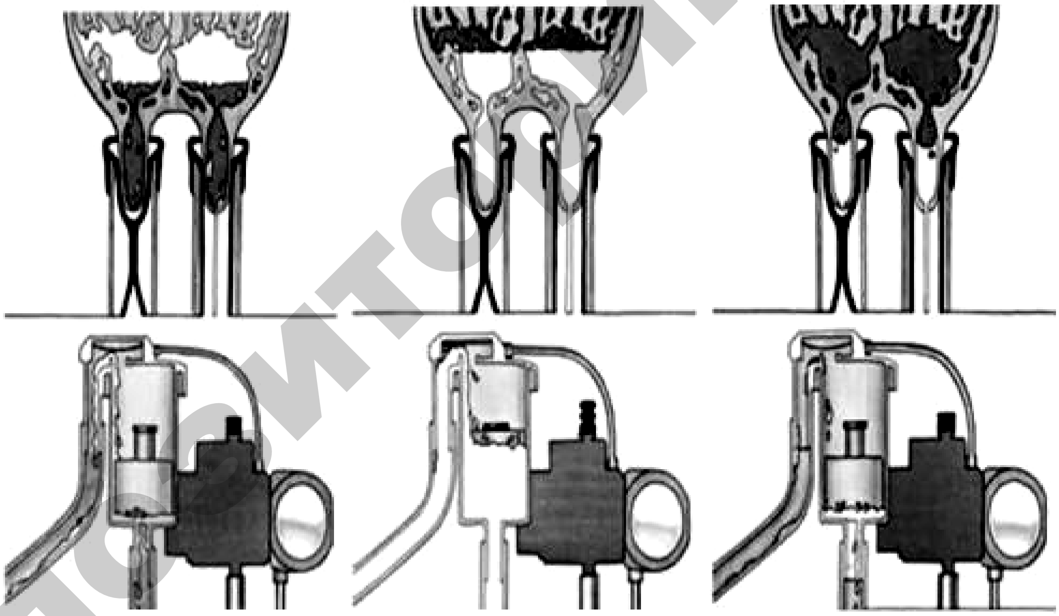
– teat cups do not have to crawl on the udder, or pinch the upper mouth of the teat canal, held in the udder without any special tools.

Three–phase milking machine does milking in three steps: Sucking, compression and relaxation. During the sucking phase in the sucking and the interspace chambers – vacuum and in the interspace – atmospheric pressure, there is a massage of the nipple. During relaxation phase in sucking and interspace chambers is atmospheric pressure. All three steps together constitute a ripple. Three–beat milking machines have a sparing effect on the teats of the udder. During the relaxation phase occurs the same positive pressure, during which the teat tissue are resting, while the circulation is restored, broken by the two previous phases. The longest phase are time sucking – 64 %, the compression phase – 14 %, relaxation phase – 22 % .

Two– phase milking machine does milking in two steps: sucking (67 %) and compression (33 %). In two– phase machines vacuum is always maintained inside teat cups (under cow's teats) during milking. By end of milking teat cup come to the base of the teat. The suction of milk by the machine slows down or stops completely. To resolve this, teat cups must be delayed by collector down. The advantage of the two– phase milking machines is that they milk cows faster. However, permanent negative pressure under the teats has an adverse effect on the udder tissue, especially with their support. In this regard, using the two–phase milking apparatuses increases the likelihood of udder disease mastitis.

Designed by «Alfa–Laval» milking apparatus "Duovak 300" stimulates the cow's udder to mitigate the constraints of animal reflexes (fig. 11.3, a). This function is performed automatically. The milking apparatus operates at low vacuum (250 mm Hg. V.) with slow pulsation (48 pulsations per minute). Thus there is a soft massage of the teats. When the milk stream reaches a certain force (fig. 113, b), the apparatus automatically switches to milking phase. The fast milking occurs under normal vacuum of 380 mm Hg and pulsation frequency of 60 pulsations per minute. Milking lasts only until the stream of milk corresponds to a predetermined level.

With decreasing milk flow below a predetermined level (fig. 11.3, c) the device automatically switches to the low level phase of vacuum and slow pulsation of about 20 seconds. In low vacuum and slow frequency fluctuations in the phase aftermilking the teats are protected from extent milking.



a b c

a – stimulation; b – milking; c – aftermilking

Fig. 11.3. Scheme of the milking apparatus"Duovak 300"

The milking machine «UNICO 1» company «S.A.C.» used on farms with stanchion cattle and milking in the milk line. The puller comes into operation when the milk flow is reduced to 230 ml/min and after a 15 sec of delay. Removing of the device happens in the open position of teat cup liner. It is powered by a rechargeable battery.

11.2 Design and schemes of teat cup

The milking apparatus comprises of four teat cups, which have two walls – an outer made of solid material and an internal – of rubber. At milking time they are put on a cow's udder teats. This produces two chambers – under the teat and between the glass walls around the taet. Through the collector and the pulsator these chambers connected by hoses to the vacuum line and a milking pail or milk line. The most common are two–chamber teat cups consisting (fig. 11.4) of the liner 1 and milking liner 2 with suction cups 3, which form sucking chamber 4 and the space between the walls 5, 6 of the vacuum and milk hoses 7. Sucking chamber 4 is connected to the collector, and through the space between the walls 5, 6 with vacuum port of pulsator.

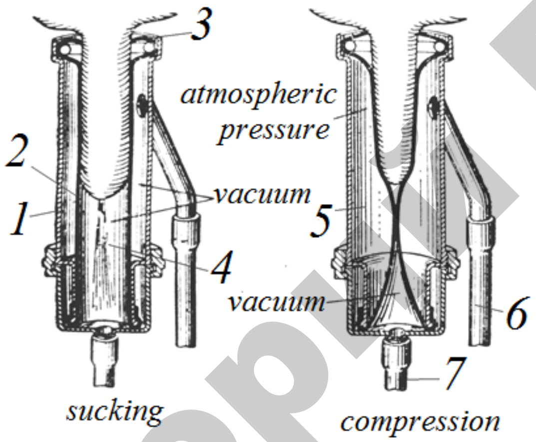


Fig. 11.4. Scheme of the teat cup

In the two–phase milking machines at the sucking phase vacuum flows in sucking chamber 4 and the space between the walls 5. Under the influence of the same distribution effort working vacuum teat cup liner 2 holds a balanced position in the sleeve 1. In the nipple operates vacuum force, which together with the existing udder pressure carries the milking process (sucking rhythm) and milk jetted in sucking chamber 4 and then flows into the reservoir, thence the milk pipe in the milking pail or milk line. As this cycle can not be long, air is supplied to the chamber 5. If the mutual action of the working vacuum in sucking chamber 4 and the air in the space 5 teat cup liner 2 is compressed around the teats compression phase occurs. Both phases together form one cycle. And the sucking rhythm lasts about 2/3 the length of the pulsation. Accordingly, the compression phase is 1/3 of pulsation. Modern milking apparatuses operate in mode 67 ± 5 min pulsation. In this two–phase milking apparatus cycle ends and the new one begins. In three– phase milking machines the third phase is added to the mentioned above when the atmospheric pressure is formed in both chambers (fig. 11.5). This is the third rest phase.

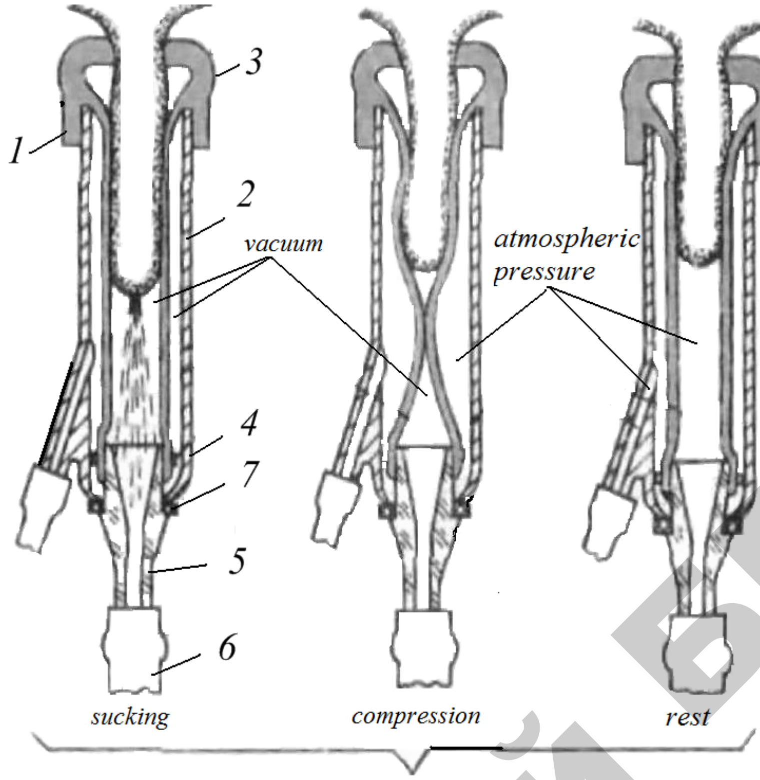


Fig. 11.5. Scheme of teat cup positions during three–phase milking:
1 – rubber sleeve; 2 – glass case; 3 – teatcup liner; 4 – connecting ring;
5 – transparent viewing tube; (cone); 6 – dairy rubber tube; 7 – sealing ring

Four–phase milking machines are developed, in which there are following sequence of phases – compression – Sucking – contraction – rest.

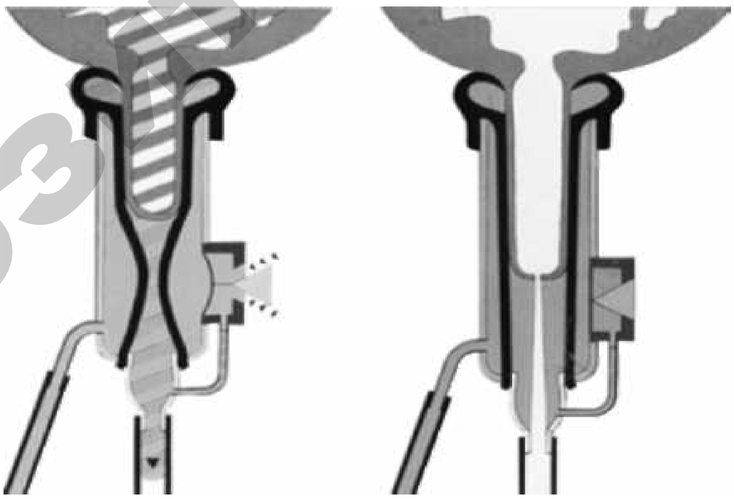


Fig. 11.6. Scheme of teat cup «BIOMILKER» unit

The design feature of the teat cup «BIOMILKER» unit of the company «Westfalia Separator» (fig. 11.6) is to promote the process of milk cows during the compression phase, increasing cow's milking efficiency above 5 %.

11.3 Types and calculation of parameters of the pulsator

Pulsator is designed to convert DC–largest vacuum in the variables needed for the teat cups.

The calculation of the parameters of the pulsator on the example of three–phase pulsator milking machine suggests (fig. 11.7).

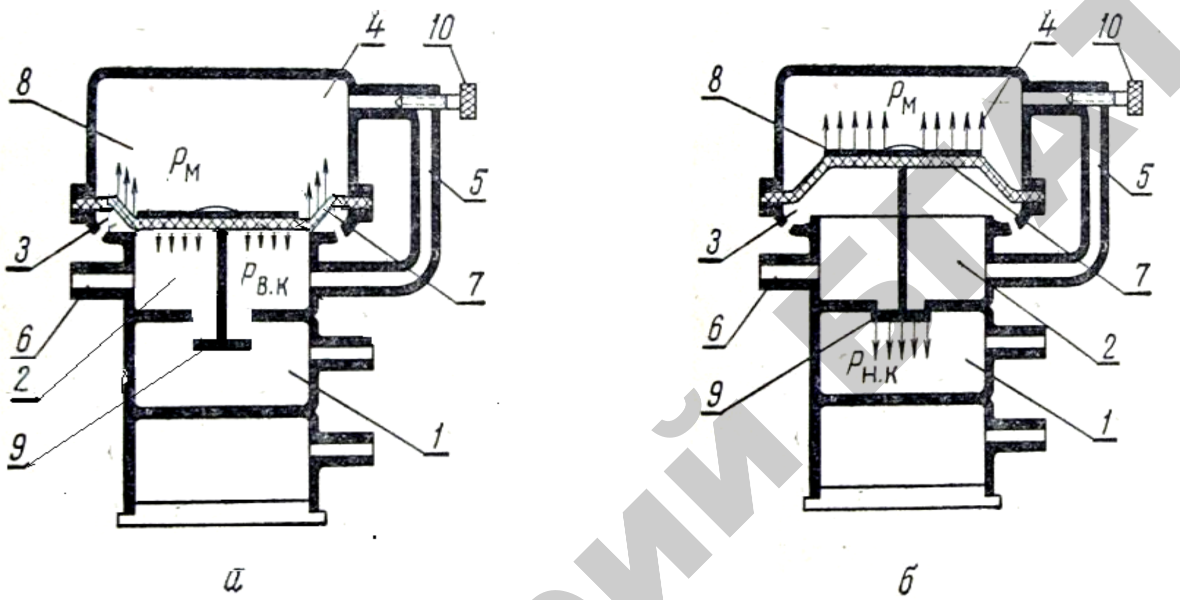


Fig. 11.7. Three–phase pulsator:
a) vacuum in chamber 2; b) atmospheric pressure in chamber 2
1 – constant vacuum chamber; 2 – variable vacuum chamber;
3 – camera with constant atmospheric pressure; 4 – AC vacuum chamber (control);
5 – air channel; 6 – tube; 7– rubber membrane; 8 – washer;
9 – valve; 10 – adjusting screw

The three–phase apparatus has four chambers:

– camera 1 is connected via a permanent vacuum hose to a vacuum line;

– alternating vacuum chamber 2 is separated from chamber 1 by lower valve membrane–valve pulsator’s device 9;

– an annular chamber 3 with constant atmospheric pressure, communicates with the ambient air through openings in the housing of the pulsator and is separated from annular projection chamber 2, which lowers the upper valve 9;

– camera 4 (top), that controls the operation of the pulsator, is separated from the chamber 3 via rubber septum 7 and with the chamber 2 communicates via a channel which section controls by screw 10.

The time during which the udder on the teat have a physiologically uniform impact teat cup is called tact.

The time during which a committed set of diverse tacts, called cycle or pulse. Pulsator calculation consists in determining the duration of cycles and air flow that occurs through the system and a vacuum pump. Pumping or sucking cycle time is defined by the formula

$$t_1 = V\Big/\left(76 - k_p\right)\ln\left(\psi_1 - h_2/h - h_1\right), \tag{11.1}$$

where V – the volume of the chamber 1 constant vacuum, m^3; h – nominal vacuum, kN/m^2 (mm Hg..); ψ_1 – variable factor–coefficient, which takes into account the time of switching valves in the pulsator; h_1 – the largest vacuum chamber 4 alternating vacuum (control) kN/m^2 (mm Hg..); h_2 – the largest vacuum chamber in a constant vacuum 1 kN/m^2 (mm Hg..).

Release time, or discharge cycle:

$$t_2 = \frac{V_n}{76 k_p}\ln\left(\psi_2 \frac{h_1}{h_2}\right), \tag{11.2}$$

where V_n – the volume of the chamber 4 alternating vacuum pulsator, m^3; k_p – coefficient of Poiseuille coefficient taking into account the dimensions of the channel and the viscosity of air.

This ratio can be determined by the formula

$$k_{\text{p}} = \frac{\pi d_0^4}{128\ l_0\ \mu_в}, \tag{11.3}$$

where d_0 and l_0 – the diameter and length of the channel connecting the two chambers and the pulsator 4, m; $\mu_в$ – the dynamic viscosity of air, which can be taken as = 0.000181 g/cm.s;

These coefficients are found by the relations:

$$\psi_1 = \frac{152 - \left(h - h_1\right)}{152 - \left(h - h_2\right)}; \tag{11.4}$$

$$\psi_2 = \frac{152 - h_2}{152 - h_1}, \tag{11.5}$$

Dividing equation (11.1) to (11.2), we find the ratio of cycles:

$$\delta_c = \frac{t_1}{t_2} = \frac{76}{76-h} \cdot \frac{\ln\left(\psi_1 \dfrac{h-h_2}{h-h_1}\right)}{\ln\left(\psi_2 \dfrac{h_2}{h_1}\right)}. \qquad (11.6)$$

In order to use the formula (11.1), (11.2) and (11.6) to calculate the length of cycles, it is necessary to determine the extreme limits of h_1 and h_2 vacuum values in the control chamber, depending on the design of the pulsator. To this end, using the calculation scheme of the pulsator shown in fig. 11.15, we form the equation of equilibrium of forces acting on the valves and diaphragm for two things:

– when a transition occurs from sucking stroke to the compression stroke and the valve is switched from the lower to the upper position;

– when the transition from the compression stroke to the sucking phase and the valve goes down by connecting the camera 2 with a constant vacuum.

1st case (fig. 11.7, a) in which there are downward force: MER – air pressure on the upper valve area Fvk and the G – weight is immovably parts.

In operation, air from the pulsator chamber 2 passes into the chamber 1, which is pumped continuously. Chamber 2 closed from above the membrane 7 and washer 8 and thereby separated from the concentric chambers 3, related to atmospheric pressure. Appears P_{vk} force acting down on the membrane 7 and the washer 8. The magnitude of this force, in this case:

$$P_{vk} = (h - h_1)\, F_{vk}\, k, \qquad (11.7)$$

where k = 1,033/76 – conversion factor; h – nominal vacuum when running pulsator kN/m^2; $F_{v.k}$ – upper valve area, m^2.

b) In a vacuum chamber 2, which is passed through conduit 6 into the manifold. Simultaneously, air from the chamber 4 enters the chamber 2 through the channel 5, causing a vacuum in the chamber 4 rises, but the camera 2 does not change because it is connected to the chamber 1.

Since the atmospheric pressure chamber 3 and a vacuum chamber 4 is increased, the force RVC, pressing down the diaphragm 7 is reduced and simultaneously occurs and grows upward force spring force P_m, membrane 7. The force is up $_{\text{Pk}}$ – air pressure from the chamber 3 to the ring membrane area F_k and R_m – the power of the elastic membrane, clamped along the perimeter. According to experimental data, the elastic force of the membrane is 1...3 N.

Where

$$P_{м} = h_1 u F_{м} k\ , \tag{11.8}$$

where $F_м$ – membrane area, m^2; h_1 – the greatest vacuum in the vacuum chamber 4 of variable kN/m^2 (mmHg..); u – the membrane activity factor considering only that part of the load, which is transmitted from the diaphragm on the valve stem;

The equation of the balance of power at the time of valve switching from the lower position to the top is as follows:

$$P_{вк} + G = P_{к} + R_{м}, \tag{11.9}$$

or

$$\left(h - h_1\right) F_{вк} k + G - R_{м} = h_1 u F_{к} k,$$

from whence

$$h_1 = h \frac{F_{вк}}{F_{вк} + uF_{к}} + \frac{G - R_{м}}{k\left(F_{вк} + uF_{к}\right)}, \tag{11.10}$$

where G – the weight of moving parts, kg; $R_м$ – membrane elasticity force, N.

Activity coefficient of the membrane,

$$u = \frac{\frac{1}{3} + \frac{d_{к}}{d_{м}} + \left(\frac{d_{к}}{d_{м}}\right)^2}{1 + 2 \cdot \frac{d_{к}}{d_{м}} + \left(\frac{d_{к}}{d_{м}}\right)^2}, \tag{11.11}$$

where $d_м$ – outer diameter of the membrane, m; $d_к$ – inner diameter of the stake–tsevoy camera, m.

Since the membrane area smaller than the area of the upper valve, i.e. $F_m < F_{v.k}$, the force acting upwards, at some point is aligned with the largest force $R_{v.k}$, and then it gets bigger, and the membrane, pulling the rod moves up. The lower valve 9 closes the lower section of the chamber 2 and it quickly filled with air from the chamber 3 to the atmospheric pressure. Air rushes into the reservoir through the conduit 6 and the channel 5 into the chamber 4. At the same time, a new, small RNA force acting on the bottom flap 9 downwards.

a) The forces acting down the $P_{нк}$ – the air pressure in the lower area of the valve F_{nk}, the G – the weight of moving parts, $R_м$ – the elastic force of the membrane.

In this position:

$$P_{нк} = hF_{нк}k \tag{11.12}$$

where $F_{н.к}$ – area of the bottom valve, m^2;

b) The forces acting up: Pm – pressure washer with an area Fsh and Pk – the pressure on the diaphragm ring area $F_к$ Wherein:

$$P_ш = h_2 F_ш k \tag{11.13}$$

$$P_к = h_2 u F_к k\ . \tag{11.14}$$

The equation of the balance of power at the time of switching valve from the top position to the bottom has the form:

$$P_{нк} + G + R_м = P_ш + P_к \tag{11.15}$$

or

$$hF_{нк}k + G + R_м = h_2\left(F_ш + uF_к\right)k,$$

from whence

$$h_2 = h\frac{F_{нк}}{F_ш + uF_к} + \frac{G + R_м}{k\left(F_ш + uF_к\right)}. \tag{11.16}$$

Formulas (11.10) and (11.16) to calculate the ratio of the duration of cycles are valid for any membrane pulsator having a control chamber 4 constant volume. However, vacuum h_1 and h_2 limit values should be determined taking into account the structural features of the scheme and the size of the pulsator. Gradually, the air from the chamber 2 into the chamber 4. The force exerted on the diaphragm 7 up until then decreases until it becomes less than the force acting on the lower valve

9 downward. Then the bottom valve, having fallen down, opens and closes at the same time the camera 2 on top of the membrane. In the chamber 2 again, a vacuum is created, and the process is repeated. Equations (11.10) and (11.16) at equilibrium forces allow limits to calculate changes in the vacuum chamber 4:

$$\frac{F_{в.к}}{F_{н.к}} = \frac{h_1}{h_2} = \frac{D^2_{в.к}}{D^2_{н.к}}. \tag{11.17}$$

For milking machines sufficient pulsation frequency 40...120 per minute, depending on the design of the device and with respect to the individual needs of the cow. In operation trehtaktnyh milking machines, for the majority of cows, the frequency is in the range 60 ... 80 pulsations per minute. The ratio of the length of ticks sucking and compression pressure is determined by the size of fields valve–diaphragm mechanism.

11.3.1 Hydraulic pulsators

Hydraulic pulsator (fig. 11.8) has constant vacuum chamber (connecting pipe via a collector chamber), two working chambers of alternating vacuum (each of which communicates with interspace of two chambers of the teat cups via nozzles), two hydraulic chambers in piston pulser.

By rubber membrane wall of one of the hydraulic chambers piston rod is rigidly attached (fig. 11.9), which moves the slide, connecting by the reciprocating motion a constant vacuum chamber with variable vacuum working chambers.

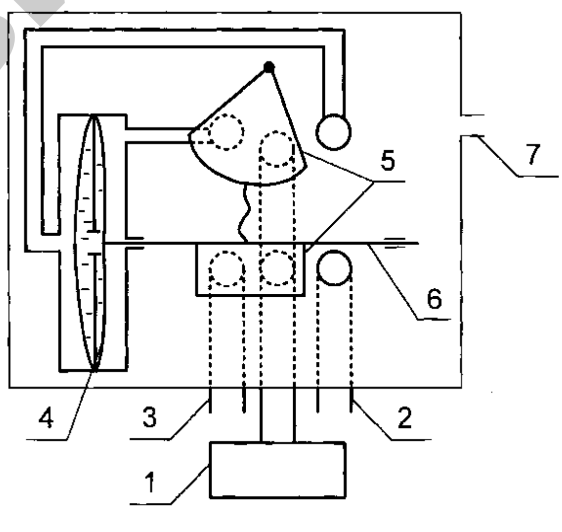


Fig. 11.8. The scheme of doubles milking hydraulic pulsator:
1 – pump; 2–3 – alternating vacuum tubes; 4 – piston; 5 – caps; 6 – rod; 7 – fitting

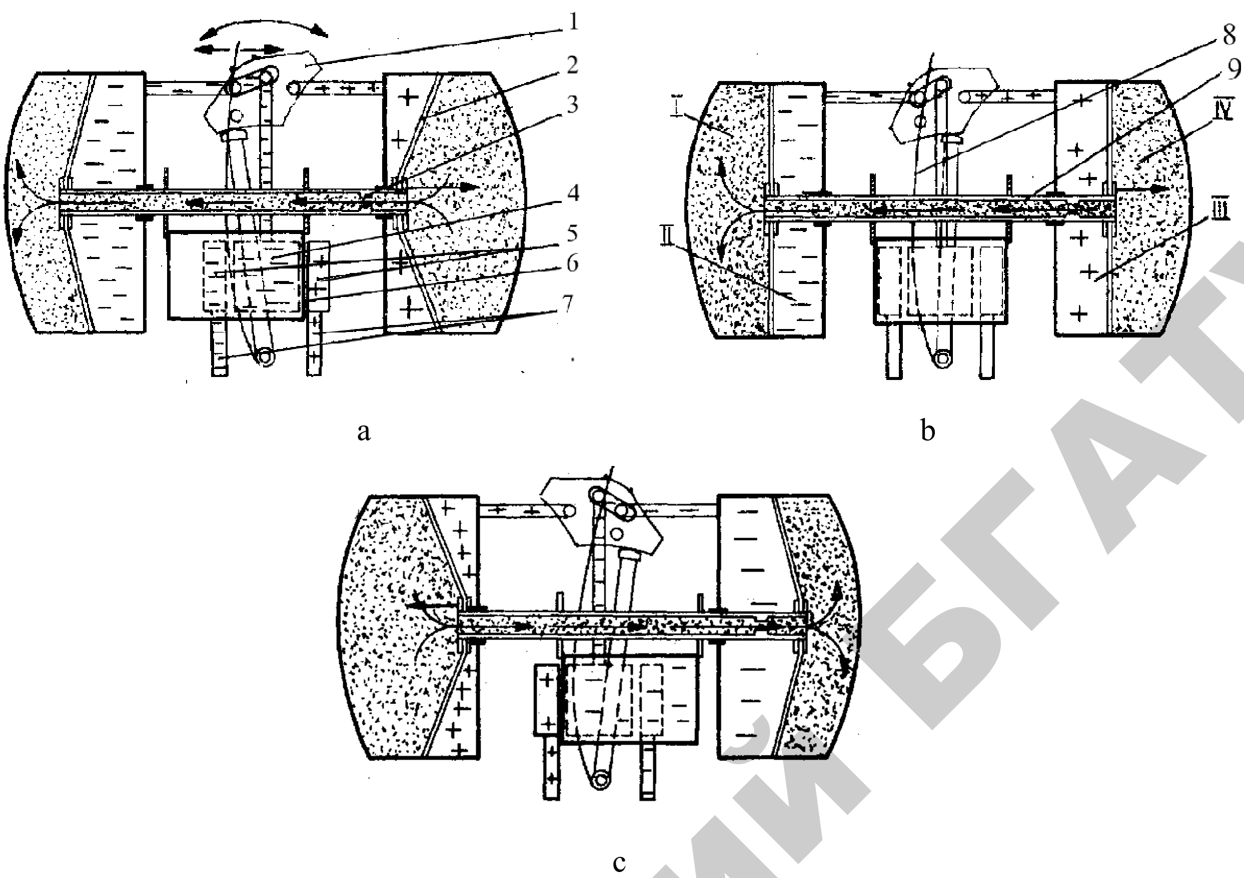
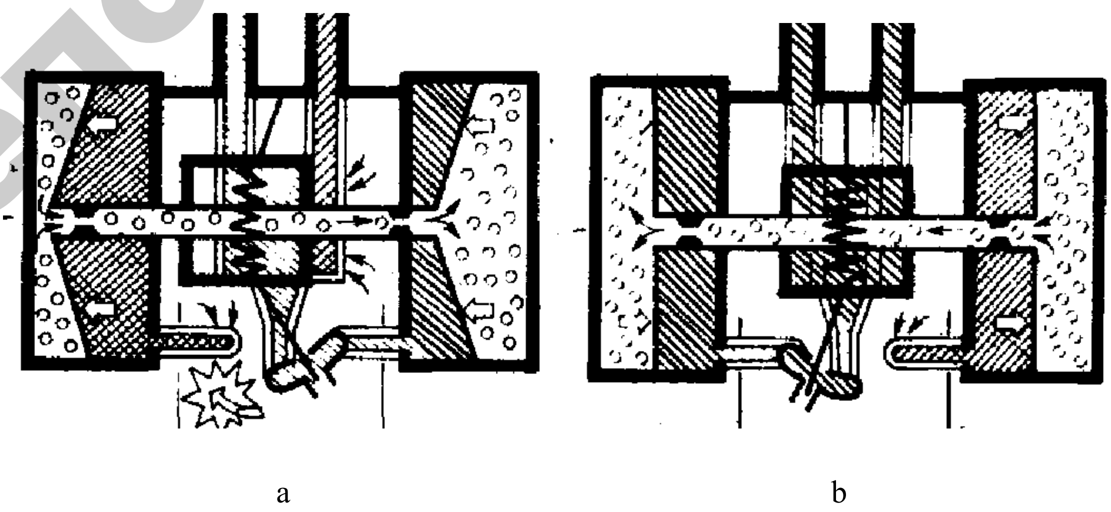
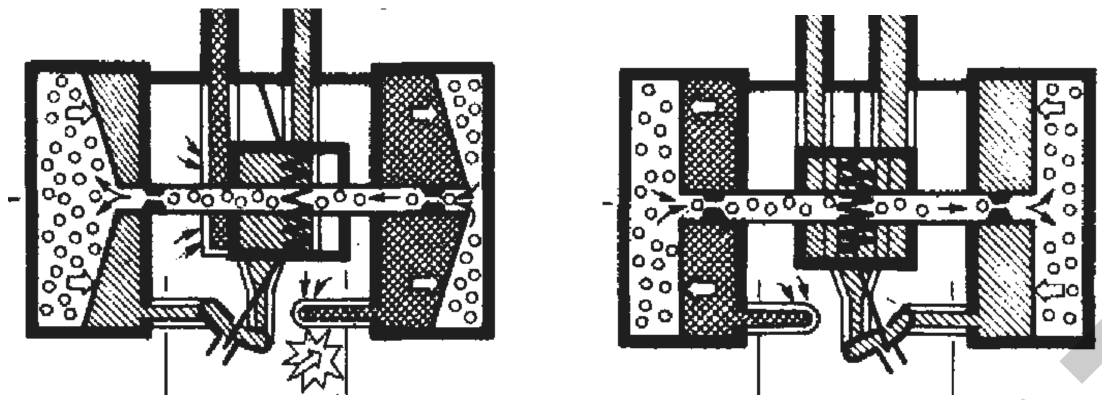


Fig. 11.9. Pairwise hydraulic milking pulsator operation working principle:
1 – switch; 2 – membrane; 3 – cover; 4 – constant vacuum chamber;
5 – AC vacuum chamber; 6 – slider; 7 – pipe; 8 – spring; 9 – tube;
I–IV – hydraulic chamber; II–III – air chamber

The control mechanism, consisting of a spring and a rotary cam is driven by a slider and connects a control chamber to atmospheric pressure, and the other – to vacuum pressure. Atmospheric pressure moves liquid from one chamber of hydraulic piston to another.

Scheme of the pulsator Alfa Laval is shown in fig. 11.10.





c d

Fig. 11.10. Scheme of hydraulic pulsator Alfa Laval:
a – compression stroke in right cup; b – vacuum in the interspace; c – compression stroke in left cup; d – vacuum in interspace of both cups

The phase distribution in the nozzles of the milking machine of pairwise milking is shown in fig. 11.11. If the vacuum (fig. 11.11, a) is created in the right nozzle 1 of pulsator, milking is made of the right half of the udder. Simultaneously, air is supplied into the left branch pipe 2 and the left half of the udder is massaged. Then, in both nozzles (fig. 11.11, b) a vacuum is supplied and thereby both udder halves are milked simultaneously. Next, air is supplied into the right nozzle 1, producing a massage of the right half of the udder and a vacuum (fig. 11.11 in) in the left nozzle 2 facilitates milking of the left half of the udder. Then, in both nozzles the vacuum is supplied and thereby (Fig.11.11 d) both halves of udder are milked simultaneously.

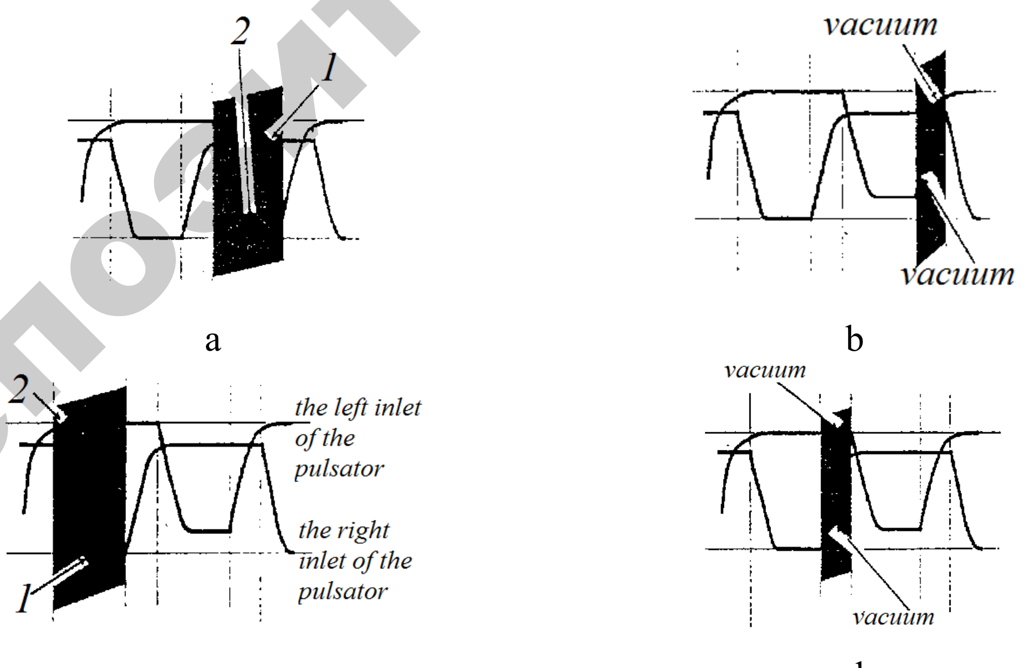
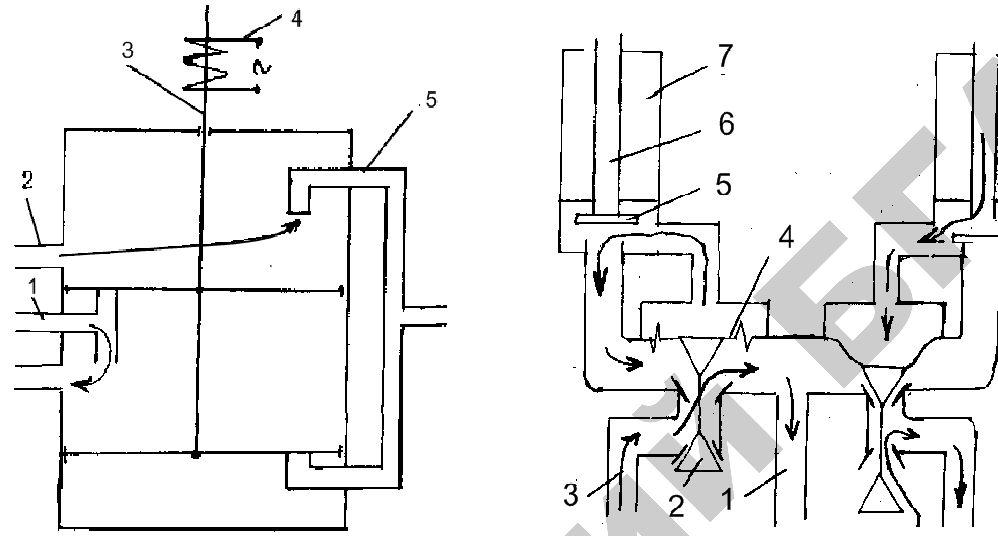


Fig. 11.11. Phase distribution in pair milking pulsator hydraulic nozzles:
1 – right branch pipe; 2 – left branch pipe

11.3.2 Electromagnetic pulsators

They are used as electromagnetic milking pulsators pair (fig. 11.12). They act by direct or alternating electric current voltage of 12 V. When electric current flows through the coil of the pulsator the core of the ferromagnetic material is drawn inside and closes the hole in the center of the pulsator, disconnecting the camera by an electromagnet by air and connecting it with a constant vacuum.



a b

Fig. 11.12. Diagram of the electromagnetic pulsators for pair milking teat:
a) one valve – (1 – air inlet; 2 – pulsating vacuum pipe, 3 – valve mechanism; 4 – electromagnet; 5 – constant vacuum tube);
b) two valve (1 – pipe constant vacuum; 2 – air valve; 3 – AC vacuum tubes; 4 – membrane; 5 – electromagnetic valve; 6 – nozzle air; 7 – electromagnet)

The design of the electromagnetic pulsator is shown in fig. 11.14. The pulsator comprises a core 1 and a coil of the electromagnet 6. The housing together with the base 2 forms a vacuum chamber 12. The variable alternating vacuum chamber connected to a vacuum pipe fitting bottom and with the atmosphere – the opening 11. When de–energized electromagnet coil anchor is in the down position, blocking the supply of vacuum. Atmospheric air entering through inlet 11 flows into the variable vacuum chamber and further through the nozzle 4 into a space between the walls of the teat cups. When the solenoid coil is energized, the armature is attracted, blocking the opening 11 in the AC of the vacuum chamber and in the space between the walls of the teat cups is supplied vacuum. Magnetic pulsator provides a predetermined frequency and duty cycle fluctuations due to the low course of the armature and the parameters of the electromagnetic actuator (a solenoid shown in fig. 11.13, b). Two air gap formed cylindrical core and the armature, and the armature and the flat end of the housing.

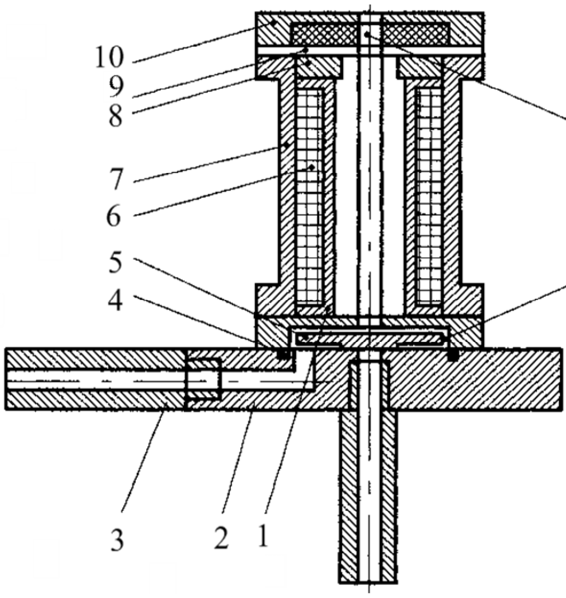


Fig. 11.13. Magnetic pulsator:

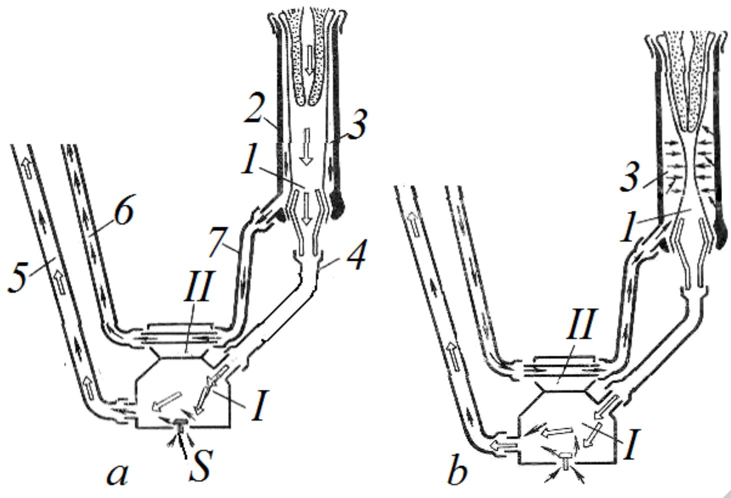
1 – Core 2 – base; 3 – fitting 4 – rubber pad, 5 – anchor, 6 – coil;
7 – the case of the electromagnet 8 – locking washer; 9 – a lining locking;
10 – a cover; 11 – hole; 12 – valve;

Pulsator «LECTRON» company «GASCOIGNE MELOTTE» has MSE–velocity pulsations from 30 to 120 min^{-1}, as well as various bars relation to the front and rear teats: 55/45 for the front teats; 60/40 for the rear. Execution pulsators can be both electronic and pneumatic. Features pulsators «UNIPULS 2» and «UNIPULS ELEKTRONIC»: infinitely adjustable frequency ripple and the balance of 30 to 120 min–1; low dependence of the frequency fluctuations of the air temperature and the vacuum level; work from an independent power source; switch when connecting a vacuum; noise level of less than 70 dB.

The company «Westfalia Separator» manufactures milking machine, equipped with an electronic pulsator «STIMOPULSC», providing for the use of the process of stimulation of the udder during milking and teat dodaivanii, taking into account the process of milk cows during milking, as well as changing the individual characteristics of animals in the interval between calving. It carries high frequency stimulation (300 min–1) in the teatcup liner pulsation milking beginning (60) and disabling pulsators in a compression stroke at the end of milking.

11.4 Purpose and construction of collector device

Collector allows to distribute the alternating pressure on the teat cups, to take them to milk the milk and send it to slubber (fig. 11.14.), Has two cameras – molokosbornuyu and distribution.



a– milking phase b – compression stroke

Fig. 11.14. The scheme of the two–stroke milking apparatus:
I – constant vacuum chamber; II – AC vacuum chamber (distribution);
1 – underteat camera; 2 – teat cup; 3 – interwall chamber; 4, 5 – milk hoses; 6; 7 – AC vacuum hoses; 8 – valve air leaks

At the bottom of the chamber I (fig. 11.15) is a rubber stopper 1, which is located in the sleeve valve pin. Through it occurs during milking air leak, which ensures quick release of the milk pipe – improves its conveying capacity. The cameras are connected to the reservoir through the connections with glasses, milking bucket and pulsator.

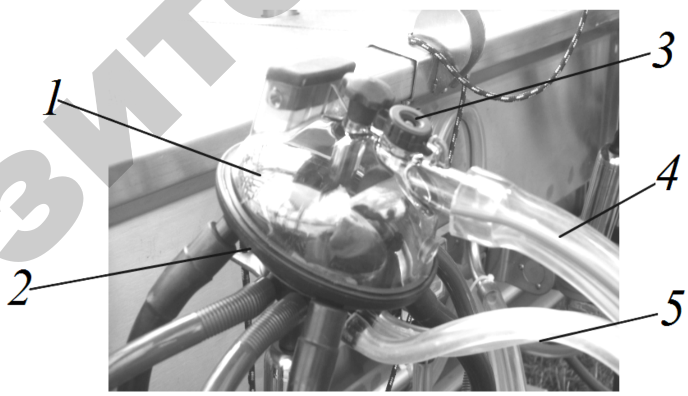


Fig. 11.15. Collector:
1 – constant vacuum chamber; 2 – camera vacuum (distribution); 3 – stopper;
4 – milk line; 5 – vacuum line

Camera constant vacuum manifold 1 is made of transparent plastic to control molokovydeleniya. Put off the vacuum valve 3, which excludes the application of the milk hose clamp. The larger the angle of inclination from the horizontal axis of the output choke manifold, about 75 °, improves milk churn and contributes to a more even weight distribution, the suspension of the milking machine to the teats of the udder cows. Capacity breast camera from 58 cm^3 to 76 cm^3. Milking machines "Eclipse" and "Mini–Orbit" (fig. 11.16) have a reservoir of 305 and 640 cm^3, respectively. The diameters of the inlet 9 mm, and output – 16 mm, provide free outflow of milk from the nipple with minimal vacuum that reduces cases of mastitis.

Fig. 11.16. The collector of the milking apparatus"Eclipse"

A feature of the milking machine collector company «S.A.C.» (Denmark) is a floating mounting design variable vacuum distributor, thereby excluded twisting pipes and improves the spatial orientation of the teat cups. The collector with integrated indicator of mastitis is shown in fig. 11.17.

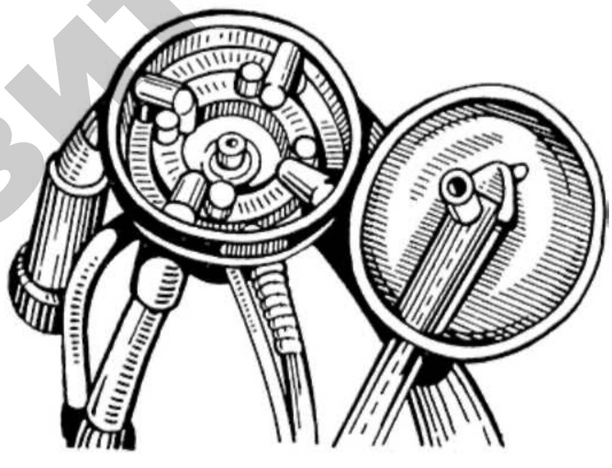


Fig. 11.17. Suspension part with mastitis indicator of the company «S.A.C.»

Each nipple has a separate sensor measuring the salt content in milk throughout milking. If the salt content exceeds the permissible level, the LED lights up, indicating that the deviation in this part of the udder. The mastitis indicator using LED also indicates the end of milking. Power is supplied from the power supply or uninterruptible power supply (battery).

11.5 Construction of milking machine systems

The vacuum pump is designed to create a discharge in the vacuum systems and provide rapid recovery of set value of the vacuum in contact with air in the system.

Vacuum balloon serves to equalize the vacuum fluctuations, prevents entering of moisture from vacuum line into the pump and used as dump tank when flushing of the vacuum system. Capacity of domestic vacuum bottle is 0,02...0,025 m^3. The bottle in the absence of vacuum in the system must be open.

V*acuum regulator* is designed to maintain a required discharge in the vacuum network. Its action is based on the intake of air in the pipe when the force of air pressure in the valve when the discharge created by the pump exceeds the weight of the load controller. Depending on the type the vacuum can be 45...58 kPa.

Flow gauge measure differential pressure (detect vacuum magnitude in system).

Vacuum conduit usually divided into several sections. The main of them – work area – part of the vacuum line, where valves are located for connecting actuators – milking machines. Vacuum pipes and fittings should be made of materials with anti–corrosion coating (galvanized pipes) and must withstand the vacuum to 700 mm. Hg. Art. (93 kPa). The diameter of the vacuum line is usually not less than 25 mm in order of the loss of pressure along the entire length does not reach more than 5 mm. Hg. Art. (670 Pa). The vacuum level should be monitored by gauge. Vacuum milking machines have grading not only in kilopascals (kPa), which is measured by the pressure in the system of units (SI), but also in millimeters of mercury (mm Hg. V.), And in kilograms of force per 1 cm 2 (kgf/cm^2).

Milk from the cups through the collector and the milk hose comes into the milkline. *The milk line* should provide a quiet passage of milk though it without excessive mixing it with air, which under certain conditions can have a negative impact on its structure and properties. The diameter of the milk line should be chosen so that the pressure loss across its length does not exceed 1330 Pa in the process of milking. The minimum diameter of the milk line is 25...30 mm.

Milk storage (fig. 11.18) is used to separate the mixture of milk and air and excretion of milk or a cleaning solution from a vacuum pressure. Milk tank consists of a frame to which milk tank with a float sensor 9, safety camera 17, a milk pump 21 and the control unit milk pump 18 are attached. A button 19 of hand milk pump control is on the control unit. Above cover 11 of milk tank the valve 12 is installed. To the upper fitting of distributor the hose is connected to wash the safety chambers and a cooler. The air is sucked from milk tank through the safety camera and vacuum line. At the bottom of the milk tank there is a milk line 2 having two fitting, a large one – for milk discharging to the pump 21 and a small one – for sucking of cleaning fluid from the protective chamber 17 during washing.

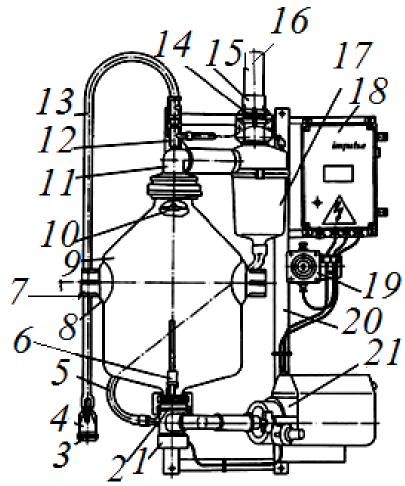


Fig. 11.18. Milk storage:
1 – float switch; 2 – milk pipe; 3 – protective cap; 4 – adapter; 5 – hose;
6 – float; 7 – milk input; 8 – seal; 9 – milk storage; 10 – sprinkler; 11 – cover;
12 – distributor; 13 – hose; 14 – valve; 15 – coupling; 16 – vacuum line;
17 – safety camera; 18 – milk pump control unit; 19 – switch;
20 – frame; 21 – milk pump

During milking and washing of the vacuum valve 14 is opened. Vacuum from vacuum line 16 extends into the fuse box 17, milk tank 9 and further into the milk line 7. Milk during milking (washing solution) flows from the milk line 7 to 9 milk tank and accumulates therein. As filling the milk tank by milk or detergent solution the float 6 goes up with magnet, connects magnetic contacts and sends a signal to the unit 18 of milk pump control 21 which turns on a pump for pumping a portion of milk or cleaning solution. The sensor of milk pump operates so that a certain portion of milk is always in milk tank, preventing air from entering the milk pump. When the milk pump failures (overflows milk tank) liquid (milk or cleaning solution) from milk tank sucked is into the *sanitary trap* (fig. 11.19).

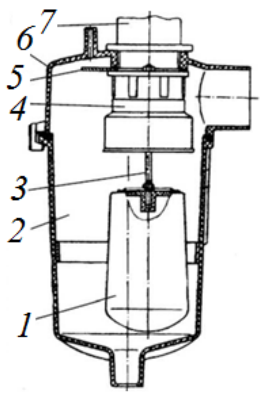


Fig. 11.19. Sanitary trap:
1 – float; 2 – chamber; 3 – stock; 4 – valve seat; 5 – sprinkler;
6 – a cover; 7 – vacuum line

When filling the protective chamber a float comes up 1 and 3 through the stock 4 moves in the valve seat, stopping the access vacuum from the vakuum line into the milk tank 7, and further into the milk line and, therefore, the milking process is terminated (flushing). Then they close the vacuum tap, press the button on the milk pump control unit. Milk or cleaning solution is pumped from milk tank (float 1 (see fig. 11.19) is lowered and opens vacuum line 7).

11.6. Determination of air flow the milking machine

Calculation conducted to substantiate the performance needs of the vacuum pump, the air flow includes the definition of milking machines and vacuum line system. Consumption of milking machines vacuum of air depends on the magnitude, frequency fluctuations, the machine type and capacity of chambers and tubes, in which an alternating vacuum. We take the total capacity of the chambers of the teat cups to one device, equal to V_a. Then the volume of air V_h after enlargement, according to Boyle's law:

$$V_h = \frac{P_б V_а}{P_h}, \tag{11.18}$$

where V_a – initial volume of air in the chamber at atmospheric pressure, m^3; $P_б$ – barometric (atmospheric) pressure, kN/m^2 (mm Hg...); P_h – absolute pressure in the chamber after evacuating kN/m^2.

Absolute pressure after the evacuation:

$$P_h = P_б - h, \tag{11.19}$$

where h – nominal vacuum, kN/m^2 (mm Hg.).

Consequently, the volume of air to be pumped out by one machine cycle of operation will be:

$$V_ц = V_h - V_a. \tag{11.20}$$

This amount must be brought to normal operating conditions, ie, to atmospheric pressure. Then the reduced volume $V_{ц.прив}$. It can be found from the equation:

$$V_{ц.прив} = \frac{V_ц P_h}{P_б}. \tag{11.21}$$

If we substitute in this formula the value of $V_ц$ (11.20) and the pressure of (11.19), the amount of air pumped in one pulsation and reduced to atmospheric pressure:

$$V_{\text{ц.прив}} = V_{\text{а}} \frac{h}{P_{\text{б}}}. \qquad (11.22)$$

From this formula it follows that when a vacuum of 52 kN/m^2 (380 mm. Hg. V), it is necessary to pump half of the air contained in the chambers cup and vacuum hoses connecting, collector and pulsator.

Milking machine –*v*– airflow With a frequency pulsations:

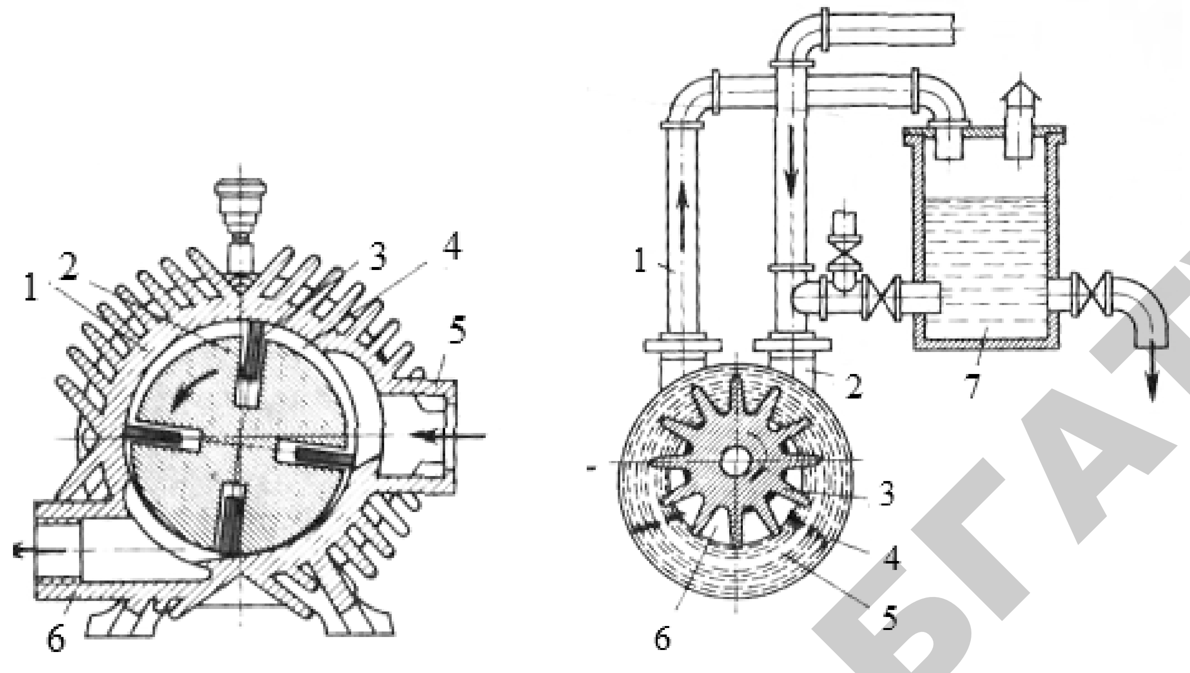
$$Q_{возд} = V_{\text{ц.прив.}} \cdot v = V_{\text{а}} \frac{vh}{P_{\text{б}}}. \qquad (11.23)$$

Performance of individual vacuum pumps should be at least 30...45 m^3/h.

Vacuum ratio. The best is the excess of the vacuum in the milk of 1...2 kPa over the corresponding values in vackuumline. Acceptable equality. At higher relative vacuum glasses fall off, wear out quickly teatcup liner, milking become dangerous to animal health. Tact sucking lasts about 2/3 the length of the pulsation. Accordingly, the compression stroke is 1/3 of pulsation. Modern milking machines operate in mode 67 ± 5 min pulsation.

Calculation of rotary vacuum pump. Currently, the vacuum on livestock farms used rotary type pumps and liquid ring vacuum pumps. Rotary vacuum pumps are identical in the device and features by performance and oil consumption. Inside permanent joint body rotates the rotor is located eccentrically with respect to the stator, but the axis (fig. 11.20, a).

The rotor has four slots disposed tangentially or radially in the plate are freely inserted blades. As the rotor rotates the centrifugal force pushes them out of the slot and presses against the inner surface of the stator. Since the stator and rotor are arranged eccentrically, that each blade forms a closed space variable in volume. Near suction volume increases and air is sucked in interscapulum. Near the exhaust manifold volume decreases, air is compressed and pushed out. For the sealing of the pump and reduce the friction of the blade liberally lubricated with oil coming from the wick lubricators.



a b

Fig. 11.20. Vacuum pumps
a) rotary blade pump (1 – case; 2 – rotor; 3 –vane;
4 – camera, 5 – inlet; 6 – exhaust pipe);
b) liquid–ring pump (1 – exhaust pipe; 2 – vacuum line; 3 – rotor; 4 – housing;
5 – water ring; 6 – variable volume chamber, 7 – container of water)

The vacuum liquid ring pump, (fig. 11.22, b) the variable volume space formed by the annular stream of water rotating the inner surface of the stator and vanes integral with the rotor. Moreover, the stator blades do not touch the walls, so lubrication is not required. The compressed air together with a small amount of water released into an exhaust pipe and further to a flow divider where cooling water is supplied again dosed into the pump together with the air sucked into it.

Volumetric flow rotary pump depends on the area AA_1B_1B ka–action suction rotor length and the frequency of rotation (fig. 11.21). Area AA_1B_1B is variable depending on the angle of rotation of the rotor –*φ*–. Rotary pumps usually do so that at the moment of maximum area, it is overlapped by the suction nozzle and opens into the discharge side. This occurs when the arc length of the stator, overlapping mezhlopastnoe space will be slightly greater than the angle between the shoulder blades, and the distance the suction and delivery pipes will be located on an axis perpendicular to the axis of eccentricity. The exact value of the maximum and minimum amounts of inter–blade can be obtained reaching the differential equation changes sectors interscapular area at the corner of the rotor rotation φ and integrating it into the appropriate range.

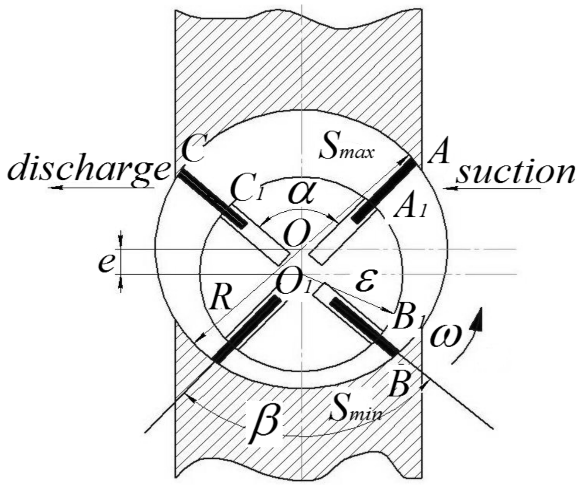


Fig. 11.21. Scheme of rotary pump

The efficiency of the rotary pump is determined by the difference between the maximum and minimum areas mezhlopastnogo space. We define this parameter for the four vane pump. Assume the radius of the stator with a maximum amount equal to the inter–blade:

$$R_1 = R + e, \tag{11.24}$$

where R – radius (actual) of the stator, m; e – eccentricity, m.

Then, with a certain excess of actual O_1CA sector area can be determined by the formula

$$S_1 = \frac{\pi R_1^2 \beta}{2\pi} \tag{11.25}$$

where β – the angle between the shoulder blades, hail.

With four blades $\beta = \pi/2$, then:

$$S_1 = \frac{\pi (R + e)^2}{4}. \tag{11.26}$$

Area $O_1C_1A_1$ rotor sector can also be determined by the formula

$$S_1 = \frac{\pi r \beta}{2\pi}, \tag{11.27}$$

where r – radius of the rotor, m.

Modern pumps are manufactured with a minimum gap (70...100 mm) between the rotor and stator. In this case, we can assume that $R – r$ = e and $r = R – e$, then:

$$S_2 = \frac{\pi\,(R-e)^2}{4}\,. \tag{11.28}$$

Then interscapular maximum volume:

$$\Delta S_{мак} = S_1 - S_2 = \frac{\pi\,(R+e)^2}{4} - \frac{\pi\,(R-e)^2}{4} = \pi\,Re\,. \tag{11.29}$$

To determine the minimum amount of interscapular also accept, with some exceeding the central angle of the stator equal to the angle between the rotor blades. Then the maximum sector interscapular area with radius R:

$$S_3 = \pi R_2/4\,, \tag{11.30}$$

and the area of the sector with a radius r_p:

$$S_4 = \frac{\pi\;r_p^{\;2}}{4} = \frac{\pi\;(R\;-\;e)^2}{4} \tag{11.31}$$

The minimum area between the shoulder blades:

$$\Delta S_{мin} = S_3 - S_4 = \frac{\pi R^2}{4} - \frac{\pi (R-e)^2}{4} = \frac{\pi (2Re - e^2)}{4}\,. \tag{11.32}$$

The efficiency of the pump is determined by the difference between the maximum and minimum interscapular area:

$$\Delta S = \Delta S_{мак} - \Delta S_{мin} = \pi\,(Re - \frac{(2Re-e^2)}{4}) = \frac{\pi}{4}(2Re + e^2) \approx 0.785e(Д + e)\,, \tag{11.33}$$

where $Д$ – diameter of the stator, m.

Useful volume interscapular camera:

$$V_n = 0{,}785\;e\;(D + \mathrm{e})\;L_{rot}, \tag{11.34}$$

where L_{rot} – rotor length, m.

With four blades and the angular speed of the rotor ω pump flow is:

$$Q_н = \frac{4V_n \omega_p}{2\pi} = 0,5e(Д + e)\, L_{рот}\ \ \omega_p\ . \tag{11.35}$$

It follows that the theoretical air pump feed is directly proportional to its geometrical dimensions (e, d, L) and the angular speed of the rotor. The actual pump performance:

$$Q_Д = Q_H\, \eta_n\, \eta_m, \tag{11.36}$$

where η_m – a gauge factor considering vacuum conditions:

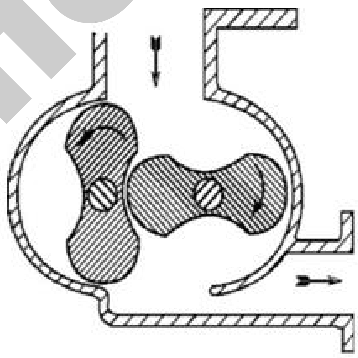
$$\eta_м = \frac{p_a - h}{p_a} = \frac{p_h}{p_a}\ . \tag{11.37}$$

In the milking vacuum units of h = 47,66 kPa, then η_m = 0,52...0,32. η_n – chamber filling rate is dependent on the design and pump rate of rotation. It can vary within wide limits, η_n = 0,3...0,9.

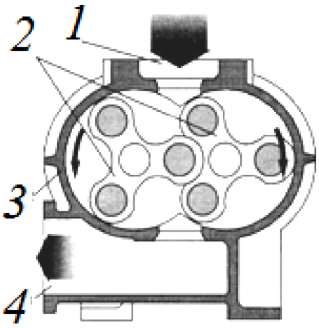
With the known feed pump and a vacuum in milking installations in the pump drive power can be determined by the formula

$$N_н = \frac{Q_н\ h}{\eta}\ . \tag{11.38}$$

Thus, setting the process conditions can be calculated con constructively parameters rotary pump. Some processes require very high pumping speed and at very low pressures. These requirements meet the tworotor volumetric pumps Roots blower. Driving such a pump is shown in fig. 11.22.



a two–lobe pump



a three–lobe pump

Fig. 11.22. Twin–rotor volumetric pump: 1 – entrance; 2 – rotor; 3 – case; 4 – exit

Two long rotor having a cross section resembling a figure eight, rotate in opposite directions without coming into contact either with one another or with the walls of the housing, so that the pump can be operated without lubrication. The need in the oil seal either, as the very small gaps between the design exactly fitting parts. The rotor rotates at up to 50 to 1, and a high pumping speed is maintained up to pressures of the order of one millionth of the atmospheric pressure. Each rotor may have two or three cams.

11.7. Classification of milking machines and milking parlors scheme

Milking machines are divided into the following groups:

1. Linear milking machines, which are used at the fastened content of dairy cattle and in its turn divided into two groups:

– milking machines for milking in portable buckets. This high proportion of manual labor (transporting milk to the dairy unit) and a low load on 1 milkmaid 20 ... 25 cows.

– milking the milk with a long 100 cows and 200 cows. This service rate increased by 2 times (50 goal.), Reduced manual labor and automated accounting of the milk from the animal groups.

Milking installation with the milk intended for milking cows in stables barns, transportation milked milk through the pipes in the milk, the group account is necessary, filtration, cooling and collecting it in a storage container.

Installation ADM–8 (fig. 11.23) consists of two closed systems – air duct to create a vacuum in the system, necessary for implementation of milking cows and the milk collection and transportation of milk milked.

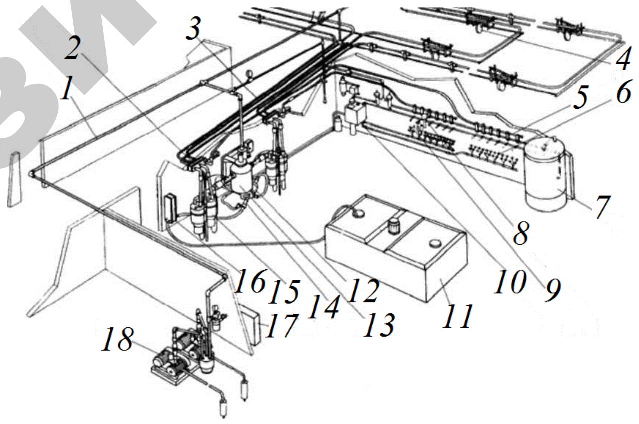


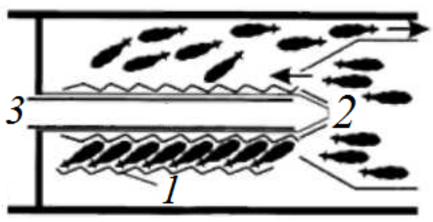
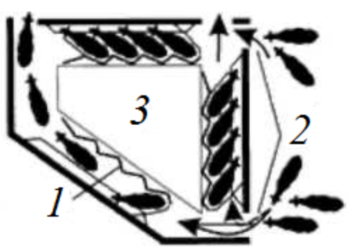
Fig. 11.23. Milking unit:
1 – vacuumline; 2 – switch; 3 – milkline; 4 – lifting device; 5 – crane; 6 – washing device; 7 – electric heater; 8 – device registration milk; 9 – milking equipment; 10 – automatic washing; 11 – milk tank; 12 – milk pump; 13 – milk storage; 14 – filter; 15 – milk dispenser; 16 – milk cooler; 17 – box for spare parts; 18 – vacuum pump.

Installation, designed for 200 cows, consists of twelve milking machines, two standardized systems for creating vacuum with air cylinders, two vacuum regulators and the duct system. Milk line consists of the milk from the glass tubes with a diameter of 45 mm, the group counter milk molokosbornika (vozduhorazdelitelya), milk pump, filter, plate dvuhpaketnogo milk coolers, automatic devices for circulatory washing of milk and milking equipment, as well as devices for the individual (zootechnical) accounting milk yield.

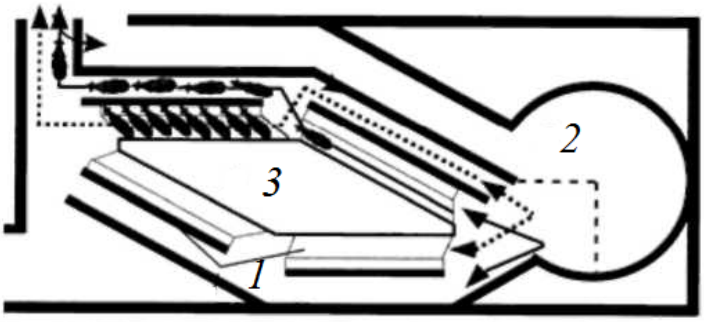
2. Milking machines are installed in special teat forcrystals. They are used primarily for cattle loose housing system content, but also can be used in tethered, especially if there is an automatic group leash. Installations of this type consist of milking machines permanently mounted indoors or move on special platforms. At milking cows is distilled from the places of their detention at a time or in groups, are admitted to the milking room and is placed in the machine. After milking the cows return to their former places, and in the milking machines are driving these cows.

The advantage of milking machines for milking parlors is a deep specialization of labor operators, excluding operations such as the distribution of food, cleaning stables and others. The presence of a buried trench removes the milker to work in an inclined position in the preparatory and finishing operations. This allows operators to increase productivity in machine milking and dairy products to obtain higher quality. Milking machines for milking cows in special halls are divided into groups: "Herringbone"; "Tandem"; "Carousel".

Type "Herringbone" (fig. 11.24). Milking machines for milking cows in special rooms such as "Herringbone" are available to the individual machine–mi, with a side entrance and independent services or a group of cows longitudinal machine tools. Installations with group machines differ in the number of seats in the machine and their location:

– "In line" – with the number of cattle stalls 2x8, 2x6 and 2x4; 
– "Trigon" with the number of cattle stalls 3x4; 

– "Range" with the number of cattle stalls 4x8;



– "Parallel" and "Europarallel"

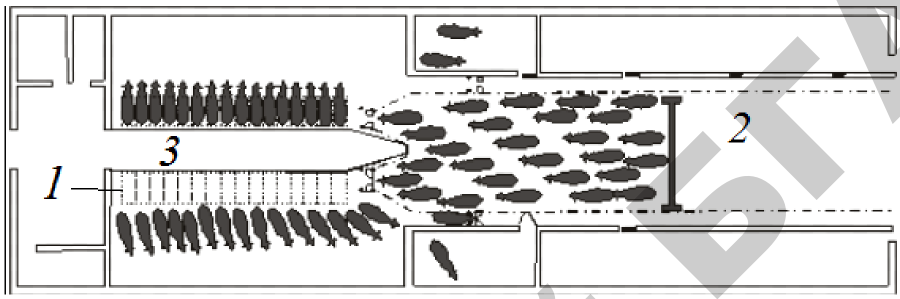


Fig. 11.24. Schemes of milking parlors such as "herringbone":
1 – machine tools; 2 – entrance gates; 3 – operator place

The composition of the stationary milking plant "Herringbone" includes two milking machine for 8 persons each, located on both sides of the working of the trench at an angle of 30...35^0, which allows the operator to carry out the processing of the trench udders and milking. The production line includes the milk, vakuum-provod, a device for cleaning the udder and milk preprocessing system. Milking unit can be equipped with the feeding system consisting of chain–shaybovogo conveyor, bins, feeders and dosing of concentrated feed. Issuance of feed – the group with the speed control portions. In plants such as "Herringbone" achieved productivity equal to 35...40 heads/hour due to a larger number of milking machines, special placement of cows and their service group. Cows are arranged at an angle of 30^0 to the trench and turned the heads of the trench, which allows you to place them in the group machine closer to each other (90...100 cm). The cows enter the milking machine and coming out of it a group rather than individually.

Characteristic features of the milking parlor of the "Polygon" peculiar configuration of the milking parlor, which is typically rectangular in shape, and each side is located six milking machines such as "Herringbone. The milking process is carried out as follows. When the input gate is open, all the feeders, but the most distant, closed. When the first cow reaches the last trough and touches it automatically opens the second feeder, the second includes the third, etc. the line to the last, closing the entrance gate. Installation is served by one operator. Type "Tandem" (fig. 11.25) have the following milking parlors scheme:

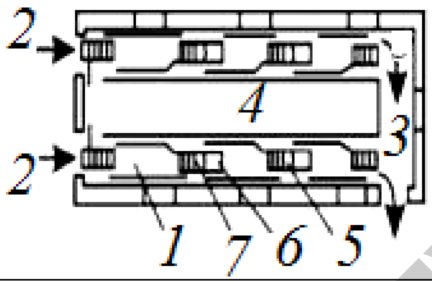
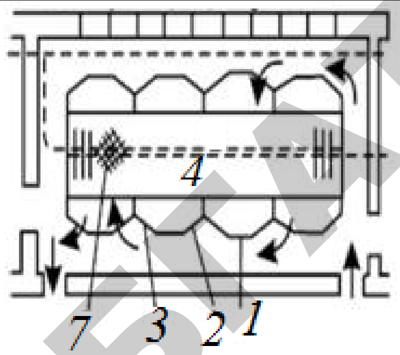
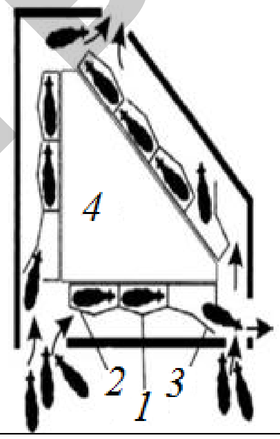
– With group longitudinal machine tools "in–line" and two–way (2x4, 2x3, 2x2) arrangement machines	
– The individual machines 'in line' and constituents (3x4) arrangement machines	
– With individual machines "Trigon"	

Fig. 11.25. Schemes parlor "tandem" type:
1 – the machine; 2 – entrance gates; 3 – exit gate; 4 – operator place;
5 – feed hopper; 6 – feeder; 7 – sewer grate

In plants such as "Tandem" as a result of a comfortable work area operators higher labor productivity than in the milk during milking stalls in barns. Operators are in a trench depth 0,6...0,75 m on each side and parallel to which there are individual machines for cows. Each machine has its own milking machine. Entry and exit of cows in the machine operator operates individually for each cow, without interfering with other machines. At the same time feeding the cows eat from feeders during milking. The main advantage of this setup – the opportunity to milk the cows in the machine of any productivity and having different duration of milking. On installations with moving machines such as "Carousel" cow continuously go on the platform, and the operator is in one place, puts or removes the teat cups. Such conveyor systems are equipped with individual machines. Known schemes milking parlors such as "Carousel" (fig. 11.26.):

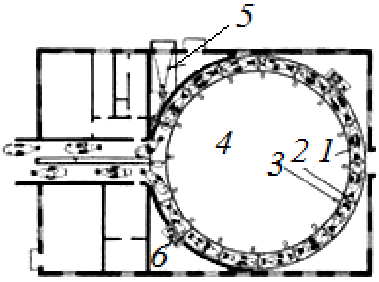
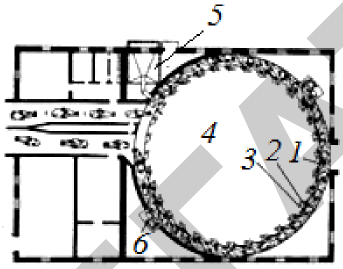
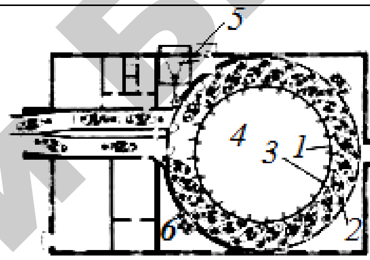
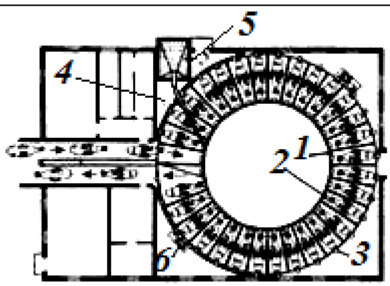
– With machines type "Tandem"	
– With machines such as "Herringbone" heads inside	
– With machines such as "Herringbone" heads out	
"Side by side" heads inside	

Fig. 11.26. Schemes milking parlors such as "Carousel":
1 – the machine; 2 – feeders; 3 – milking machine; 4 – operator place;
5 – feed hopper; 6 – station

Milking installation "Carousel" is a rotating platform in the form of an annular disk – the inner diameter of 12 m, the outer 15 m, which are mounted on machines with feeders such as "Herringbone." Power of Platform 4 kW of gear motor with stepless variator providing the platform speed within one revolution of 6...14 minutes. It applies to complexes and farms of industrial type with a well–chosen on livestock productivity and designed to continuously stream milking cows, milked milk transport milk in the room, filtering, cooling and feeding it into a storage tank. It allows you to milk the cows in the rhythm of the assembly line, which creates conditions for the automation of the milking process. Operators serving the population are inside the installation. The best conditions for the application installation – on farms with loose housing animals.

When milking the milk from the teat cup apparatus through the meter is sucked into vozduhorazdelitel. Further, milk pump and filter it through a plate cooler is fed into a container to collect and store. Sanitization udder cows produce four sprinklers is filled with warm water, which is supplied from the cylinder through the pipeline. Sprinklers are only for washing the udder. Installation can be ex–pluatirovatsya in two modes: milking and washing of milking machines. Aerial system operates as during milking and during automatic flushing circulation system. To reduce the ripple in the air system, and protect the pump from getting into the water and foreign bodies between him and the trunk duct mounted air tank. To prevent possible damage the air pump blade as it rotates in the opposite direction (by the vacuum in the line and the air cylinder upon termination of the milking machine) air line fuse installed before the pump. During the installation operation in a milk it maintained a constant level of vacuum – 48 kPa, and the air duct – 45 kPa. To ensure stable operation of the milk and the air line installation are combined in one process line, which lay on the edges of the trench. Adjusting the air mode produces the vacuum regulator.

3. Universal mobile milking machines are used in stanchion and loose housing cows. In winter, these installations are used in milking areas as fixed, and in the summer on the pastures as mobile. These include universal milking station (fig. 11.27), which section has two parallel through–machines installed on runners. The rupture between the machines are for dry feed hopper. They are displayed in the trough feeders machines attached to the doors, which are designed to release the animals from the machine and connected to the drive lever system handle. The station encompasses setup with vakuumsilovaya elektrodvi–motors, petrol engine, water and milk pumps, alternator with lighting equipment and installation of hot water.

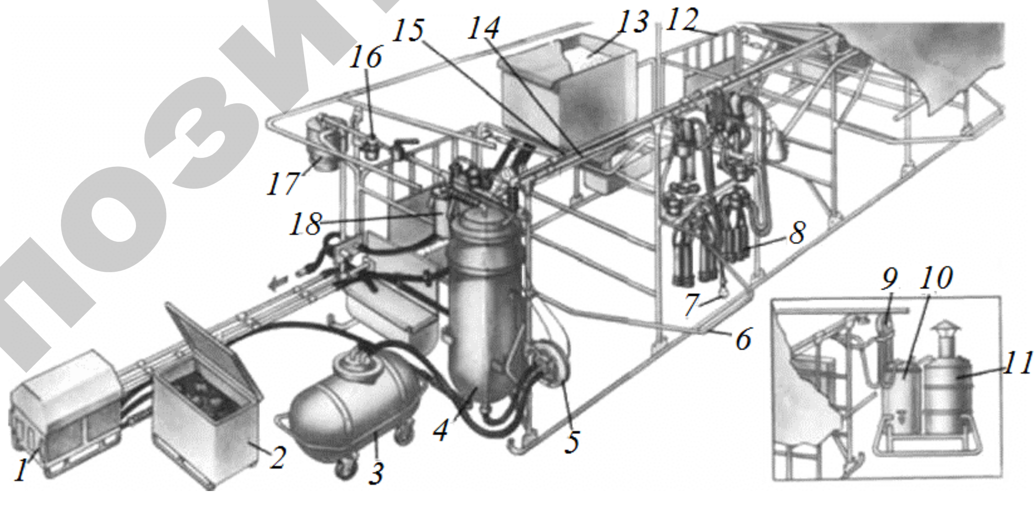


Fig. 11.27. General view of the milking machine:
1 – power unit; 2 – icebox; 3 – tank; 4 – filter–cooler; 5 – diaphragm pump; 6 – arc–lock; 7 – sprinkler; 8 – milking machine; 9 – pump–mixer; 10 – tank of cold water; 11 – boiler; 12 – exit door; 13 – hopper feeder; 14 – vacuumline; 15 – milk; 16 – vacuumregulator; 17 – vacuumballon; 18 – safety camera

At a distance of 100 m installation can be transported on runners at a rate of no more than 8...10 km/h. At large distances, milking machines are transported on special trucks and other equipment to other means of transport. For milking cows with loose housing company «Alfa–Laval» provides the following types of milking parlors:

– "Herringbone 30" (standard series: single–sided machines – from 1x3 to 1x6; bilateral – from 2x3 to 2x12);

– "Evroparallel" (standard series: single–sided machines from 1x4 to 1x12; bilateral – from 2x4 to 2x20);

– "Carousel" (modification of "Herringbone" from 16 to 40 stoylomest and "Parallel").

Milking parlors are modular and allow you to vary the amount of skotomest from livestock animals and milking parlors sizes. In the milking parlor "Evroparallel" cows are placed parallel to each other, back to the milking pit. Milking machines are connected between the hind legs. Group of cows leave the milking parlor at the same time, immediately after milking is finished. This increases the productivity of the plant. or any livestock milking cows at the fastened content of animals by «Alfa–Laval» comes milking machine having a modular design. The installation kit includes:

– Vacuumlines (core of plastic pipes 50 and 75 mm and a tap–st of precision steel tubes 50 mm) and milk pipes from nerzhaveyu–ing of steel 52 mm;

– Vacuum unit with the vacuum level controller;

– Milking "the MU 200 DeLaval – Duovac» (milking process is regulated by the flow of milk), «MU 100 the DeLaval» or «MU 350 the DeLaval – Milkmaster" (with an electronic pulsator, digital display with milk flow display, weight of milk and the time the main milking phase , automatic milking system selection mode and automatic removal of the suspension of the unit to the udder of the animal);

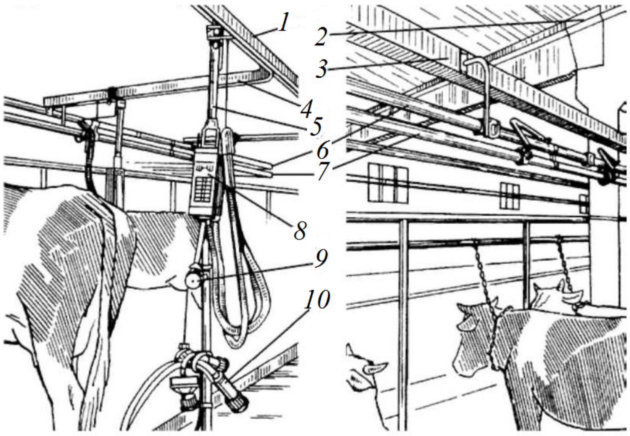
– Milk–receiver;

– Measuring device in milk yield during milking control – "Milkskop";

– Automatic washing the milking equipment;

– Equipment cooling and storing milk.

In an automated milking parlor with the milk «Milk Master» there is cableway and mobile milking machines. Cableway precludes portability of milking machines and facilitates the work of the milker. Milker runs with four milking machines 10 (fig. 11.28, a), move the bracket 5 on the suspension line 1. In order to avoid downtime milking machines, for filming a milking machine to others, provided the inner rail 4. Milking cows from the front. Milk and vacuum lines are mounted on the universal arm 2 (fig. 11.28, b), which is attached to the suspended channel (3). The milk is placed under vacuumpline 6. The milking machine is connected to the milk–vacuum line with a crane.



a) *b)*

Fig. 11.28. Fragments of milking machines of the company «Alfa–Laval Agri»:
a – with the milk on the brackets; b – with the milk «Milk Master»;
1 – suspension line; 2, 5 – an arm; 3 – channel; 4 – guide; 6 – vacuumline;
7 – milkline; 8 – control unit; 9 – removing milking device; 10 – milking apparatus

The milking machine with an electronic control unit 8 has a built–Flow–type electrode, which in reducing milk flow rate up to 0,2 kg/min gives a signal device 9 removing the milking machine. Milking machines can be equipped with stationary feeders, which are designed for transportation and dispensing of dry, free–flowing granules of animal feed with the size up to 14 mm. Feeders can be operated in manual and automatic filling dispensers. Bulk concentrate feed dispensing system comprises a drive station 1 (fig. 11.29) from the hopper 2, chain–shaybovy conveyor 8 disposed in the pipe, drives feed 6 dispensers 5, dosing control unit, the pneumatic system comprising pulsousiliteli 4 and vakuumprovody 3.

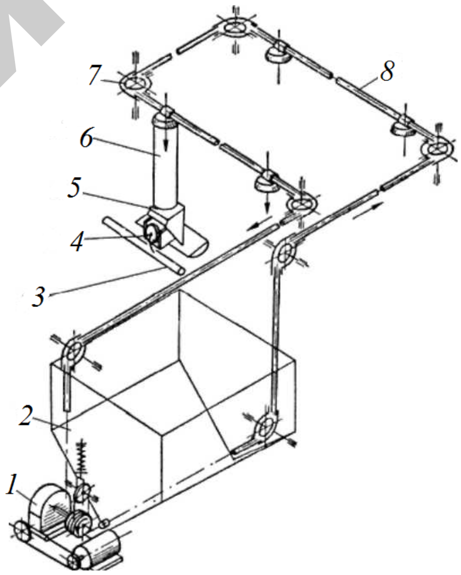


Fig. 11.29. Technological scheme of the feeder dry food:
1 – drive station; 2 – bunker; 3 – vacuumline; 4 – pneumatic chamber;
5 – dispenser; 6 – tubular drive; 7 – rotary unit; 8 – chain–conveyor feeder

When the drive motor sprocket chain conveyor extends through the hopper 2. Washers chain conveyor passing through the hopper, grab the food and deliver it to the pipe 5 drives, consistently filling them through a hole in the bottom of the tube on each drive. After filling the last drive of the corresponding micro drive off the conveyor drive. To determine the degree of filling of food drives them there are peepholes.

11.8 Robotic milking machines

The development of automated milking systems goes in two directions – one box with a robot arm and a robotic system of several milking boxes served with one hand. Performance odnoboksovyh robotic milking cows to 60 per day. odnoboksovyh several robots or one mnogoboksovaya robotic milking system can be used for large dairy farms. The use of robots takes into account the individual circadian rhythms of each cow. The cow herself is milking box where she, along with the milking, issued daily rate of concentrates. Animals quickly get used to the milking robots and independently attend boxing. Thus production of the cows rises to 15 %. Using robots enables 4 times reduce labor costs compared to milking milking installations like "Carousel." The milking robot «Astronaut» company «Lely» consists of a milking box with dimensions 4,5x2,5x2,5 m (fig. 11.30).

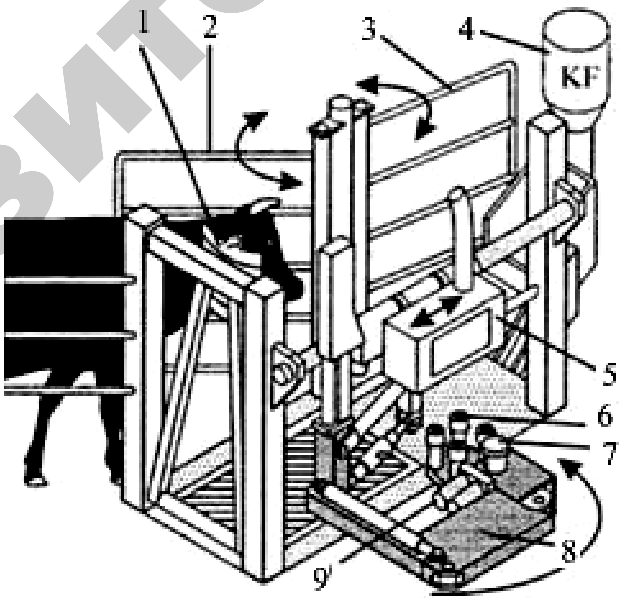


Fig. 11.30. «Astronaut» Driving milking robot:
1 – manipulator positioning the animal; 2 – front door; 3 – output door;
4 – Automatic Feeding Station; 5 – unit controlling the movements of the hands;
6 – teat cups; 7 – laser sensors; 8 – Robot Arm; 9 – rollers washing the udder

At the entrance there is a cow in a box of its identification and the computer determines the need for milking cows now, or to immediately release her from boxing. If you want to milk a cow, in the feeder portion of the issued (1,5...2,5 kg) of concentrated feed. animal movement behind the limited special manipulator 1. Approximately 10 seconds after positioning the cow robot arm 8 captures device 9 for washing the udder with two rollers, covered with a cotton cloth moistened with water and brings under the animal's udder. It determines the location of the nipple and begins the process of cleaning rotating rollers in opposite directions. After cleaning, the robot arm removes commercials in a special recess, where they are washing with water and disinfection disinfectant solutions.

Robot arm again brought under the cow, but the milking machine with 6 and 7 with a laser positioning it begins. To position as a reference point are the front teats. By positioning the end of the robot begins to consistently wear the teat cups on the teats, starting from the rear udder quarters. This mobile test plate transmits the motion to the cow by means of ultrasonic sensors of the robot arm, which follows the movement of the cow. In an unsuccessful attempt to wear glasses milking robot makes two more additional attempts. If unsuccessful third attempt, the robot produces cow, beeps and a message on the computer screen. The first trickle of milk remove in a special tank. The amount of milk produced and the electrical conductivity of each quarter of the udder of the animal goes through a separate milk line. The teat cups are removed from each teat individually, as the discontinuation of milk from it.

Robotic milking machine «Liberty» company «Prolion» includes up to 4 boxes served with one hand. At the entrance there is a cow of its identity and a decision about the need for milking. If a positive decision is issued in the corresponding feeder portion of the concentrated feed. Preliminary animal positioning is performed by moving the front wall of a trough (the size of the animal entered into the computer). Then, the robot arm is moved to the milking box, grabs the side of the console to the milking machine and brings it under the animal's udder. teat Locations are defined by two ultrasonic sensors. In this case the relative reference point is front–right nipple, which determines the coordinates of one of the ultrasonic sensors. Another sensor, moving downwards, determines the relative distance between the point and the other nipples. When moving the movable unit of the animal changes its position accordingly. Upon completion of positioning consistently put on the nipple teat cups and teat washing process begins in a glass of water jets. Used water together with the first trickle of milk is given in a special tank. After 8...10 seconds. after milking process starts. Quantitative indicators are monitored on each nipple.

After donning the teat cup the robot arm is returned to its original position and can be used in the other boxes. Under the udder of the animal milking unit is supported by a special side console. At the end of milking the teat cups to the teats of the udder subside and the console returned to its original position. Front wall with trough moves away from the cow, the door is opened, and the cow comes out of the box.

Milking robot «Merlin», manufactured by «Full–wood», aimed at robot «Astronaut» «Lely» company. «Merlin» from «Astronaut» Robot Robot The main differences are: the identification and registration of animal movements is carried out using a pedometer, mounted on the foot of a cow; the availability of energy–efficient systems six–minute cleaning and disinfection boiling water milking equipment.

The milking robot company «alfa–laval agri» brand «vms» (voluntary milking system) has the following features: the use of a pneumatic system to drive some of the elements of the robot (including mechanisms for donning and removal of the teat cups); application of four–point suspension mechanism teat cups to ensure their displacement in a horizontal plane.

Robotic milking system «Duvelsdorf» (company «Westfalia») includes 2 ... 4 Boxing tandem mounted in a row. They connected an additional box washing and breeding cows, thereby increasing the efficiency of the entire installation. The system works as follows. To prepare cows for milking member of the milking parlor the cow is kept in a box, where it is washed udders. a robot arm with a round brush extends under the cow. To clean the udder and teats with the brush washing water supplied to it rotates, moves back and forth. When the set ahead of time the water supply is stopped and the brush dries udder. Then the cow enters the milking machines one is identified and the computer decides whether the milking animal. Then the hand of the robot grasps the milking machine and moves under the udder, with the help of ultrasonic and optical sensors determines the location of the nipple (when contaminated optical date–sors are automatically washed with a damp sponge). The teat cups are put in series. Moreover, if the cow changes its position, the robot arm moves too. One robot arm serves all milking parlors, moving on a special guide. The process of milking, the teat cups removal, quality control and the amount of milk is performed similar to other automated systems milking.

The company «Gascoigne Melotte» developed a robotic «Zenith» system consisting of a milking box constructed as a combined section for milking and distribution of concentrated feed. Milking is done through the hind legs of the animal. Blows to the milking machine are eliminated with the help of two special brackets. A robotic arm holds the milking unit during the entire milking process.

The use of robots is constrained by their high cost. Also for milking robots requires careful selection of cows in the formation of the herd.

11.9 The technological design of the line machine milking cows

Organize a milking machine – so efficiently provide people to work, to choose and technically use the equipment correctly, pick up the cows, the most appropriate machine milking, and careful use of their level of productivity. Cows are well remember the daily routine, and they even milking sequence is created. The presence of unauthorized persons during milking of cows, the change of time and order processing and milking rough treatment (pain, fear), various kinds of noises have a negative effect on the animals and lead to premature fading of reflex milk ejection. Milking cows should be in the following order: first, the young, the old and then healthy, then treated and finally the patients. Technology calculation machine milking cows line comes down to the definition of the necessary number of units, number of operators to serve the whole population, the number of milking machines and their performance.

The task of calculating the production line includes the establishment of relationships between the set time of milking cows, the necessary number of units and machines, the number of milkmaids. Whole cow milking process is divided into three groups: preparatory operations, proper milking machine, final operations. In addition to the operations carried out by the operator with a cow, it is necessary to begin the process with the evaluation of the vacuum state, serviceability milking machines and other milking units. The preparatory operations include: udder washing, wiping his Forestripping first jets, massage. Washing the udder before milking, it is a potent stimulator of milk reflex prevents the transfer of germs from sick to healthy animals and reduce the likelihood of their falling into the milk during milking. Apply two ways of washing the udder before milking: 1) from a bucket; 2) a warm jet of water from a spray hopper.

The first processing method of the udder to be used in extreme cases as the water in the bucket becomes dirty quickly and mechanical particles and microorganisms require constant change. Jet wash the udder – is the most promising–tion to adapt quickly to launder even heavily soiled udders and also massage it. The water temperature (40...50 °C) should be constant, as fluctuations it can fail to stimulate milk flow, and its inhibition. Especially effective stimulation of milk, if using a pulsating flow of warm water. After washing the udder by any means wipe it dry with a paper or cloth, or a clean, well–wrung cloth. Simultaneously with the rubdown conduct vigorous massage of the udder, which greatly contributes to the well–expressed milk ejection reflex. In this case it is advisable to slightly compress the nipple in his fists and push them the base of the udder, as does a calf sucking the udder. Additional massage can be performed by compressing each teat at its base – the so–called false milking (without isolation of milk) for 4...5 seconds. At the same time wash, rubbing and massage of the udder should last no more than 35...40.

Before donning the teat cups on the teats, you must first streams of milk from each teat remove in a separate bowl. Appearances in the milk of cows cheesy clots, blood, pus, indicating one or another disease of the udder. From the patient sdaivayut quarter milk in a separate bowl. Milking of the first streams of milk from each teat is held for 8...12. All preparatory operations, in conjunction with the individual characteristics of cows, do not last more than 30...60 seconds. Only in this case the milk ejection reflex will be used most efficiently. In prepared udder (wash, wipe, massage, Forestripping first streams) immediately put the teat cups, warmed in a bucket of hot water (45...55 °C). Cold glass of milk inhibit reflex. In the process of milking is necessary to monitor the milk let down. When decay cups with teats must disconnect the vacuum cups to rinse contaminated water and gently massage the cow's udder, once again put them on the nipples. It is necessary to strictly maintain a constant vacuum and the number of pulsations, avoiding overexposure milking machines. The duration of one milking cows should be not more than 7 minutes, but can be different.

Final steps include a final massage milking, the machine shutdown. Final massage and milking (for 15...20) is performed to retrieve the latest, most doses of milk fat of the upper sections of the udder. In operation suitable to machine milking cows with good milk ejection reflex milking usually not required. Overexposure milking machines should not exceed 2 minutes and stroke – 1 minute. Over a long period of "idle" milking irritated teats and udder cisterns by "creeping" of the nipple, which has a negative effect on their condition, causing irritation and further inflammation.

Milking Reference Manual, determined immediately after removing the teat cups, to be not more than 200 ml, and from separate quarters of not more than 100 ml. In order to prevent disease of mastitis cows, after removal of the teat cups on teats dipped 2...3 seconds in a 1 % solution of iodine monochloride or chlorine preparations. Multiplicity milking set such that in the intervals between milkings the udder filled with milk, and not inhibited by molokoobrazovanie. Cows are usually milked 2...3 times a day, and high–calved 3...4 times. Before running the number of milkings gradually reduce. By reducing the number of milkings from three to two labor costs are reduced by 25...30 %. Do not allow alternate milking cows is trehtaktnym, the two–stroke machines, used flawed or improperly working and having great wear milking machines, join of different types of milking machines, remodel trehtaktnye devices on the push–pull mode of operation in terms of dairy farms and complexes. This leads to an increase in the number of cows that undergo disease mastitis. To calculate the cow milking process must be set number of animals cows. Milking time of the herd, or a particular group of cows at a shift–in–line equipment for zootechnical requirements of *Tg* = 2:00. In the large industrial complexes, which uses shift–threading system of keeping animals, the milking of the herd reaches 5...6 hours.

Number of milking machines, need for maintenance of the herd:

$$Z_{ап} = \frac{n_{ж} t_{маш}}{T_{д}}, \tag{11.39}$$

where $t_{маш}$ – number of cows on the farm; $t_{маш}$ – computer time milking a cow, minutes, $t_{маш}$ = 240...300; $T_{д}$ – the duration of the entire milking herd minutes.

When you receive a fractional number of milking machines, the resulting knowledge chenie rounded down.

One unit during milking of cows may serve:

$$n'_{ж} = \frac{T_{д}}{t_{ц}} \tag{11.40}$$

where $t_{ц}$ – fulltime milking cycle, s.

The number of cows, which can cater for one operator during milking (operator loading), defined by the equation:

$$n''_{ж} = \frac{T_{д} - t_{ц}}{t_{осн}} + 1. \tag{11.41}$$

To properly arrange machine milking cows, determine the number of staff:

$$n_{обс} = \frac{n_{ж} t_{р}}{T_{з} 60} \tag{11.42}$$

where $n_{ж}$ – number of cows on the farm, taking into account the planned development; $T_{з}$ – allowed the milking and processing milk (1,5...2 h); $t_{р}$ – time to perform manual operations on a per cow about 1...4 minutes.

Or determined by the expression

$$t_{р} = t_{осп} + t_{есп} + t_{тр}$$

where; $t_{осп}$ – time to perform the basic technological operations, c; $t_{есп}$ – time on auxiliary operations at milking a cow with; $t_{тр}$ – time transport operation, p.

The number of milking machines per operator:

$$Z_{an}^{/} = \frac{t_{ц}}{t_{p}} = \frac{t_{маш} - t_{p}}{t_{p}}. \tag{11.43}$$

Design capacity of the milking unit is given by the formula

$$Q_{д} = n_{жс}/T_{д}. \tag{11.44}$$

The required performance of the milking line, select the type of milking machine and determine their number:

$$Q_{ду} = Q_{д}/Q_{дуч}, \tag{11.45}$$

where $Q_{дуч}$ – hour performance of the milking plant.

Number of milking machines:

$$Z_{an} = \frac{z_{an}^{общ}}{z_{an}^{уст}}, \tag{11.46}$$

where $z_{an}^{общ}$ – number of milking machines on the same installation, pc; $z_{an}^{уст}$ – the total number of vehicles on all cows to be milking, pcs.

Selection of a milking plant for specific conditions is to select the type of milking machine used for the flock, and the installation itself, an appropriate detention conditions. It is important to avoid dry milking. This is possible with long duration of milking cows (productivity) and dual–mode devices using low vacuum during the final phases of the milking. The operator is obliged to develop a clear rhythm machines service and maintain it during milking.

Milking units are selected depending on the content of the cows system. The group selected cows' physiological condition: calved (1...3 months after calving), the first half of lactation (3...6 months), the second half of lactation (6 months or more). Cow group formed by the length of time milking and milk flow rate. The order of movement of cows for milking should be organized taking into account their physiological state: calved at the beginning, then the first half of lactation, and after the second half of lactation. Compliance with the rules of milking equipment helps to ensure a maximum milk yield. If timing data are very different from scientific evidence, the calculation can be carried out on the installation and start–up period, the second payment – after appropriate training of staff.

Chapter 12

Mechanization of primary processing of milk, cold formation

12.1 The primary processing operations and processing of milk

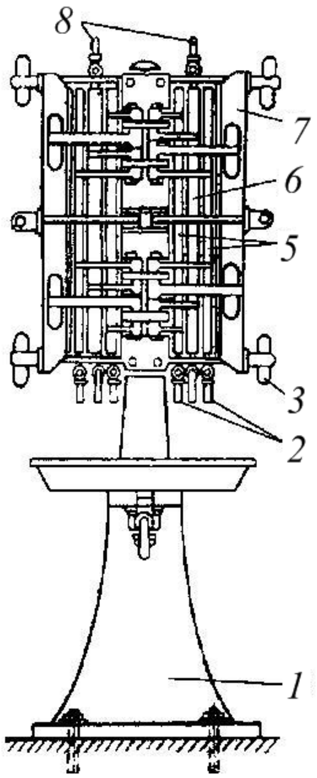
Milk quality is largely dependent on the timeliness of handling and processing, as milk is a perishable product. In order to maintain a fresh milk, it is subjected directly to the primary processing farms. This processing comprises the following process steps: filtering, cooling and storage. In some cases, they added pasteurization, separation and normalization. Milk processing is performed on dairy farms supplying milk products directly to the retail network, as well as the dairy industry. Cleaning the milk of mechanical impurities (residues of litter, food particles, hair, etc.) produced by centrifugal filters and milk purifiers.

Normalization of milk for fat content provides a product with a predetermined fat content. On farms normalization is done by separation. Fresh milk has antibacterial properties, which remain a certain time. By lowering the temperature of the milk, increase the validity of its bactericidal properties. In the non–refrigerated fresh milk at t = 30 ^{0}C bactericidal phase is 3 hours, the temperature drops to 16 °C –76 h to 10 °C ... 13…36 h to 4...5 °C – the vital activity of bacteria almost stops. The first step in the technological scheme of primary processing of milk is to clean it from mechanical impurities, for which the milk is passed through a mesh, gauze and flannel filters or using centrifugal cleaners.

When milking buckets in the milk is filtered during discharge it into the jar. As the filter elements using cotton pads, gauze, mylar cloth and disposable filters. Their disadvantage is the heavy pollution. Best results are obtained lavsan cloth or enanth which washed and disinfected after use. Cotton pads changed every 50...60 liters of filtered milk. Gauze does not provide a complete cleaning of milk from small impurities.

In the dairy industry apply filters to the metal (sieve) and tissue septa. Producing metal partitions wickerout and stamped, with the number of holes per square centimeter of from 25 to 100 units, ranging in size from 0,5 to 1,5 mm. The open area re–campus up to 50 %. canvas using different density tissue septa, and enanth and Dacron (one centimeter from 2 to 255 cells). The highest degree of purification is obtained, while the use–tion of wire mesh and filter cloth.

Apparatus closed filter plate shown in fig. 10.1. The filter consists of two sections 6. Each section consists of three chambers constituting a frame 5 with mesh and filter cloth 4. Milk alternately through the tubing 10, every 10...30 minutes, enters one of the chambers and the cells distributed parallel sections 6 . The supply of milk switch the three–way valve 9. When one section of the work, the other is cleaned and ready for operation (changing the filter cloth). The temperature of the milk in filtering 30...35 °C.



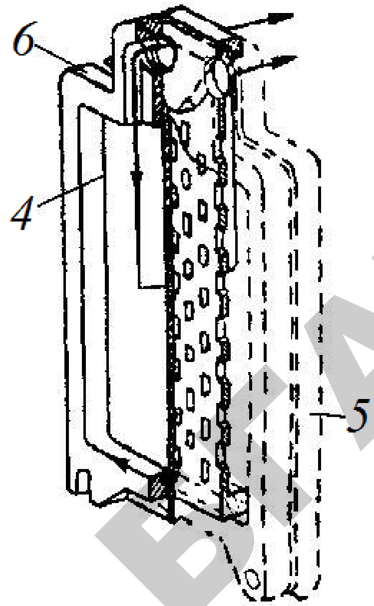


Fig. 12.1. Closed plate filter for milk:
1 – frame; 2 – drain cocks milk; 3 – clips; 4 – filter cloth; 5 – frame with a grid;
6 – chamber; 7 – cover sections; 8 – valves for air exhaust

For the production accumulated during the operation of the air from time to time–kryvayut cranes 8. seep through leaks in the collection of the milk gets 12. There also comes the rest of the milk in the sections at dismantling the sections. Disc indoor filter (fig. 12.2) consists of a steel casing 3, the cover 7, the valve 8, a set of filter discs 9 with holes 10, Th–cut which passes the milk, insert shims 11, inner cup 5, the cage 6, uprights 12 and a tap 2 for the descent of milk residues. Milk enters the Filter through pipe 4 passes through the opening 10 within through filter pads 11 and exits into the tube 1.

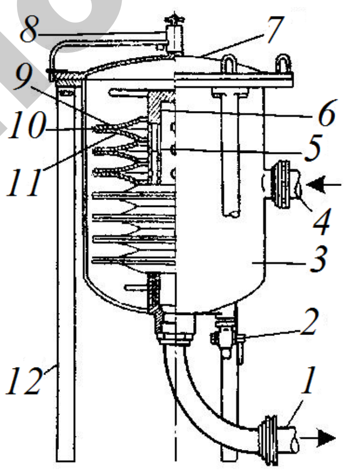


Fig. 12.2. Private disk cleaner for milk

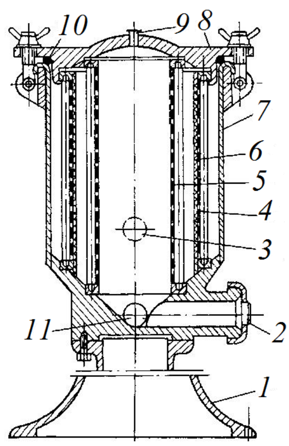


Fig. 12.3. Closed cylindrical filter

Cylindrical closed filter for purification of milk is shown in fig. 12.3. The milk enters the cylindrical filter is under a pressure of 200 MPa, six passes filter cloth stretched over the inner and outer screens 4 and 5. A milk inlet 3 leaves the filter. The tightness of the filter assembly is achieved by installing a cover 8 rubber pads 10. Trapped air is discharged through the tap on the pipe 9. 11 pressure gauge, which is controlled by the pressure. Before filtering the milk is heated to a temperature of 30 ... 40 ° C. The filter cloth must be replaced after 15...30 minutes. Equipment for cleaning the milk is usually part of the milking variables ency—units with the milk.

Milky main filter (fig. 12.4) consists of a body 1, the filter element 2, the sealing rings 3, 6 and 7. Milk guide passing under pressure or vacuum through the filter element 2, is cleaned from impurities. Last disassembled for cleaning and replacement of the filter elements (polyester or enanth).

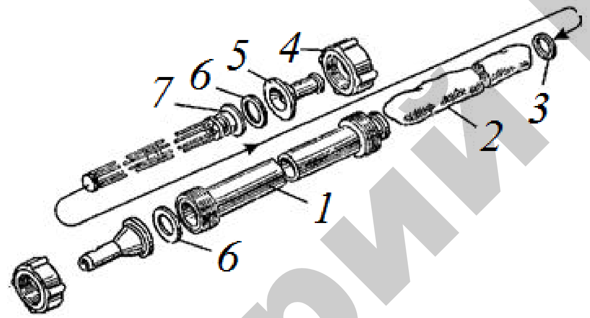


Fig. 12.4. Milk filter:
1 – the case; 2 – filter element–ment; 3, 6 – a sealing ring; 4 – a nut;
5 – adapter; 7 – directing

For the continuous operation of the milking plant installed in parallel and are connected via a three way valve. Performance filters is determined with a known working in surface. Number $V_ц$ product can be passed through the filter within one working cycle:

$$V_ц = qF, \tag{12.1}$$

wherein q – 1 m^2 load on the filter during the cycle of operation, L; F – the filter surface, m^2.

The performance of the filter is determined by the formula

$$Q_ф = qF / \Sigma\tau, \tag{12.2}$$

where Στ – the duration of one cycle of the filter, hour;

$$\Sigma\tau = \tau_f + \tau_{pp} + \tau_p, \qquad (12.3)$$

where τ_f – the duration of filtration hour; τ_{pp} – the duration of washing, hour; τ_p – the duration of loading and preparing the filter hour.
resistance partitions

$$R = 0{,}16\ R_0\ \mu, \qquad (12.4)$$

where R_0 – coefficient of resistance (for a dense canvas R_0 = (20...30) 10^{10}; for canvas medium density R_0 = (10...20) 10^{10}, for a rare canvas R_0 = (6...10) 10^{10}; for copper dense weave mesh R_0 = (7...12) 10^{10}; 50 % R_0 = 5 10^{10}) for extruded mesh with the living section.

Filtering surface is calculated according to the formula

$$F = V_l / Q_1, \qquad (12.5)$$

where V_l – the volume of liquid to be filtered, m^3; Q_1 – the performance of filtered surface, m^3/h.

Given the duration of the filtration surface of the filter installation:

$$F = (V_l/Q_1)\ t_{общ} / \tau_p, \qquad (12.6)$$

where $t_{общ}$ – the total duration of the filtration cycle, h.

12.2 Workflow and calculation of separator– milk purifiers

Separator–milkpurifiers used for cleaning milk from the outside in the impurities, the weight of which is higher than the proportion of milk plasma. Extraneous solids of different density vyde–lyayutsya with milk in a centrifugal force field. Thus the heavier components moving towards the periphery of the rotating rotor, lighter – they are displaced toward the center. The drum cleaners (fig. 12.5) differs increased mud space, its plates have no openings, no upper partition plate.

This separator consists of a frame 13, in the neck of which the spindle 12 is drum 10. Above the drum is closed drum 7. In–Water from the electric motor through a friction clutch and a pair of worm. For lubrication of worm gear inside the frame has an oil bath. The presence of lubricating oil is controlled by a level indicator window 3. Acceleration Time 3 ... 6 min depending on the type of separator.

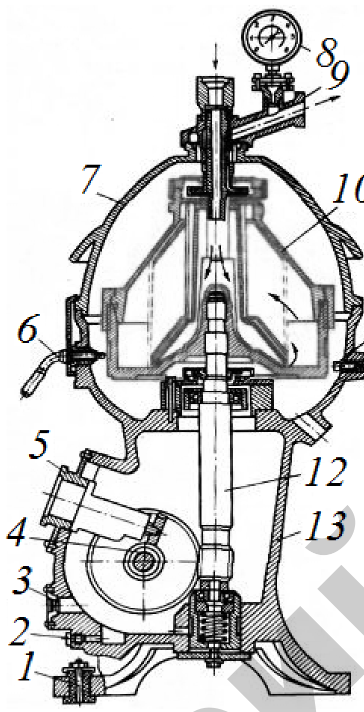


Fig. 12.5. Hermetic separator:
1 – frame; 2 – channel to drain the oil from the bearing; 3 – oil level indicator; 4 – shaft attachment the worm wheel; 5 – tachometer; 6 – clamp; 7 – the top plate; 8 – cap; 9 – milk receiver; 10 – disc stack; 11 – brake; 12 – vertical shaft (spindle); 13 – bed

Workflow–separator molokoochistitelya proceeds as follows (fig. 12.6). The drum cleaner rotates at a speed of about 8000 rev./min. Milk is dispensed through the throttle pump milk enters the receiving tube 8. It moves under tarelkoderzhatel 9 and under pressure enters the drum periphery. Since in this area the distance from the center of rotation significantly milk centrifugal force, and impurities having a specific gravity greater than milk, this power from the bulk milk pulled out and discarded in the on–board mud volume 10, where it accumulates in the form of so–called the separation of mucus. Purified milk thus, again under pressure entering the drum passes in the gaps between the cone plates, suitable for paring disc 11 and is output from the drum. Next comes the milk cooling.

In some drums separators–cleaners distance between the one–burner 1–2 mm, in other 8...10 mm. In platespace separator–purifier drum can be conventionally delimit the impurities precipitate separation zone 1 and zone reset them in the product 2 (fig. 12.7).

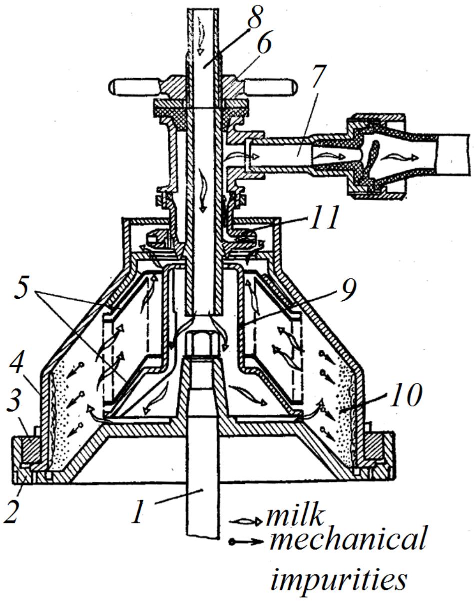


Fig. 12.6. Technological scheme of the workflow separator–purifier:
1 – drum shaft; 2 – housing base; 3 – nut; 4 – drum corps; 5 – plates;
6 – nut of the milk; 7 – milk tube; 8 – receiving tube; 9 –plate holder;
10 – mud camera; 11 – paring disc

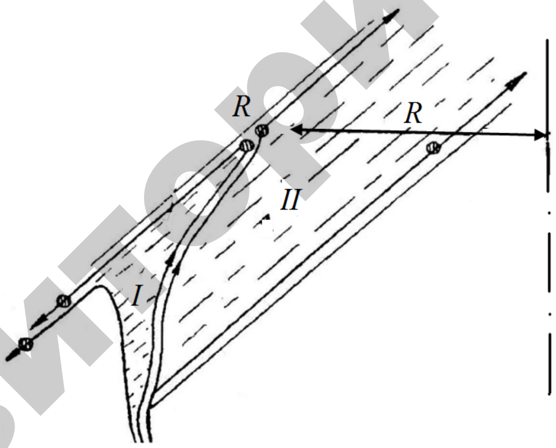


Fig. 12.7. The trajectories of the fluid flows in separator– milk purifiers:
1 – separation zone impurities in the sediment;
2 – reset impurities in the product zone

The theoretical performance of the drum plate can be determined by the formula

$$Q_{cen} = 0{,}116\omega^2 z tg\alpha \left(R^3_{max} - R^3_{min}\right)\frac{\left(\rho_c - \rho_ч\right)}{\mu_c}\mathrm{d}_ч \qquad (12.7)$$

where ω – the angular velocity of the drum, c^{-1}; z – number of plates within the drum; α – angle of lifting the cone plates, hail; R_{max} – the maximum current range plates, m; R_{min} – the minimum radius of the settlement plates, m; μ_c – dynamic viscosity of the dispersion medium, Ns/m^2; $d_ч$ – the minimum diameter of the dispersed particles secreted, m; ρ_c – density of the dispersion medium (plasma), kg/m^3; $\rho_ч$ – the density of the dispersed phase particle, kg/m^3.

This formula gives a good agreement between the theoretical performance of a cleaner with a passport with an estimated diameter of particles of 2 to 2.5 microns. When the distance between the plates, δ = 1...2 mm:

$$Q_{cen} = \frac{z\mathrm{V}_{расч}\omega^2}{35\cdot 10^6}, \tag{12.8}$$

where $\mathrm{V}_{расч}$ – the calculated volume of the drum, m^3.

When the distance between the plates δ = 8...10 mm:

$$Q_{cen} = \frac{z\mathrm{V}_{расч}\omega^2}{11\cdot 10^6}. \tag{12.9}$$

Theoretical performance:

$$Q_{cen} = 0{,}116\omega^2 R_{min}^2 \delta^2 z \frac{(\rho_c - \rho_ч)}{\mu_c} \mathrm{d}_ч cos\alpha, \tag{12.10}$$

where δ – the distance between the plates, m.

The optimal distance between the plates cleaner

$$\delta_{опт} = \frac{1{,}71}{R_{мин}} \sqrt[4]{\frac{Q_д \mu_c \left(R^3_{max} - R^3_{min}\right) tg\alpha}{\omega^2 z(\rho_c - \rho_ч) cos^2\alpha}}, \tag{12.11}$$

where $Q_д$ – separated flow rate of fluid through the drum, m^3/s.

Stir in the milk platespace defined by the relationship:

$$Q'_{cen} = \frac{Q_{cen}}{z} < \frac{442}{\delta} + 64, \tag{12.12}$$

where Q'_{cen} – the actual average load on platespac channel cm^3/s.

The milk flow, coming from the drum periphery, carries only those particles of dirt, the Stokes velocity at the boundary plates which is less flow rate at the entrance to the disc stack. Critical size entrained in the dispersed particles package:

$$d_{кр} \le \frac{1,69}{\omega R_{max}} \sqrt{\frac{Q_{д} \mu_{c} cos\alpha}{\delta z (\rho_{c} - \rho_{ч})}}, \tag{12.13}$$

where R_{max} – outer radius plates, m.

The minimum particle size is allocated to cleaner:

$$d_{min} = \frac{2,93}{\omega} \sqrt{\frac{Q_{д} \mu_{c}}{z (R^{3}_{max} - R^{3}_{min})(\rho_{c} - \rho_{ч}) tg\alpha}}. \tag{12.14}$$

The minimum size of the separated particles:

$$d^{/}_{min} = \frac{8,58 Q_{д} \mu_{c}}{z \omega^{2} R^{2}_{min} \delta^{2} (\rho_{c} - \rho_{ч}) cos\alpha}. \tag{12.15}$$

The time needed for the radial movement of particles calculated by R_{max} to R_{min}:

$$\tau_{расч} = 41,5 \frac{\mu_{c}}{d^{2}_{ч} \omega^{2} (\rho_{c} - \rho_{ч})} lg \frac{R_{max}}{R_{min}}. \tag{12.16}$$

The residence time in the drum of milk:

$$\tau_{б} = \frac{V_{б}}{Q_{д}} = \frac{H}{\upsilon_{св}}, \tag{12.17}$$

where $V_{б}$ – the volume of the drum, m^{3}; H – working height of the liquid ring in the bath–bar, m.

Critical size dirt particles released from the conditions $\tau_{b} = \tau_{расч}$:

$$d_{кр} = \frac{3,65}{\omega} \sqrt{\frac{Q_{д} \mu_{c} lg \frac{R_{макс}}{R_{мин}}}{(R^{2}_{макс} - R^{2}_{мин}) H (\rho_{c} - \rho_{ч})}}. \tag{12.18}$$

Length–purifier separator must ensure continuous operation of the milk processing time for one milking td without disassembly of the separator:

$$t_d = V_{гр}\ 100/P. \qquad (12.19)$$

where $V_{гр}$ – the volume of the mud drum space, m^3; P – percentage of fat separator mucus purified from total milk, P = 0,03…0,06 %.

The distance between the plates should not approach the critical value, at which the flow can turbulence. The amount of space taken Milk purifier mud at the rate of 1 liter per 1000 liters hour Flow rate. Milk purifier can work continuously 3 ... 4 hour. In this case, the deposition space in the mud reaches 0,03 % of the milk passed. The angle forming the plates to the horizontal shall be equal to 60...50^0. The energy expended in the separator is used to post the kinetic energy of the ejected liquid N_1, overcoming the air friction drum N_2, bearings N_3, in the gear N_4. Power required to report the kinetic energy of the ejected liquid is determined by the formula (for open separators):

$$N_1 = \varphi_c Q_c r^2 \omega_б^2, \qquad (12.20)$$

where Q_c – separator performance, kg/s; $\omega_б$ – the angular acceleration of the drum, s^{-1}; r – the distance from the rotational axis to the outlet, m; φ_c – coefficient taking into account the radial velocity of the jet (φ_c =1,1...1,2).

When calculating the power necessary for the operation of hermetic and semi–hermetic separators, instead of throwing power in the separation of the drum products address the needs of power for the pressure discs. The power needs to overcome the friction of the drum air is calculated by the formula

$$N_2 = \rho_в F_{бар} v_{бар}^3, \qquad (12.21)$$

where $\rho_в$ – the specific gravity of air in kg/m^3; $F_{бар}$ – The area of the lateral surface of the drum, m^2; $v_{бар}$ – Drum peripheral speed, m/s.

Power used to overcome the friction in the bearings supporting the drum shaft is calculated by the formula

$$N_3 = \frac{C_{тр} G_i v_{бар}^3}{d_i}, \qquad (12.22)$$

where G_i – bearing load in kilograms; d_i – trunnion diameter, m; $C_{тр}$ – coefficient ($C_{тр}$ = 0,002).

The power to overcome friction in the gearbox is determined on the basis of efficiency actuator η.

When worm gear:

$$\eta = \frac{tg(\alpha_л + \varphi)}{tg\alpha}(1 - 0,26f)\eta_0, \tag{12.23}$$

where $\alpha_л$ – the angle of ascent helix, hail; φ – the angle of friction (for bronze and steel shaft wheel φ = 40); f – the friction coefficient (f = 0,07); η_0 – coefficient taking into account the friction losses in the bearings (with ball bearings η_0 = 0,97).

Total power consumption:

$$N_{общ} = \frac{N_1 + N_2 + N_3}{\eta}. \tag{12.24}$$

The work required for the kinetic energy of the drum messages in acceleration between the separator is determined by the formula

$$A = \frac{I_{бар}\omega_{бар}^2}{2}, \tag{12.25}$$

where the $I_{бар}$ – the moment of inertia of the drum, kg/m c^2, $I_{бар} = m r_{ин}^2$.

or

$$I_{бар} = \frac{G_б}{g} r_{ин}^2, \tag{12.26}$$

where $r_{ин}$ – the radius of gyration, m; $G_б$ – drum weight, kg.

If the drum continues crackdown seconds, the average power requirement for a drum of kinetic energy during acceleration:

$$N_{ср} = \frac{A}{102\tau_1}. \tag{12.27}$$

In the acceleration period of the separator and the power needed to overcome friction in the starting mechanism (friction clutch with shoes or by sliding cams, or when the belt is slid to the idler pulley on the right). On average, the separator acceleration period starting device is absorbed about 40 % of the required power. Power, needs a separator between the power stroke, is distributed approximately as follows:

– friction in the starting mechanism – 25 %;

– by air friction drum – 50 %;

– on the kinetic energy of the ejected liquid message and pre–overcoming hydraulic resistance – 25 %.

The number of revolutions of modern separators are usually below the upper critical number. The actual number of revolutions of the drum in the range from 6,000 to 12,000 min^{-1}. In the presence of the masses debalansiruyuschih (pollution), incorrect or incomplete assembly, be sure there are significant drum beats, which can lead to an accident separators.

12.3 Workflow and calculation parameters, cream separator

Some dairy farms, as well as the dairy industry, milk processing is carried out, to which the separation of milk into cream and skim. Milk is known to be a mixture of fat density of 877...961 kg/m^3 and plasma (protein, water, minerals) density of 1006...1036 kg/m^3. Such dispersed mixture can be separated by separation in a centrifuge for mechanical centrifugal force field into two fractions – cream and skim milk (skim). In this case the heavier components of the mixture move to peripherical rotating rotor, a lighter – they are forced out to the center. The separation in centrifugal force field intensifies the process. Thus there is the possibility of separating the quality of regulation by changing the force field. In this connection, a mechanical separation of milk received its primary application in the treatment .

Quality and efficiency of separation depend on the following factors:

– purity and freshness of milk. The lower the acidity and contamination, the longer it can operate non–stop for the separator cleaning. Maximum acidity of milk – 220 T;

– size of the fat globules. The bigger the balls, the faster the separation;

– fat content of milk and cream. feeding it shall be reduced by increasing the fat content of milk. The maximum fat content of the cream is limited to 30...35 %, the residual fat content of less than skimmed milk 0.05 %;

– drum rotational speed;

– milk temperature. Optimal milk temperature 45...50 °C.

The physical nature of the process of separation of milk based on the precipitation of the dispersed phase by centrifugal force. The dispersed phase is milk plasma and milk fat is dispersed in the form of tiny balls (diameter: 0.1...0.01 mm). The conditions under which the concentration achieved the fat globules in milk plasma (cream, cream) and perhaps more complete degreasing of the rest of its parts, identify calculation.

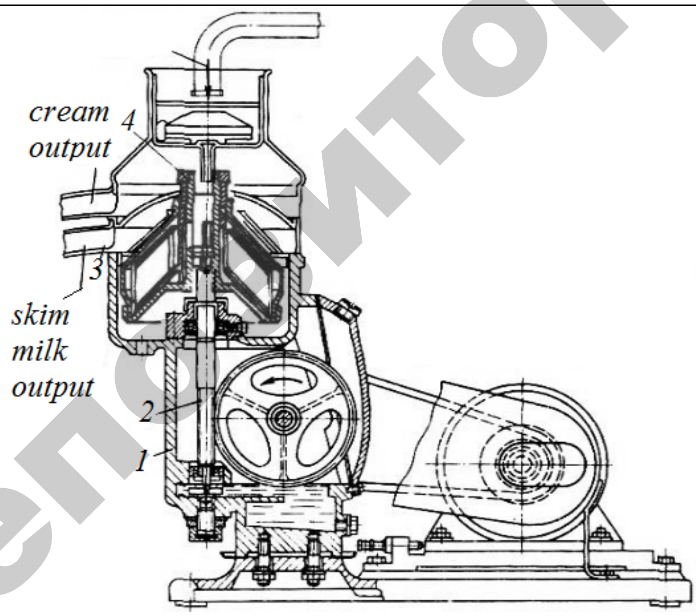
Separators design features may be:

– open – the milk and the removal of the separation products wasp–mented in contact with ambient air. Feed no more than 0,3 kg/s;

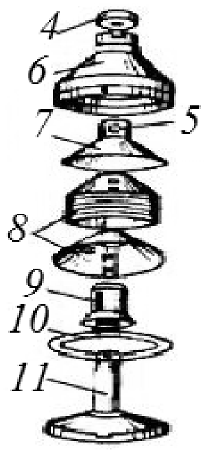
– half–closed – the supply of milk is carried out in contact with air, and the retraction – without contact. Feed 0,5...1,0 kg/s;

– sealed – characterized in that the supply of milk products and the removal of the separation occur without air pressure. They are used in pasteurization, cooling equipment. Feeding more than 1 kg s.

On dairy farms use cages open and semi–closed type. Separator cream separator–open type is shown in fig. 12,8. Its body is mounted on a base with an electric motor. The rotation of the motor is transferred to the drum freely planted on a vertical shaft (spindle), through a *V*–belt transmission, friction clutch and worm pair. separator shaft was placed in a frame on two pillars. The upper elastic support vertical shaft provides a self–adjustment of the drum at high speeds. The elastic support of the spindle is a ball bearing, enclosed in a cage. With the help of the adjusting screw thrust spindle can be moved in height, ensuring correct positioning of the drain holes of the drum with respect to dairy dishes. Wrong in the installation leading to penetration of the cream into skim.



– general form;



– details of the drum;

Fig. 12.8. Separators, cream separators open type:
1 – frame; 2 – a vertical shaft (spindle); 3 – dishes; 4 – Union nut; 5 – adjustment screw–fat cream; 6 – the case; 7 – the top plate; 8 – a package separating plates; 9 – plates holder; 10 – a sealing ring; 11 – drum base with a central tube

Cream separator drum (fig. 12.9) consists of a housing, packet dividing inserts plates holder, cover, seal–ring, and a tightening nut. The upper partition plate–IME is in the central part of the cylindrical hood, which is placed on the side with the hole soldered onto the adjusting screw. Turn of the Screw alter yield and fat content of the cream.

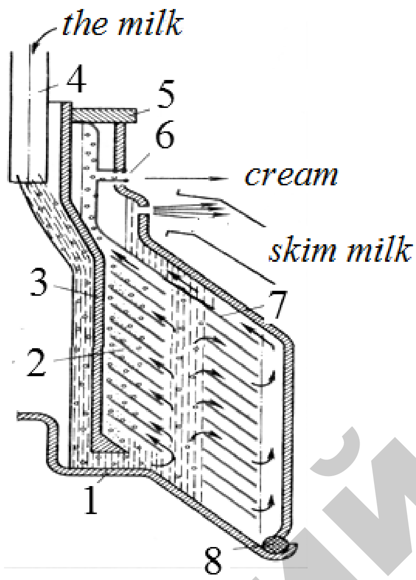


Fig. 12.9. Scheme of the cream separator drum separator:
1 – bottom; 2 – package of plates, 3 – plates holder;
4 – calibrated tube float chamber; 5 – union nut; 6 – adjusting screw–fat cream;
7 – the top separating plate; 8 – a rubber ring

Partition plates are provided with holes forming a three holes for passage of milk. The free space between the package of plates and the housing cover forms a sump. The gap between the pairs of plates in different designs is within 0.5...0.35 mm. Milk from the float chamber through the central tube and is supplied to the channels disc stack and moves plates gaps. Since the movement of the flow gaps mezhtarelochnyh Lamy–stationary–milk and milk particles – plane–parallel without twists (Stokes movement), then isolated from the milk fat globules, penetrate the surface of the plates and the flow of milk, by moving it in the direction of the drum axis.

Skim milk (skim) the centrifugal force directed perpendicularly to the axis of rotation (horizontal) moving toward the periphery of the drum, where it allocated sump from mechanical impurities. The purified milk (skim) passes over the separating plate to the holes for the ejection in dairy dishes. Thus, the milk trajectory consists of skimmed movement path in the direction opposite to the general flow (path 1) and the path of movement of the fat globules in the general direction of flow (path 2) (fig. 12.10).

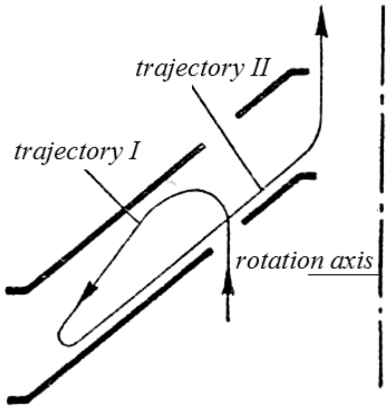


Fig. 12.10. The trajectory of the particles in mezhtarelochnom space, cream separator

One of the necessary conditions for the separation is the penetration of the fat globules through the thickness of the liquid. The particles (fat globules) have not reached the surface of the plates are carried into streams to address. Fat globules which penetrated the surface plates are moved in it.

Obviously, if skimmed milk flow rate at the surface speed of the plates over the fat globules are blown they flow (fig. 12.11). If skimmed milk flow velocity less than the velocity caused by the centrifugal force, the particles will move towards the axis of rotation and will fall into the stream of the fat globules (cream).

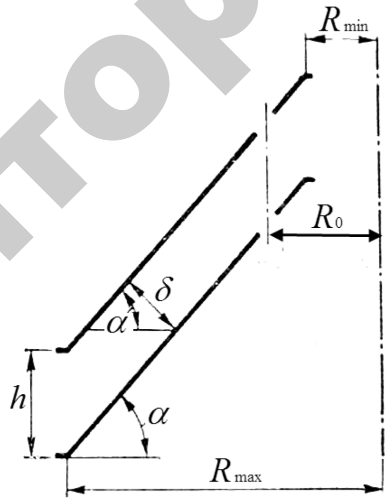


Fig. 12.11. Driving section

The radial velocity of the particles in gap is defined by the formula

$$\upsilon = \frac{1}{18}\omega^2 R d^2 \frac{(\rho_c - \rho_ч)}{\mu_c}, \tag{12.28}$$

where ω – the angular velocity of the drum, c^{-1}; R – the radial distance from the rotational axis of the particle, m; d – diameter of the fat globules (the dispersed phase) m; ρ_c – density of cream (dispersion medium), kg/m^3; $\rho_ч$ – skimmed density particles (dispersed phase), kg/m^3; μ_c – cream viscosity, Ns/m^2.

Ultimately the minimum size allocated cage Ms. rovyh–balls:

$$d_{мин} = \frac{2,94}{\omega}\sqrt{\frac{Q_д \mu_c}{\beta z tg\alpha \left(R^3_{макс} - R^3_{мин}\right)\left(\rho_c - \rho_ч\right)}}\,. \tag{12.29}$$

The minimum size of the fat globules, which reached the cream layer is not washed away in the flow of skimmed mud drum space:

$$d^{/}_{мин} = \frac{17,2 Q_д \mu_c}{\omega^2 z \delta^2 R^2_{мак}\left(\rho_c - \rho_ч\right) cos\alpha}\,. \tag{12.30}$$

From the equality of dmin = d’min δ is determined by the distance between the plates

$$\delta = \frac{2,43}{R_{мак}}\sqrt[4]{\frac{Q_д \mu_c \beta\left(R^3_{мак} - R^3_{мин}\right) tg\alpha}{\omega^2 z\left(\rho_c - \rho_ч\right) cos\alpha}}\,, \tag{12.31}$$

where z – number of plates in the drum units; α – angle of lifting the cone plates, hail; $R_{мак}$ – the maximum current range tareliki, m; $R_{мин}$ – the minimum radius of the settlement plates, m.

The theoretical performance of the separator is determined by formula.

$$Q_т = 0,116\omega^2 z tg\alpha\left(R^3_{макс} - R^3_{мин}\right)\frac{\left(\rho_c - \rho_ч\right)}{\mu_c} d^2_ч\,, \tag{12.32}$$

where $d_ч$ – the minimum diameter of the fat globules excreted. m.

Formula proposed performance of the conditions of the particle retention on the periphery of the plate in the form of:

$$Q_д = 0,058\omega^2\delta^2 R_{мак} z d_ч cos\alpha\,\frac{\left(\rho_c - \rho_ч\right)}{\mu_c}\,, \tag{12.33}$$

where δ – gap normal to the generator, m.

The actual performance of the separator is less than the theoretical. The ratio of actual performance to the theoretical:

$$\frac{Q_{\partial}}{Q_{\tau}} = \beta = 0,5...0,7\,, \tag{12.34}$$

where β – the process, the efficiency of the separator.

The optimal distance between the axle and axle plate holes in it is determined by formula:

$$R_0 = \sqrt{\frac{\varphi\, R^2_{мак} + R^2_{мин}}{1+\varphi}}\,, \tag{12.35}$$

where φ – volume ratio of light phase to heavy.

The overall performance, cream separator has a degree of degreasing:

$$\sigma = 100\frac{C_{сл}(C_{м} - C_{об})}{C_{м}(C_{сл} - C_{об})},$$

where $C_{об},\ C_{сл}$ and $C_{м}$ – the fat content of skimmed milk, cream and milk, %.

The amount of cream allocated cage:

$$Q_{сл} = Q_{м}\frac{C_{м} - C_{об}}{C_{сл} - C_{об}}\,. \tag{12.36}$$

On the separation quality significantly affected by the following constructions–tively–mechanical factors:

– number of holes for entry of the product, their position and the radius. The number of holes should be minimal (3...4), but sufficient to ensure that the liquid at the periphery of the plates spread throughout their circumference. The distance between the holes should provide unhindered by current–cream;

– channels for the passage of the original product and fractions separirova–tion should be streamlined to minimize the possibility of flow disturbance in order to avoid deterioration of the separation process and homogenization effect;

– angle forming the plates and dishes on the periphery of the limb take from 40 to 60^0. It should provide sliding masses are deposited on the surface of the plates;

– the process of separating milk significantly affects the edge of plates. It provides the necessary lockup liquid at the periphery of plates which should spread over their entire circumference. Form bending edge plates should not promote mixing;

– separator drum must be carefully is balanced, because only under this condition the high quality of separation. With the reduction, reduced wear and power requirement of the drum vibrations significantly improves the quality of the separation of liquids;

– mud drum space must accommodate the sediment deposited during operation of the separator.

12.4 Calculation of the main parameters of milk pasteurizers

In order to destroy bacteria in milk it is heated to a certain temperature. all pasteurizers meet the following requirements:

– the complete destruction of all forms of microbes;

– processing should not impair properties of milk;

– simplicity of the device and operation;

– the surface in contact with the milk must be resistant to chemical attack of milk and cleaning liquids.

Pasteurizers of milk divided by:

– energy supply into the steam, electrical with induction heating, radiative;

– the character of a process – into continuous and periodic action;

– the constructive implementation – plate, tubular, centrifugal with displacement drum, capacitive with jacket and shackle;

– the number of sections – single, double, multi–sectional or combined;

– the direction of the fluid and coolant – direct–flow and counterflow;

– the Processing method – thermal, in which the milk is heated below the boiling temperature and the cold in which bacteria are killed by various physical effects – by irradiation with ultraviolet or infrared rays, radiation, sonication etc.

The most widely used in the processing of milk are thermo–cal pasteurizers, which by a work schedule are divided into three types:

– continuous milk pasteurization apparatus in which heating is done to 63...65 °C and held at this temperature for 30 minutes;

– short pasteurization apparatus in which heating is done in a thin layer to a temperature of 76 ± 2 °C, held for 20 seconds;

– flash pasteurization apparatus in which the milk is heated for a few seconds up to 85...87 °C without further exposure.

The devices of continuous pasteurization are used for milk heating before its separation or fermentation. Long pasteurization has the greatest effect on the physico–mechanical properties of milk, but it provides reliable destruction of all types of microorganisms except for heat–resistant bacteria. Long pasteurization of milk is carried out in Baths of prolonged pasteurization that are identical in design, have a *water jacket* around the working container and a stirrer with a drive. The pasteurization bath (fig. 12.12) consists of an inner stainless steel case enclosed in a double–walled outer case. Under the inner case there is a steam device, to the outlet branch of which a collector is connected. The branch for draining water from the interstice space is brought down and connected to the shutoff valve.

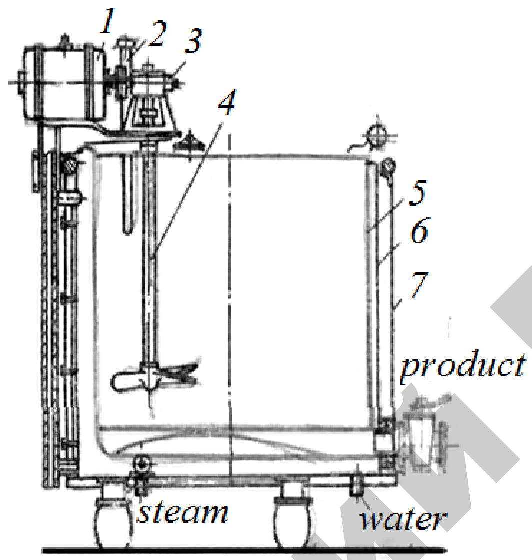


Fig. 12.12. The scheme of a bath of long pasteurization:
1 – electric motor; 2 – thermometer; 3 – reducer; 4 – agitator; 5 – internal reservoir; 6 – external reservoir; 7 – cladding

Mixing of the product is carried out by a stirrer rotating from the electric motor fixed to the plate. Draining of the finished product is produced through the shut–off. Temperature control of the product and water in the inter–wall space is carried out by thermometers. The bath is filled with milk. Then the inter–wall space is filled with water to the level of the overflow pipe. Water is heated by steam and by heat exchange through the walls of the inner shell it heats the milk. To increase heat transfer, the milk is mixed with a stirrer. Cooling of the product in the bath is accomplished by filling the interstice space with ice water. After everything is done, the product is removed through a dairy faucet, the water from the interstice space is removed through the drainage pipe.

Short–term pasteurization is carried out in steam pasteurizers with a displacement drum and in plate heaters with heating by means of hot water. Plate pasteurizers do not have moving parts. the heat exchange occurs there between the streams of hot water and milk, separated by thin plates of stainless steel. Between the plates, milk and water alternate in countercurrent. Milk and water pumps create the pressure necessary for traffic flow. In the automated plate–milling pasteurizing–

cooling plant (fig. 12.13), the working process proceeds in the following order. Milk from the milk collector is fed by gravity or by means of a pump into the equalizer tank 4. The milk level must be at least 300 mm in order to avoid sucking air into the milk pump. By means of Pump 3 milk gets to the section I of the plate device (regeneration section) where it is heated by heat exchange with hot milk from the pasteurization section through the holder 6. The milk heated to 37…40 °C leaves the section into the milk cleaner, And from there it gets to the second regeneration section, where it is further heated by the pasteurized milk that has passed the heat exchange in the regeneration section I. From the regeneration section II, the milk passes to the pasteurization section III, where it is heated by heat exchange with hot water to 76 °C or 90 °C.

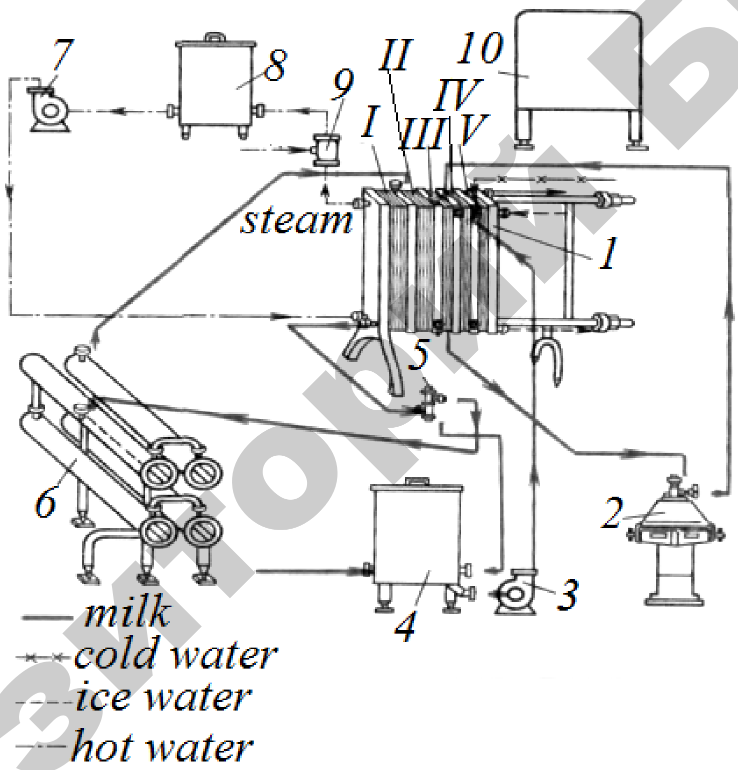


Fig. 12.13. The scheme of the automated plate–type pasteurization–cooling plant: I – first regeneration section; II – second regeneration section; III – pasteurization section; IV – water cooling section; V – section of brine cooling.
1 – plate device; 2 – separator–milk cleaner; 3 – the centrifugal pump; 4 – equalization tank; 5 – bypass valve; 6 – stand holder; 7 – hot water pump; 8 – boiler; 9 – injector; 10 – control panel

Pasteurized milk passes through a stand in the I and II regeneration sections, where it gives off some of the heat to the cold milk and its temperature drops to 20…25 °C. Further, this milk passes successively the sections of the coolant, after which its temperature drops to 5...8 °C, depending on the initial temperature of the cooling water or brine. Cold milk comes into tanks for storage. Two factors are detrimental to microorganisms: the temperature of heating and the duration of its exposure. At the same time, the higher the temperature, the less exposure is needed. There is an upper temperature threshold to which there is no change in the physico–mechanical properties of milk. In this regard, in order to ensure pasteurization and not deteriorate the quality of milk, it is necessary to strictly adhere to the boundaries between the lower and upper temperature limits. In pasteurizers of long–term and short–term exposure, hot water is used as a source of heat, saturated steam in instantaneous pasteurizers. For any type of pasteurizer, their thermal performance:

$$T_п = Q_м C_м (t_{км} - t_{нм}) = k_{ст} \ F_{тепл} \Delta t_{ср} , \tag{12.37}$$

where $Q_м$ – milk supply, kg/s; $C_м$ – heat capacity of milk, J/kg deg; $t_{н\ м}$ and $t_{к\ м}$ – the initial and final temperatures of milk, respectively, degrees; – coefficient of heat transfer through a flat wall, W/m^2 deg; $F_{тепл}$ – total heat exchange surface, m^2; $\Delta t_{ср}$ –is the average temperature gradient between heat exchange media, deg.

The thermal performance of the pasteurizer (kcal/h) depends on the size of its heating surface, heat transfer coefficient and average temperature difference between the steam in the jacket and the product and is characterized by the equation:

$$\theta = F_{тепл} k_{ст} \ \Delta t_{ср} = Q_{охл} C_м (t_{к.м} - t_{н.м}). \tag{12.38}$$

For water pasteurizers:

$$\Delta t_{ср} = \frac{\Delta t_{\max} - \Delta t_{\min}}{\ln \dfrac{\Delta t_{\max}}{\Delta t_{\min}}}, \tag{12.39}$$

And for steam pasteurizers:

$$\Delta t_{ср} = \frac{\Delta t_{км} - \Delta t_{нм}}{\ln \dfrac{\Delta t_{\max}}{\Delta t_{\min}}}. \tag{12.40}$$

Total heat exchange area of the pasteurizer:

$$F_{тепл} = \frac{Q_м C_м (t_{км} - t_{нм})}{k \ \ \Delta t_{ср}}. \tag{12.41}$$

And for steam pasteurizers:

$$F_{тепл} = \frac{1}{k} Q_м C_м ln \frac{\Delta t_{max}}{\Delta t_{min}}, \qquad (12.42)$$

where for the steam pasteurizer k ≈ 105 MJ/m^2.h.deg.

Instant pasteurization of milk is carried out on apparatuses in which milk is heated for a few seconds to 85...87 °C without further exposure. These include devices with a displacement drum. Milk in them passes in the gap between the walls of the displacement drum and the stationary reservoir, which has a paraboloid shape. Steam enters inside of the drum and outside the tank. The drum is driven by the electric motor. In a steam pasteurizer with two–sided heating (fig. 12.14), low–pressure steam gets into the space of the steam jacket and into the cavity of the displacement drum.

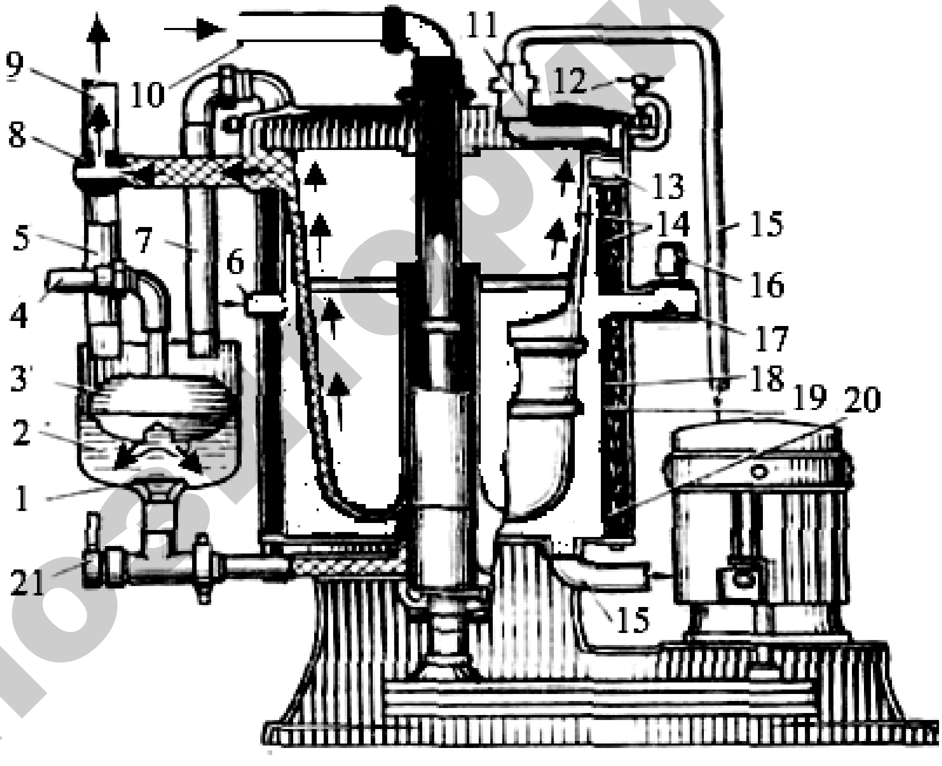


Fig. 12.14. The Scheme of steam pasteurizer:
1 – removable insert; 2 – funnel of the milk collector; 3 – float pressure regulator; 4 – milk supply pipe; 5 – drain pipe; 6 – steam inlet pipe; 7 – overflow pipe; 8 – crane; 9 – milk drain pipe; 10 – branch pipe of a supply of steam in a drum; 11 – upper condensate collector; 12 – screw; 13 – scapula; 14 – steam jacket; 15 – condensate drain pipes; 16 – steam valve; 17 – air valve; 18–bath; 19 – drip rings; 20 – lower condensate collector; 21 – drain cock

He gives his heat to the milk, passing from the jellied funnel through the gap between the working surfaces of the pasteurizer. Heated milk, rising in the gap, falls under the action of the blades of the drum cover and is pumped through the discharge pipe for further processing. The steam jacket has two nozzles. The upper serves for connecting the safety valve, the lower one for the supply of steam. The bottom of the steam jacket is provided with a condensate outlet. At the inlet of the bath there is a receiving funnel with a float device, which ensures a uniform supply of milk to the pasteurizer.

The productivity of the pasteurizer is regulated by interchangeable inserts in the funnel. When the drum rotates, pressure is created, and the condensate is withdrawn through the tube into the condensate drain. In pasteurizers with a displacer, the main process is heating, which results in a pasteurizing effect. However, these devices also work as pumps. The pasteurized liquid, induced by the rotating propellant, acquires a constant angular velocity and forms a paraboloid of revolution. its depth is determined by the formula

$$h = \frac{4\pi R_{мак}^{2} n_{выт}^{2}}{2g}, \tag{12.43}$$

where $n_{выт}$ – is the number of revolutions of the displacer, min^{-1}; $R_{мак}$ – is the maximum radius of the propellant blades in the enlarged part, m.

Knowing n and R, you can calculate the height of the rise of the liquid:

$$h_{ж} = \frac{\omega_{ж}^{2} h_{0}^{2}}{2g} = (R_{мак}^{2} - r_{0}^{2}), \tag{12.44}$$

where r_0 – is the radius of the cavity formed under the cover, m. The minimum value of r_0 is equal to the distance from the center of the lid to the overflow pipe, over which the excess of pasteurized liquid returns to the receiving funnel; h_0 – is the total value of the actual lifting height and resistance in the pipeline, m; $\omega_{ж}$ – angular velocity of rotation of the liquid, s^{-1}.

The rotating liquid exerts considerable pressure on the lid and the walls of the pasteurizer. The force applied to the lid of the pasteurizer on the liquid side is determined by the formula

$$P_{к} = \frac{\pi g h_{0}^{2} \gamma}{\omega_{ж}^{2}}. \tag{12.45}$$

The pressure reaches significant values, so the fixing and sealing between the lid and the body of the pasteurizer must be reliable. Pressure on the walls of the reservoir is calculated by the formula

$$p = \frac{\omega_{жс}^{2}\gamma}{2g}(R_{мак}^{2} - r_0^2). \tag{12.46}$$

The wall thickness is determined by the equation:

$$\delta = \frac{pr}{\sigma}. \tag{12.47}$$

In machines with a rotating propellant, the pasteurization effect is achieved during the heating of the liquid to the pasteurization temperature in the gap between the reservoir and the propellant and during its holding in the expanded part of the pasteurizer. In this regard, the following two conditions must be observed:

1. For a given temperature regime, the heating surface of the pasteurizer must correspond to the required capacity, i.e.

$$k_{ст}F_{выт}\Delta t_{ср} = Q_{ап}C_{с}(t_{к.м} - t_{н.м}), \tag{12.48}$$

where $k_{ст}$ – is the coefficient of heat transfer from the steam to the wall; $F_{выт}$– displacer surface, m^2.

Determined by geometric formulas based on measurement data; $\Delta t_{\text{ĭð}}$ – мean temperature of heating. It is assumed, based on the conditions that the temperature of the vapor is constant, and the liquid temperature rises from t_1 to t_2.

2. The holding time must satisfy the equation:

$$\tau = \tau_1 + \tau_2, \tag{12.49}$$

where τ_1 – is the duration of heating of milk in the gap between the reservoir and the pasteurizer displacer, h; τ_2 – the duration of the product in the expanded part of the pasteurizer, h.

In its turn:

$$\tau_1 = \frac{3600 V_1 \gamma}{Q_n}; \tag{12.50}$$

$$\tau_2 = \frac{3600 V_2 \gamma}{Q_n}, \tag{12.51}$$

where V_1 – the volume of fluid in the gap, m^3; Q_i – productivity of the pasteurizer, kg/h; V_2 – is the volume of liquid in the expanded part of the pasteurizer, m^3,

$$V_2 = \pi (R^2_{мак} - r_0^2) H_p, \tag{12.52}$$

where H_p – is the height of the widened part, m.

Productivity of the pasteurizer:

$$Q_n = \frac{k_{ст} F_{выт} \Delta t_{ср}}{C_м (t_{к.м} - t_{н.м})}. \tag{12.53}$$

The steam consumption for pasteurization is determined by the heat balance equation

$$Q_{пара}(C_n - C_к t_{конд})\eta = Q_{расч} C_м (t_{к.м} - t_{н.м}), \tag{12.54}$$

where C_n – heat content of steam, kcal/kg; $C_к$ – is the heat capacity of the condensate, kcal/kg deg; $t_{конд}$ – condensate temperature, degree (condensate temperature is 4...6^0 below the heating steam temperature); $t_{к.м}$,$t_{н.м}$– the final and initial temperatures of milk, degrees; $Q_{расч}$ – the amount of milk to be cooked, kg; $C_м$ – heat capacity of milk, kcal/kg hail; η – is the thermal efficiency of the apparatus (η = 0,8...0,95).

In instantaneous pasteurizers with a displacement drum, the milk, under the action of the constituent of the centrifugal force, rises upward along the paraboloid surface of the drum. In the upper part of the drum there are blades, under which the milk is injected into the pipeline.

The pressure developed by the blades can be determined by the formula

$$H_л = \frac{v^2_{бар}}{2g}, \tag{12.55}$$

where $v_{бар}$ – is the circumferential speed of the blades of the drum, m/s.

The developed pressure can be spent on the overcoming of the hydraulic resistances of the milk pipeline and the geodetic lifting of milk. If we take into account that $v_{бар} = 2\pi rnb$, where r is the radius of the drum blades, the required drum rotation frequency for overcoming the hydraulic resistance, equal to $H_л$, can be determined by formula

$$n_б = \sqrt{\frac{2gH_л}{4\pi^2 r^2}} = \frac{1}{2\pi r}\sqrt{2gH_л}. \qquad (12.56)$$

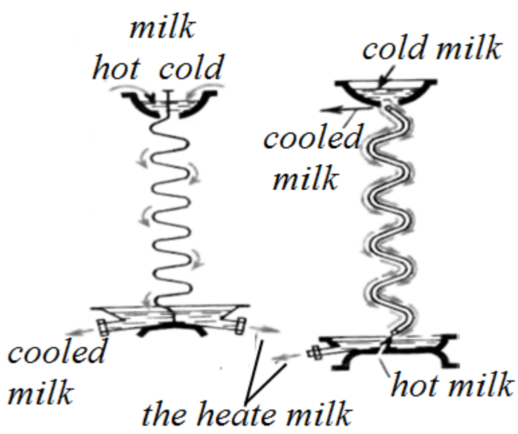
The total power for the pasteurizer drive will be determined:

$$N = N_{бар} + N_{сопр}, \qquad (12.57)$$

where $N_{сопр}$ – the power to overcome the hydraulic resistance, equal to $N_{сопр}$ = g $Q_м$ N, kW; $N_{бар}$ – power to drive the drum, $N_{бар} = P_{окр}\ v_{бар}$, kW; Rockr is the circumferential force on the drum drive, with steady motion predetermined by frictional forces in bearings, N.

12.5 Purpose and calculation of regenerators–heat exchangers

The purpose of the regenerator is to use the heat of the pasteurized milk to heat the cold milk going to pasteurization. At the same time, the heat consumption for pasteurization is reduced, since the milk is already preheated into the pasteurizer. Regenerators–heat exchangers with a parallel current of milk and countercurrent are used in lamellar pasteurization–cooling units, in the form of separate apparatuses (fig. 12.15).



a в

Fig. 12.15. Open milk regenerators:
a – direct–flow ("zigzag"); в – countercurrent

The coefficient of regeneration shows the ratio of the amount of heat given off on the regenerator by hot milk to the cold, to the amount of heat that is necessary to heat the cold milk to the pasteurization temperature. Almost for recurrent regenerators, the value of regeneration coefficient is 0,3, for countercurrent 0.5...0.8, in plate devices due to the developed surface of heat exchange, reaches values of 0,9...0,93.

The amount of heat obtained by milk going to pasteurization:

$$\mathrm{G}_{\mathrm{p}} = Q_{м} C_{м} (t_{рм} - t_{нм}). \tag{12.58}$$

The amount of heat required for the pasteurization of milk is determined by the equation:

$$\mathrm{G} = Q_{м} C_{м} (t_{км} - t_{нм}), \tag{12.59}$$

The ratio of the amount of heat received in the regenerator to the amount of heat needed for pasteurization is called the regeneration factor:

$$\xi = \frac{\mathrm{G}_{\mathrm{p}}}{\mathrm{G}} = \frac{Q_{м} C_{м} (t_{рм} - t_{нм})}{Q_{м} C_{м} (t_{км} - t_{нм})} = \frac{t_{рм} - t_{нм}}{t_{км} - t_{нм}}. \tag{12.60}$$

Since the consumption of milk that goes to pasteurization and is cooled after the pasteurizer is the same, in the counterflow regenerative heat exchanger, the temperature head along the entire path of milk movement can be determined by the formula:

$$t_{н} = t_{км} - t_{рм} = \Delta t_{ср}. \tag{12.61}$$

We substitute this value in the regeneration factor:

$$\xi = \frac{t_{рм} - \Delta t_{ср} - t_{нм}}{t_{км} - t_{нм}}, \tag{12.62}$$

then:

$$t_{ср} = (1 - \xi)(t_{к\,м} - t_{н\,м}). \tag{12.63}$$

The area of the heat exchange surface of the regenerator is determined from the previously given formulas, starting from the heat balance equation:

$$F_{тепл} = \frac{Q_м C_м \left[(t_{км} - t_{нм}) \right] - (1-\xi)(t_{км} - t_{нм})}{k\,(1-\xi)\,(t_{км} - t_{нм})} = \frac{Q_м C_м \xi}{k\,(1-\xi)}. \qquad (12.64)$$

Analysis of the formula shows that with an increase in the regeneration factor, the area of the heat exchange surface increases over the hyperbola. Pasteurizing plants, as a rule, are equipped with heat exchangers, in which, besides the pasteurization of milk, i.e. Bringing it to the desired temperature with the subsequent holding, provides for the regeneration of heat in one or two sections and cooling the milk in two sections with water and brine or ice water.

Number of plates in the section of the lamellar pasteurizer:

$$z_{пл} = \frac{F_{тепл}}{f_{пл}}, \qquad (12.65)$$

where $f_{пл}$ – is the working surface of one plate, m^2.

The consumption of hot water or steam for pasteurization (heating) of milk is determined from the heat balance condition:

$$Q_{п.в} = \frac{Q_м C_м (t_{км} - t_{нм})}{(C_{гв} - C_к)\eta}, \qquad (12.66)$$

where $C_{гв}$ – heat content of hot water or steam, J/kg; $C_к$ – heat content of waste water or condensate, J/kg; η – is the thermal efficiency of the pasteurizer.

12.6 Classification and calculation of parameters of milk coolers

Milk is a perishable product, so its cooling is a mandatory operation in the initial treatment. Milk must be cooled to 8 ^{0}C. Milk coolers can be classified according to the following parameters:

– by the nature of contact with the surrounding air – open (irrigation) and closed (flowing);

– by the number of sections – single and multi–section;

– by shape – flat and round;

– by effects on heat exchange media – pressure, vacuum and gravity;

– by the relative direction of heat exchange media – flow, countercurrent and cross–flow.

In fig. 12.16 shown flat open countercurrent coolers. In the operation of this type of cooler, the milk passes through a mesh of holes in the bottom of the upper chute 2 and drains a thin film along the coolant working surface 3 consisting of a series of horizontal pipes connected by a collector 1 in the form of a coil over which the coolant moves countercurrent. Further, the cooled milk flows into the lower trough 5, from where it flows through the drainage pipe to the milk collection container. Coolers can consist of several parallel sections, on which the product comes simultaneously from a common chute. To prevent splashing and contamination of milk, the pack coolers on both sides are closed with lids.

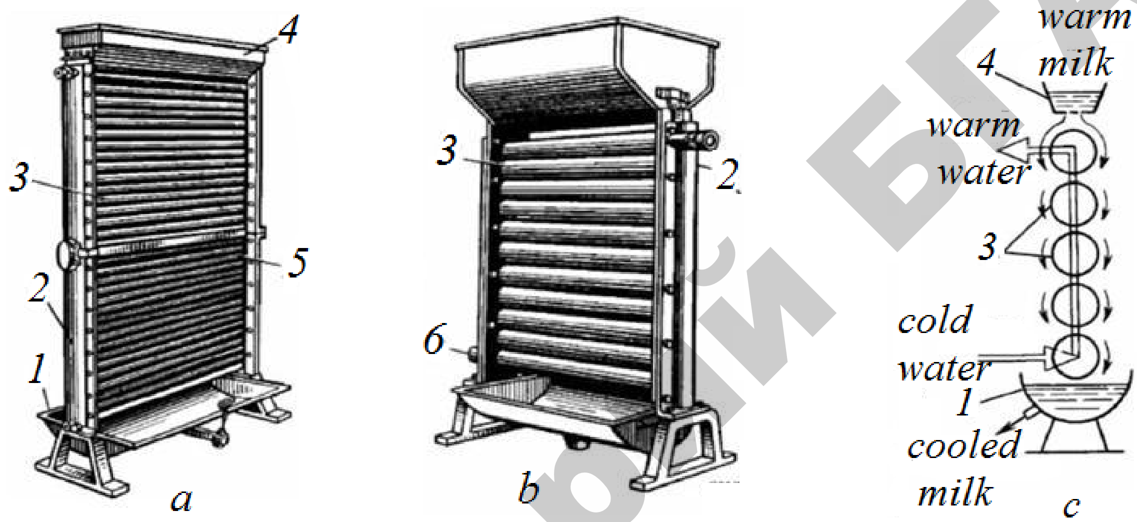


Fig. 12.16. Flat Irrigation Countercurrent Coolers Open type:
A – two–section; B – single–section; C – scheme of operation of the cooler; 1 – lower trough; 2 – collector; 3 – cooling surface; 4 – upper trough; 5 – working surface of brine section; 6 – inlet branch pipe of a cooling liquid

Profiles of the most common working parts of the surfaces of irrigation coolers are shown in fig. 12.17.

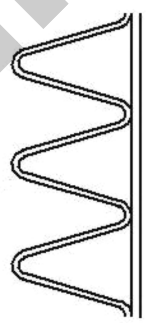
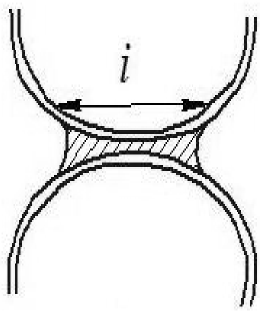
– cylindrical;	– tubular;	– Shaped.
		

Fig. 12.17. Working elements of irrigation coolers

The thickness of the flowing film can be determined from the formula (fig. 12.18)

$$\delta_{пл} = \frac{K\ m}{\sqrt{r_{тр}}\,2g} + \frac{1}{sin\dfrac{\theta}{2}}, \tag{12.67}$$

where K – is a constant determined experimentally; $r_{тр}$ – radius of the pipe, m; θ – is the angle measured from the vertical, deg.

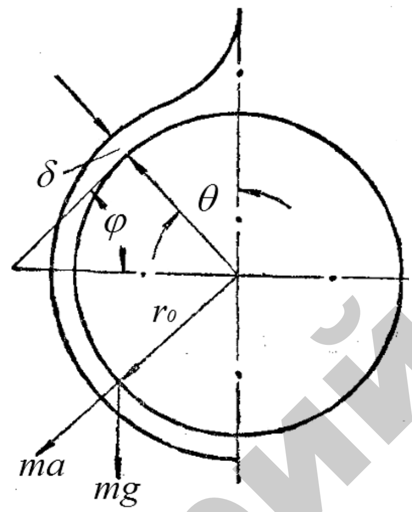


Fig. 12.18. Scheme of forces acting on a fluid running down the tube

With increasing angle θ, the thickness of the flowing film decreases. For flat coolers, the thickness is in the range 0,3…0,6 mm. The smaller the tube radius, the smaller the thickness of the liquid film, and, consequently, the higher the effectiveness of the cooler, provided that the fluid flows uniformly.

The heat flux taken from the milk by the coolant is determined by the formula:

$$G_x = Q_M C_м (t_{н.м} - t_{к.м}), \tag{12.68}$$

where Q_M – milk supply, kg/s; $C_м$ – heat capacity of milk, J/kg deg; $t_{н\,х}$ and $t_{к\,х}$ – the initial and final temperatures of the milk, respectively.

If we neglect the loss of heat to the environment, then the same amount of heat acquires a coolant:

$$G_x = Q_x C_м (t_{н.х} - t_{к.х}), \tag{12.69}$$

where Q_x – is the refrigerant supply, kg/s; $C_м$ – is the heat capacity of the refrigerant, J/kg deg; $t_{н\,х}$ and $t_{к\,х}$ – are the final and initial coolant temperatures, respectively.

In the plate coolers the supply of milk and refrigerant is carried out continuously. But, depending on the temperature regime of the coolant, more is required than the cooled one. The ratio of the refrigerant charge to the flow rate of the cooled liquid is called the coefficient of the refrigerant flow rate:

$$K_{\text{k}} = \frac{Q_x}{Q_м}, \qquad \text{or} \qquad Q_x = K_{\text{k}}\, Q_м. \tag{12.70}$$

Substituting in equation (12.70) and equating (12.68) and (12.69), we can determine the multiplicity factor:

$$K_{\text{k}} = \frac{C_м (t_{нм} - t_{км})}{C_x (t_{нх} - t_{кх})}. \tag{12.71}$$

The value of K_k for the water sections of the coolers is in the range 2.5...3, for the brine sections – 1,5...2,5. If the cooler is two–section (water and brine), then the total heat flow from the milk to the refrigerants:

$$G_{общ} = G_в + G_р, \tag{12.72}$$

where $G_в$ – is the heat flux obtained by water, W; $G_р$ – is the heat flux obtained by brine, W.

The heat flux passing through the walls of the cooler can be expressed by the Newton equation:

$$G_{охл} = k_{ст}\, F\, \Delta t_{ср}, \tag{12.73}$$

where $k_{ст}$ – is the coefficient of heat transfer through a flat wall, W/m^2 deg; F – total heat exchange surface, m; $\Delta t_{ср}$ – is the average temperature gradient between heat exchange media, deg.

The cooling surface is determined from the relationship:

$$F_{охл} = \frac{G C_м (t_н - t_к)}{k_{ст}\, \Delta t_{ср}}, \tag{12.74}$$

where $t_н$, $t_к$ – initial and final product temperatures, respectively, °C; $C_м$ – is the heat capacity of milk, J/kg °C; $\Delta t_{ср}$ – is the average temperature difference, °C.

The average temperature gradient or temperature head is defined as the average logarithmic:

$$\Delta t_{cp} = \frac{(\Delta t_{\max} - \Delta t_{\min})}{\ln \frac{\Delta t_{\max}}{\Delta t_{\min}}}, \tag{12.75}$$

where Δt_{max} and Δt_{min} – are the maximum and minimum temperature regimes.

The coefficient kst can be determined:

$$k_{cm} = \frac{1}{\frac{1}{\alpha_1} + \frac{1}{\alpha_2}}, \tag{12.76}$$

where α_1 – is the coefficient of heat transfer from the wall to the cooling liquid, W/m^2 °C; α_2 – is the coefficient of heat transfer from the laminar external flow to the wall, W/m^2 °C.

Approximately, it can be assumed that when transferred from water to water = 4.18 MJ/m. H.

The cooling surface, depending on the type of coolant surface, can be determined from one of the formulas:

– cylindrical:

$$F_{охл} = \pi \frac{D_{мак} + d_{мин}}{2} S\, n + \frac{\pi}{4}(D^2_{мак} - d^2_{мин}) + \pi D_{мак} \frac{h_1 + h_2}{2}, \tag{12.77}$$

– flat with round pipes:

$$F_{охл} = (\pi D_{мак} - 2h_i)\, n_в\, l, \tag{12.78}$$

– flat with pipes of shaped section:

$$F_{охл} = 2\, S\, n_в\, l_{охл}, \tag{12.79}$$

where S – is the outer generatrix of the turn, m; $n_в$ – number of turns, $\frac{h_1 + h_2}{2}$ – average height of the lower cylindrical part, m; $d_{мин}$, $d_{мах}$, $d_{мах}$ – maximum and minimum diameters, m; h_1 – the width of the soldering between the pipes, m; h_2 – length of pipe, m.

Cylindrical coolers are manufactured with a cooling surface of 0,5...4,5 m and a capacity of up to 1000 *l/h*, tubular coolers with a surface from 2,4 to 14,5 m and a capacity of up to 4000 *l/h*. As coolant, irrigation coolers use artesian water or brine. Allowable pressure of the coolant in cylindrical coolers with channels of shaped section should not exceed 150 MPa, for tubular coolers 150...300 MPa. The capacity of the cooler must correspond to the capacity of the coolant distributor:

$$\frac{G}{\rho}=\frac{k_{\text{ст}}\ F_{\text{охл}}\ \Delta t_{\text{ср}}}{C_{\text{м}}\,(t_{\text{н}}-t_{\text{к}})\,\rho}\,\varphi\, f_0\sqrt{2\,g\,H_{\text{ж}}}\ , \tag{12.80}$$

where f_0 – is the area of the hole, m^2; φ– coefficient of expiration; $H_{\text{ж}}$ – liquid level in the distributor, m; ρ is the density of the liquid, kg/m^3.

The amount of moisture evaporated from the surface of the irrigation cooler can be determined from formula

$$K_{\text{в}} = FT_{\text{охл}}C_o\left(p_{\text{ж}}-\varphi p_{\text{в}}\right)760\ , \tag{12.81}$$

where $T_{\text{охл}}$ – duration of cooling, s; $p_{\text{ж}}$ – is the vapor pressure in the saturation state at liquid temperature, mm Hg. P.; $p_{\text{в}}$ – vapor pressure at ambient air temperature, mm Hg. P.; C_0 – is the experimental coefficient, which depends on the speed of air movement, m/s.

The amount of heat that loses the liquid due to evaporation is calculated from the formula

$$Q_m{=}K_{\text{в}}\ C_{\text{и}}, \tag{12.82}$$

where $C_{\text{и}}$ – is the heat of vaporization at evaporation temperature, J/kg.

The flow rate of water or brine is established from the heat balance equation:

$$m_{\text{пр}}C_{\text{м}}\ \ (t_{\text{к.в}}-t_{\text{к.м}}) = G_{\text{в}}C_{\text{в}}\ \ (t_{\text{н.в}}-t_{\text{н.м}})\ , \tag{12.83}$$

where $m_{\text{пр}}$ – is the product quantity, kg; $C_{\text{м}}$ – is the heat capacity of the product; J/kg °C; $G_{\text{в}}$ – amount of water, kg; C is the heat capacity of water, J/kg °C.

Expressing through the amount of heat, according to the formula of Newton, we determine the time of the milk in the cooler:

$$t_{охл} = \frac{S_{пл}\ \Delta l z_{пл} \rho_{м} C_{м} (t_{нм} - t_{км})}{2\ k_{ст}\ F \Delta t_{ср}}, \tag{12.84}$$

The same time can be determined by the speed of movement of milk in the cooler:

$$t_{охл} = \frac{h}{v_{м}}. \tag{12.85}$$

The speed of milk movement in the cooler is determined by:

$$v_{м} = \frac{2\ k_{ст}\ F \Delta t_{ср} h}{S_{пл}\ \Delta l z_{пл} \rho_{м} C_{м} (t_{нм} - t_{км})}. \tag{12.86}$$

Cooling capacity:

$$Q_{м} = v_{м} S_{пл}\ C_{м}\ \Delta l \frac{z_{пл}}{2} = \frac{k_{ст}\ F\ h\ \ \Delta t_{ср}}{C_{м}(t_{нм} - t_{км})}. \tag{12.87}$$

When cooled with water or brine, the amount of heat released (kcal) is determined:

$$\theta = m_{охл} C_{м} (t_{к.м} - t_{н.м}), \tag{12.88}$$

where $m_{охл}$ – weight of cooled milk, kg; $C_{м}$ – is the heat capacity of milk equal to 0.94 kcal/kg · deg; $t_{н.м}$ and $t_{к.м}$ – the initial and final temperatures of milk, deg.

The flow rate of the refrigerant is determined by the equation:

$$B = m_{охл} n_2, \tag{12.89}$$

where n_2 – is the multiplicity factor of the flow. For brine 1,5...2,5, for water 2,5...3.

In the general case of the cooler, the following heat balance is obtained:

– cooling with water:

$$\theta_{м} = \theta_{в} = m_{охл} C (t_{к.м} - t_{н.м}) = n_2 m_{охл} (t_{к}' - t_{н}') C_{в},. \tag{12.90}$$

– cooling with brine:

$$\theta_м = \theta_р = m_{охл} C(t_{к.м} - t_{н.м}) = n_2 m_{охл} (t_к^{"} - t_н^{"}) C_р\,, \qquad (12.91)$$

where C_p – is the heat capacity of the brine, C_p = 0,87 kcal/kg deg.

Values are generally known or specified, and the final temperature of water and brine is calculated by the formulas

$$t_к^{'} = \frac{C_м}{n_2}(t_{к.м} - t_{н.м}); \qquad (12.92)$$

$$t_н^{"} = \frac{C_м}{C_р - n_2}(t_{к.м} - t_{н.м}) + t_н^{"}. \qquad (12.93)$$

If the cooler is multi–sectional (water or brine sections), the calculation is made for each section separately. In this case, the final temperature of the previous section is the initial temperature for the subsequent section. In all designs of coolers, milk flows on the working surface in a thin layer, which ensures rapid and uniform heat transfer. The ratio of the amount of spent coolant to the amount of milk is called the coefficient of water flow or brine. It is equal for water sections 2...3, and for brine sections – 1,5...2,5.

12.7 Process and calculation of a countercurrent milk cooler of closed type

Closed coolers are tubular and plate–like. They have the following advantages of sealed devices:

– continuity of the flow and more intensive heat transfer

– improvement of Sanitary and hygienic treatment conditions.

The closed vacuum cooler (fig. 12.19) is used for milk cooling in the milk line. It consists of a body 2, an inner corrugated cylinder 3, a pallet and branch pipes for feeding and discharging milk, water, and also a connection to a vacuum line.

The cooler works as follows. Milk from two main milk pipes 14 and 11 goes first to the horizontal filter 15 and then to the distributor 5, which has a number of holes in the circumference to get the milk to the inner corrugated surface of the cylinder 3. After that, the milk is collected in a pan and drained from the cooler through the branch pipe 1 the milk enters the cooler due to a vacuum formed in the interior of the corrugated cylinder by connecting the pipe 12 to the vacuum line of the milking plant.

Water is supplied through the lower branch pipe 4 and passes countercurrently along a spiral channel upward between the inner cylinder of the cooler body 2 and the outside of the corrugated cylinder 3 and exits through the branch pipe 4.

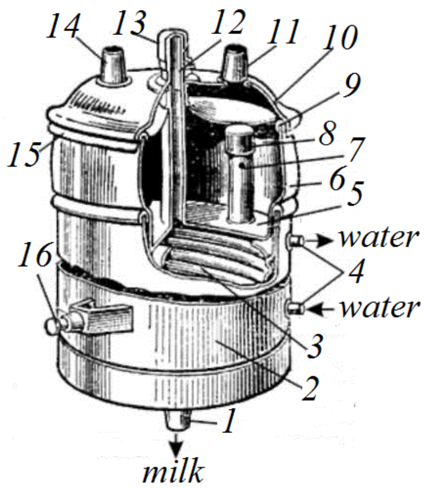


Fig. 12.19. general view of the closed vacuum sprinkler:
1 – branch pipe for drainage of milk; 2 – housing; 3 – corrugated cylinder;
4 – pipe for water; 5 – distributor; 6 – milk receiver; 7 – overflow pipe;
8, 13 – cap; 9 – lining; 10 – cover; 11, 14 – branch pipes with a cap;
12 – vacuum pipe; 15 – gauze filter, 16 – suspension axis

The tubular cooler consists of a set of double tubes inserted one into the other and rolled into collectors (fig. 12.20). The collectors are closed with lids provided with rubber gaskets. The cooler has a casing with thermal isolation.

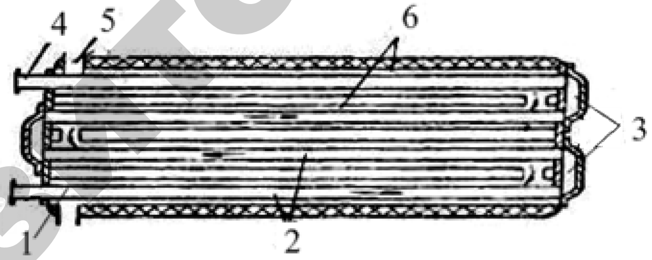


Fig. 12.20 Tubular closed cooler:
1 – outlet for milk: 2 – internal pipes; 3 – coupling kalachi; 4 – outlet for milk;
5 – pipe for refrigerant inlet; 6 – external pipes

Milk comes through the pipe 1 into the inner pipes 2, connected by the rollers 3. After passing through all the pipes, it leaves through the pipe 4. The chloride agent circulates along the annular space counter–current from top to bottom. The heat and coolant consumption is found from the heat balance equation:

$$QC(t_2 - t_1) = Fk\Delta t_{cp}, \tag{12.94}$$

where Q – is the capacity of the cooler, kg/s; t_2, t_1 – the final and initial temperature of the cooled liquid (milk), degrees; C – heat capacity of milk, J/kg hail; F – heat transfer surface, m^2; k – is the heat transfer coefficient, W/m^{2}grade; Δt – is the average temperature difference of the refrigerant, deg.

Denoting the degree of milk cooling δt, the temperature criterion, in general form, is determined by the formula

$$k_\tau = \frac{t_2 - t_1}{\Delta t_{cp}} = \frac{\delta t}{\Delta t_{cp}}; \tag{12.95}$$

– when cooled with cold water

$$k_\tau = 2{,}3 \ lg \frac{t_1 - t_0}{t_2 - t_0}; \tag{12.96}$$

– with counterflow motion of two liquids

$$k_\tau = \frac{t_2 - t_1}{(t_1 - t_в) - (t_2 - t_0)} 2{,}3 \ lg \frac{t_1 - t_в}{t_2 - t_0}, \tag{12.97}$$

where t_0 – is the average temperature of the refrigerant, deg; $t_в$ – the final temperature of the refrigerant, deg.

Then equation 12.94 can be represented in the following form

$$QmCk_\tau = Fk. \tag{12.98}$$

The coefficient of heat transfer for a plane wall of thin–walled pipes is determined by the formula

$$k = \frac{1}{\frac{1}{\alpha_1} + \frac{\delta_1}{\lambda_1} + \frac{\delta}{\lambda} + \frac{\delta_2}{\lambda_2} + \frac{1}{\alpha_2}}, \tag{12.99}$$

where α_1 – is the coefficient of heat transfer from the medium to the wall, W/m^2 deg; α_2 – coefficient of the heat transfer from the wall to the medium, W/m^2 deg; λ – thermal conductivity of the pipe wall, W/m degree; δ – pipe thickness, m; δ_1/λ_1, δ_2/λ_2 – thermal resistance of heat–conducting layers, m^2 deg/W; δ / λ – thermal resistance of the pipe wall, m^2 deg/W.

The resulting formula (10.98) allows us to solve two problems:

– determine its productivity With known heat exchanger parameters.

– determine the area of the heat exchanger for a given capacity and temperature regime.

The capacity of the flow heat exchanger is determined by the equation of flow continuity.

– for a ring device

$$Q = \frac{\pi\left(D^2 - d^2\right)\omega\rho}{4}, \quad (12.100)$$

where D – is the diameter of the ring, m; d – is the diameter of the pipe, m; ω – speed of milk movement in the pipe, m/s; ρ – density of milk, kg/m^3.

– for parallel tube bundle

$$Q = \frac{\pi d^2 \omega\rho z}{4}, \quad (10.101)$$

where z – is the number of tubes in the bundle, pcs.

Length of flow machine channel

$$L = r^{/}\omega\rho C\frac{k_\tau}{k}, \quad (12.102)$$

where $r^{/}$ – hydraulic radius – the ratio of the cross–sectional area of the channel to the heated perimeter, m.

Cooling fluid flow rate

$$\omega = \sqrt[3]{\frac{8k\Delta p}{k_c\rho^2 k_\tau C}}, \quad (12.103)$$

where k_c – is the drag coefficient; Δp – loss of pressure of the moving fluid, N/m^2.

The most widespread in agricultural production was closed flowing with counterflow movement of milk and cooler, the working elements of which are heat exchange plates (fig. 12.21). More modern are stamping machines with vertical or horizontal channels. Stamped plates are made of a thin sheet (about 1 mm thick) of stainless chromium–nickel or chromium–nickel–molybdenum steel, resistant to corrosion at high humidity, and also using alkaline and acid detergent solutions. Heat transferring (working) surface of one plate from 0,2...0,25 to 0,3...0,35 m^2.

In some units, the stamped plates alternate with smooth ones. The depth of the channels (the thickness of the liquid layer with the compressed plate pack) is 3 ... 6 mm. For each of the plates of the packet, part of the total flow passes, while in all the plates of one package the product moves in one direction.

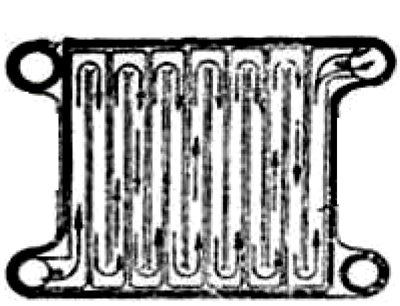
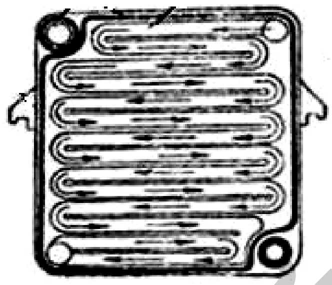
– with vertical milling channels	– With horizontal stamped channels	– With vertical stamped channels
		

Fig. 12.21. Heat exchanger plates

The cooler is a set of heat–exchanging single–type plates, two separation plates and one extreme plate mounted on the clamping plate. Separating and end plates, unlike the others, have two holes. Each plate, except for the extreme, has a sealed gasket (fig. 12.22).

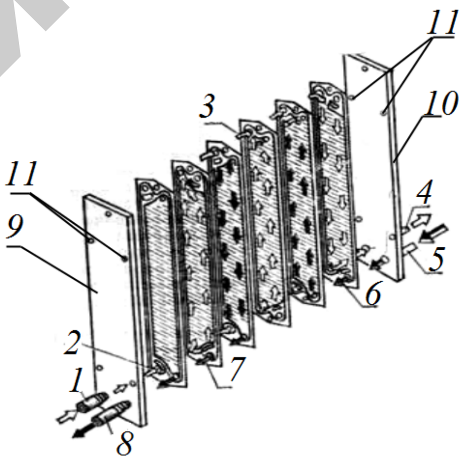


Fig. 12.22. Diagram of the motion of heat exchange media in a plate cooler:
1, 4 – branches for supply and discharge of milk, 2, 3 – respectively, the lower and upper longitudinal channels of milk movement, 5, 8 – nozzles for supply and discharge of the coolant, 6, 7 – lower longitudinal channels of movement of the coolant; 9 – pressure plate; 10 – thrust plate; 11 – holes for tightening bolts

The plates are clamped between the plates by tightening bolts. Rubber rings are installed in the stop plate, and in the clamping plate – sealing rings. Assembly of the cooler must be carried out according to the layout of the plate. All odd–numbered plates should be placed against the "milk" fitting with the ends marked A, and all even ends with a "B" mark, with the exception of the last plate that is to be installed with the "A" stamp. Set the spacer plates in order, as the 1st and 22nd. After assembling the cooler, the plates are tightened.

When working, the milk to be cooled from the separator – the milk cleaner enters the cooler through the union of the stop plate and enters the longitudinal collector formed by the holes of the plates. The coolant (water) comes through the nipple of the clamping plate, moves in the direction opposite to the direction of movement of the milk and exits the cooler through the fitting of the thrust plate. Packages of plates form sections, the arrangement of which is one–tier and two–tier, one–sided and two–sided. To the massive plates separating the sections, as well as to the pressure plate and the machine bed, pipelines for milk and working fluids are connected. Heat exchange between heat exchange media in plate coolers occurs through walls, the intensity of which depends on the temperature difference between the cooling and cooled media and the heat transfer coefficient. To ensure the best conditions for heat transfer, the mode of movement of the coolant must be turbulent (Re> 2320), and on the coolant surface, across the horizontal pipes, the liquid must move laminarly.

When the coolant moves inside the channels, significant hydraulic drags arise. The necessary head in front of the apparatus is calculated by the formula

$$H_{an} = H_1 + \frac{v_{хл}^2}{2g}\left(1 + \lambda_{TP}\frac{l_{охл}}{d_{тр}} + \sum \lambda_{MC}\right), \qquad (12.104)$$

where H_1 – is the height of the installation of the cooler receiver above the level of the pump supplying water or brine to the cooler, m; $v_{хл}$ – speed of movement of the coolant, m/s; $l_{охл}$ – length of coolant pipes, m; $d_{тр}$ – diameter of pipes, m; λ_{TP} – coefficient of friction resistance; λ_{MC} – is the local resistance coefficient.

For channels of non–circular cross section, it is necessary to determine the equivalent diameter:

$$d_{экв} = \frac{4F_к}{P_к}, \qquad (12.105)$$

where $F_к$ – is the cross–sectional area of the channel, m^2; $P_к$ – perimeter of the cross section of the channel, m.

Number of working plates in the section (heat exchange surfaces):

$$z_{пл} = \frac{F}{S_{пл}}, \tag{12.106}$$

where $S_{пл}$ – is the area of the working surface of one plate, m^2.

In order to cool down the milk to the set temperature, it must be in the cooler for a certain time and it can be taken as the necessary amount of heat to cool the milk that is simultaneously in the cooler (the number of channels for milk Zm:

$$\theta_m = S_{пл}\, \Delta l \frac{z_{пл}}{2} \rho_м C_м (t_{нм} - t_{км}). \tag{12.107}$$

Expressing the amount of heat by the formula of Newton, we determine the time of the milk in the cooler:

$$\tau = \frac{S_{пл}\, \Delta l z_{пл} \rho_м C_м (t_{нм} - t_{км})}{2\, k_{ст}\, F \Delta t_{ср}}, \tag{12.108}$$

The same time can be determined by the speed of movement of milk in the cooler:

$$\tau = \frac{h}{v_м}. \tag{12.109}$$

The speed movement of milk in the cooler:

$$v_м = \frac{2\, k_{ст}\, F \Delta t_{ср} h}{S_{пл}\, \Delta l z_{пл} \rho_м C_м (t_{нм} - t_{км})}. \tag{12.110}$$

Cooling capacity:

$$Q_м = v_м S_{пл}\, C_м\, \Delta l \frac{z_{пл}}{2} = \frac{k_{ст}\, F\, h\, \Delta t_{ср}}{C_м (t_{нм} - t_{км})}. \tag{12.111}$$

Using the formulas given, for the given productivity it is possible to determine the parameters of the cooler. If the cooler is multi–section (water sections, brine), the calculation is made for each section separately. In this case, the final temperature of the previous section is the initial temperature for the subsequent section.

The capacity of the cooler must correspond to the capacity of the coolant distributor:

$$\frac{G}{\rho} = \frac{k_{ст} \ F_{охл} \ \Delta t_{ср}}{C_м \ (t_1 - t_2) \ \rho} \varphi \ f_0 \sqrt{2 \ g \ H_{ж}} \ , \qquad (12.112)$$

where f_0 – is the area of the hole, m^2; – coefficient of expiration; $H_{ж}$ – liquid level in the distributor, m.

The flow rate of water or brine is established from the heat balance equation

$$G_{пр} C_м \ (t_{к.в} - t_{к.м}) = G_в C_в \ (t_{н.в} - t_{н.м}) , \qquad (12.113)$$

where $G_{пр}$ – is the product quantity, kg; $G_в$ – amount of water, kg.

12.8 Purpose, structure and operation of the refrigeration system

To cool and store milk on livestock farms and complexes, a refrigeration unit is used. They are stainless steel containers with chilled milk inside. Outside the container has a hermetic casing and a thermal insulating jacket. Between the container and the cladding flows a refrigerant or coolant, which takes heat away from the milk, stirred in the tank with a special stirrer. Such machines operate on the principle of quenching certain technical fluids during their evaporation during the abrupt transition from a high pressure state to an atmospheric one. Such a liquid is chladone. When using water cooled by the evaporator in the bath to wash the container with milk, water is a coolant. Cooling water is produced by refrigerating machines, where the refrigerant circulates in a closed loop, undergoing phase transformations, evaporating and taking heat from the cooled water, condensing and giving off heat to ambient air or running water.

The energy for phase transitions is obtained from the compressor 1, which compresses the gaseous refrigerant to the condensing pressure. In the refrigeration machine, in addition, there is a thermal cylinder 2, a condenser 3, a heat exchanger 4, a thermal valve 5, an evaporator 6, a water tank 7, a milk tank 8, a stirrer 9, a thermostat 10, a pressure switch 11, a filter drier 12, a receiver 13 and pipelines , Connecting individual aggregates into a single closed system (fig. 12.23).

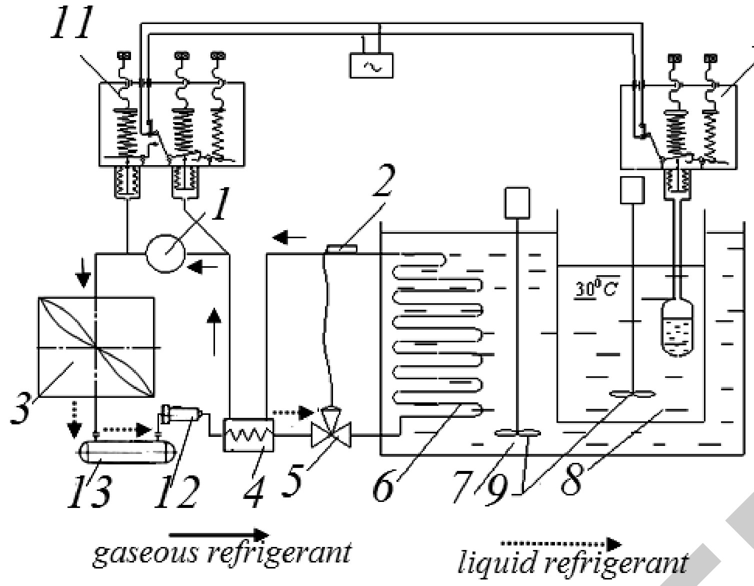


Fig. 12.23. Schematic diagram of the refrigeration system:
1 – compressor; 2 – thermoballoon; 3 – condenser; 4 – heat exchanger;
5 – thermostatic valve; 6 – evaporator; 7 – water tank; 8 – milk tank; 9 – agitator;
10 – thermostat; 11 – pressure switch; 12 – filter–dehumidifier; 13 – receiver

Liquid refrigerant, which has the ability to boil at a low temperature, enters the evaporator 6, where it boils and transforms into a vapor state. The heat necessary for the boiling of the refrigerant is perceived by it from the cooled object of the coolant – the water that irrigates the evaporator. The compressor 1 sucks the refrigerant vapor from the evaporator 6, compresses it to the condensing pressure and pumps it into the condenser 3. Compressing the refrigerant vapor in compressor 1 from the boiling pressure to the condensing pressure is accompanied by an increase in their pressure and temperature. The temperature of the final compression of the vapor in the compressor depends on the difference in pressure at the inlet and outlet of the compressor and reaches 60...80 °C.

In the condenser 3, three processes consistently occur: the cooling of the compressed vapor to the saturation state, their condensation and supercooling of the liquid refrigerant. The pressure and condensation temperature depend on the temperature of the cooling medium, the heat transfer surface of the condenser and the intensity of heat transfer. As a rule, the condensation temperature is 5 ... 20 ° C above the ambient temperature. The liquid refrigerant comes from the condenser 3 through the receiver 13, the heat exchanger 4 and the filter drier 12 to the thermal control valve (TRV) 5, when passing through the throttling, the pressure of the coolant drops sharply to the boiling point. The refrigerant in the form of a vapor–liquid mixture enters the evaporator 10, where it boils at low temperatures, taking away heat from the coolant that irradiates the evaporator. The vapor of the refrigerant formed during boiling is sucked off by the compressor and the refrigeration cycle is repeated. Thus, the refrigerant, moving in a closed cycle, removes heat from the water in the cold accumulator and gives it to the air that blows the condenser.

The number of refrigeration units for storing milk on the cattle–breeding farm is determined by the formula

$$n = \frac{V_{xp}}{V_m}, \tag{12.114}$$

where V_{xp} – is the amount of milk needed for storage within a given period of time (shift or day), m^3; V_t – is the capacity of the milk tank, m^3.

The amount of milk needed for storage for a given period of time:

$$V_{xp} = \frac{M}{\rho_м} n_к, \tag{12.115}$$

where M – is the mass of milk coming from one animal, kg; $\rho_м$ – milk density, kg/m^3; $n_к$ – the number of dairy animals on the farm, pcs.

The evaporator cools the intermediate coolant (water) contained in the cold accumulator as a result of heat exchange with the boiling refrigerant. Through the surface of the evaporator water gives its heat to the refrigerant, which in this case turns into steam. Thus, in the evaporator, the refrigerant boils at a low temperature, taking heat from the cooled water.

The compressor sucks the refrigerant vapor from the evaporator and maintains a low pressure in it, which ensures a low boiling point. In addition, the compressor pumps the vapors into the condenser and compresses them to such a high pressure, at which they turn into a liquid, provided that they are cooled by the surrounding medium at a temperature of 20…30 °C. In refrigerating machines, compressors of two types are used:

– piston reciprocating pistons in cylinders;

– rotary, screw and spiral – with rotational movement of working parts

Reciprocating compressors are used most often in high–power machines and are designed solely by an indirect method (fig. 12.24), when the suction and discharge valves are located side by side in the lid and the vapor stream rotates 180 °.

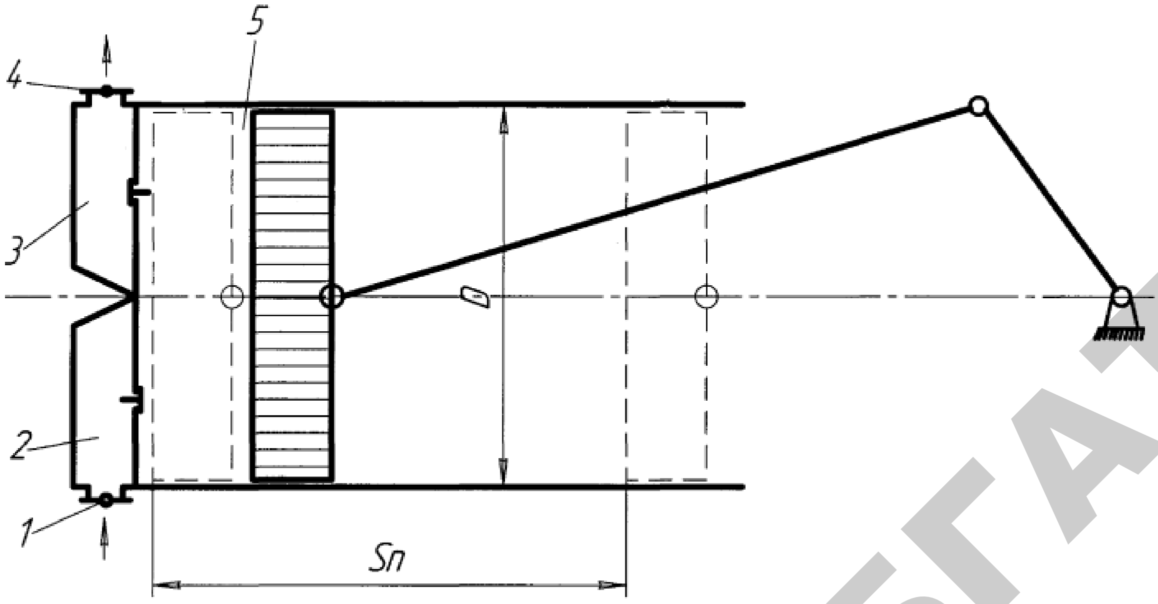


Fig. 12.24. Piston compressor:
1 – standard suction point; 2 – the suction cavity; 3 – injection cavity;
4 – standard injection point; 5 – working cavity of the cylinder

This is due to the fact that the piston of indirect motors compared with straight–flow compressors is much shorter and easier, which makes it possible to make them more compact and high–speed. The following basic modifications of piston compressors are known:

– hermetic compressors;

– semi–hermetic compressors;

– open compressors.

Hermetic compressors: used in refrigeration machines of low power (1,5...35 kW). The motor is located inside the hermetic compressor casing. The motor is cooled by the intake refrigerant itself. Semi–hermetic compressors: used in medium–duty refrigerators (30...300 kW). In semi–hermetic compressors, the motor and compressor are connected directly and placed in a single demountable container. The advantage of this type of compressor is that in the event of damage, it is possible to remove the engine in order to repair the valves, piston and other parts of the compressor. The motor is cooled by the intake refrigerant itself.

Open compressors: they have an external electric motor that has been driven out of the housing and connected to the compressor directly or via a transmission. The power of many refrigeration units can be smoothly regulated by means of inverters – special devices that change the speed of rotation of the compressor. In semi–hermetic compressors, another way of controlling the power is possible: bypassing the steam from the outlet to the inlet or by closing a portion of the suction valves.

The main drawbacks of piston compressors:

– pulsations of the vapor pressure of the refrigerant at the output, leading to a high noise level;

– large loads at start–up, requiring a large power reserve and leading to compressor wear.

The principle of operation of rotary compressors is based on the suction and compression of the gas when the plates rotate. Their advantage over piston compressors is low pressure pulsations and a decrease of current at start–up. There are two modifications of rotary compressors:

– with stationary plates;

– with rotating plates.

In a compressor with stationary plates, the refrigerant is compressed by an eccentric mounted on the rotor of the engine (fig. 12.25). As the rotor rotates, the eccentric rolls along the inner surface of the compressor cylinder, and the refrigerant vapor in front of it is compressed and then ejected through the compressor discharge valve. The plates separate the high and low pressure regions of the refrigerant vapor inside the compressor cylinder.

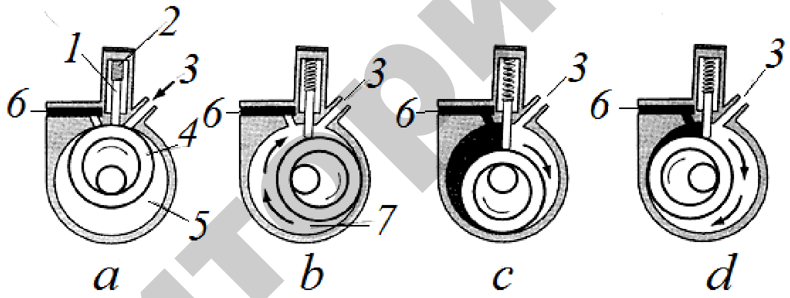


Fig. 12.25. Compressor with stationary plates:
1 – plate; 2 – a spring; 3 – suction channel; 4 – rotor; 5 – the receiving chamber; 6 – the exhaust channel; 7 – compression chamber

In a compressor with rotating plates, the refrigerant is compressed by means of plates fixed to the rotating rotor (fig. 12.26). The rotor axis is offset from the axis of the cylinder of the compressor. The edges of the plates fit tightly to the surface of the cylinder, separating the high and low pressure regions. The diagram shows the cycle of suction and compression of steam. Steam fills the available space. Compression of steam inside the compressor begins and suction of a new portion of the refrigerant begins. Compression and suction is completed. A new cycle of suction and compression begins. The compressors, on one shaft of which two rotors are located, are called two–rotor.

In that compressor (A) – steam fills available space and the compression of the steam within the compressor begins and the suction of a new portion of the refrigerant begins; (B) – the compression of steam within the compressor begins and the absorption of a new portion of the refrigerant begins; (B) – compression and suction continues; (D) – Compression completed, the steam finally filled the space inside the compressor cylinder.

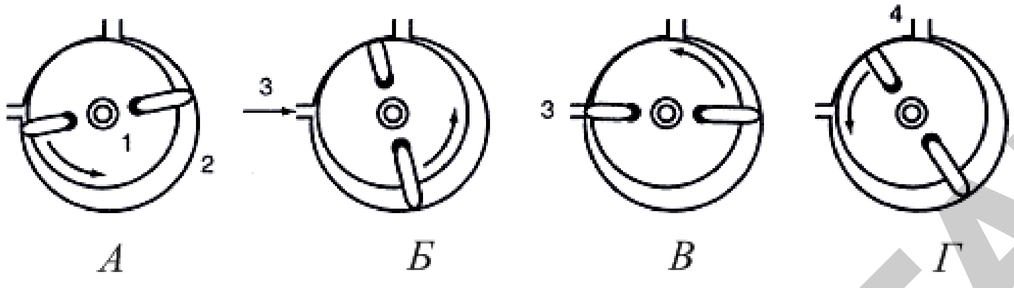


Fig. 12.26. Compressor with rotating plates:
1 – rotor; 2 – working chamber; 3 – the inlet channel; 4 – outlet channel

Scroll compressors are used in small and medium capacity refrigerating machines. Such a compressor consists of two steel spirals (fig. 12.27). They are inserted one into the other and expand from the center to the edge of the compressor cylinder. The inner spiral is fixed, and the outer spiral revolves around it.

а) $\varphi = 0°$; б) $\varphi = 90°$; в) $\varphi = 180°$; г) $\varphi = 270°$;

Fig. 12.27. Moving spiral along a circular orbit:
1 – stationary spiral; 2 – movable spiral

Spirals have a special profile (involute), which allows rolling without slipping. The moving coil of the compressor is mounted on the eccentric and rolls along the inner surface of the other spiral. The point of contact of the spirals gradually moves from the edge to the center. The refrigerant vapor in front of the touching line is compressed and pushed out into the central hole in the compressor cover. Touch points are located on each turn of the inner spiral, so the pairs are compressed more smoothly, in smaller portions than in other types of compressors. As a result, the load on the compressor motor is reduced, especially when the compressor starts. Refrigerant vapor enters through the inlet in the cylindrical part of the casing, cools the engine, then it is compressed between the spirals and exits through the outlet at the top of the compressor's casing.

The suction cycle is performed per revolution of the compressor shaft. The cycle of compression and ejection of refrigerant vapors lasts 2...3 turns, depending on the angle of the helix and the size of the discharge window. The designs of spiral compressors require high accuracy and cleanliness of the surface of the spirals, precision machine tools and appropriate equipment. They are more reliable in operation, contain 40 % less parts than piston ones, produce less noise and have a longer service life.

Screw compressors are used in large–capacity refrigerators (150...3500 kW). There are two modifications of this type:

– with single screw;

– with a double screw.

Models of a compressor with a single screw have one or two satellite gears connected to the rotor from the sides. Compression of refrigerant vapors occurs by means of rotors rotating in different directions. Their rotation provides a central rotor in the form of a screw. The refrigerant vapor flows through the compressor inlet, cool the engine, then enter the outer sector of the rotating gears of the rotors, compress and exit through the sliding valve into the outlet. The compressor screws must be sealed, therefore, a lubricating oil is used. Subsequently, the oil is separated from the coolant in a special compressor separator. The models of the compressor with a double screw are distinguished by the use of two rotors – the main rotor and the drive one. Screw compressors do not have inlet and outlet valves. Absorption of refrigerant constantly occurs from one side of the compressor, and its release – on the other hand. With this method of vapor compression, the noise level is much lower than that of piston compressors. Screw compressors allow smooth adjustment of the capacity of the chiller by changing the engine speed.

The condenser is the main unit of the refrigeration unit next to the compressor. The condenser receives a gaseous refrigerant with a temperature of about 80 °C at a pressure of 0,8 MPa. At this pressure, the condensation temperature of the refrigerant is 40...50 °C. The condenser provides cooling of the compressed refrigerant vapor with ambient air in order to lower the vapor temperature to the condensation temperature (saturation state) and to condense the saturated vapor into a liquid state. After condensation of all refrigerant, the liquid is cooled by the air, which cools the condenser with the help of a fan.

The receiver creates a reserve of liquid refrigerant, which is necessary to ensure a uniform supply of the evaporative system (fig. 12.28). In addition, the receiver is an additional capacity of the condenser, which prevents the latter from overflowing with a liquid refrigerant. The receiver of the operating refrigeration machine must be filled with a liquid refrigerant by 50 % of its volume.

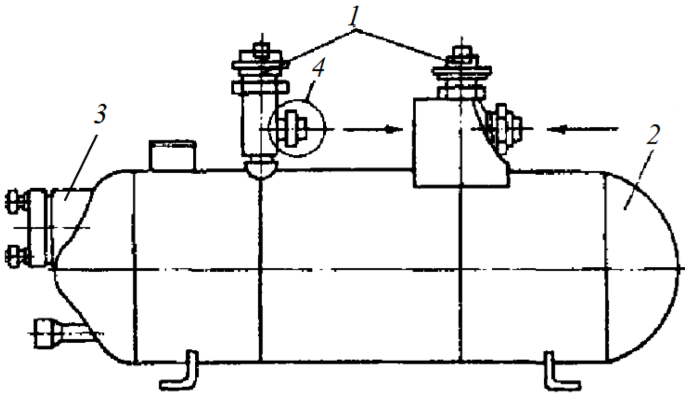


Fig. 12.28. General view of the receiver:
1 – valve; 2 – vessel; 3, – sight glass; 4 – fusible plug

During operation of the refrigeration system, refrigerant leaks may occur, which leads to a decrease in the cooling capacity. Due to the volume of the receiver, the amount of refrigerant refilled into the system is increased. The cylindrical receiver of the refrigerating machine includes a steel vessel on which a fusible plug is installed to release the refrigerant in the event of an emergency temperature rise above 70 °C.

The filter drier catches various mechanical impurities (sawdust, rust, etc.) of the refrigerant and absorbs the moisture present in the system. It is installed, as a rule, after the receiver. A typical design scheme of filter–driers for trapping mechanical impurities and moisture is shown in fig. 12.29. Liquefied refrigerant from the receiver enters a strainer, in which the largest mechanical impurities are trapped. The filtering element of the dehumidifiers is a wire mesh made of phosphor bronze and suede with a thickness of up to 2 mm. With further movement, the hladon is cleaned of moisture, which is adsorbed by grains of silica gel with a grain size of 3 to 5 mm. Its absorbing capacity is 10 to 40 % moisture by weight of silica gel or zeolite.

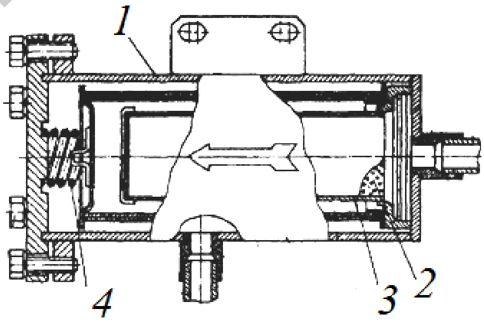


Fig. 12.29. Drier filter:
1 – housing; 2–silicagel; 3 – filter mesh; 4 – spring

To improve the economy and prevent liquid refrigerant from entering the compressor into the compressor, modern refrigeration units are equipped with a heat exchanger, which allows increasing the cooling capacity by 12...14 %. It is installed after the desiccant filter, before the thermostatic valve in the high pressure line. The device and the scheme of the heat exchanger operation of the refrigerating machine are shown in fig. 12.30.

In it, there is a heat exchange between the liquid coming from the filter–drier to the thermoregulating valve and the steam coming from the evaporator to the compressor. Passing through the heat exchanger, the cold suction steam absorbs heat from the liquid and overheats, and the liquid is supercooled, which reduces throttle losses.Overheating of the steam in front of the compressor is necessary for the safe operation of the compressor, although it leads to an increase in compression work, an increase in the final discharge temperature and an increase in the thermal load on the condenser. In addition, for refrigerators operating on chladones, it is advantageous to maintain a higher superheat temperature, as this reduces volumetric losses and increases the cooling capacity of the compressor.

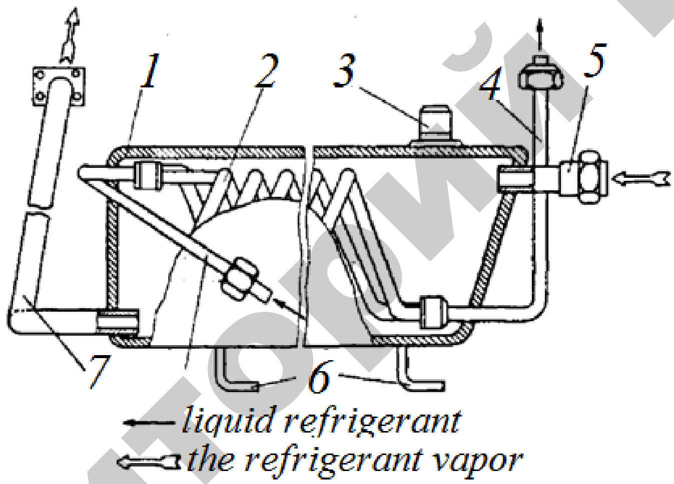


Fig. 12.30. Heat exchanger circuit:
1 – shell; 2 – coil; 3 – fitting; 4.7 – tube; 5, 8 – the pipe; 6 – paws

The refrigeration unit works most effectively when the entire heat transfer surface of the evaporator is washed with boiling refrigerant, i.e. Its boiling occurs on the entire surface of the evaporator. Both increase and decrease in the amount of refrigerant supplied to the evaporator reduces the cooling capacity of the installation. To meet this requirement, a thermostatic expansion valve is designed to throttle the liquid refrigerant entering the evaporator and to regulate its flow, i. E. Giving in a unit of time so much liquid, how many vapors have time to suck the compressor during this time.

Valves with variable hydraulic resistance include valves with internal or external equalization. Thermoregulating valves with internal equalization are mainly used only in small refrigeration systems at a boiling point above –30 °C, where the resistance to movement of refrigerant in the cooling device is small. In such valves, the evaporator is generally short and the pressure from the evaporator to the compressor is almost the same. The diagram of the thermostatic valve and the flow of the process for regulating the degree of filling of the evaporator with the refrigerant are shown in fig. 12.31. The degree of filling of evaporators with a refrigerant in which the vapor–liquid mixture moves (dry type of evaporators) is judged by overheating of steam at the outlet of the evaporator. Overheating is understood as the difference between the superheated steam temperature at the outlet of the evaporator t and the boiling point t o corresponding to the vapor pressure at the evaporator outlet.

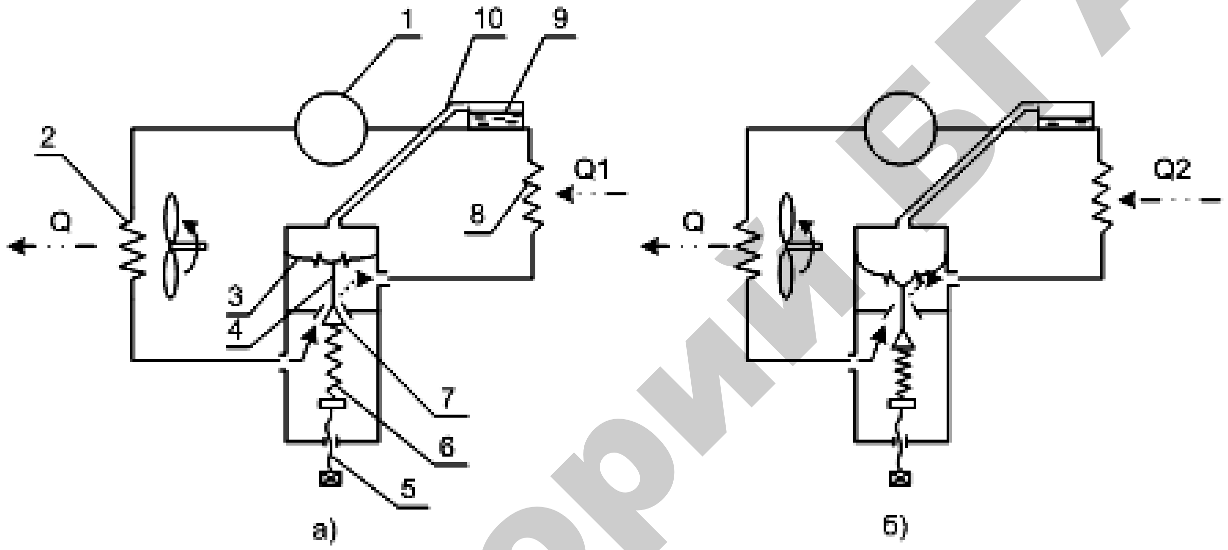


Fig. 12.31. Thermo–ventilator with internal alignment:
a) heat load is average; б) the thermal load is large (Q1 <Q2)
1 – compressor; 2 – condenser; 3 – membrane; 4–stem; 5 – screw;
6 – spring; 7 – valve; 8 – evaporator; 9 – thermoballoon; 10 – capillary tube

The principle of the action of the TRV is based on a comparison of the boiling point of the refrigerant in the evaporator with the temperature of the vapors emerging from it and maintaining a constant difference of these temperatures. If the overheating is increased, indicating that the evaporator is not full, the TRV valve automatically opens, increasing the supply of refrigerant to the evaporator, and conversely, if the superheat is reduced, which results from excessive refrigerant entering the evaporator, the valve automatically closes and thereby reduces the flow of refrigerant.

The power–sensitive thermosensitive part of the device is a closed hermetic system consisting of a thermoball, a capillary tube, a cavity above the membrane. This sealed system is filled with the same refrigerant on which this refrigerator works (or another substance that is close in its thermodynamic properties to the refrigerant).

The thermo–balloon is attached to the pipeline at the outlet of the evaporator. The diaphragm through the pushers is connected to the valve's needle holder, which covers the passage section of the thermal valve seat.

The liquid refrigerant from the receiver enters the TRV under pressure and, when passing through the annular section between the seat and the valve, drastically reduces the pressure that is maintained in the evaporator by the compressor. At the same time, part of the liquid refrigerant is converted to steam and a vapor–liquid mixture moves along the evaporator. At some point the liquid turns into steam.

At the exit from the evaporator, on the suction pipe, a thermo–balloon is sensed, which senses the temperature t of the outlets. At this cylinder temperature, the pressure in the TRV power system is set, which is perceived by the membrane. So in case of overheating of refrigerant fumes, if there is difference pressure in the evaporator and pressure maintained by the compressor, membrane flexes downward and pushes the needle holder through the pusher, opening the valve. The valve opens until the force of the compressed spring balances the pressure on the membrane. With this valve position, the entire (required) filling of the evaporator with the refrigerant occurs.

With decreasing temperature in the cooled medium, the heat influx to the evaporator decreases. Then the path of the vapor refrigerant flow is reduced and the steam overheating decreases. Now the thermal cylinder perceives a lower temperature and a lower pressure is established in the power system of the TRV and the valve moves upwards under the action of the spring, thus reducing the throttling valve cross–section and the supply of refrigerant to the evaporator. With a smaller supply of refrigerant to the evaporator, the boiling ends earlier and the overheating assumes a value close to the original value.

The set initial overheating value, which ensures the required opening of the valve, is set by appropriate compression of the spring by turning the screw. As the heat load increases the pressure in the thermoballoon also is increased, which is transferred to the membrane, and through the pushers the opening of the valve opens. The flow of liquid through the expansion valve increases and overheating begins to fall.

In refrigeration machines with a large cooling capacity, the evaporators have a considerable length and large hydraulic resistance. In such evaporators, the pressure of the coolant at the outlet is lower than at the inlet. Therefore, in refrigerating machines having evaporators in which the pressure drop is more than 0.2 · 105 Pa, a TWR with an equalizing tube is used (fig. 12.32).

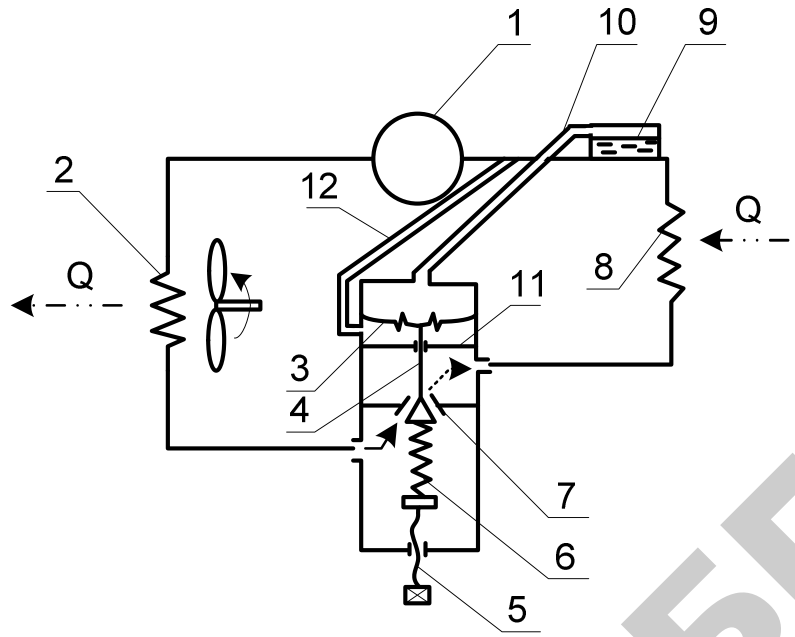


Fig. 12.32. Thermogauge with external lining:
1 – compressor; 2 – condenser; 3 – diaphragm; 4–stem; 5 – adjusting screw; 6 – spring; 7 – valve; 8 – evaporator; 9 – thermoballoon; 10 – capillary tube; 11 – partition; 12 – equalization pipe

The diaphragm 3 is installed in the body of such a thermowell. Due to this, the diaphragm is supplied with pressure not from the inlet side, but from the evaporator outlet side via the equalizing tube 12. The presence of the diaphragm allows an additional choke to be installed at the outlet of the expansion valve. This gives the following advantages:

– increasing pressure behind the valve, due to the presence of an additional throttle section, relieves the operation of the valve and allows increasing its diameter;

– suppling of superheated steam under the membrane and the transfer of the boiling point of the refrigerant reduces the cooling of the entire device and the possible condensation of the vapor over the membrane.

The operation of refrigeration units in automatic mode and their protection from emergencies is provided by automation devices, which include a temperature sensor and a pressure switch.

The temperature relay in refrigeration units serves to control the process of frosting ice on the outer surface of the evaporator panels, to regulate the temperature in the refrigerating compartment of the coolant at the outlet from the tank (water basin).

The principle of operation of the device is based on changing the pressure of the vapor–liquid mixture of coolant in the device's thermo system (fig. 12.33), depending on the temperature change of the thermoball.

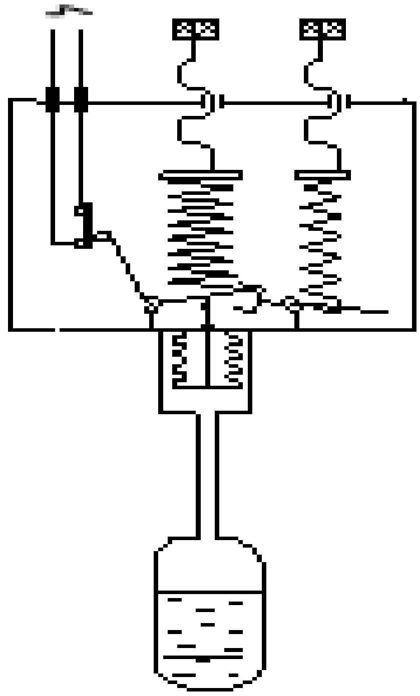
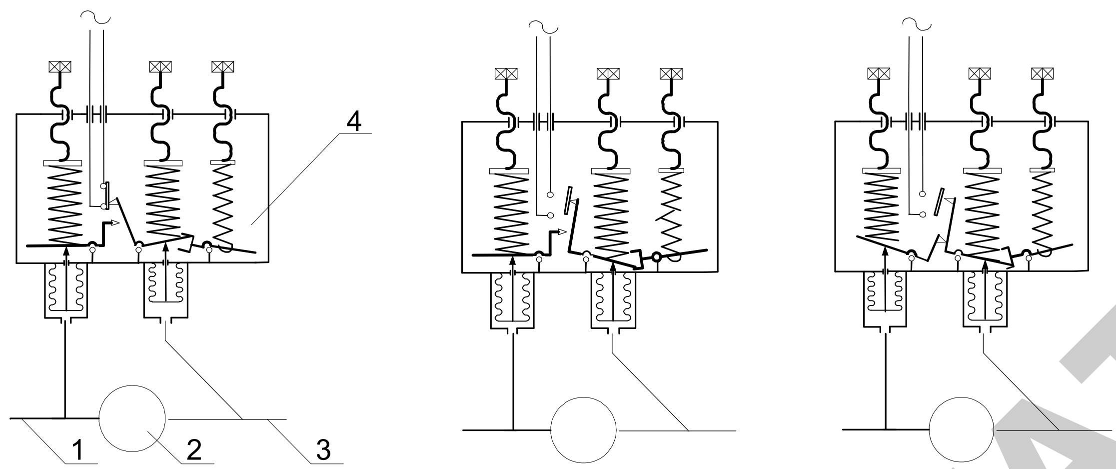


Fig. 12.33. Temperature relay circuit

Increasing the temperature of the thermo ball causes a corresponding increase in the pressure of the coolant and, acting through the capillary tube on the bellows, compresses it. The bellows push rod acts on the main lever, aiming to rotate it clockwise. This is prevented by compressed spring acting on the lever from above. Turning the main lever clockwise closes the compressor start contacts. Force of compression of a spring is regulated by the screw. The instrument is controlled by the position of the scale pointer. Compression of the differential spring strengthens counteraction to the rotation of the main lever clockwise, and consequently, the closing of the contacts of the device will correspond to the greater temperature of the object being monitored The differential unit is designed to be installed with a differential screw of a certain temperature difference between the direct operation of the device (the contact is opened at the same time) and the reverse trip (contact closure). The main control device of the refrigeration unit is a pressure switch. It is designed to maintain a given pressure in the refrigeration circuit of the plant by turning the compressor on or off.

The device consists of low and high pressure units (fig. 12.34). When the pressure in the suction line is increased above the permissible level, the bellows compresses and, overcoming the force of the springs, turns the lever and turns on the electric drive. If during the operation of the refrigeration system the pressure in the discharge line has increased beyond the permissible range, the high–pressure bellows is compressed and turns the lever counterclockwise through the pusher, turning off the electric drive.



a) inclusion; b–c) disconnection

Fig. 12.34. Principle of operation of the pressure switch:
1 – high pressure line; 2 – compressor; 3 – low pressure line; 4 – pressure switch

The suction sensor switches off the compressor when the refrigerant vapor pressure in the suction line is lower than the permissible limits. Such limits for equipment operating on refrigerant–22, is the pressure below 0,1 MPa. The pressure switch on the discharge pressure switches off the compressor when the refrigerant vapor pressure in the discharge line rises above the permissible limits. Such limits for equipment operating on refrigerant–22 are pressure above 1,68 MPa. Agitators allow mixing of water and milk, since the last one, and are arranged in such a way that they are evenly cooled throughout the volume of the respective tanks. Blending and propeller mixers are used for mixing.

When the agitator is used, the power is spent to overcome the frictional forces in the mixed medium, which can be determined from formula

$$N_p = k_c d_л^5 \rho_{жс} n^3 z, \qquad (12.116)$$

where k_c – is the drag coefficient; $d_л$ – is the diameter of the circumscribed circle, m; $\rho_{жс}$ – density of the stirred liquid, kg/m^3; n – number of revolutions of the mixer, s^{-1}; z – number of agitator blades, pcs.

Coefficient of resistance to movement of liquid with a significant immersion of the mixer

$$k_c = \frac{A}{Re_м^k}, \qquad (12.117)$$

where k_c – factors that depend on the tank and mixer. Presented in Table; $Re^k_м$ – is the Reynolds criterion. Determined by the formula:

$$Re_м = \frac{nd_л^2}{k_в}, \tag{12.118}$$

where $k_в$ – is the coefficient of kinematic viscosity of the stirred liquid, m^2/s.

Required electric motor power to mixer drive

$$N_{эл} = \beta \frac{N_p}{\eta_м}, \tag{12.119}$$

where β – is the power reserve factor, $\beta \approx 1{,}1...2$; $\eta_м$ – is the mechanical efficiency, $\eta_м = 0{,}6...0{,}85$.

Starting power of the electric motor can be determined by the formula

$$N_n = N_p\left(\frac{k_u}{k_c} + 1\right), \tag{12.120}$$

where k_u – coefficient that takes into account the influence of inertial forces when starting the electric motor, k_u = 3,87 (h/dl); h – tip of the agitator blade, m.

Table. Values of coefficients for different types of agitators

Agitator type	Basic Correlations		Values Coefficients	
	$\frac{D}{d_ё}$	$\frac{H}{d_ё}$	$\grave{A}$	k_c
Two–bladed with straight blades	3	3	6,8	0,2
Two–bladed with blades at an angle of 450	3	3	4,05	0,2
Four–bladed with straight blades	3	3	8,52	0,2
Four–bladed with blades at an angle of 450 (up)	3	3	5,05	0,2
Propeller two–blade with blades at an angle of 22.50	3	3	0,985	0,15
Propeller three–blade	3,8	3,5	1,19	0,15

* D, H – diameter and height of the milk tank, m.

The starting capacity is approximately 2...4 times the required power. Equipment for cooling milk type MTCO DIAN 2500/2 is designed to cool milk on livestock farms, points of reception and collection of milk for the purpose of storing it before transport to the place of further processing. The refrigeration unit can be equipped with a system for heating the process water (recuperation) with refrigerant vapors coming from the compressor to the condenser. The refrigeration unit is charged with refrigerant R22 (boiling point at atmospheric pressure –40,8 °C) in the amount of 14,1 kg. The equipment is made in the form of a closed milk tank with indirect cooling. Milk cooling is performed by irrigating the outer side walls of the milk container with ice water by a pump through the piping system. The milk container is installed in a water bath and is above the level of ice water. Cooling of water occurs due to the melting of ice, which is formed on the coil evaporator of the cooling system. The compressor–condensate unit is installed separately on the foundation. Used compressor GRNQ–050E–TFQ–552 firm Gopeland, the installed capacity of which is 11,5 kW. The principle of operation of the refrigeration plant (fig. 12.35) is as follows.

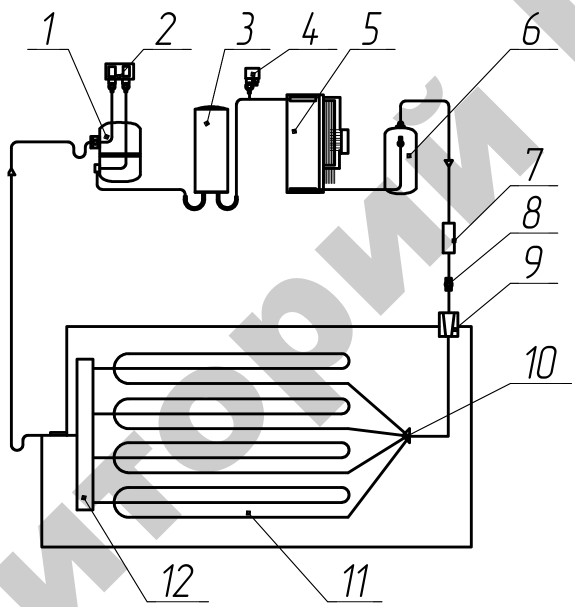


Fig. 12.35. Schematic diagram of the cooling system MTCO DIAN 2500/2:
1 – compressor; 2 – low and high pressure switch; 3 – heat exchanger;
4 – pressure switch; 5 – condenser with fans; 6 – the refrigerant receiver;
7 – filter–dehumidifier; 8 – viewing eye; 9 – thermoregulating
10 – refrigerant distributor; 11 – evaporator; 12 – collector

The gaseous refrigerant by the compressor 1 is pumped into the heat exchanger 3, in which some of the heat is supplied to the cold water passing between the plates, and then enters the condenser 5. The condenser provides cooling of compressed refrigerant vapors with ambient air and their condensation. From the condenser liquid refrigerant enters the receiver 6, the filter drier 7, where it is cleaned of mechanical impurities and dehydrated, and further through the solenoid valve 8 into the thermostatic expansion valve (TRV) 9. The receiver creates a supply of liquid refrigerant necessary to ensure a uniform supply of the evaporating system, in addition, it is an additional capacity of the condenser, which prevents overflow of the liquid refrigerant. The refrigerant is throttled in the TRV. Throttling is accompanied by a decrease in the pressure of the refrigerant from the condensation pressure to the boiling pressure. A part of the liquid passed through the TEV is converted to saturated steam, while cooling the rest of the refrigerant to the boiling point, i. E. A mixture of liquid and saturated steam (wet steam) emerges from the TRV.

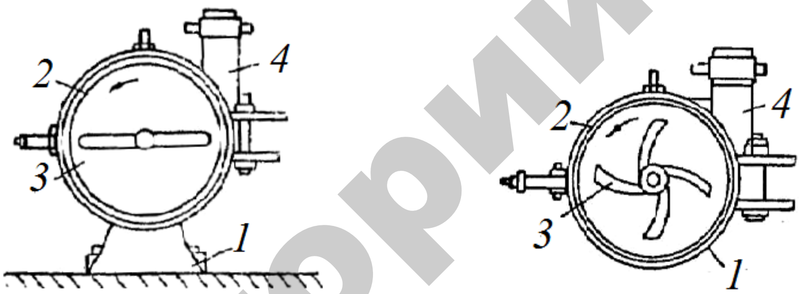
Liquid refrigerant passes the distributor 10, entering the evaporator 11, boils, absorbing heat from the walls of the milk container. The refrigerant vapor drawn off by the compressor 1, as it passes through the evaporator 11, is additionally heated as a result of heat exchange through the walls of the latter. Therefore, their temperature at the outlet of the evaporator is usually higher than the boiling point. And the cycle repeats. The control unit serves to monitor and display the milk temperature, as well as to provide an automatic cooling mode and maintain the temperature of the cooled milk.

Chapter 13

PUMPS FOR TRANSPORTATION OF MILK

13.1 Classification and operation of a centrifugal pump

Centrifugal and rotary pumps are used to transport milk and dairy products through pipes. Advantages of centrifugal pumps: compactness, simplicity of the device, ease of assembly and disassembly, availability of washing, the drive is carried out directly from the motor shaft. However, the correct operation of centrifugal pumps is possible only under the level of milk filling, although they can create the same suction height as piston pumps. However, the air penetrating the system causes a strong foaming of the liquid. In addition, centrifugal pumps have a low efficiency (0,4...0,7). In pumps in which there are no guides, η_{ob} = 0,2...0,4. Blade and disk centrifugal pumps are used. Rotary centrifugal pumps are divided into single–bladed pumps with straight blades and multi–bladed ones with blades curved backwards (fig. 13.1, a, b).

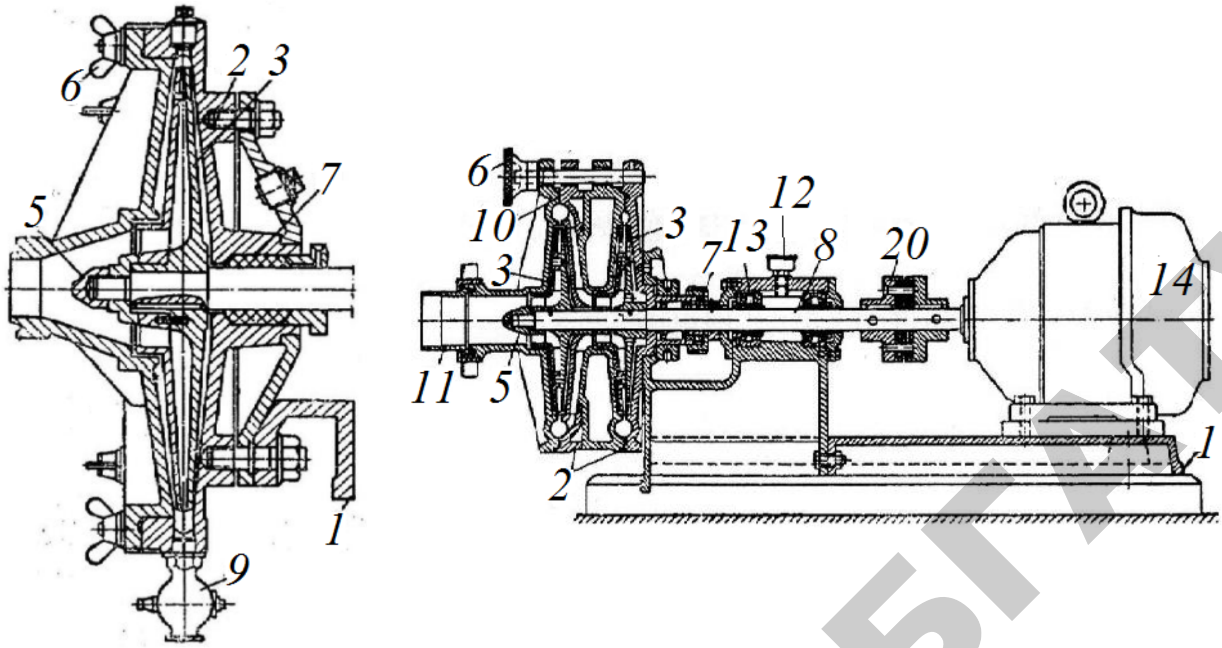


a b

Fig. 13.1. Rotary centrifugal pump:
A – one–bladed; B – multiblade; 1 – frame; 2 – pump chamber;
3 – impeller (blade); 4 – milk outlet

Multiple–lobe centrifugal pumps have a higher coefficient of efficiency; They are used to inject liquid into a relatively small height. The pressure created by them does not exceed 5...10 m. In cases where it is necessary to overcome a relatively large resistance, one–stage and two–stage disk centrifugal pumps are also used for transporting milk (fig. 13.2, a, б).

The pressure created by them reaches 30 m and more. The efficiency of disk pumps is higher than that of blade pumps and varies between 0,3 and 0,5. Centrifugal pumps work as follows. Through the branch pipe 11 milk enters the chamber of the pump 2, where it acquires a rotational motion from the impeller or blades 3. The impeller (blade) makes a minute 1000...3000 revolutions per minute. Under the influence of the centrifugal force, the milk is forced into the discharge pipeline, connected to the chamber 2 through the branch pipe 4.



а)   б)

Fig. 13.2. Disc centrifugal pumps:
A – single–stage disk; B – two–stage disk;
1 – frame; 2 – pump chamber; 3 – impeller (blade); 4 – outlet for milk;
5 – nut of an impeller; 6 – clamping screws; 7 – gland; 8 – shaft; 9 – cock;
10 – a rubber lining; 11 – branch pipe for milk input; 12 – oil can;
13 – bearings; 14 – electric motor; 15 – cover; 16 – bushing; 17 – valve;
18 – limiter; 19 – lining; 20 – coupling

Centrifugal pumps are used mainly for pumping milk and such low–viscosity dairy products as skimmed milk, buttermilk, whey. However, they can also be used for transportation of more viscous products – condensed, whole and skimmed milk without sugar and in cases where intensive mixing of the product in the pump does not affect its quality.

13.1.1 Calculation of the centrifugal pump

In fig. 13.3 shows one of the channels of the impeller, bounded on both sides by blades. The liquid enters the channel through the suction pipe at a speed c_1. Its unwise input is provided under the condition that the velocity of the fluid at the inlet does not vary either in magnitude or in direction. From the impeller channel, the fluid exits at a speed of ω_2 (relative to the impeller). The absolute exit velocity C_2 is found as the diagonal of the parallelogram.

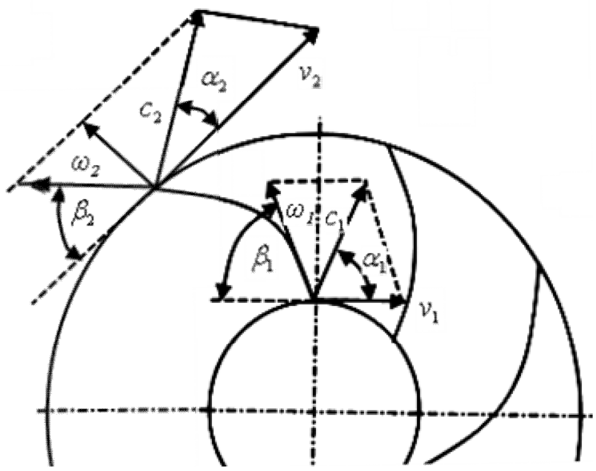


Fig. 13.3. Calculation of centrifugal pump velocity

Very important for centrifugal pumps is the correct choice of the number of pump blades. The most advantageous number of blades located at an angle to the radius of the impeller can be determined by Pfleiderer's formula:

$$z_л = 2K^{/} \frac{r_T}{l_{cp}} \sin \frac{\beta_1 + \beta_2}{2}, \tag{13.1}$$

where r_T – is the radius of the center of gravity of the middle line of the blade, m; l_{cp} – length of the middle line, m; $\beta_1 + \beta_2$ – angles of inclination of the blades at the inlet and outlet of the liquid, deg; $K^{/}$ – a factor of 6,5.

For radial blades

$$z_л = 6,5 \frac{D_2 + D_1}{D_2 - D_1} \sin \frac{\beta_1 + \beta_2}{2}, \tag{13.2}$$

where D_2, D_1 – outer and inner diameters of the impeller, m.

With an increase in the number of blades compared to the calculated, the efficiency of the pump decreases. With a decrease in the head, cavitation is observed, in which the change in the quality of the product, and in particular the degree of dispersion of the fat phase, is very likely.

There is a relationship between the parameters of a centrifugal pump:

$$b_I(\tau D_1 - \delta z_л) v_1 \eta_{ob} = b_2(\tau D_2 - \delta_л z_л) v_2 \eta_{ob} \ , \tag{13.3}$$

where b_I – is the width of the wheel on its inner circumference, m; b_2 – is the width of the wheel on its outer circumference, m; $\delta_л$ – thickness of wheel blades, m;

$z_л$ – is the number of blades in the impeller, pcs; v_1, v_2 – respectively, the fluid velocity on the inner and outer side of the disk, m/min; η_{ob} – volumetric efficiency, $\eta_{ob}=(0,6...0,8)$.

With the radial arrangement of the blades

$$v_1 = c_1 \sin\alpha_1 \qquad и \qquad v_2 = c_2 \sin\alpha_2 , \tag{13.4}$$

where α_1, α_2 – is the angle between the tangent and the vectors of the velocity components c_1 and c_2 when fluid is separated from the disk, deg. The speed c_1 is assumed to be equal to the velocity of the liquid in the suction line.

The head H_H produced by the pumps is determined by the following formula

$$\omega_2 c_2 \cos\alpha_2 = \frac{V_{ж} H_н}{\eta_r} \,, \tag{13.5}$$

where η_r – is the hydraulic efficiency; $V_ж$ – the amount of liquid supplied per second, m^3/s; $H_н$ – height of lifting of a liquid, m.

The practically created head is determined by the formula

$$H_н = \varphi_H \frac{v_{ок}}{2g} , \tag{13.6}$$

where $v_{ок}$ – is the peripheral speed of the impeller, m/s.

Between the performance, the height of the fluid supply, the power consumption and the number of revolutions, there is the following relationship:

$$\frac{V_1}{V_2} = \frac{n_1}{n_2}; \;\; \frac{H_{н1}}{H_{н2}} = \frac{n^2_1}{n^2_2}; \;\; \frac{N_1}{N_2} = \frac{n_1^3}{n_2^3}. \tag{13.7}$$

The critical number of revolutions is calculated by the formula:

$$n_{кр} = \frac{30}{\pi} 4,76 \sqrt{\frac{EI_в g}{m_в l_в^3}} , \tag{13.8}$$

where E – is the modulus of elasticity of the material from which the shaft is made, kg/cm^3; $m_в$ – weight of a unit of shaft length, kg; $l_в$ – shaft length, m; $I_в$ – moment of shaft inertia, kg m/s^2; g – gravity acceleration, m/s^2.

With increasing head or with the aim of increasing productivity, it is necessary to change the number of revolutions of the impeller of the centrifugal pump. It is necessary to check the critical number of revolutions of the impeller, especially in multi–stage pumps. Actual turnover should not come close to critical.

13.2 Rotary pumps

Rotary pumps are designed to move both low–viscosity and high–viscosity dairy products. They are divided into sliding, to which one can include self–priming pumps, gear and screw. The main parts of the slide pumps (fig. 11.4) are the rotor 1, the housing 2 and the contactors 3. The rotor is driven from the drive shaft, the housing is usually stationary and has a receiving chamber 4 and a pressure chamber 5.

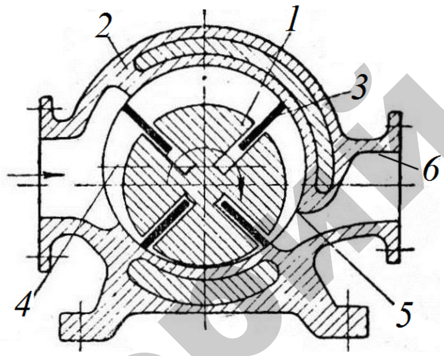


Fig. 13.4. Rotary vane pump:
1 – rotor; 2 – housing; 3 – blade; 4 – receiving chamber;
5 – pressure chamber; 6 – outlet

Closers are the blades that extend out of the radial cuts. The rotating rotor is installed eccentrically with respect to the fixed housing, as a result of the displacement of the rotor, a working process is provided. The rotor of the slide pumps performs 400...1000 min^{-1}. Depending on the direction of rotation of the rotor, the direction of movement of the liquid in the pump can vary. Slot pumps are used to move milk, buttermilk, condensed milk, cream.

The volumetric efficiency of the slide pump is determined by the formula

$$V_н = 2 l_п e n_п (\pi D_ц - \delta_п z_{пл}) 60, \qquad (13.9)$$

where is the length of the piston, m; E – relative displacement of piston and cylinder, m; – the number of revolutions per minute. The number of revolutions of the piston is selected based on the permissible (circumferential) speed of the rotor, which should not exceed 12...15 m/s; Dц – internal diameter of the cylinder, m; – number of plates, pcs; – thickness of plates, m.

Self–priming pumps (such as water–ring pumps) (fig. 13.5) are used to transport whey, buttermilk, milk. The working body of the self–priming pump is made in the form of a rotor 1 with straight blades, installed eccentrically to the housing 2. Before the first start–up, the chamber 4 of the pump is filled with liquid through the funnel 7. With small interruptions in operation, the liquid from the pump can not be removed and no re–filling is required. This is different from centrifugal pumps.

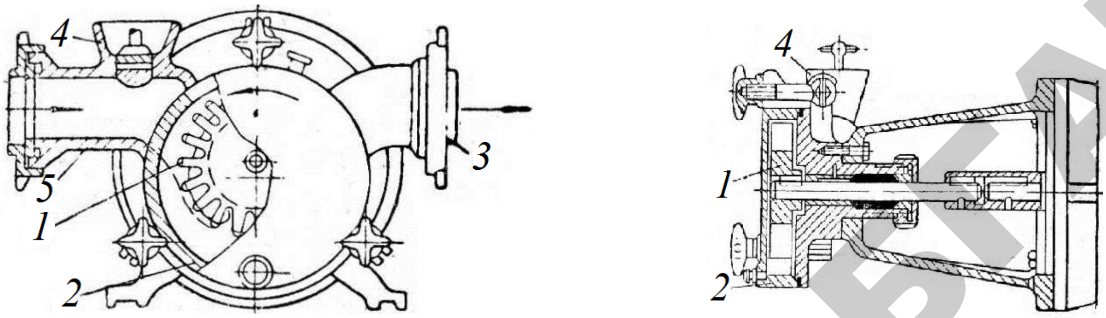


Fig. 13.5. Self–priming rotary pump:
1 – rotor; 2 – housing; 3 – contactor; 4 – receiving chamber; 5 – pressure chamber; 6 – milk outlet pipe; 7 – milk filling funnel; 8 – milk inlet pipe

When the impeller rotates, the liquid is pushed to the walls of the housing by the action of a centrifugal force and forms a liquid ring. When rotating, the impeller blades alternately sink into the fluid ring, depending on the location, and exit it. As a result, a vacuum or compression is created in the space between the blades. The side on which the discharge is created is suction and there is a suction hole in it, on the opposite side – the discharge one. At the beginning of operation, the pump draws air from the suction pipe, resulting in the supply of liquid. Therefore, for example, when emptying jars, it is possible to suck off all the milk, because when the air enters the pipeline, the milk flow does not stop. Self–priming pump with a capacity of 13 m^3/h even at 80 °C can suck milk to a height of 3 m and create a head of 9...11 m of water.

The volume of delivery of self–priming pumps is calculated by the formula

$$V_н = \left\{ \frac{\pi}{4} \left[(D_а - \mathrm{h}_л)^2 - D_ц^2 \right] - z_{лоп}(l_1 - a)\delta_п \right\} b_л \frac{\pi}{60} \eta_{ob}, \qquad (13.10)$$

where – is the diameter of the rotor with blades, m; – depth of immersion of blades in the lower part of the liquid ring, m; – number of blades, pcs; l_1 – length of blades, m; – blade width in the axial direction, m; – volumetric efficiency.

Toothed pumps are also called rotary or gear. They are used mainly for pumping whole and skim milk, buttermilk, whey, as well as cream and condensed milk. Distinguished gear pumps with external and internal gearing. A gear pump with external gearing (fig. 13.6) works as follows. The drive pinion 3, rotating counterclockwise, transmits the motion to the driven gear 4, which rotates clockwise.

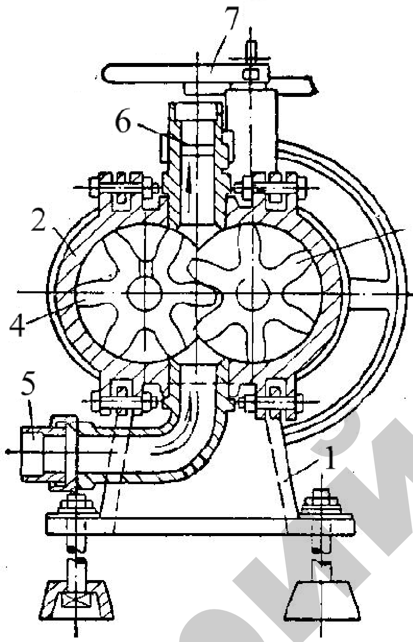


Fig. 13.6. Rotary gear pump with external gearing:
1 – frame; 2 – housing; 3 – leading gear wheel; 4 – conducted gear wheel;
5 – suction pipe; 6 – delivery branch pipe; 7 – belt removal

Near the suction nozzle, the teeth of one gear come out of engagement with the other, creating a vacuum, and the liquid is sucked. The fluid in the cavities between the teeth of the gears is transferred to the delivery nozzle. Here, the teeth of the gears begin to mesh and the liquid is forced into the discharge line. At the pump with internal gearing (fig. 13.7), the drive gear 2 is the driving wheel. Rotating, it drives the internal gear 1. The liquid from the suction nozzle 5 enters the space between the teeth of the gears and moves them to the discharge nozzle. As a result of meshing gears, liquid is forced into the delivery pipe 6. Crescent liner 3 can be removable or mounted in the pump cover. It prevents reverse fluid movement, gear shifting and facilitates the assembly of the pump. If the smooth surface of the teeth is broken or the inaccurate manufacturing of the teeth occurs, the clamping of the liquid occurs when the teeth are attached, which can lead to an accident. To avoid this, in some pumps in the troughs between the teeth of the gears there are special channels for withdrawing the clamped liquid. The number of turns of the gear is from 200...400 to 1200...1400 rpm, efficiency – 0,6...0,75.

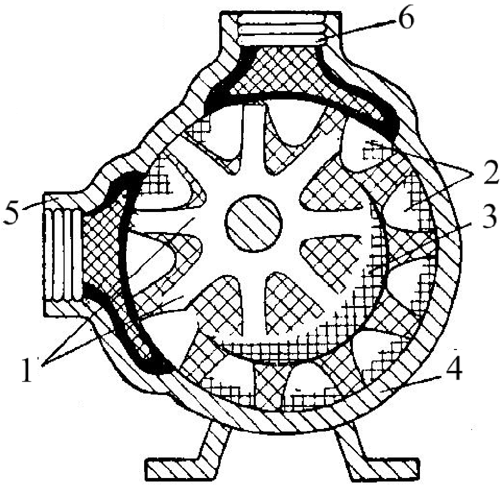


Fig. 13.7. Rotary pump with internal gearing:
1 – cogwheel; 2 – cogged disk; 3 – crescent liner; 4 – pump housing;
5 – suction pipe; 6 – discharge pipe

Gear pumps are used primarily for injection. Depending on the direction of rotation of the gears, they can inject liquid in two directions. The discharge height can reach 200...250 mm of water. Art. The volume capacity of a gear pump with external gearing with sufficient accuracy for practice can be determined by formula

$$V_н = 2\pi 60 D_H m_з b_ш n_ш \eta_{ob}, \qquad (13.11)$$

where Dn is the diameter of the initial circle of gears, m; – the engagement module, m; – width of gears, m; Nш – number of revolutions of gears per minute; – volumetric efficiency. ().

The volume capacity of a gear pump with internal gearing is recommended to be determined by formula

$$V_н = b_ш \frac{\omega_1}{2}\left[2r_{1ш}(h_{B1}+h_{B2}) + h_{B1}^2 - \frac{r_{1ш}}{r_{2ш}}h_{B2}^2 - \left(1-\frac{r_{1ш}}{r_{2ш}}\right)\frac{l_3^2}{12}\right]\eta_{ob}, \qquad (13.12)$$

where is the angular velocity of the driving gear, c^{-1}; – radius of the initial circle of the drive gear, m; – radius of the initial circle of the driven gear, m; HB1 – the height of the heads of the drive gear, m; HB2 – head height of the driven gear, m; L3 – half the length of the mesh line, m; – volumetric efficiency. Pump.

According to the principle of operation, screw pumps are adjacent to the gear pumps (fig. 13.8). The screw threading of the driven screw 2 has a direction opposite to the screw threading of the lead screw 1. The suction holes are located at the ends of the housing, and the discharge holes are in the middle. As a result of the symmetrical arrangement of both pairs of screws, the rotors are unloaded from the axial forces.

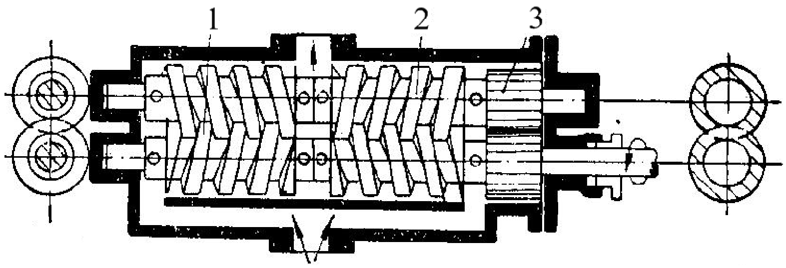


Fig. 13.8. Screw rotary pump:
1 – leading screw; 2 – driven screw; 3 – gears

A three–screw pump is similarly arranged. In it, the lead screw is located between the two followers, so that the amount of liquid supplied is almost 1.5 times increased. The speed of the screw pumps can reach 10,000 per minute, the pressure they develop is 200 atm. And the amount of liquid supplied is from 3 to 300 m^3/h.

The volume capacity of a twin–screw rotary pump can be approximated by the formula

$$V_н = \frac{3\pi t_в n_в}{16}(D_в^2 - d_в^2), \tag{13.13}$$

where $t_в$ – is the pitch of the screw, m; $D_в$ – outer diameter of the screw, m; $d_в$ – internal diameter of the screw, m;

Theoretical performance of a three–screw pump with a cycloidal profile is determined by the formula

$$V_н = 4n_в d_н^3, \tag{13.14}$$

where $d_н$ – is the main diameter of the screws (external driven screw), m; $n_в$ – number of revolutions of propellers per minute.

The power required for the operation of the pumps, based on the capacity V_H, the liquid feed rate H and the sum of the inertia forces, frictional resistance and local resistance, is determined by the formula

$$N_н = \frac{V_н(H + h_0)}{3600\gamma_м}, \tag{13.15}$$

where γ_i – is the mechanical efficiency.

Correct operation of pumps is possible provided that the flow of liquid is continuous in all areas of its movement. Flow ruptures not only disrupt the operation of pumps, but also impair the quality of products transported through the pumping system. Due to the rupture of the flow, strong foaming is observed, which is undesirable, since the fat and protein parts of dairy products are changing; It is possible to simultaneously coarsening and fragmenting fat and protein particles.

The continuity of the flow is provided provided:

$$\frac{p_{пр}}{\gamma} \geq \frac{p_0}{\gamma} - l_{ж} - \frac{\upsilon^2_{жит}}{2g} - h_0, \tag{13.16}$$

where $p_{пр}$ is the limiting pressure corresponding to the boiling of the liquid at the temperatures considered; p_0 is the fluid pressure at the zero mark (usually the largest); $l_{ж}$ – height of liquid column, m; $\upsilon_{жит}$ is the flow velocity, m/s; h_0 – hydraulic resistance on the way from the zero mark at the outlet of the branch pipe.

The rupture of the flow is possible both on the suction line and on the discharge line. However, it is more likely to rupture it on the suction line. Therefore, the following relation should be observed:

$$H_{BC} \leq P_{атм} - h_t - \sum h_c - h_{кав}, \tag{13.17}$$

where $P_{атм}$ – is the atmospheric pressure, mm of water; h_t – is the saturated vapor pressure of the intake liquid, mm of water; $\sum h_c$ – is hydraulic resistance, including pressure, increasing liquid flow velocity, mm of water; $h_{кав}$ – correction for cavitation (decrease in suction height to avoid cavitation), depending on the capacity V and the speed n of the pump.

For centrifugal pump:

$$h_{кав} = 0,00125(V_н n_p^2)^{0.67}. \tag{13.18}$$

The tightness of all pump connections (especially on the suction side) is an extremely important condition for its operation.

Chapter 14

OPTIMIZATION OF ENERGY SAVINGS TECHNOLOGICAL PROCESSES IN LIVESTOCKING AGRICULTURE

14.1. Optimization of the number of machines and equipment of Technological lines

In order to obtain a positive effect from a complex production chain in animal husbandry, it is necessary to consider micro– and macro–processes as a single system working for one economic result. This is possible with the introduction of in–line production.

The main requirements for in–line production include the following:

– synchronization of technological processes;

– simultaneous execution of various technological operations on all lines of component processing;

– division of labor (transport, transportation of feed, milking, management of the technological process, etc.);

– high level of working capacity of machines;

– high level of unification of equipment.

One of the main factors of on–line production is saving time for preparing each machine, for preparing and executing each technological operation.

Consequently, under the flow mechanized technology, a set of machines and equipment should be adopted, arranged in the order of the sequence of technological operations with the required (given) capacity.

The basis for all work on the organization of a flow mechanized technology should be based on optimal options for prospective, current and operational interlinked plans.

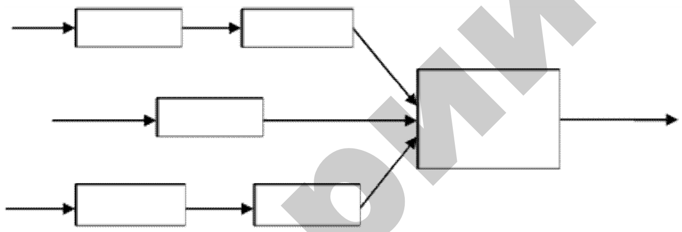
A perspective approach to the definition of material flows provides for the determination of their size and structure on the basis of an optimized dynamic balance in the livestock production of an agricultural business entity. For the rational creation of *production* units, it is necessary to optimize the production structure of the system. The infrastructure of the economy available at the time of optimization is the starting point for locating and determining the size of production sites. At the same time, it can be transformed in the right direction.

Current planning serves as the scientific justification for the relationship between the production processes on the farm. At the level of the current planning, the study of technical means ensuring the performance of technological processes deserves attention. The data obtained make it possible to smooth the dependence of the dynamics of machines that ensure the operation of the entire system. On a single basis, the problem of interconnected planning of all links of the technological chain is solved.

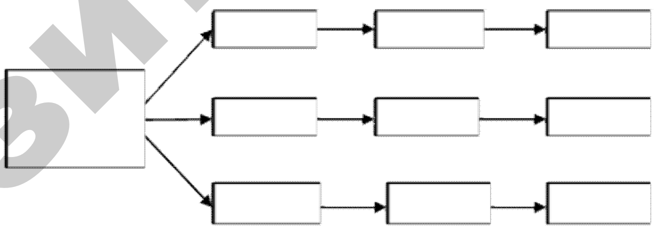
After solving the task of forming the technological links of the system, the task of operational control in each subsystem arises. At the stage of operational management, the rate of consumption of resources necessary to ensure the continuity of technological processes is clarified. Among the reasons that cause such fluctuations are: change in the number of animals, weather conditions, seasonality of work, random factors, miscalculations in planning, etc. As a result of a comprehensive analysis of material flows, a harmoniously matched material–conducting system with specified parameters at the output is obtained. This system is distinguished by a high degree of consistency in its production lines.

According to the flow structure, the lines can be single–threaded, multithreaded and mixed. Single–flow lines usually process one kind of raw material and the machines in them are connected in series one after another. Multithreaded lines can be with convergent, divergent and parallel streams.

Convergent flows make it possible to produce one kind of product from several types of raw materials (for example, to prepare multicomponent feed mixtures).



Diverging streams on the contrary, from one type of raw material allow you to manufacture different types of products.



Parallel flows are used in cases where the line includes machines that have a performance significantly lower than the performance of the entire line.

Along with the flow structure, the type of communication between machines or sections of the line is important for characterizing the production line. The connection between machines in the production lines of the equipment set can be of several types:

– rigid (fig. 14.1), when all machines from the first to the last should work with the productivity, the same or multiple of the main machine of the machine set, for example, when preparing mixed fodders or when distributing feed, removing manure, etc.

$$Q_A = Q_1 \le Q_2 \le Q_3 \ldots \le Q_n.$$

With a rigid connection, the failure of any mechanism or device leads to the termination of the entire line at once.

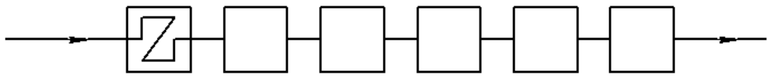


Fig. 14.1. Schematic diagram of rigid connection of machines in production lines

Given the low organization of maintenance of machines in animal husbandry, the low reliability of feed–preparation machines, as well as the mandatory observance of feeding time, milking animals, production lines with rigid communication machines will not find wide application;

– Flexible (fig. 14.2), when there is a storage capacity after each machine. Consequently, the performance of each machine is strictly dependent on the performance ahead of the standing or subsequent machine. A certain, sometimes considerable, deviation is allowed, since the presence of operational capacitances must compensate for the difference in performance;

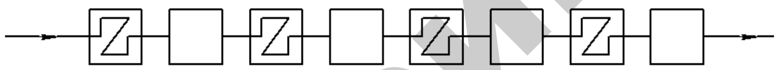


Fig. 14.2. Schematic diagram of flexible connection of machines in production lines

Lines with flexible communication in practice are cumbersome, metal–consuming and expensive. They include operational capacities of large volumes that cause line failures:

– mixed (fig. 14.3), when the equipment set of the feed mill is divided into separate lines (sections), consisting of a group of machines with rigid connections between them. In turn, the lines (sections) are interconnected by flexible connections in the form of storage–dispensers.

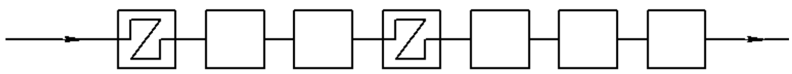


Fig. 14.3. Schematic diagram of mixed connection of machines in production lines

In lines with a mixed connection, in the event of a malfunction in the operation of a machine, not all, but only rigidly connected with it, stop, the rest of the machines continue to operate. If the fault is quickly eliminated, the line can operate almost without interruption, i.e the downtime of neighboring machines and sites is mutually compensated.

Thus, the layout of lines should be based on the solution of a number of such fundamental questions as: choosing a rational number of machines and equipment; determination of the optimal composition of the machines for each operation; rational arrangement of machines in a set of equipment.

It is obvious that equipment sets with parallel connection of machines in the line are more reliable than sets with serial connection. With the successive installation of machines, the reliability of the line or set operation is determined, according to probability theory, as the probability Pc of the joint occurrence of n independent events and is equal to the product of the probability Pi of these events, ie.

$$P_C = \prod_{i=1}^{n} P_i . \tag{14.1}$$

Since the reliability of each line in the set is always ≤ 1, the reliability of the kit is lower than that of the most unreliable machine in the line. If the machines are connected in parallel, the complete failure of the set will occur if all parallel sections of the line are simultaneously stopped. Since the probability of a single failure as a random event, the opposite of non–failure, is:

$$Q\ (t) = 1 - P(t), \tag{14.2}$$

The probability of failure–free operation of a set of machines:

$$P_c = 1 - \prod_{\mathrm{i}=1}^{\mathrm{n}} \left(1 - P_{\mathrm{i}}\right). \tag{14.3}$$

Thus, the probability of failure–free operation of a production line or a set with parallel flows is higher than the production line

To synchronize the operation of machines of different lines, the duration of individual technological processes must be the same. If the machines entering the production line for the production of components have the same or adjustable output, then single–threaded arrangements can be used with transport devices that transfer the necessary components from one machine to another. When analyzing the operation of machines and mechanisms in the production of livestock products, it can be seen that each machine operating as part of technological processes has a direct, direct, and indirect influence on the operation of other machines and units. For an objective evaluation of the effectiveness of a particular machine, it is advisable to consider them in connection with the general technological process and the general system of machines. Often a separate machine can have sufficiently high technical and economic indicators outside this system. However, in the process line, it can sometimes give even a negative effect. When choosing a machine, one should proceed from the fact that the number of machines entering into each line should be minimal. So the probability of failure–free operation of a set of machines, aggregates for a time τ (ie the probability of obtaining a failure after the expiration of the time τ) can be determined by the formula

$$P(\tau) = \int_{\tau}^{\infty} f(\tau)\, d\tau \quad . \tag{14.4}$$

Then the average number of well–functioning machines in the kit

$$N(\tau) = N_0 P(\tau). \tag{14.5}$$

where N_0 – is the number of machines in the kit.

Through an arbitrarily small time interval $\Delta\tau$ from the moment τ to $\tau + \Delta\tau$, the number of failures ΔN of machines will be written in the form:

$$\Delta N = N_0 f(\tau)\, \Delta\tau\,. \tag{14.6}$$

It can be seen from the formula that the fewer N_0 machines in the kit, the lower the failure rate. Therefore, it is necessary to simplify the kit as much as possible. The number of failures per unit time on average will be

$$\frac{\Delta N}{\Delta \tau} = N_0 f(\tau). \tag{14.7}$$

Consequently, as a parameter for optimizing the number of machines, it is advisable to accept a minimum of machines capable of performing the technological process. The basis for determining such a minimum can be a detailed operating scheme. If there is a detailed characteristic of each operation (zootechnical requirements, scope of work, labor intensity, duration, etc.) and analyzing them together with the technological scheme, one can combine the operations in one machine.

14.2 Determining the continuity condition of the production line

With the streamlined organization of the production process, the product obtained as a result of the work of the previous machine is the starting material for the subsequent machine. In this case, operations at all workplaces are performed at intervals equal to or multiple to the rhythm of the flow with continuous movement of the processed product.

The rhythm r, or step, of a flow of a production line is the time interval through which a production line or a separate machine produces a unit of finished product.

The tempo of the flow, is called the inverse of the rhythm. The cycle characterizes the intensity of the work, showing how many units of finished products the line produces per set time unit.

With a continuous flow, the magnitude of the clock and rhythm is the same for all operations of the process, with a discontinuous cycle and rhythm are different for the individual links, therefore, for each link of the process it is necessary to perform their independent calculation. The size of the clock and rhythm set for the link is the same for all operations that are included in it.

When assessing the performance of machines and process equipment, productivity is considered as the main technical and economic indicator, which makes it possible to judge the effectiveness of the use of technical means in this technological process.

The productivity of machines during operation does not remain constant. It depends on the organization of production, the quality of raw materials, the development of technology, the conditions of its operation and a number of other factors. In this connection, the following types of productivity are distinguished in the calculations:

1) Theoretical productivity Qt is the calculated or planned quantity of output received per unit time. For machines that provide technological processes that are not related to direct impacts on animals, theoretical performance is determined by means of design parameters and the established kinematic regime, and therefore it is sometimes called calculated, or nominal.

For machines that are in direct contact with animals, theoretical performance often does not lend itself to rigorous analytical analysis. In these cases, the production program and the planned productivity of animals are established, based on previously achieved production indicators (milk yield).

2) *Technological productivity Qtechn* is determined by the quantity of production received per unit of time, i.e. per hour of clean work of the machine. It does not take into account the time spent on stops and idling.

Technological productivity per hour of clean work is real, and not calculated, as it is determined experimentally based on the results of state tests on MIS and is usually indicated in the technical characteristics of the machines.

3) *Cycle productivity Qc* of the machine is characterized by the quantity of product obtained per unit of cycle time.

4) *Technical performance QTech* is calculated taking into account the time spent on stops due to the need for maintenance and preparatory–final operations with the machine in good working order.

5) *Operational performance Qo* is determined taking into account all the time losses: for preparatory–final operations, maintenance and downtime for organizational and technical and other reasons. It is often called the actual Qf or Qe.

The productivity of the production line can be represented in a form that satisfies the flow condition:

$$Q_{np} = \sum_{j=1}^{n_i} q_{ij}\eta_{ij} \le \sum_{j=1}^{n_{i+1}} q_{(i+1)j}\eta_{(i+1)j}, \tag{14.8}$$

Or in the reduced form for calculation of each link of a stream we receive:

$$Q_{np} = n_{м} q\eta, \tag{14.9}$$

where $n_{м}$ – is the number of machines, pcs; q – productivity of machines, t/h; η – is the coefficient of using the working time of the machine.

The rhythm of the production line is determined from the ratio:

$$R = \frac{1}{Q_{np}}. \tag{14.10}$$

Knowing the productivity of the production line and the flow link machines, we determine the need for them:

$$n_{м} = \frac{Q_{np}}{q\eta}. \tag{14.11}$$

For newly designed production lines, the productivity of machines is found from the equation:

$$q = \frac{Q_{np}}{n_{м}\eta}. \tag{14.12}$$

Since the productivity of machines in the flow links is not always possible to equalize, it is necessary to adhere to its multiplicity of productivity of the basic base link.

The most advantageous use of the productivity of machines in the stream link is obtained when the flow coefficient is 1:

$$K_n = \frac{Q_{np}}{n_{ij} q_{ij} \eta_{ij}} \approx 1. \qquad (14.13)$$

Used as indicators, that characterizes reliability of machines and process – the availability and technical use of machinery and equipment.

The probability that the facility will be operational at an arbitrary time, in addition to the planned periods during which the use of the facility for its intended purpose is not provided, is called the availability factor:

$$K_r = \frac{\tau}{\tau + \tau_в}, \qquad (14.14)$$

where τ – is the time between failures, hour; $\tau_в$ – average time of forced downtime for repair of failures, hour.

The ratio of the mathematical expectation of the time of the object's stay in an operational state for a certain period of operation to the sum of the mathematical expectation of the time of the object's stay in operable condition, the time of maintenance–related downtime and the repair time for the same period of operation is the technical utilization factor. It is determined for an individual machine by the formula

$$\eta_m = \frac{\Sigma \tau_z}{\Sigma \tau_z + \tau_p + \tau_{об}}, \qquad (14.15)$$

where Στz is the total operating time of the considered time interval, h; Tp, ττо – respectively, the time to eliminate the downtime associated with repair and maintenance, hour.

The coefficient of technical use of the entire set of equipment can be determined through the coefficients of technical use of each machine included in the kit. It is difficult to determine the downtime of each machine in a set of equipment at the design stage. It is more expedient to solve this problem by methods of probability theory. For such a solution, it is necessary to have a large set of statistics on the frequency of idle time, the law of their distribution.

The formula (14.15) taking into account the average idle time will take the form:

$$\eta_{т.и} = \frac{\Sigma \tau_z}{\Sigma \tau_z + N_м (\tau_{ср.пр} + \tau_{ср.об})}, \qquad (14.16)$$

where $\tau_{ср.пр}$, $\tau_{ср.об}$ – accordingly, average time for elimination of idle times for technical reasons and maintenance of machines included in the set of equipment, hour; $N_м$ – the number of machines in the kit, pcs.

Since the set of equipment for mechanization of technological processes in the flow will, in general, use mixed connections between machines and lines, the coefficient of technical use will be slightly higher. In this case, the set of equipment will be idle only when machines that are connected by rigid connection fail. These machines are mixing lines, distributing feeds and those that are rigidly connected to the mixing line.

The coefficient of technical use of the set of equipment with mixed connections will take the form:

$$\eta_{т.и} = \frac{\Sigma\tau_z}{\Sigma\tau_z + N_{м.ж}\tau_{ср.пр} + N_м\tau_{ср.об}}, \tag{14.17}$$

where $N_{м.ж}$ – number of machines with a rigid connection, pcs.

Considering the above, the actual performance of any set of equipment at the design stage can be determined:

$$\mathrm{Q} = \frac{B_ф}{\tau} \cdot \eta_{т.и}, \tag{14.18}$$

where $B_ф$ – is the volume of production received per shift; τ – is a change of time.

Taking this into account, the continuity condition of the production line can be written as follows:

$$\frac{Q_i}{\sum_{j=1}^{n_i} q_{ij}\eta_{ij}} \geq \frac{Q_{i+1}}{\sum_{j=1}^{n_{i+1}} q\left(i+1\right) j\eta\,(i+1)\,j}, (i = 1, 2, 3). \tag{14.19}$$

where Q_i – is the volume of output, kg.

It can be seen from the formula that the residence time of the product, which inhibits the process inside the machine of each subsequent link, should be less than or equal to the time of the previous link of the flow. To assess the technological lines, you need to know the actual performance. It is especially important to know the actual performance at the concentration of the industry, which is explained by the great damage in case of equipment failures, the complexity of its operation, etc.

14.3 Determination of the capacity of the operational capacity and the number of feed transport machines

In operating conditions, equipment often makes unforeseen stops caused by clogging of working elements, failure of individual parts, absence of any component, etc. Therefore, in order to ensure a continuous flow, in a number of cases it is economically necessary to install accumulating control tanks that smooth out the difference in the operation of consecutive line elements.

In this case, several options are possible for the operation of the line or sections:

b) loading capacity of the container is higher than the discharge rate (fig. 12.4, a): $Q_{заг} > Q_{выг}$;

For the case $Q_{заг} > Q_{выг}$, the minimum capacitance of the tank is selected from (fig. 12.4, a) and are determined from the equation:

$$V_{min} = Q_{выг}(\tau_{выг} - \tau_{заг}), \tag{14.20}$$

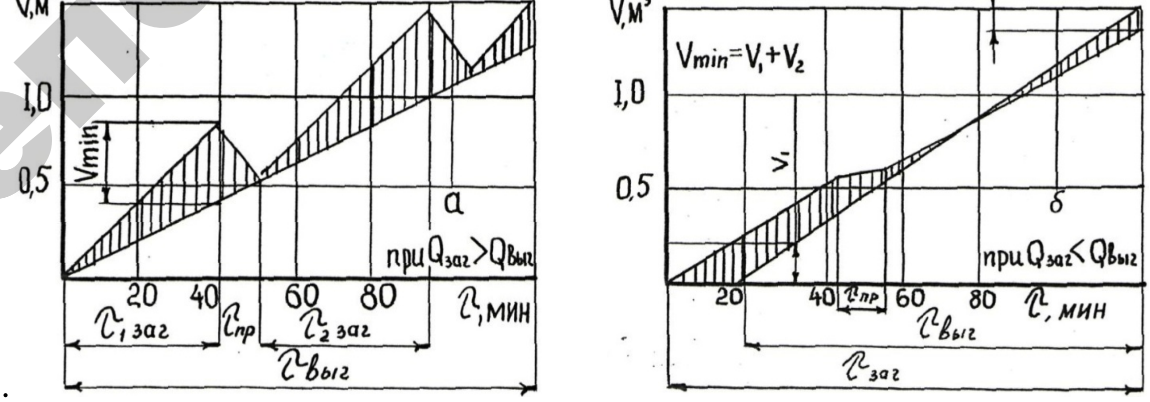
where $\tau_{выг}$ – time of issue of the component, hour; $\tau_{заг}$ – the time of component loading, hour.

c) loading line capacity is equal to the flow rate: $Q_{заг} = Q_{выг}$;

C) loading line capacity is less than the flow rate: $Q_{заг} > Q_{выг}$ (Fig. 14.4, b).
For the case Q < Qrg

$$Vmin = Q_{заг}\,(\tau_{заг} - \tau_{выг}) \tag{14.21}$$

A very important factor that also affects the capacity of the operational capacity is the permissible duration of the storage period. This indicator is determined by the stability of the preservation of its properties



а б

Fig. 14.4. Calculation capacity of the hopper–doser

For the case when Qзаг>Qвыг (fig. 14.4, a) with unit reliability of the machines in the line (section), the purpose of the operational capacity is reduced only to maintaining the necessary productivity, established by the ration of feeding. The capacity of such a capacity is limited by the permissible number of inclusions of machines per unit of time, i.e. Reliability of starting and other equipment. Since the reliability of machines is always less than one, the capacity of the operational capacity is expedient to be calculated through the average downtime of the ahead (loaded) line (section). So, if on such a line segment the total idle time is $\Sigma\tau_{1np}$ with the corresponding number of downtime N_{1np}, the average time

$$\tau_{1cp} = \frac{\Sigma\,\tau_{1np}}{N_{1np}}\,. \tag{14.22}$$

If the first section of the line is stopped, the reserve in the hopper should ensure the operation of the second section during the time τ_{2cp}. For this case in the hopper there should be a food in the quantity G = $Q_{выг}\,\tau_{2cp}$, or taking into account the average time of idle time

$$V_g = \frac{Q_{выг}\left[\tau_{ncp} + (\tau_{выг} - \tau_{заг})\right]}{\rho_к \kappa_з}, \tag{14.23}$$

$$V_g = \frac{Q_{выг}(\tau_{заг} - \tau_{выг}) + \tau_{ncp}}{\rho_к \kappa_з}\,. \tag{14.24}$$

Formulas (14.23) and (14.24) make it possible to determine more rational values of hopper capacity, since they take into account the average idle times of the corresponding lines.

When determining the number of machines for transportation from the places where feed is stored to the places where they are prepared for feeding, the guidelines for the technological design of livestock enterprises and zootechnical requirements are used as guidance materials.

The required number of vehicles for transportation of feed can be determined by formula

$$n_T = \frac{Q_л\,T_ц}{W_T\,\eta_T}, \tag{14.25}$$

where $Q_л$ – hour productivity of a technological line of preparation of forages to feeding, т/ч; $T_ц$ – is the duration of the transport cycle, h; W_T – carrying capacity of the vehicle, t; η_T – coefficient of time use shift.

The transport cycle consists of the following main parts:

$$T_ц = T_1 + T_2 + T_3 , \qquad (14.26)$$

where T_1 – is the vehicle's running time, h; T_2 and T_3 – respectively, the time taken to load and unload fodder from the vehicle, h;

The running time of the vehicle can be determined from the formula

$$T_1 = \frac{2L}{V_{cp}} , \qquad (14.27)$$

where L – is the path length, km; V_{cp} – average speed of movement of a vehicle, km/h.

Maximum vehicle performance is reached when

$$L = \frac{W_T V_{cp}}{Q_{з\text{-}в}} . \qquad (14.28)$$

From the formula (14.28) it is possible to obtain the value of the vehicle's carrying capacity when the complex performance indicator has a maximum:

$$W_T = \frac{LQ_{з\text{-}в}}{V_{cp}} . \qquad (14.29)$$

To minimize transport costs, various variants of the organization of traffic routes can be applied. The simplest of them is a pendulum route with a back–loaded vehicle run. With this version of the organization of transportation of feed, the vehicle is half loaded.

When organizing the movement of a vehicle, it is necessary to minimize the formula

$$L_x = \sum_{i=1}^{n_i} (L^1{}_x - L_{гр})\, n_T , \qquad (14.30)$$

where $L^1{}_x$ – is the distance covered by the vehicle without load, km; n_T – number of routes.

When solving a problem, routes are ordered in ascending order of differences (). In this case the formula

$$(L^{1}_{x1} - L_{гр1}) \leq (L^{1}_{x2} - L_{гр2}) \leq \dots \leq (L^{1}_{xn_i} - L_{грn_i}). \qquad (14.31)$$

Then the optimal solution will look like:

$$x_1 = min\,(m_{к1}, n_T), \qquad (14.32)$$

$$x_2 = min\,(m_{к2}, n_T - x_1),$$

$$x_3 = min\,(m_{к3}, n_T - x_1 - x_2),$$

$$x_n = min\,(m_{кm}, n_T - \sum_{i=1}^{n-1} x_i).$$

To solve the problem, the initial data are entered in table 14.1.

Solving the problem, we will get the data, allowing to determine the most rational route of the vehicle's movement. The best value will be for the minimum difference $L_{xn} - L_{грn}$.

Table 14.1. Initial data for the determination of a rational pendulum route

Fodder storage area	Number of loaded riders	Difference Column
$Б_1$	L_{x1} $L_{гр1}$ m_{r1}	$L_{x1} - L_{гр1}$
$Б_2$	L_{x2} $L_{гр2}$ m_{r2}	$L_{x2} - L_{гр2}$
$Б_3$	L_{x3} $L_{гр3}$ m_{r3}	$L_{x3} - L_{гр3}$
$Б_n$	L_{xn} $L_{грn}$ m_{rn}	$L_{xn} - L_{грn}$

The average value of the speed of movement of the unit during transportation of feeds is recommended to be chosen depending on the distance between the places of their storage. The speed of the unit should not be more than 7,2 km/h. With an increase in the transportation distance of feed (), this figure should be increased to 22 km/h. When choosing the speed of movement, it is necessary to take into account the nature of the road surface – on roads with hard surface, the average speed of the unit increases. The speed of movement of the unit when distributing feed is recommended to 5 km/h. Thus, various variants of routes can be compiled for transportation of feed. The problem of transportation planning includes the definition of a route with a minimum idling of the vehicle, the distribution of rolling stock and loading means along the routes of work, which must be closely linked to the productivity of stationary vehicles. The number of feed distributors required to maintain the livestock population,

$$n_p = \frac{m_{жс} q}{Q_c}, \tag{14.33}$$

where $m_{жс}$ – is the number of animals on the farm; Q_c – productivity of the feeder for 1 hour of shifting time, kg/h.

Productivity of a feed dispenser for 1 hour of changeable time, kg/h,

$$Q_c = Qk_p, \tag{14.34}$$

where Q – is the productivity of the feed distributor per hour of net time, kg/h; $Q = qv_a K_{v,}$ k_p – coefficient of working time use:

$$k_p = \frac{t}{t + t_0}, \tag{14.35}$$

where t – time spent on direct distribution of feed, h; t_0 – time spent on non–productive (auxiliary) operations, h.

Then

$$t_0 = t_1 + t_2 + t_3 + t_4 + t_5 + t_6 + t_7, \tag{14.36}$$

where t_1 – the time of delivery of an empty feed dispenser from the place of keeping the animals to the place of loading and back, h; t_2 – loading time, h; t_3 – time for distribution of feed, h; t_4 – time spent on a simple for technological reasons, h; t_5 – time spent on maintenance, h; t_6 – time spent on repairing the machine, h; t_7 – time of moving from one distribution line to another, if the capacity of the body (bunker) provides distribution of food in several lines, h.

The running time of the vehicle can be determined from the formula

$$t_1 = \frac{2L}{v_{cp}}, \tag{14.37}$$

where L – is the path length, km; v_{cp} – average speed of the vehicle, km/h.

The time for performing loading and unloading operations is determined by the formula

$$t_2 + t_3 = \frac{W_T K_T}{Q_{загр}} + \frac{W_T K_T}{Q_{выгр}} = \frac{2W_T K_T}{Q_{з\text{-}в}} \ , \tag{14.38}$$

where CT is the utilization factor of the vehicle's carrying capacity; $Q_{з\text{-}в}$ – average harmonic productivity of the loading–unloading process, t/h, $Q_{з\text{-}в} = \frac{2Q_{загр}Q_{выгр}}{Q_{загр} + Q_{выгр}}$.

The correct choice of machines for loading feeds largely determines the efficiency of the livestock farm. The productivity of the loader can be determined by the formula

$$Q_{загр} = m_п \, n_ц \, , \tag{14.39}$$

where m_n – is the mass of the cargo as it rises, t; $n_ц$ – number of machine cycles per 1h of continuous operation, depends on the duration of one cycle, h;

$$n_ц = \frac{3600}{T_ц^{'}} \, ,$$

– cycle time, h;

$$T_ц^{'} = \sum_{i=1}^{n_n} t_{on} \ ,$$

t_{on} – time taken to perform certain operations during loading, hour; n_n – number of elements of the loader work.

The time spent on distributing feed to animals can be determined by the formula

$$t_{разд} = \frac{n_{ж} \quad L_{разд}}{V_{разд}} \, , \tag{14.40}$$

where $n_{ж}$ – is the number of animals served per cycle, pcs; $L_{раз}$ – the length of the feed distribution front, km; $V_{раз}$ – speed of movement of the unit at distribution of forages, km/h.

According to the above technique, a techno–economic calculation of self–propelled mixer–distributors has been made (table 14.2) for different sizes of trusses.

Based on the data in table 14.1, a graph (fig. 14.5) is plotted for the dependence of the given unit costs, from which it can be seen that the rational capacity of the mixer–distributor hopper for a dairy farm for 600 cows is within 11 m^3.

Table 14.2 Technical and economic characteristics of self–propelled

Index	Capacity of bunkers Mixer dispensers, m^3				
	6	10	12	14	20
<u>Farm 200 cows</u>					
Specific capital investments, $ / t	1,46	2,23	3,86	4,36	8,62
Direct costs, USD / t	3,38	4,36	7,33	8,42	15,51
The resulted expenses, dollars / t	3,67	4,92	7,8	9,07	17,23
<u>Farm 400 cows</u>					
Specific capital investments, $/t	0,73	1,11	1,92	2,16	4,28
Direct costs, USD t	2,33	2,74	4,14	4,43	7,22
The resulted expenses, dollars/t	2,47	2,96	4,52	4,86	8,08

Mixers–distributors of fodder at an annual milk yield of 6000 liters

<u>Farm 600 cows</u>					
Specific capital investments, $/t	0,97*	0,74	1,28	1,44	2,85
Direct costs, USD/t	4,52*	2,26	3,21	3,54	5,92
The resulted expenses, dollars/t	4,71*	2,41	3,47	3,83	6,49
<u>Farm 800 cows</u>					
Specific capital investments, $/t	–	1,11*	1,98*	2,30*	2,14
Direct costs, USD/t	–	2,94*	4,00*	4,74*	4,84
The resulted expenses, dollars/t	–	3,16*	4,40*	5,10*	5,27

* Indicators with two mixer dispensers.

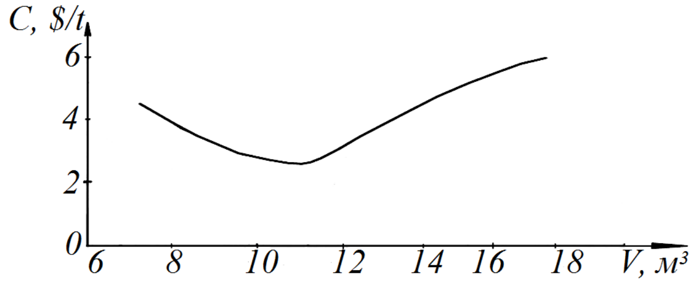


Fig. 14.5. Dependence of unit cost per capacity
Bunker of mobile feed mixer–distributor

Based on the above, we can draw conclusions:

– for dairy farms up to 200 goals. And up to 1000 goals. Feeding capacity of the hopper of the mixer–distributor should not be more than 6 m^3;

– for farms with large livestock it is advisable to produce mixer–distributors with a hopper capacity of about 11 m^3;

– a combination of two mixers–distributors with bins of 6 m^3 and 11 m^3 will allow servicing any dairy farms available in the republic with the least operating costs.

14.4 Organization of technological processes Milking and primary processing of milk on a livestock farm

With machine milking, a number of important requirements must be fulfilled, which determine its successful application and create favorable conditions for the activity of the mammary glands of cows. First of all, it is necessary for an animal to develop a full reflex of milk yield, that is, an active response of the cow to milking. To do this, the udder is washed with warm water and massaged. The irritation (massage) of the nerve endings in the nipples before milking promotes more complete milk yield.

At every milking from the udder of the cow, all milk must be extracted, otherwise the secretory activity of the breast will be disturbed. In addition, incomplete extrusion causes a premature start of the cow. Do it regularly and at least twice a day. Milking should be fast – within 7 min. Rapid milking and the removal of a manual dodo make the cow give milk completely; On the contrary, after slow milking, it must be dosed. In conditions of machine milking, it is necessary to eliminate the causes that inhibit the reflex of milk yield, since the response to stimuli of milking can be interrupted by the nervous state of the animal.

It should also take into account the individual characteristics of cows and their habits. Selection of cows in the group, similar in many ways and, in particular, in the type of nervous activity, is an obligatory measure and provides a more rational and high–performance use of milking areas. Multiplicity of milking is set such that between milking intervals the udder is filled with milk and the milk forming is not inhibited. Usually, cows are milked 2...3 times a day, highly productive and novative 3...4 times. Before starting the number of milking is gradually reduced.

Produce two and three–fold milking cows during the day. With a three–fold milking, in a number of cases, 10 % more milk is received than twice. But this is typical for cows with a small udder capacity. Cows with a large udder capacity do not increase milk yields in such cases. With a reduction in the same amount of milking from three to two, labor costs are reduced by 25…30 %. Fresh milk has high taste and nutritional qualities. However, under favorable conditions, bacteria rapidly multiply in milk, contributing to its souring. The rate of reproduction of bacteria depends mainly on the temperature of the milk. Thus, for fresh milk at a temperature of 30 °C, the period of delay in the development of bacteria lasts 3 hours, for rapidly cooled to 10 °C – 24 hours, and for a cooled to 5 °C – 36 hours. To maintain the quality of fresh milk for a long period, It is necessary to observe the rules of hygiene during milking, to clean (filter) it in a timely manner after milking, to cool and store at a temperature of 3...5 °C. To completely destroy the microorganisms, the milk is pasteurized. It is established that 100 % of bacteria die when the milk is heated to 80 °C.

The implementation of technological processes in dairy cattle breeding is inextricably linked with the transport processes performed to move the product without changing its state in the path. With the machine method of milking and primary milk processing, all consecutive operations are combined into an uninterrupted technological stream, i.e. Line production mechanized and automated lines are created. The flow–technological lines must: to carry out the technological process with the least expenditure of labor, energy, resources and time; meet the zootechnical requirements for the quality of work and be as reliable as possible; to service all livestock of animals on the farm. The construction of the technological process begins with the definition of the composition and sequence of operations, which are included in this or that line, represented as a diagram. Technological (or operational) schemes are a brief description of the order and sequence of execution of individual operations of the line–technological lines (PTL) without specifying the type and brand of the machine that carries out this or that operation. The scheme is a list of operations connected with one another arrows indicating the direction of the technological (material) flow (fig. 14.6). The composition and sequence of operations are chosen taking into account zootechnical requirements to the quality of final products, the latest achievements of science and technology.

Structural and technological schemes reflect the specific composition of the machines included in the PTL and are presented in the design technical documentation showing the types of machines and technological processes (fig. 14.7).

Structural schemes of PTL reflect the internal structure of production flows, the co–ordination of individual elements, sections or sections, show the direction of movement of material flows, control actions and commands, the presence and location of regulatory or reserve tanks and reserving facilities (fig. 14.8).

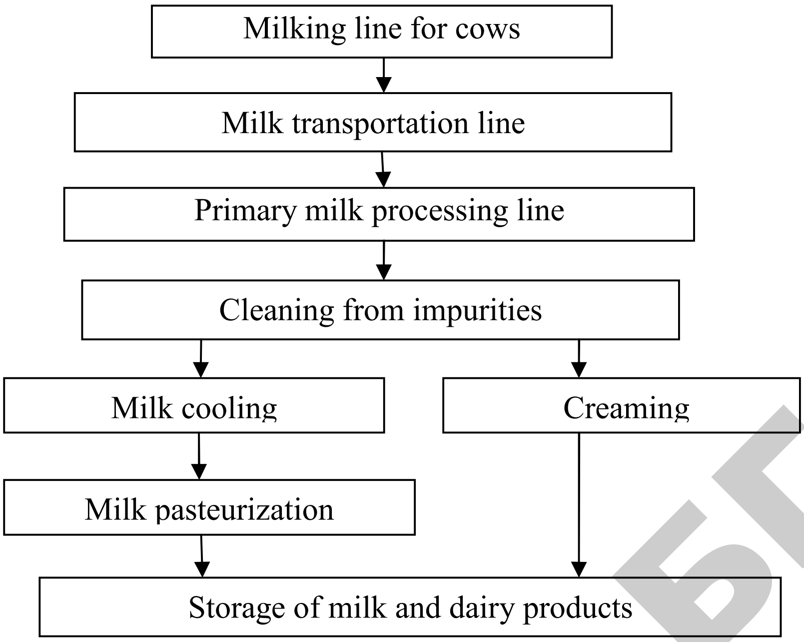


Fig. 14.6. Technological scheme of the process of milking cows and primary processing of milk on a dairy farm

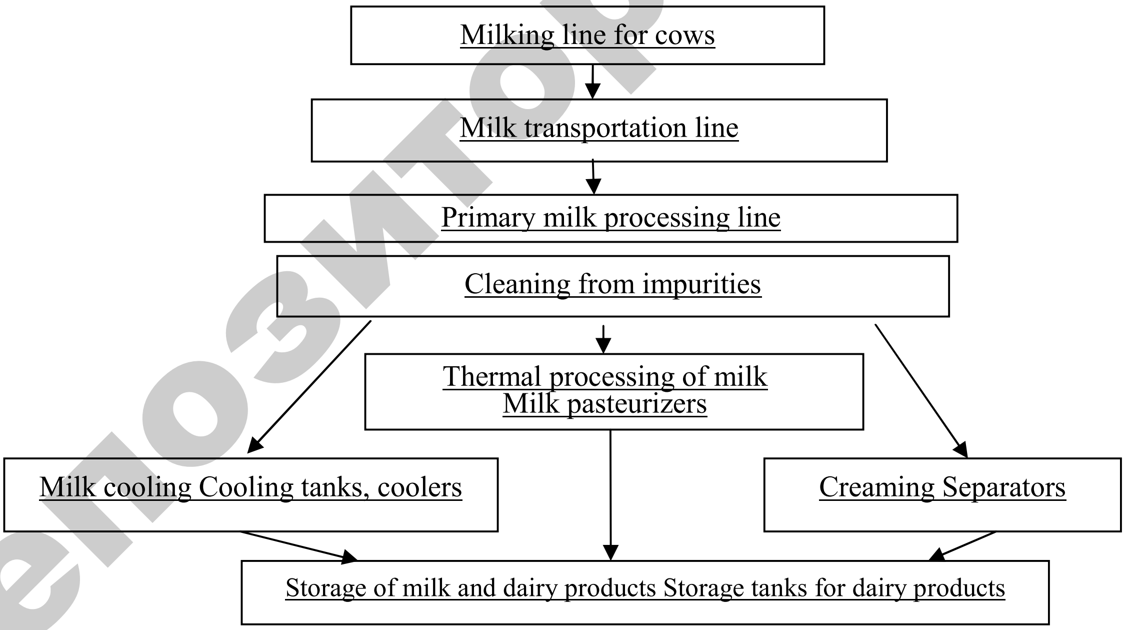


Fig. 14.7. Structural and technological scheme of milk production and primary milk processing on a dairy farm

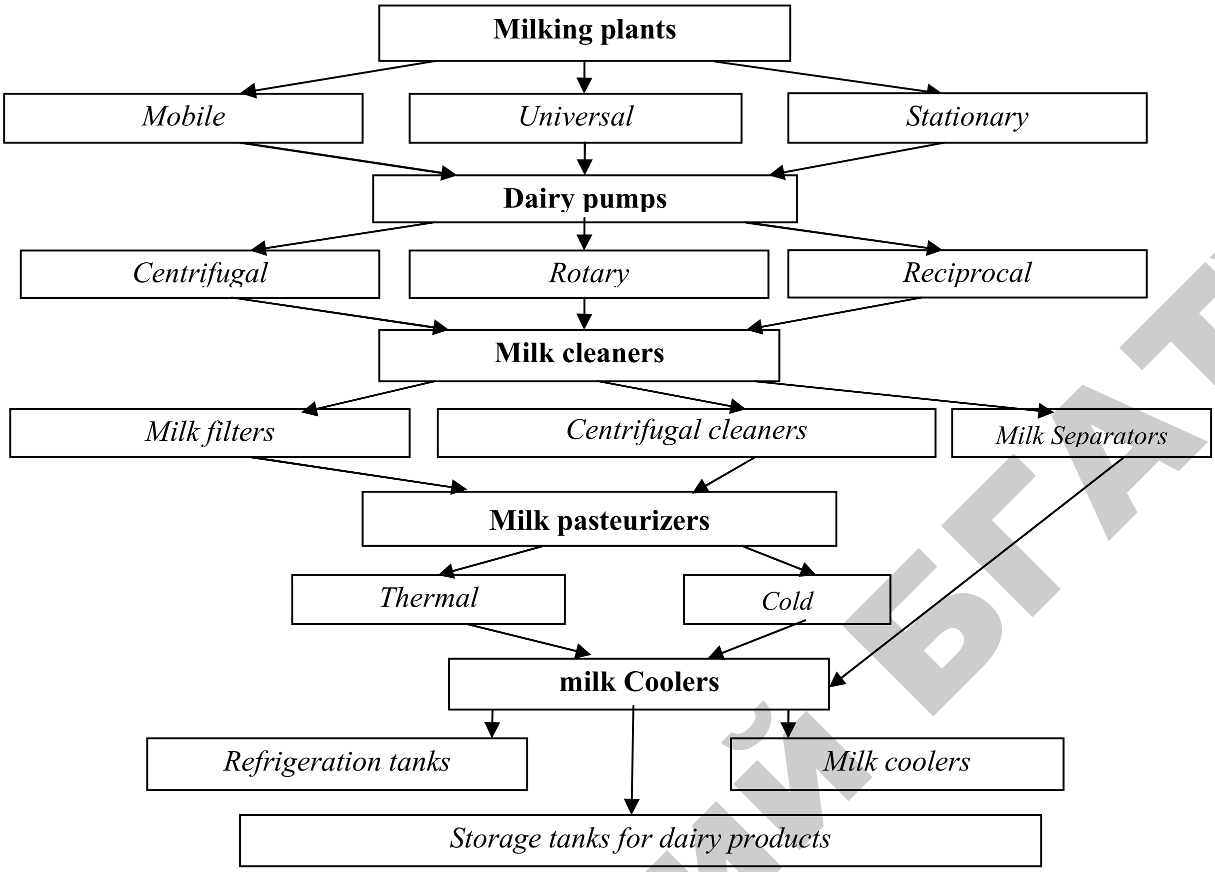


Fig. 14.8. Flow chart of the milking line of cows
And primary processing of milk

From the correct choice of the structure of the PTL depends, first of all, the reliability of the entire line and its technical and economic indicators.

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