MINISTRY OF EDUCATION AND SCIENCE OF UKRAINE

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DESIGNING OF AIRCRAFT GEAR PUMPS

Tutorial

Kharkiv “KhAI” 2021

UDC 629.7.036.34
B44

Наведено основні відомості щодо конструкції і розрахунків деяких шестеренних насосів, що застосовують в системах паливоживлення, змащення, гідравлічних системах літальних апаратів із різними приводами: як від роторів двигуна, так і від електродвигуна. Розглянуто особливості конструктивного виконання, принципи роботи шестеренних насосів. Викладено основи їх розрахунку і проектування.

Для англомовних студентів, які вивчають системи і агрегати авіаційних силових і енергетичних установок.

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Tutorial provides basic information about the designs and calculations of some gear pumps used in fuel-supply systems, lubrication, hydraulic systems of aircraft with various drives: both from the engine rotors and from the electric motor.

The features of the design, the principles of operation of gear pumps are considered, the foundations of their calculation and design are outlined.

For students studying the courses "Aircraft Power Plants and Units", "Design of Aircraft Engines and Units", "Design of Power Plants" and a pre-diploma course.

This tutorial will be interesting for English-speaking students who study the systems and units of aircraft engines and power plants.

Figs 20. Tables 4. Bibliography: 6 names

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CONTENT

OPERATIONAL PRINCIPLE

In this section, an example for each type of gear pump is briefly illustrated. In all figures, the driving gear rotates clockwise and the delivery port is located on the upper side.

Gear pumps belong to a positive displacement rotary group, and are made by enclosing two or more gears in a close-fitting housing. A driver turns a shaft connected to one of the gears, causing it to rotate. This gear drives the other gear through the meshing of the teeth of the two gears, just as with power transmission gears.

In Figure 1 an example of an *external gear pump* is shown. The fluid is transported in the inter-teeth volumes (also called tooth-space volumes or carry-over volumes) along the periphery of the two gears. Two grooves are machined on the balance plates for avoiding trapped fluid in the meshing region. The balance plates house the bushings and are mounted with clearance on the casing so that a small axial movement is allowed. On the rear of the balance plates, the delivery pressure acts on a proper surface limited by a seal in order to generate a clamping force on the gears; in this way the compensation of the axial clearances is obtained.

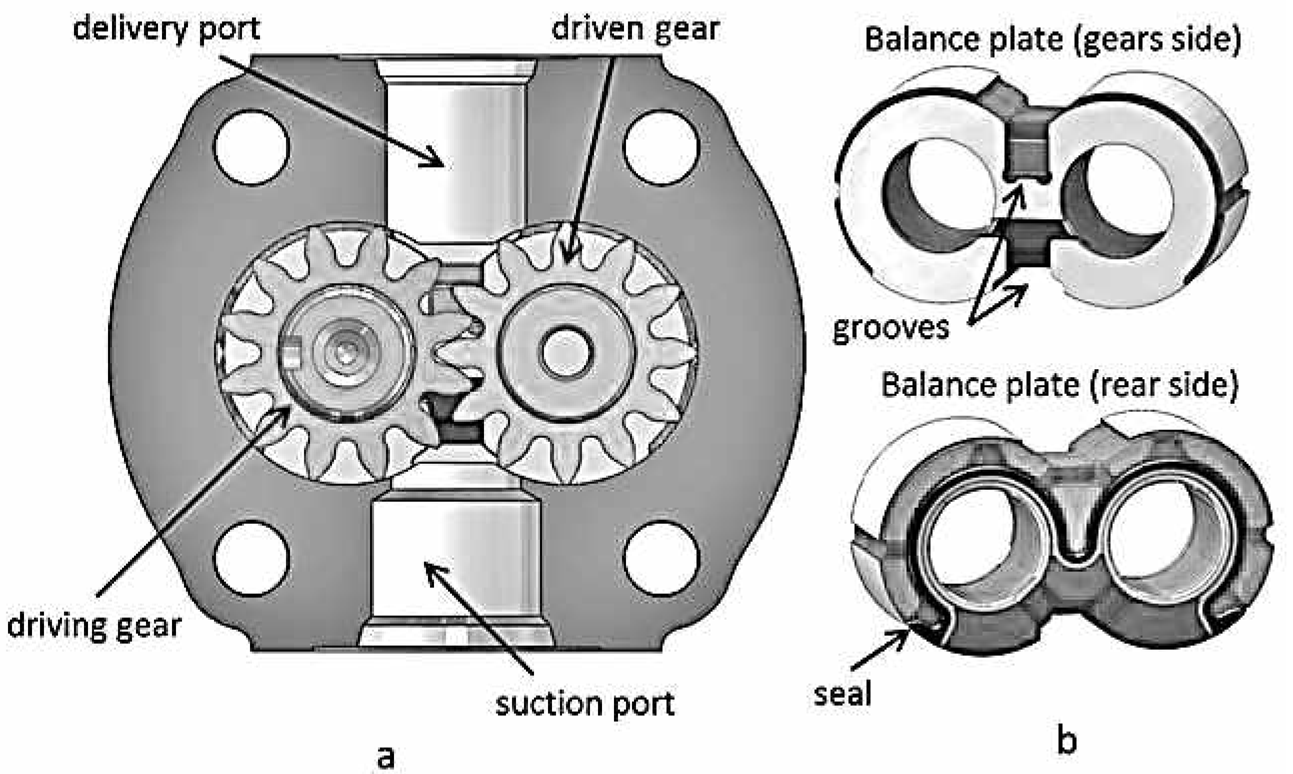


Figure 1 – External gear pump:
a – example of an external gear pump; b – detail of the balance plates

As the gears rotate, on one side, the teeth are coming out of mesh with each other (see Figure 1). As a tooth is pulled out of the space between two teeth of the other gear, it creates a vacuum. Since the housing forms a seal all around the set of gears, the liquid that rushes into this space to fill this void has to come in through the pump's suction port. Once the spaces between gear teeth are filled with liquid, the liquid rides in these pockets, trapped in place by the housing, until it reaches the discharge side of the pump. The liquid stays in place between the teeth until it reaches the other side of the gear mesh, where the teeth are coming together. Then, when a tooth from the other gear comes into the space between the teeth, the liquid there is forced out. Since the housing still forms a seal around the gears, the only place for the displaced liquid to go is out the pump's discharge port. The pump thus operates like a conveyor belt, with the pockets of liquid between the gear teeth being picked up at the gear mesh, carried to the other side, and dropped off at the other side of the mesh.

Internal gear pumps have one larger (outer) gear with the teeth turned inward, meshing with a smaller gear with external teeth.

Figure 2 shows a low pressure *gerotor pump* used in a lubricating circuit for internal combustion engines. Ideally, the conjugated profiles of the rotors have as many contact points as the number of teeth of the outer gear.

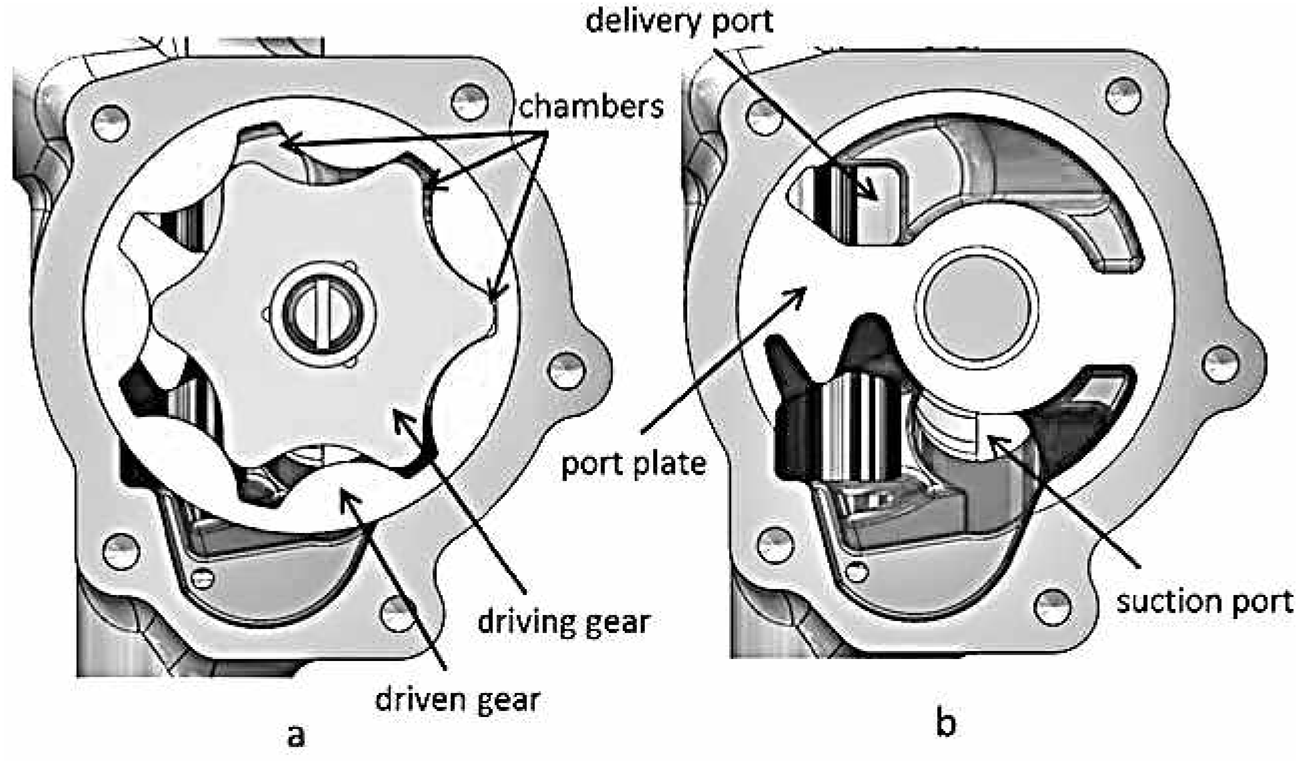


Figure 2 – Internal gerotor pump:
a – example of a lubricating gerotor pump; b – detail of the port plate

Besides these pumps have a special tooth structure: the inner gear of these pumps has one tooth less than the external one (see Figure 2). The two gears form a seal by themselves.

The rotors are mounted directly in the housing without any automatic compensation of the gaps. The shape of the port plate can be quite complex for improving the filling of the chambers at high speed.

If the larger gear has at least two teeth more than the smaller gear, then a crescent-shaped projection of the housing goes between the two gears to help form a seal.

In the *crescent-type internal gear pumps* the difference between the number of teeth of the gears is greater than one. For this reason to ensure the sealing between the suction and delivery volume, a fixed element, integral with the port plate, is needed. A medium pressure industrial pump in depicted in Figure 3.

Internal gear pumps are more compact and immune to cavitation. The disadvantage of pump with the internal gearing is a high manufacturing cost.

The operating principle is the same for all of these types of pumps, and they operate in similar fashion.

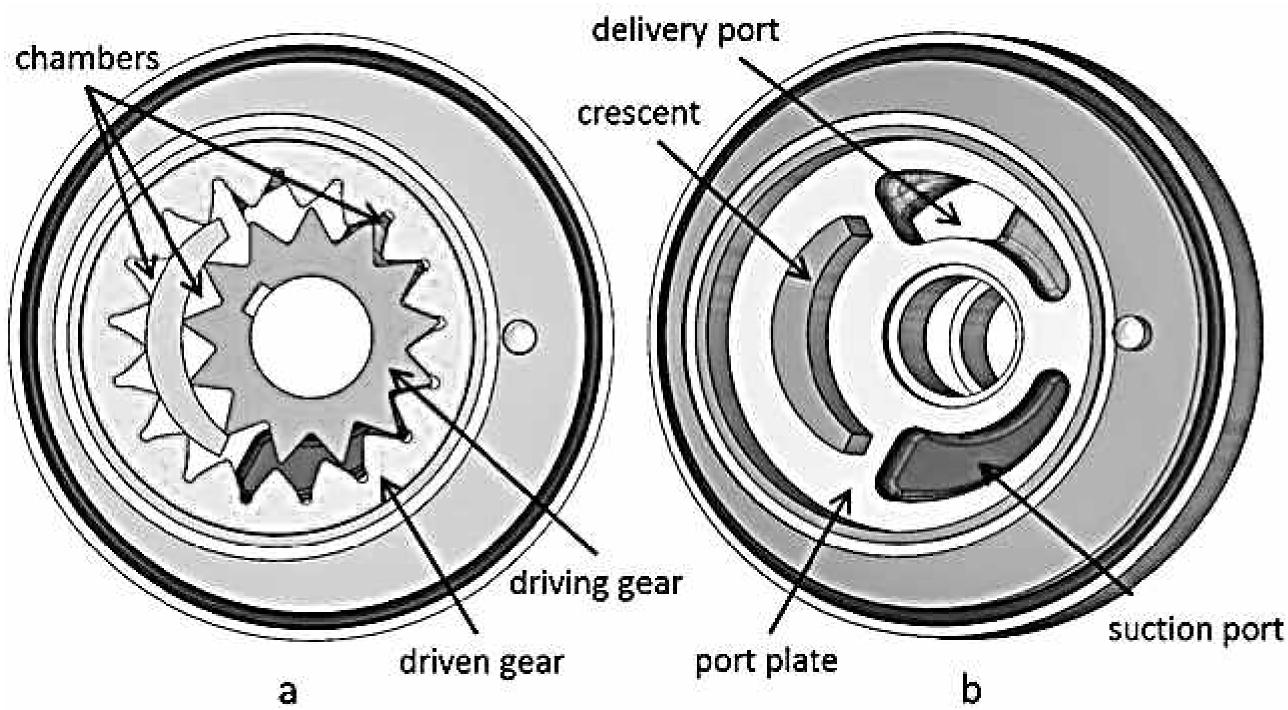


Figure 3 – Crescent-type internal gear pump:
a – example of a crescent pump; b – detail of the port plate

CLASSIFICATION

There are gear pumps main classifications (Figure 4) according to:

- number of stages;
- number of sections;
- number of gears;
- teeth shape;
- gearing type;
- working pressure value.

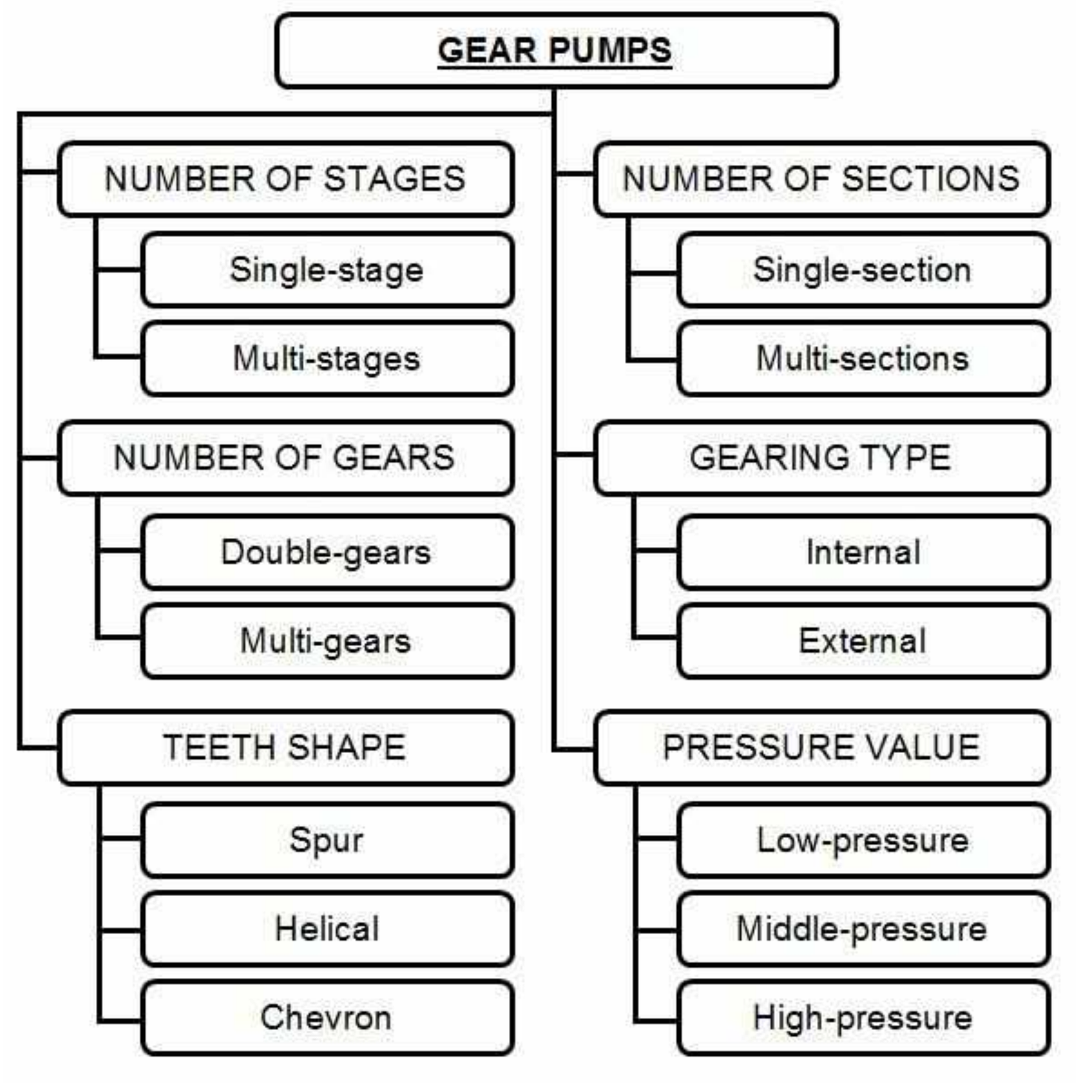


Figure 4 – Gear pumps classification

According to **number of sections** in a single unit there are single-section and multi-section pumps (Figure 5, a). Multi-section pump presents a few sections which are connected in parallel. Such scheme gives possibility to increase working liquid flow and decrease overall sizes and structural mass of unit. It is very important because of it is very difficult to provide contact line in case big gears diameter.

Depending on **number of stages** there are single-stage and multi-stage pumps (Figure 5, b). Multi-stage pump presents a few stages, which are connected in series. Each subsequent stage has a higher capacity than the previous. Liquid excesses behind stage are withdrawn to inlet cavity through reducing valve that is adjusted to some pressure value. This scheme makes it possible to increase the outlet pressure by 2 - 3 times without increasing the size of the gears. An important disadvantage of multistage pumps is a decrease in volumetric efficiency because each stage, with the exception of the last, is designed for a higher flow rate than necessary.

According to **number of gears** in a single unit there are double-gear (Figure 6, a) and multi-gear (usually three-gear) pumps (Figure 6, b). Three-gear pump scheme gives possibility to reduce overall sizes and structural mass. Besides this, drive gear bearings are unloaded from liquid pressure.

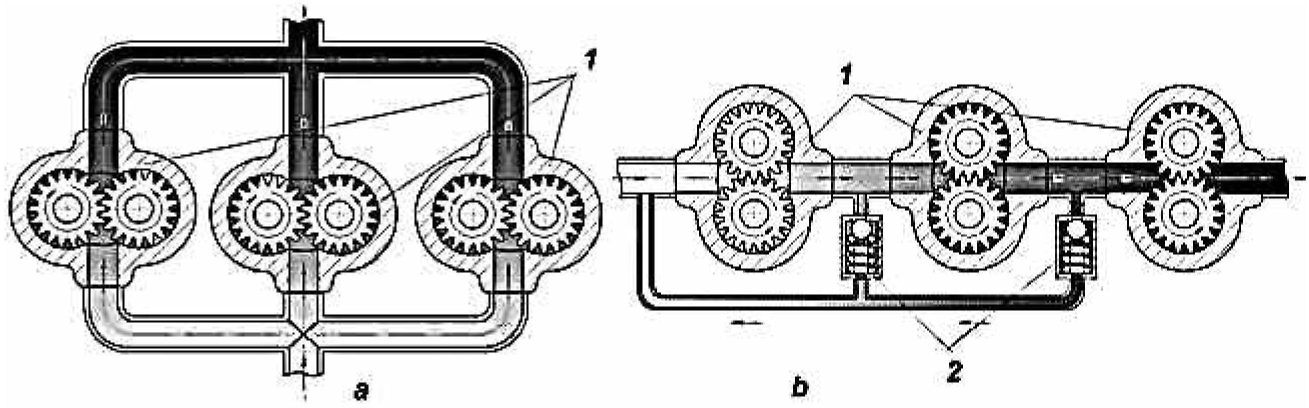


Figure 5 – Gear pump connection types:
a – multi-sections pump; b – multi-stages pump
(1 – pumps; 2 – pressure reducing valves)

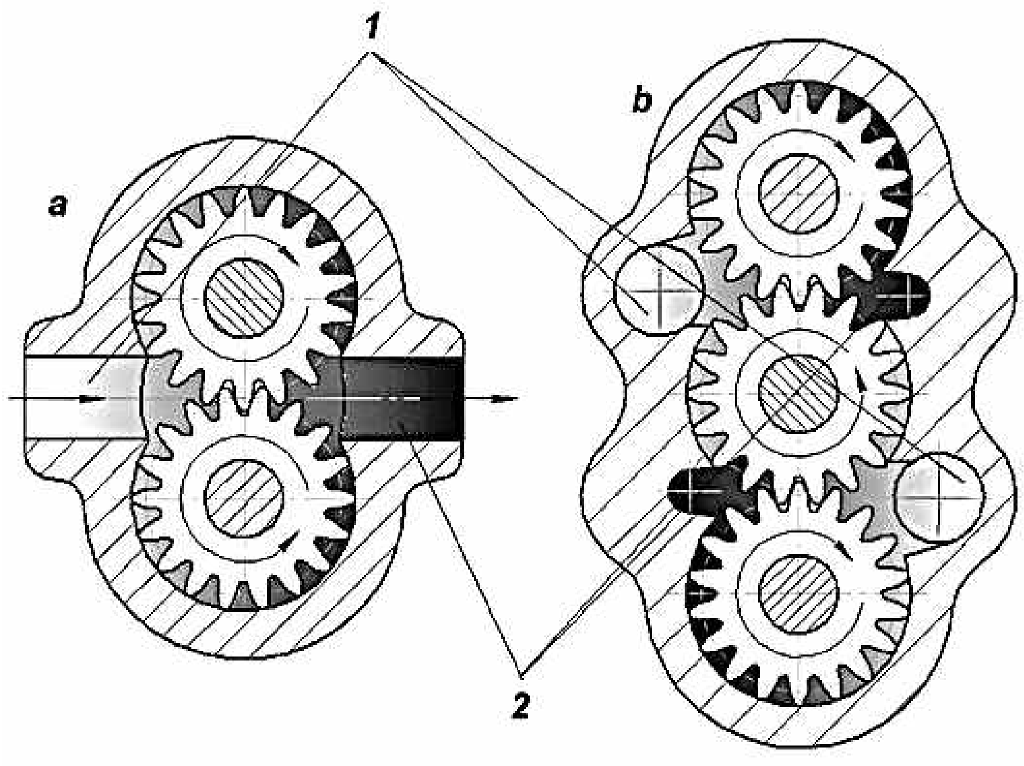


Figure 6 – Gear pumps according to gears number:
a – double-gear pump; b – multi-gear pump *(1 – inlet cavities; 2 – outlet cavities)*

Depending on **teeth shape,** there are spur, helical and chevron gear pumps. **Spur** gears (Figure 7, a) are simple and cheap in design and manufacture but this gear type is characterized by rapid wear of teeth, significant noise in operation and pulsation during working liquid supply.

Helical (see Figure 7, b) gears design and manufacturing are more difficult and expensive but these gear types give possibility to provide smooth operation, reduce noise value and prevent pulsation and blocking of working liquid in gear cavities.

Disadvantage of helical gears is additional axial forces, which press gear to end. This problem is solved in **chevron** (herringbones) gears (see Figure 7, c) but its manufacture is more difficult and expensive that spur and helical gears.

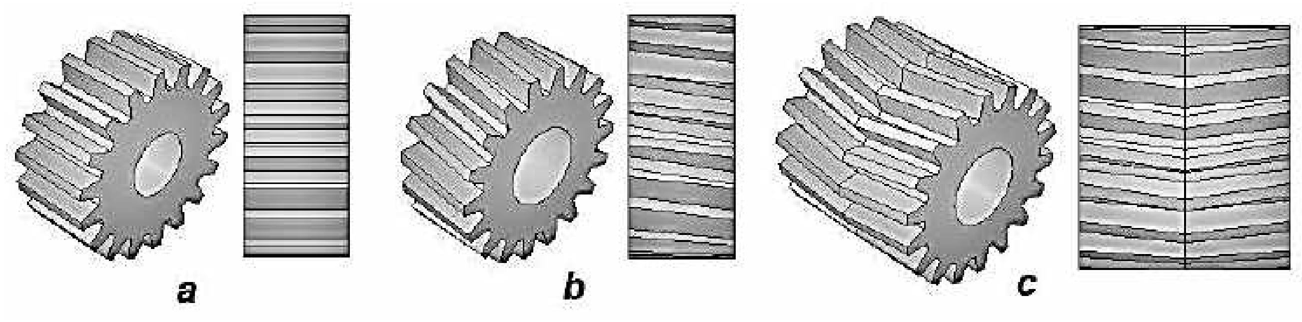


Figure 7 – Teeth shapes types: a – spur-teeth; b – helical-teeth; c – chevron-teeth

Depending on **gearing type,** there are external and internal gearing pumps. First pump type is common because of simplicity in manufacture.

Internal gearing (see Figures 2 and 3) includes **outer** and **inner gears**. This pump type is more difficult and expensive but its scheme has some significant advantages. First of all, it has less overall sizes and mass in compared with external gearing. Besides this, internal gearing pumps have better cavitation characteristics.

According to **working pressure value,** there are three gear pump types: low pressure (less 2 MPa), middle pressure (2 - 10 MPa) and high pressure (more than 10 MPa). Low-pressure gear pumps are usually used in power plant lubrication systems. This system doesn't need in large pressure difference because of oil spraying is provided by centrifugal forces of bearing rolling bodies. High-pressure gear pumps are used in fuel system because it is necessary to provide significant pressure difference at fuel nozzles.

PUMP CAPACITY

A tooth space volume, a number of pockets (spaces), the rotational speed and the volumetric efficiency determine the capacity (output flow) of a gear pump.

The displacement of a pump is the volume of liquid moved in those pockets between gear teeth. It is the theoretical output of the pump before any losses are subtracted. The instantaneous mode of displacement varies slightly as the teeth move through different positions in the mesh, so displacement per shaft revolution cannot be calculated exactly. However, there are some good approximations. For example, if the cavity area between the gear teeth is assumed to be approximately equal to the area of the teeth themselves (i. e. a cavity is an inverse of a tooth), then the displacement per revolution would be equal to the volume occupied by half the space between the gear outside diameter (addendum circle) D_o and a root diameter (clearance circle) D_r minus gearing radial clearance volumes ($0.25m$), multiplied by two (to account for two gears), times gear width:

$$q \approx 2\pi D_p m b,$$

where m – gearing modulus;
b – gear width (a tooth length);
$D_p = mz$ – pitch circle diameter of drive (pinion) gear;
z – drive gear teeth number.

In fact cavities area greater than the teeth one, therefore the formula above gives a little bit underestimated values.

Nomenclature of spur gears is shown in Figure 8.

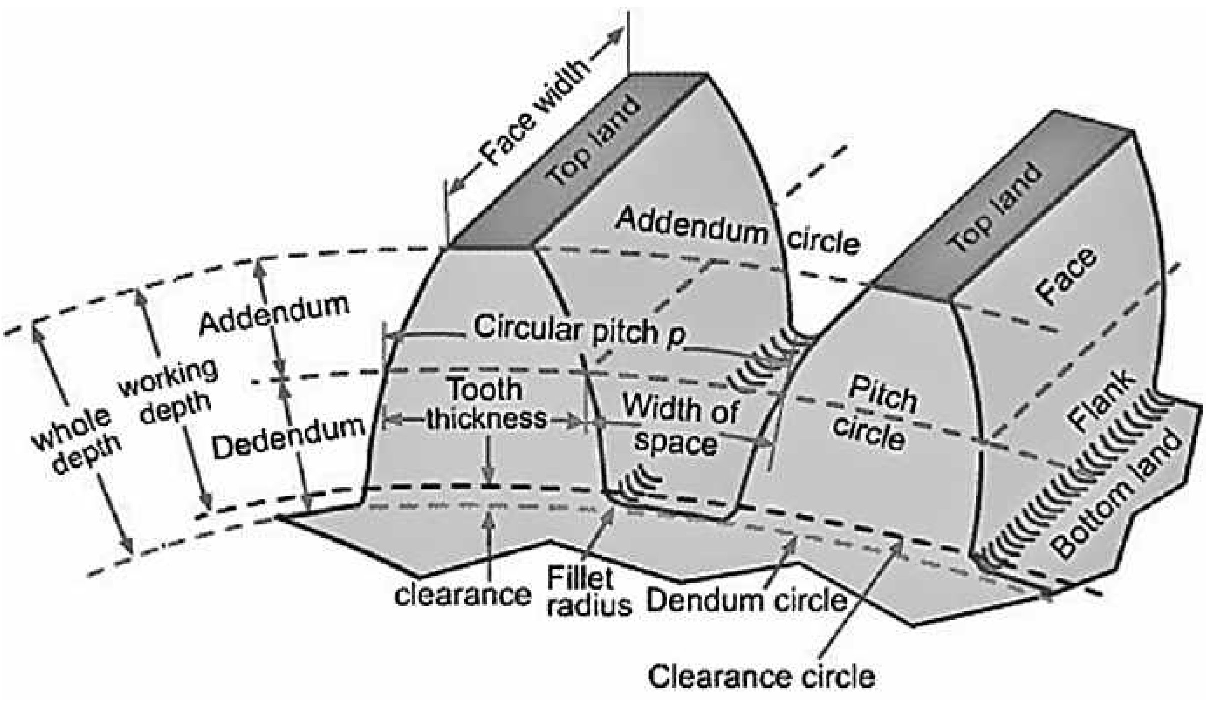


Figure 8 – Nomenclature of pumping spur gears

If the number of teeth is z = 7 - 16, then the capacity of such pump can be evaluated by the empirical formula that gives result that is more precise:

$$q \approx 6.5 D_p m b.$$

An even more accurate result is given by the formula

$$q \approx 2\pi K D_p m b\,.$$

where K – is a correction factor which depends on the teeth number, a gearing angle and a tooth formation method. This factor can be considered 1.07 – 1.15 for the approximate analysis.

The ideal capacity in the general case is

$$Q_{id} = qn\,,$$

where n – is a drive gear rotational speed, rps.

These formulas assume that both gears have the same outside diameter and number of teeth. The addendum of gears for pumps is often extended when compared to power transmission gears. This is to increase the pump's displacement. Gears with smaller numbers of teeth have larger addendum for a given center distance. Therefore, most gear pumps have 12 or less teeth on the gears.

Volumetric efficiency

Slip is the difference between the theoretical flow (displacement × speed) and actual flow, assuming that there is no cavitation. Slip is the leakage of liquid from the high pressure side of the pump back to the low pressure side. There are a number of separate slip paths in any gear pump, including any liquid from the outlet of the pump that is bled off to flush a seal chamber or lubricate bearings. Three paths are common to all gear pumps: between the ends of the gears and the endplates (known as lateral clearance), between the tips of the gear teeth and the inside of the casing (known as radial clearance), and between the profiles of the meshing teeth. The slip through this last path is very small and is usually ignored.

Slip varies strongly with differential pressure and viscosity and, to some extent, with speed. Slip is directly proportional to differential pressure. It varies inversely, but not proportionally, with viscosity. Slip varies asymptotically with viscosity, approaching zero slip at high viscosities. This means that at low viscosities, small changes can mean large differences in slip. Slip varies inversely with speed to a small extent, but this is normally ignored, and predictions are made slightly conservatively at higher speeds. There is also a strong relationship between clearances and slip. Slip, through a particular clearance, varies directly with the cube of that clearance. This is similar to the oil flow in a hydrodynamic bearing, and, indeed, the interaction between gear ends and the casing wall (or wear-plate) is also similar, producing a like hydrodynamic bearing effect. This means that if you double the lateral clearance, you will get eight times as much slip through that clearance. The percentage of slip through each slip path varies with pump design, but in most gear pumps, over half of the slip goes through the lateral clearance; this is because it is usually the largest clearance and it has the shortest distance from high to low pressure. This is why some high-pressure pumps eliminate lateral clearance altogether by using discharge pressure to hold movable endplates against the gear faces while the pump is running.

The positive displacement pump of any type has some inner volumetric losses (slips). They result in reduction of the effective pump capacity.

The reduction of the effective capacity Q_p relative to the ideal one occurs because of the internal leakages (slip) ΔQ_{leak} over the radial (tip) ΔQ_r and lateral (face) ΔQ_f clearances between the exhaust and inlet cavities and because of the volumetric losses during the suction ΔQ_{in} in case of the cavitation:

$$Q_p = Q_{id} - \Delta Q_r - \Delta Q_f - \Delta Q_{in}.$$

The volumetric efficiency η_v defines volumetric losses in the pump. The volumetric efficiency is a relation between the effective and ideal capacities:

$$\eta_v = \frac{Q_p}{Q_{id}} = 1 - \frac{\Delta Q_r}{Q_{id}} - \frac{\Delta Q_f}{Q_{id}} - \frac{\Delta Q_{in}}{Q_{id}} = 1 - \delta Q_r - \delta Q_f - \delta Q_{in}.$$

The volumetric efficiency depends on the pump design and its operational conditions: pumping unit sealing, conditions at the pump inlet, pressure drop, rotational speed and liquid temperature.

The volumetric efficiency cannot get outside the 0 - 1 range. Its magnitude depends on influence of these parameters.

In the absence of cavitation, the volumetric efficiency is estimated taking into account only internal leaks using the formula

$$\eta_v = 1 - \frac{\Delta Q_r}{Q_{id}} - \frac{\Delta Q_f}{Q_{id}}.$$

CAVITATION IN GEAR PUMPS

Cavitation is the formation of voids or bubbles in a liquid as the pressure drops below the vapor pressure of the liquid in the pump's inlet. These bubbles then collapse when they reach the high pressure side of the pump. This collapse can, over time, damage the pump and erode hard surfaces. Cavitation causes a drop in output flow that can sometimes be mistaken for slip, but cavitation can usually be identified by its distinctive sound. Significant cavitation will usually sound like gravel rattling around inside the pump. A rule of thumb is that the liquid velocity in the inlet port should be no more than 1.5 m/sec for low-required net inlet pressure. When pumping viscous fluids, the rotational speed of the pump must be such that the fluid has enough time to fill the voids between gear teeth at the inlet. In other words, the pump can only move the fluid out if there is sufficient suction pressure to push the liquid into the pump inlet. Otherwise, the voids are not filled completely, effectively reducing actual flow through the pump. Therefore, minimum allowable suction pressure depends on the rotating speed, size (pitch diameter), number of gear teeth, and viscosity of the fluid. An approximate relationship for suction pressure following from the equation of radial equilibrium of liquid in a rotating tooth space is

$$p_{\mathrm{in}} \geq p_0; \; p_0 = p_d + \frac{\rho}{8}\left(\mathrm{D}_{\mathrm{o}}^2 - \mathrm{D}_{\mathrm{d}}^2\right)\omega^2,$$

where p_d is the minimum pressure at the dendum circle diameter D_d in the inlet cavity that provides cavitation free operation.

In general, the minimal cavitation-free pressure is a vapor pressure at the constant temperature $p_t(t)$ plus some pressure reserve (NPSH) $\Delta p_{c.\min}$

$$p_d = p_t(t) + \Delta p_{c.\min}.$$

If we know p_d and p_{in} we can find the maximum rotational speed from $p_{in} = p_o$ condition:

$$\omega_{\max} \leq \sqrt{\frac{8}{\rho}\frac{p_{in} - p_{r\min}}{D_o^2 - D_r^2}}.$$

Therefore, pump suction pressure must be greater than the minimum allowable value. If this condition is not maintained, the pump flow will decrease, accompanied by noise, vibrations, and possible damage to the equipment. The cavitation damage in gear pumps, however, is not as severe as in centrifugal pumps. Typically, gear pumps are used for oil and similar liquids which have a significantly lower cavitation (boiling) intensity. The resulting bubbles implode less vigorously than in the case of cold water, and their impact against the equipment's internal boundaries is, therefore, less severe.

The angular speed of rotors in modern gear pumps is set to limit the circumferential velocity of the gear tip u_0 (less than 10 m/s). The typical values are 6 – 8 m/s (for pumps without pressure head at the inlet).

If *n* breaks the limit, then some part of a tooth slot is filled with the fluid vapor. Hence, Q_p diminishes causing an emulsification and enlarged wear out of the pump parts.

Pump drives are designed with the great gear ratio to ensure the *n* variation within the allowable range. Usually $n_{\max} = 5000 - 6000\ rpm$. The reliability of filling the tooth slot also depends on other factors:

– an inlet chamber size;
– the shape and size of ducts feeding the slots;
– presence of dissolved air and gases in the fluid;
– dead space in a tooth slot.

To eliminate the losses at the inlet more than the quarter of the gear circumference must be drawn to the inlet cavity. The fluid is fed to slots along the full gear width and also from the face of teeth root.

The most effective method to prevent the cavitation in the cavities is to pressurize the inlet. It is perspective to use centrifugal-gear pumps where fluid gets to the cavities being forced by the centrifugal forces. Such pumps can operate at high rotational speed without any booster pump.

Centrifugal-gear pumps

A centrifugal - gear pump is a combination of a gear pump with a centrifugal boost pump, the impellers of which are built into the pump gears (Figure 9).

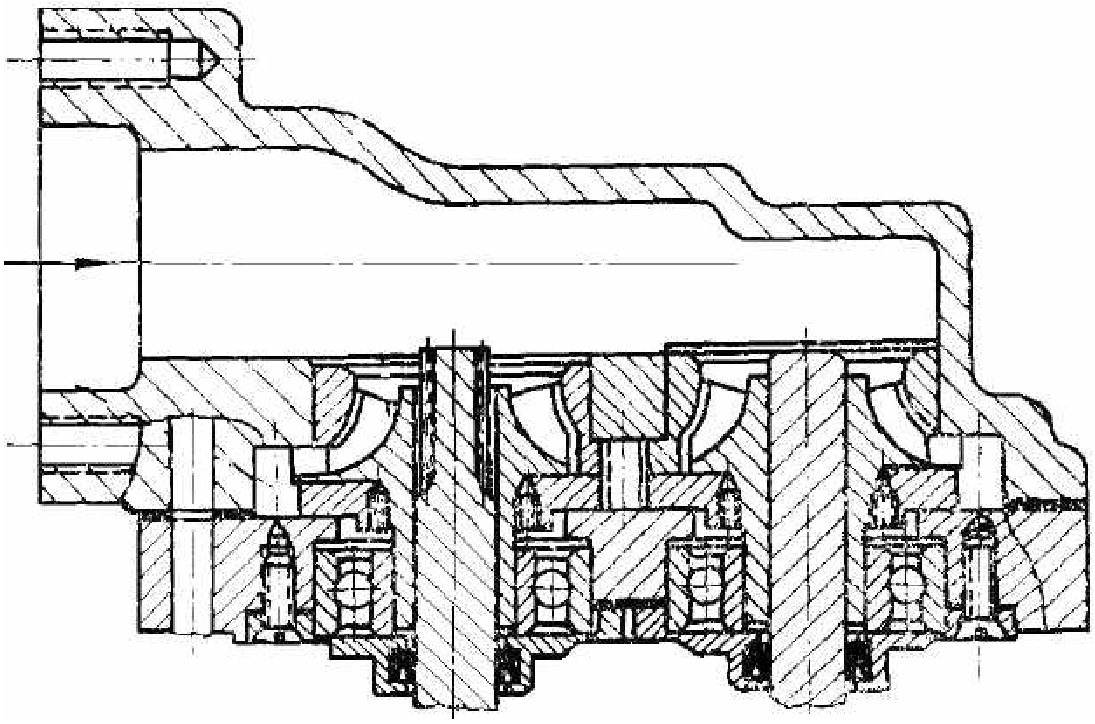


Figure 9 – Centrifugal-gear pump

The centrifugal pump helps to fill the cavities of the gears with oil, because of which the flow rate, pump capacity and the altitude performance of the lubrication system increase. The maximum permissible speed on the tips of the gear teeth of such a pump reaches 30 – 35 m / s with a volumetric efficiency of 0.85 – 0.92.

To increase the volumetric efficiency, the duct for supplying oil to the gear teeth is often positioned so that the oil does not flow to the tooth tips, but to the bottom of the cavity through the hollow axles or to the faces of the gears (Figure 10).

In this case, centrifugal forces contribute to filling the cavities of the gears with liquid, as a result, the volumetric efficiency, pump capacity and altitude performance increase, and the maximum permissible peripheral speed along the tooth tips can be increased to 30 m / s.

The width of the inlet in the housing is must be not less than 0.15 – 0.20 of the circumference of the tooth tips, and in the case of oil supply to the ends of the gears - up to 0.4 – 0.6 of this length. With a narrow inlet, the cavities may not have time to fill with oil, and with an excessive increase in the width of the inlet, the flow of oil through the radial clearances increases.

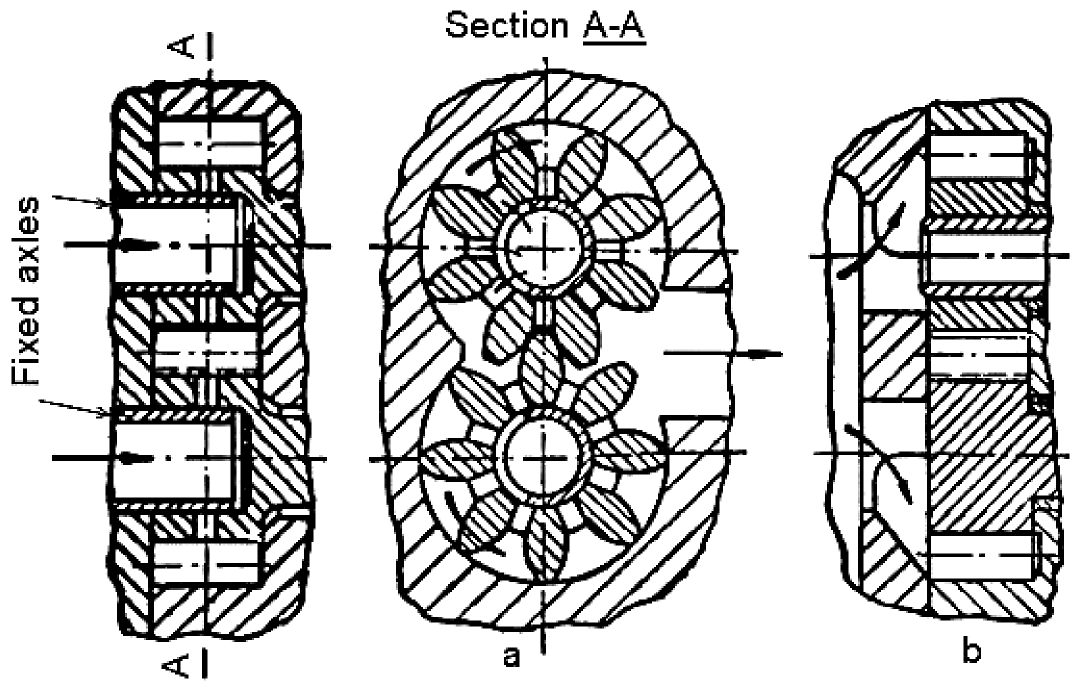


Figure 10 – Scheme of oil supply to the bottom of the space (a) and gears faces (b)

FORCES APPLIED TO THE GEAR PUMP SUPPORTS

Forces that act on the pump bearings are (Figure 11):

– lateral force acting on a gear P_L (the force appears due to the pressure drop between the outlet and inlet);

– reaction P_M (the reaction appears because of a torsion torque applied to the gear);

– pressure forces between the meshed gears (the force appears due to the fluid compression) etc.

If we assume that the pressure in a radial clearance changes linearly (Figure 12, that is fair only when $z \to \infty$) and the length of the inlet and the exhaust cavities is the same, then

$$P_L = a \Delta p D_o b \,,$$

where a is a proportionality factor, which depends on the angular length of the inlet and the exhaust cavities.

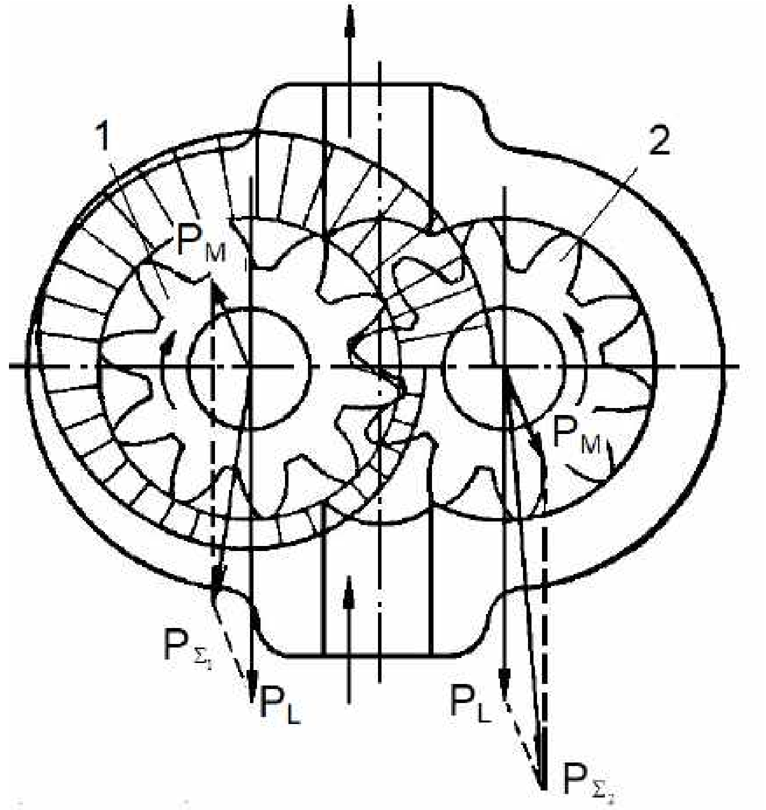


Figure 11 – Forces acting on the bearings: 1 – driving gear; 2 – driven gear

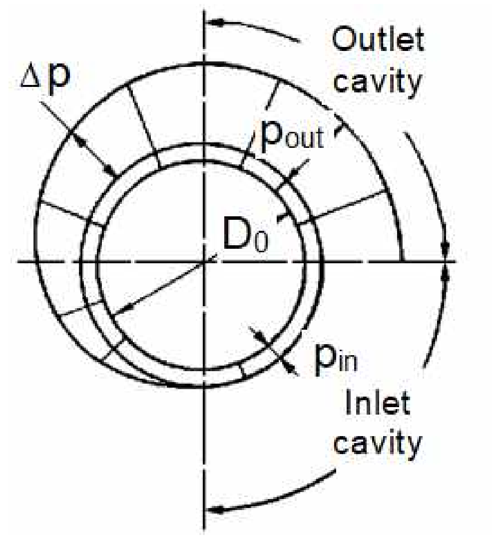


Figure 12 – Pressure distribution in the radial clearance (the inlet and discharge cavities angular length is 45 °)

Therefore, the extreme case is when each cavity occupies 1/2 of circle. The value of a in the extreme case is equal to unity (if 1/4 then a = 0.818; if 1/6 then a = 0.71; if 1/8 then a = 0.65).

The force P_L is perpendicular to the line that connects the centers of gears.

If we know the power consumed by the pump and its sizes we can find the reaction P_M. The reaction P_M depends on the torque, gear diameter, position of a meshing point (or gear rotation angle). The gear angle and friction of sliding teeth determine the direction of the reaction. The nominal reaction is accepted for the analysis:

$$P_M = \frac{2M}{D_{in}\cos\alpha}, \; M = \frac{30N}{\pi n}, \; \alpha = 20^\circ.$$

After summing up P_L and P_M, we can conclude that P_Σ is less for the driving gear than for the driven one, hence the driving gear bearing is less loaded than driven gear bearing.

The total force acting on the bearing of the normal size is usually 0.9 P_L – for driving gear and 1.1 P_L – for driven gear, i. e. approximately 20 % more.

It is obvious that the bearing of the driven gear must be more powerful to keep the bearings equally loaded. In case of sliding bearings, to keep the bearings equally loaded, the bearing of the driven gear must be of greater length or diameter. Both gear bearings are often similar for the sake of manufacturability.

As the lateral pressure applied to gear determines forces acting on the bearing, so it is necessary to reduce the length of the exhaust chamber, the gear diameter and the width when designing a high-pressure pump. However, you must always keep in mind the gear pump with short gears has an extremely low volumetric efficiency.

Taking into account the pressure forces between the meshed gears and the pressure pulsations, the evaluated value of force for the *drive gear* in predesign:

$$P_{d_1} = P_L.$$

For the *driven gear*

$$P_{d_2} = P_L + P_M \text{ or } P_{d_2} = 1.2 P_L.$$

The ***sliding bearings of low-pressure pumps*** are designed using the specific load acting on the bearings:

$$K = \frac{P_d}{d_b \; L_b},$$

where d_b is the diameter of a bearing journal;

L_b is the length of a bearing support surface.

The acceptable specific load for tin-and-leaden bronze bearings for oil pumps must be less than 8 – 10 MPa, for fuel pumps – less than 2 – 2.5 MPa. At the same time the circumferential velocity of the journal must not exceed 5 m/s. If the specific load is out of the specified range then the designer must increase the bearing size or apply hydraulic unloading of the supports, or to use the rolling bearings.

The hydraulic unloading is a process when the pressure is delivered to the chamber located diametrically opposite to the outlet cavity. Choosing dimensions of the channel and the chamber it is possible to unload bearings up to the acceptable limits (Figure 13). But in this case volumetric efficiency is less.

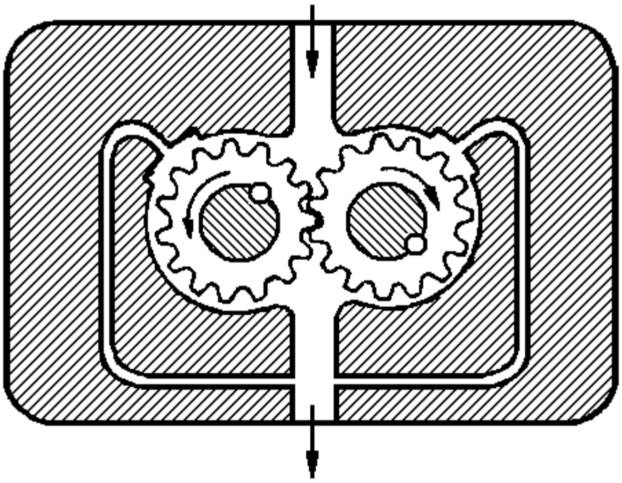


Figure 13 – Hydraulic unloading by the fluid pressure

The rolling bearings and especially needle bearings are widespread in aircraft pumps. It is a very hard task to produce many needles of the similar size. That is why the needles must be selectively assembled to meet the following requirements:

– the difference of the diametrical size must not exceed 0.002 mm;

– the total circumferential clearance between needles in the assembled bearing must be within the range 0.2 to 0.4 mm;

– the total axial clearance between needles in the assembled bearing must be within the range 0.2 to 0.4 mm.

The shafts must have the rigidity enough to provide the sag less than the radial clearance between gears and the gear chamber. If the rigidity is not enough then the normal gearing condition is violated resulting in the case damages.

When calculating a driving shaft for torsion it is necessary to remember that the pressure fluctuations result in a significant bursts of the acting load. Therefore, drives of gears should be 20 – 25 % more durable than the analytically determined average torsion torque.

STRUCTURE AND DESIGN

The main structural parts of gear pump are casing, pumping unit, drive shaft, and sealing elements. Simplified structural scheme of gear pump is shown at Figure 14.

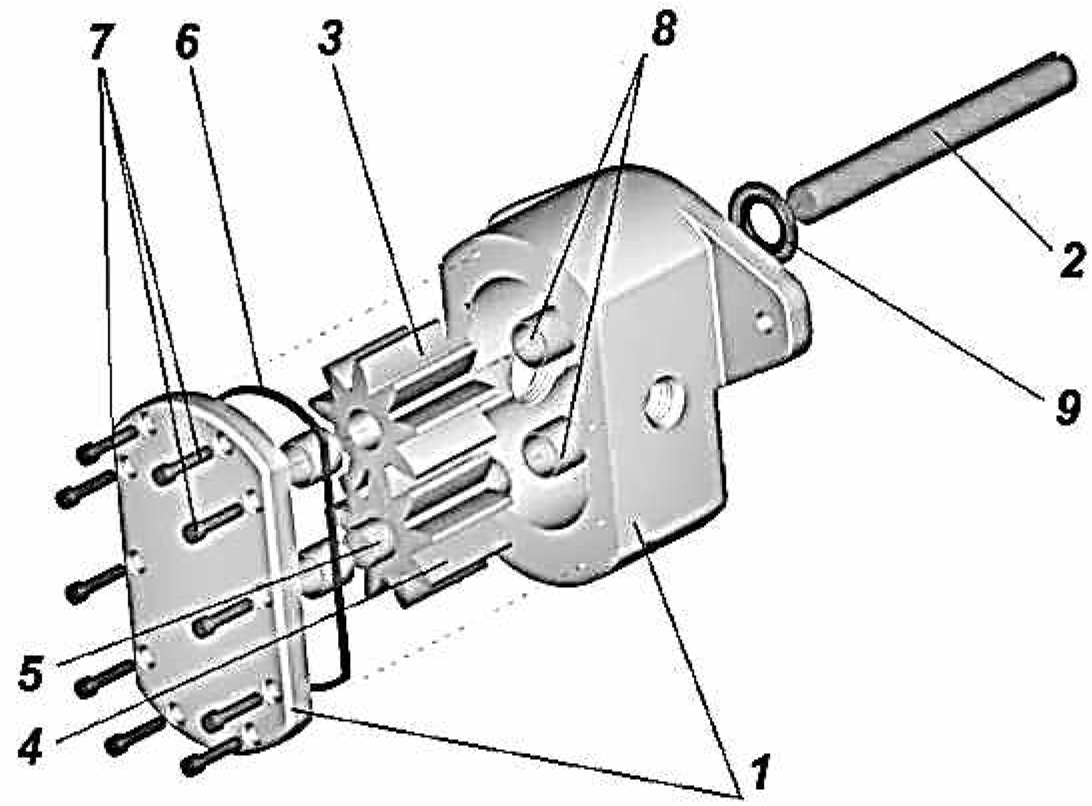


Figure 14 – Main structural part of gear pump:
1 – pump casing; 2 – drive shaft; 3 – drive gear; 4 – driven gear; 5 – axle;
6 – sealing gasket; 7 – connecting bolts; 8 – bearings; 9 – sealing element

Pump casing is intended to locate all pump structural parts, inlet and outlet cavities etc. It usually consists of few parts that are constructed from magnesium, aluminum alloys or steel. Contact surfaces are processed with significant accuracy. Casing parts are centered by pins and connected by studs or bolts. Sealing gaskets are located between casing parts.

External surfaces of casing parts are usually equipped with ribs that increase structure stiffness and convective heat transfer between pump inside space and environment. It gives possibility to reduce pump parts temperature by cooling.

Drive shaft is used for rotation transmission to pumping unit.

Pumping unit presents separate pumping elements that provide supply of working liquid by conversion mechanical energy to energy of fluid. It is usually consists of two (three or four sometimes) gears in gearing which are located on shafts (in case drive gear) or axle. Shaft arrangement is provided by supports with bearings.

Gears are located in special borings of pump casing with minimal clearances. These borings are connected with inlet and outlet cavities. Gears can be performed as a single structure with axle or such separate parts which are connected by pin, keyed or ball joint. Keyed joints give possibility to transfer large torque. Pin joints are used in case narrow gears. Ball joints provide self-adjustment of gears.

Gear which is installed on drive shaft usually is called **drive gear**. Gears which are driven by drive gear are called **driven gears**. In some cases there are two driven gears in one pump (one of them can be *idler* gear).

Design of gear pumps and materials

As the gear ratio exceeds unity, some fluid can be blocked in the cavities between the meshed teeth causing the high fluid pressure. This causes the additional force acting on the bearings and fluid heating.

Fluid compression in tooth slots appears due to the clearanceless closure of one or several slots. Especially, when the tooth enters the cavity during rotation, it closely contacts with the opposite gear in two points c and d (Figure 15, a), some fluid is blocked in the considered volume. Gear turning makes the cavity diminish, thereby compressing the fluid inside.

Figure 15, a shows that the minimal closed volume corresponds to a position when the tooth is relatively symmetric to a center line.

The smaller tooth thickness eliminates the contact in point c and prevents fluid from being blocked in the meshed teeth cavity. Thinner teeth provide the clearance S everywhere along the normal line to a profile (Figure 15, b).

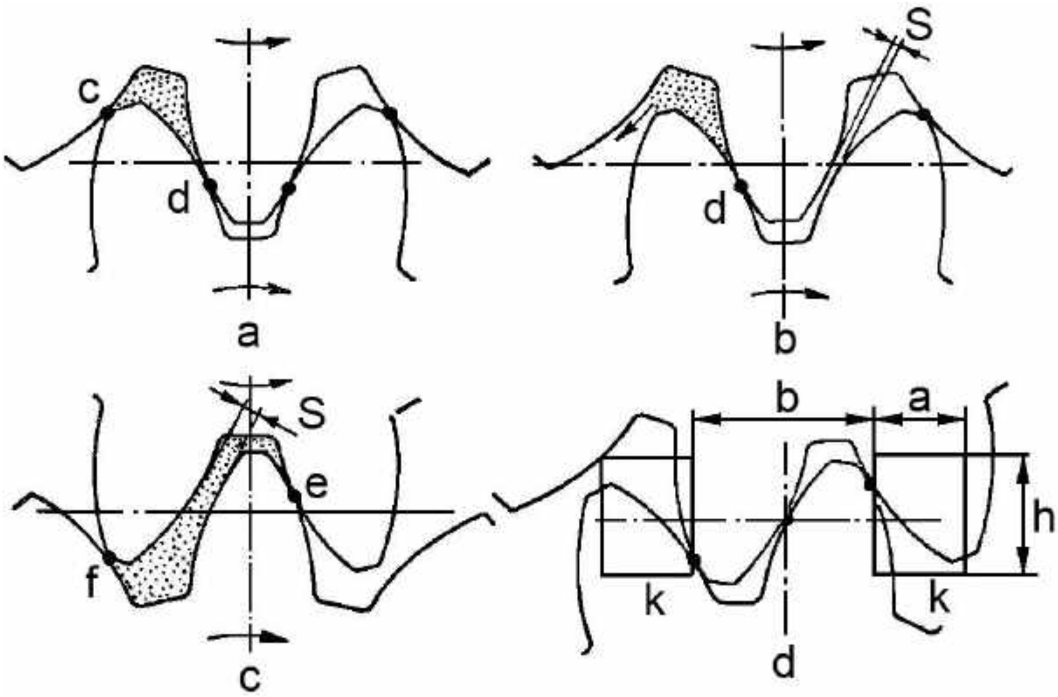


Figure 15 – The liquid blockage in the gear cavities

However, when the contact ratio exceeds unity, close contact of the second pair of gearing teeth generates the cavity with the blocked fluid. The cavity is greater, so the problem is not as urgent as in the previous case. The locked cavity is formed by the closed contact of two geared teeth pairs in points e and f (see Figure 15, c).

It consists of two cavities formed by the geared teeth of driving and driven gears and connected by the gap S. When the gear rotates as it is shown in the figure, the bottom part of this cavity decreases and the top one – increases. It is obvious, that if total volume of closed cavity does not change then the fluid stays uncompressed. Actually, the closed cavity varies achieving the minimal volume when the mechanical center of the closed area coincides with a centerline (see Figure 15, d).

Special drain grooves *k* are milled in face covers. They serve to escape the blocked fluid from the cavities. The examples of the unloading grooves in gear cavities are in Figure 16.

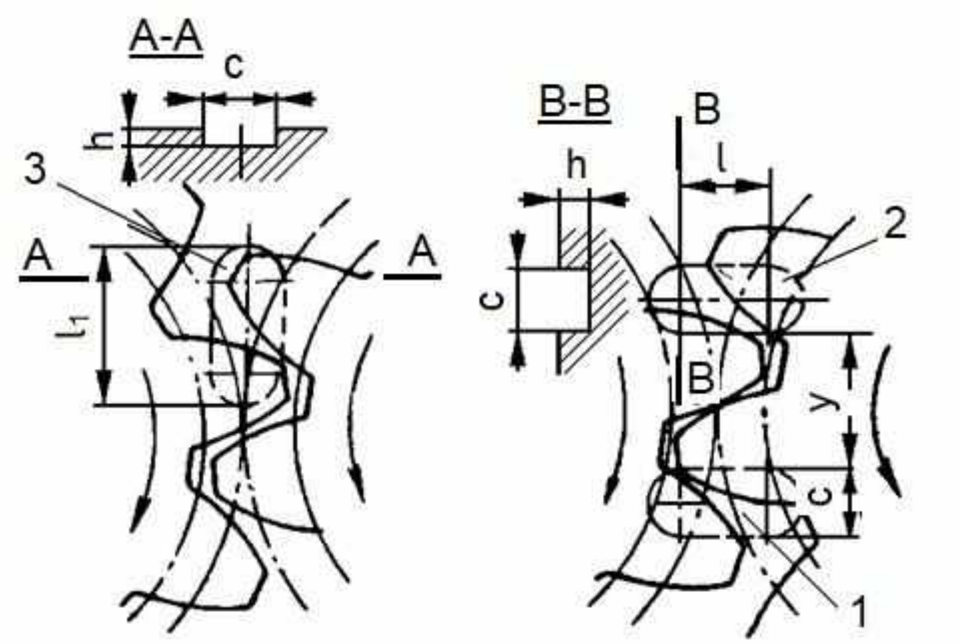


Figure 16 – The drainage of the blocked fluid through the sewer grooves :
1 – inlet chamber; 2 and 3 – exhaust chamber;
s – is the distance between gear axles

The positions of the drain grooves must be chosen to meet the following requirements:

- when the cavity diminishes, it must be interconnected to the outlet zone;
- when the cavity grows, it must be interconnected to the suction zone (this prevents the cavitation);
- the grooves must be distant enough to prevent the contact between the sewer grooves through the tooth cavity.

All mentioned requirements are met when the partition value b between the grooves k is equal or close to the circular pitch, i. e.

$$b = \frac{\pi D_{in}}{z} = \pi m.$$

The grooves for the fluid blockage prevention are arranged at the exhaust part of the pump. The similar grooves are made at the inlet side to prevent the discharge from a cavity volume.

In case of identical gears, the distance between gear axes is equal to a pitch diameter. The characteristic dimensions from the Figure 16 are determined by the semi-empirical relations:

$$h \approx 0.5m; \quad c \approx (1-1.2)m; \quad l_1 = (1.5-2)m;$$

$$l = 2.95m\sqrt{1-0.88\frac{m^2z^2}{s^2}}; \quad y = 2.78\frac{m^2z}{s}.$$

Figure 17 shows less effective and rarely used draining ways of gear cavities.

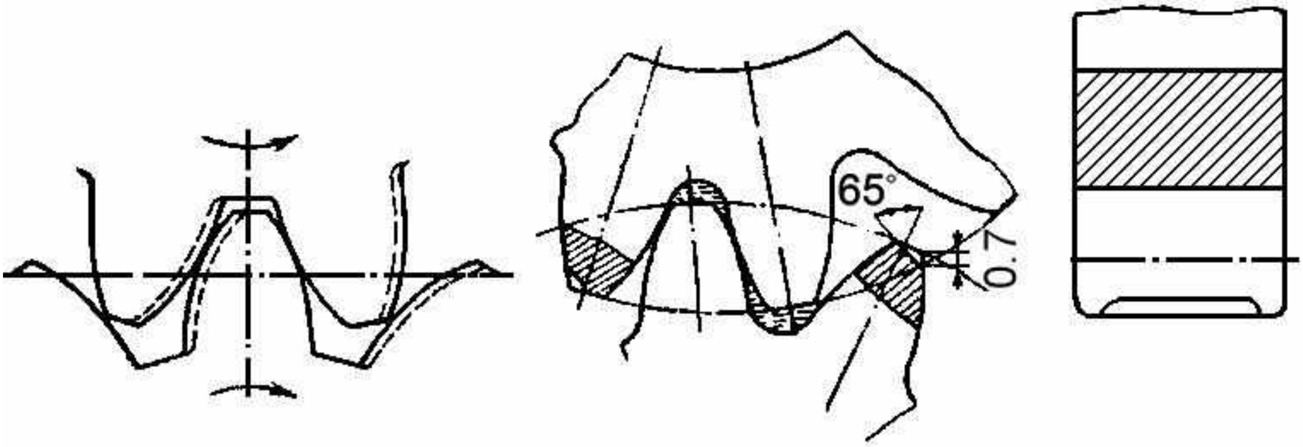


Figure 17 – The closed cavity is drained through grooves at the tip of the teeth (to the right) and grooves along the teeth (to the left) on the dwell track of teeth

Gear and shaft can be manufactured separately or as a single piece. In the first case the design and manufacturing becomes simpler. In the second case the treatment of the mounting surfaces and gear faces (both gears for one pass) becomes simpler.

The gear mounting on the shafts can be carried out using key, pins and balls. Key transfers the large torque. The pins fit the narrow gears the best, and the balls provide a gear self-adjustment.

Gears are manufactured of the alloyed steel (12ХН3А, 18ХНВА, 38ХМШ).

The teeth and faces of gears are polished. Faces are sometimes grounded in. Sometimes gears are made of bronze and austenitic steels (i. e. from materials with increased linear expansion factor) in order to prevent the large variation of clearances generally in case of manufacturing gear chamber from light-metal alloys. The shafts are made of the alloyed cemented and nitrated steels. The axles are made of alloyed steels and also of a cast-iron, a bronze or a duralumin.

Gear chambers are usually made of light-metal alloys - aluminum or magnesium (Мл-2, Мл-5, Ал-5, Ал-9). Sometimes the cast iron is used to reduce the clearances between the heated parts.

Shafts are usually manufactured from carbonized or nitrated alloy steels. **Bearing** type depends on working pressure value. Low-pressure pumps are usually equipped with sliding (lain) bearings. Their liners are usually designed from bronze. Middle- and high-pressure pumps are usually equipped with rolling bearings. In some cases, high-pressure pumps use needle bearings without internal rings. Needles are located in special slots on shaft surface. External ring is pressed in casing.

Sealing elements provide inside cavity seal on the drive shaft side and reduce internal leakage of working liquid. It is usually sealed by lip seal which are produced from resistant to aggressive substances (such as fuel or oil) rubber. Working liquid that seeps through lip seal is discharged into drainage system.

End seal presents a movable endplate, which provides minimal clearance between gear face and casing surface. Movable endplate is pressed down to end gear by springs. Springs provide a primary contact. Additional pressing is ensured by working liquid, which bring to endplate by special ducts. Pressure value should provide sealing but not cause significant friction between pump parts.

Axial face seal. ***Movable endplates*** - bushings (Figures 18 and 19, parts 10 and 17) of the axial seal ensure that the end clearances between the gears and the casing are practically zero throughout the entire service life of the pump.

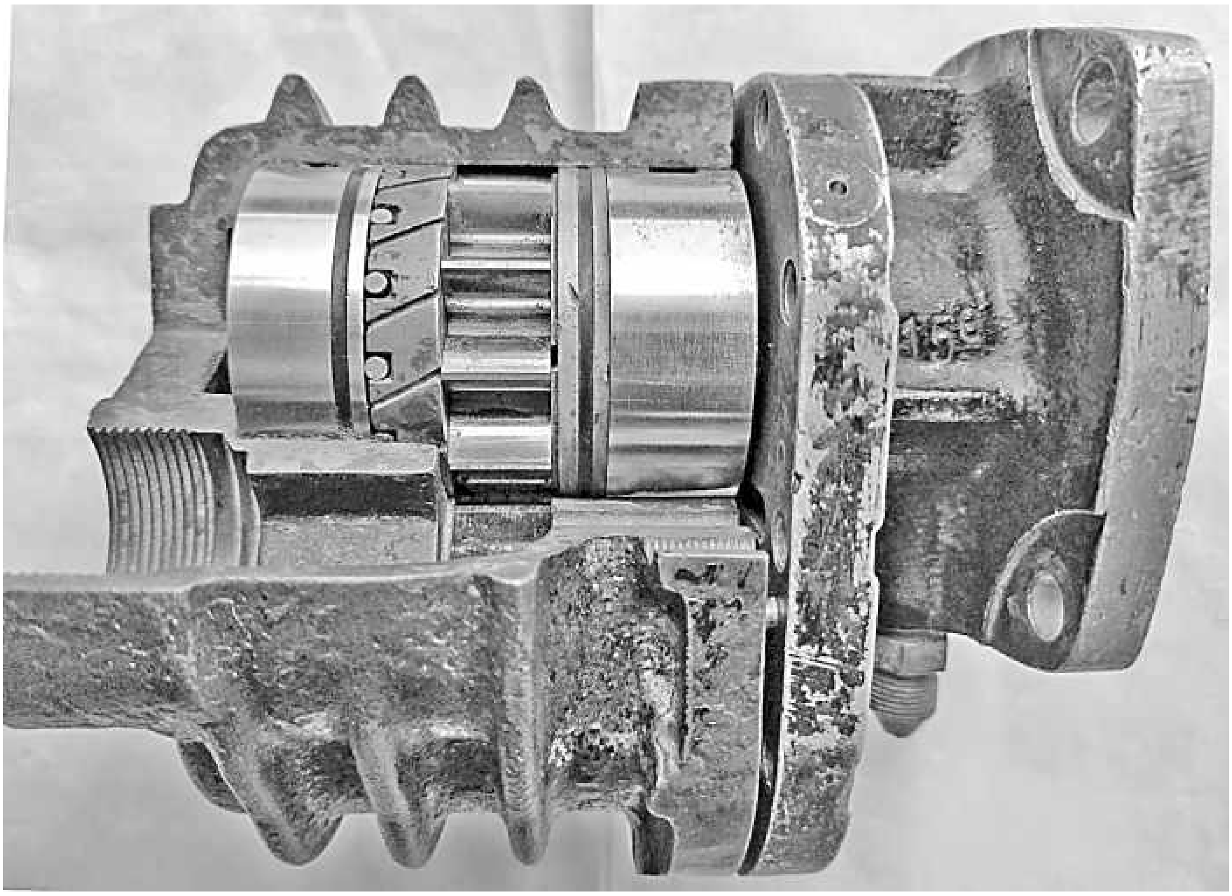
Movable endplates 10 and 17 have the ability to move along the bore of the pump casing under the influence of springs 16 and hydraulic forces arising from fluid pressure on the end area of the endplates during operation.

Endplates are always pressed against the end face of the gears, while eliminating the clearances between the contacting end surfaces of the gears, endplates and front bearing racers. Movable endplates on the cylindrical surfaces have a number of grooves, with the help of which the gears teeth spaces connect with the cavity formed by the end planes of the endplates and rear bearing racers.

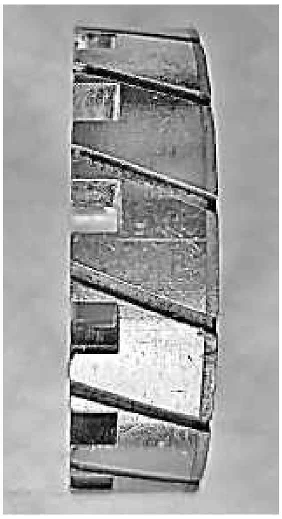
In order for the liquid pressure in this area to be balanced with the variable pressure in the cavities between the teeth, the cavity between the ends of the endplates and the rear racers is divided into several compartments. Adjacent compartments are separated from each other by radially arranged brass cylinders 11, and along the flats of the endplates - by a block (pin).

Sealing cylinders and block are placed in rectangular grooves of movable endplates, in the same grooves there are holes where springs 16 are installed. The use of a cast-iron housing allows maintaining the constancy of the hydraulic parameters of the pump at various ambient and working liquid temperatures.

Figure 18, a shows a photo of a gear pump with movable endplates, and Figures 18, b and 18, c – a photo of a movable endplate.



a



b

c

Figure 18 – Axial face seal (movable endplates)

CALCULATIONS OF GEAR PUMPS

Pumps calculations are based on their appointment and conditions of work. The calculations of the main fuel pump are slightly different from the calculations of the pumps of the oil systems. High-pressure fuel pumps operate in more severe conditions than oil system pumps. Therefore, we focus on fuel pumps.

Initial data for gear pump calculations

1. Determine the required flow rate of the main fuel pump (high-pressure pump, HPP), based on the specified volumetric fuel flow rate in the engine. For turbojet (turbofan) engines

$$Q_{en} = \frac{C_{sp}\ \mathrm{P}}{3600\ \rho_f},$$

where P – thrust, N;

C_{sp} – specific mass flow rate, kg/(N·h);

ρ_f – fuel density, kg/m^3;

for turboprop or turbo-shaft ones

$$Q_{en} = \frac{C_{spN}\ \mathrm{N}}{3600\ \rho_f},$$

where N – power, kW;

2. Determine the volumetric capacity of the pump, taking into account an adequate supply of liquid flow

$$Q_p = \mathrm{Q_{en}}\ \mathrm{K_1},$$

where $K_1 = 1.3 - 2$ – safety factor of liquid flow.

3. Sets the gear pump typical values of volumetric and mechanical efficiency

$$\eta_v = 0.75 - 0.85;$$

$$\eta_m = 0.97 - 0.99.$$

Sets the gear pump total efficiency value

$$\eta_p = \eta_v \eta_m .$$

4. Determine saturated vapor pressure of fuel, depending on its temperature at the inlet of the pump and the grade of kerosene according to Table A. To convert to Pa to the tabulated value multiplied by 133.

Table A – Physical properties of fuels

Fuel	T-1	TC-1	T2	T5
Density, kg/m^3	800	775	775	845
Pressure $p_{t\,4/1}$, mm Hg	$12.1\left(\dfrac{T}{221}\right)^{2.08}$	$12.4\left(\dfrac{T}{221}\right)^{3.46}$	$12.9\left(\dfrac{T}{221}\right)^{4.13}$	$14\left(\dfrac{T}{221}\right)^{5.67}$
Viscosity ν, mm^2/s	$\dfrac{85.6}{\left(T/221\right)^{11.8}}$	$\dfrac{14.9}{\left(T/221\right)^{8.5}}$	$\dfrac{7.77}{\left(T/221\right)^{6.7}}$	$\dfrac{5.58}{\left(T/221\right)^{6.13}}$

Set the pressure at the pump inlet from the condition

$$p_{in} \geq p_t\left(t\right) + \Delta p_{c.\min} + \frac{\rho}{8}\left(\mathrm{D}_\mathrm{o}^2 - \mathrm{D}_\mathrm{d}^2\right)\omega^2 ,$$

where $\Delta p_{c.\min}$ = 0.15 – 0.3 MPa – net positive suction head (cavitation margin).

5. Drive power is determined from the outlet pressure of engine - prototype and the pump capacity

$$N_p = \frac{\Delta p_p Q_p}{\eta_p} ,$$

where the pressure drop at the pump $\Delta p_p = p_p - p_{in}$.

Pressure at the outlet of the pump can be evaluated as

$$\mathrm{p_p} = K_2\left(\pi_c p_H + \Delta p_n\right),$$

where Δp_n= 3 – 6 MPa – pressure difference at the fuel spray nozzle;

π_c – compressor pressure ratio;

K_2= 1.5 – 2 – safety factor.

Approximate values $\mathrm{p_p}$ = 8 – 15 MPa.

6. According to the known desired pump capacity and volumetric efficiency of the pump determine the theoretical performance is necessary for determining the design parameters

$$Q_{id} = \frac{Q_p}{\eta_v}.$$

For gear pumps, the average volumetric efficiency is 0.75 – 0.85.

The choice of pump parameters

The choice of the basic parameters of a gear pump can be made as follows:

1. Sets the value of the coefficient of proportionality c, the limit is usually not exceed ten modules; this limit is defined by technological considerations. As a general rule $c = 4 - 10$.
2. Specifies the number of gear teeth z within 7 – 12.
3. Specifies the rotational speed within $n =$ 5500 – 6000 rpm.

Calculation of gear parameters

4. Hence we find the fractional value of the required module gear pump designed according to the raw data:

$$m = \sqrt[3]{\frac{30\,Q_{id}}{\pi \mathrm{Kzcn}}}, \text{ m.}$$

5. Rounding the resulting value of the module to the nearest standard value, we find the pitch diameter and the width of the gears. Depending on the initial data determine a rotational speed of the designed pump $n = \frac{u_{in}}{\pi D_{in}}$ or circumferential velocity u_{in}.
6. Check the performance of the pump according to the equation

$$Q_{id} = \frac{\pi K \mathrm{D_{in} mbn}}{30},$$

substituting the found and selected values K, m, z, b, n.

Since the resulting capacity is not the same as given, it is necessary to adjust the parameters of the pump to obtain the desired value Q_{id}. The easiest

way to adjust to meet due to change in the width of the gears in the acceptable range, or a change in speed, if permitted by the specifications.
For example, from the last equation:

$$\mathrm{b}' = \frac{30\,Q_{id}}{\pi K \mathrm{D}_{\mathrm{in}} \mathrm{mn}}.$$

Check condition $\mathrm{b}'/\mathrm{m} = 4...10$.

7. Define the outer diameter of gear (addendum circle)

$$D_o = mz + 2m$$

and the diameter of the slots (dendum circle)

$$D_d = mz - 2.5m.$$

8. Determine the pressure drop along a slot height

$$\Delta p_{sl} = \frac{\rho}{8}\left(\mathrm{D}_\mathrm{o}^2\text{-}\mathrm{D}_\mathrm{d}^2\right)\left(\frac{\pi n}{30}\right)^2$$

and check on the condition of the absence of cavitation

$$p_{in} > \Delta p_{sl} + p_{t\,4/1}(t) + \Delta p_{c.\min}.$$

If the last inequality is not met, either increase the inlet pressure p_{in} or decrease the rotation speed according to the condition

$$\omega_{\max} \le \sqrt{\frac{8}{\rho}\frac{\Delta p_{sl}}{D_o^2 - D_d^2}}.$$

CHOICE OF BEARINGS

Determining the forces acting on the support

Forces that act on the pump bearings are (see Figure 11):

– side (lateral) force acting on a gear P_L (the force appears due to the pressure drop between the outlet and inlet pressure);

– reaction P_M (the reaction appears because of a torsion torque applied to the gear);

– pressure forces between the meshed gears (the force appears due to the fluid compression) etc.

If we assume that the pressure in a radial clearance changes linearly (see Figure 12) (that is fair only when $z \to \infty$) and the length of the inlet and the exhaust cavities is the same, then

$$P_L = a \Delta p D_o b \,,$$

where a is a proportionality factor, which depends on the length of the inlet and the exhaust cavities. So, the extreme case is when each cavity occupies 1/2 of circle. The value of a in the extreme case is equal to unity (if 1/4 then a = 0.818; if 1/6 then a = 0.71; if 1/8 then a = 0.65).

The force P_L is perpendicular to the line that connects the centers of gears.

If we know the power consumed by the pump and its sizes we can find the reaction P_M. The reaction P_M depends on the torque, gear diameter, position of a meshing point (or gear rotation angle). The gear angle and friction of sliding teeth determine the direction of the reaction. The nominal reaction is accepted for the analysis.

$$P_M = \frac{2M}{D_{in} \cos\alpha}, \; M = \frac{30N}{\pi n}, \; \alpha = 20^\circ .$$

After summing up P_L and P_M, we can conclude that P_Σ is less for the driving gear than for the driven one, hence the driving gear bearing is less loaded than driven gear bearing.

The total force acting on the bearing of the normal size is usually 0.9 P_L – for driving gear and 1.1 P_L – for driven gear, i. e. approximately 20 % more.

It is obvious that the bearing of the driven gear must be more powerful to keep the bearings equally loaded. In case of sliding bearings, to keep the bearings equally loaded, the bearing of the driven gear must be of greater length or diameter. Both gear bearings are often similar for the sake of manufacturability.

As the gear side pressure basically determines forces acting on the bearing, so it is necessary to reduce the length of the exhaust chamber, the gear diameter and the width when designing a high-pressure pump. However, you must always keep in mind the gearbox with short gears has an extremely low volumetric efficiency.

Taking into account the pressure forces between the meshed gears and the pressure pulsations, the evaluated value of force for the driving gear in predesign:

$$P_{d_1} = P_L .$$

For the driven gear

$$P_{d_2} = P_L + P_M \text{ or } P_{d_2} = 1.2 P_L .$$

Selection and calculation of sliding bearings

Sliding bearings are widely used in ***oil pumps***.

Compressive load on the *bearing surface of sliding bearings* is calculated by the formula

$$K_L = \frac{P_\Sigma}{d_b \left(l_1 + l_2\right)}, \text{ MPa,}$$

where d_s – the diameter of support pins;

l_1 and l_2 – the length of pins.

Typically, for indirect estimation of wear resistance of sliding bearings using the following parameters.

– Specific support load (unit pressure) $\mathrm{K_L}$, MPa

– pins tip speed u_l, m / s;

– product $K_L \times u_l$, MPa, m / s.

Valid values for the above parameters for sliding bearings made of bronze pumps are shown in Table B.

If the load exceeds a specified amount, you must either increase the size of the bearings or to apply hydraulic unloading supports, or use the bearings.

Table B - Values of permissible modes of bearings semi fluid friction

Bearings material	K_L, MPa	$[u_l]$, m/s	$[K_L \times u_l]$
Бр С30	25	12	30
Бр О10Ф1	15	10	15
Бр А9Ж4Л	15	4	12

Calculation of bearings on dynamic load capacity

In the aircraft ***fuel pumps*** are widely used frictionless bearings and particularly *roller and needle bearings.*

Selection and calculation of bearings carried by the method described in the Bibliography.

As a radial loading P_r the value of force $P_{d_2}/2 = \left(P_L + P_M\right)/2$ must be selected.

Main dimensions:

B – width of the clips (selected from the loading conditions);

d – the diameter of the holder;

D – diameter of the outer cage;

r – radius of curvature.

The type of bearings' capacity calculation depends on a ring rotational speed. Bearings are selected on static load capacity if external load is motionless or rotates slow ($n \le 1$ rpm). At ring rotational speed $n > 10$ rpm, bearings are calculated on dynamic load capacity. The bearings working at ring rotational speed $n > 10$ rpm and under sharply variable load, also it is necessary to check on static load capacity.

Bearings type and the positioning scheme are selected preliminary for a both shaft supports. Bearings of one type and one size according to the load capacity of the most loaded support are usually used for general-purpose gearboxes. If it is impossible to decide, what support is more loaded, both supports are calculated in parallel up to the values of equivalent load and then more loaded support is selected.

Calculation is based on the known equation of fatigue curve

$$F^p L = C^P, \tag{1}$$

where F is the equivalent load, N; L is the service life in millions revolutions;

C is the required dynamic load capacity, N; p is the exponent: for ball bearings p = 3, for roller bearings p =10/3.

It is possible to define dynamic load capacity from equation (1) if F and L are known:

$$C = \left(\frac{L}{a_1 a_{23}} \right)^{\frac{1}{p}} F, \qquad (2)$$

and then select a bearing which satisfies condition

$$C \le C_c, \qquad (3)$$

where C_c is the bearing dynamic load capacity taken from the catalogue or calculated on empirical dependences.

Service life in millions revolutions L and in hours L_h are connected by equation

$$L = \frac{60 n L_h}{10^6}, \qquad (4)$$

where n is the rotating ring rotational speed, rpm.

In formula (2) a_1 is the factor considering of reliability (Table C).

Table C

Reliability	0.90	0.95	0.96	0.97	0.98	0.99
a_1	1	0.62	0.53	0.44	0.33	0.21

The factor a_{23} considers bearings' material, lubricant and operation conditions:

1) usual conditions;
2) without bearing axes shift and with an lubricant film in contacts;
3) the same, as in 2), but at manufacturing of rings and rolling elements from electroslag or vacuum steel.

Values of factor a_{23} are shown in Table D.

Table D

Bearing type	Conditions		
	1	2	3
Ball bearings (except spherical)	0.7 – 0.8	1.0	1.2 – 1.4
Cylindrical roller bearings and spherical ball one	0.5 – 0.6	0.8	1.0 – 1.2
Tapered roller bearings	0.6 – 0.7	0.9	1.1 – 1.3
Spherical roller bearings	0.3 – 0.4	0.6	0.8 – 1.0

Equivalent load F for the radial type of bearings can be defined under the following formula:

$$F = P_r K_b K_t \,,$$

where P_r is the radial load on a bearing, N; K_b is the safety factor considering load non-uniformity (in case of light load K_b = 1, with medium non-uniformity K_b = 1,3 – 1,8, with heavy load K_b = 2 – 3); K_T is the temperature factor (K_T = 1 at t <125 °C and K_T = 1,05; 1,1; 1,25; 1,4 accordingly at t = 125, 150, 200 and 250 °C).

If condition (3) is not satisfied or the service life is too small, it is necessary:

- select bearings of heavier series or other bearing type;
- or increase a shaft diameter;
- or use two identical radial bearings in one support. In such case it may be considered as one two-row bearing and support dynamic radial load capacity for roller bearings should be assumed as $C_\Sigma = 1.714\,C$; equivalent load is defined as for two-rows bearings.

Some bearing data are given in the Appendix. For more complete information on bearings (see Bibliography).

DESIGN EXAMPLES OF GEAR FUEL PUMPS

Figure 19 shows a drawing of a medium pressure fuel gear pump **348-И**. This pump is used in turboprop engine

The gear fuel pump consists of the following main parts and assemblies: a housing, a support flange of two gears, roller bearings, a drive shaft sealing assembly, movable endplates and a drive shaft.

In the case 29 of special cast iron of the CHM1 brand, in two adjacent cylindrical bores there are two pump gears 9 and 18, bronze fixed 8 and 19 and movable 10 and 17 endplates, two front 7 and 20, and two rear 14 and 27 roller bearing races, rollers 26 and cages 15.

The oval-shaped housing with transverse stiffeners has a flange for attaching the support flange 2, cast from AL-3 aluminum alloy. On the sides on the housing, there are two cast lugs in the form of branch pipes, the holes of which are directed towards the bottom of the housing. The branch pipes are designed for the inlet and outlet of the working liquid and are connected by linear channels with the working cavity of the pump. The nipples are screwed on the threads of the fuel inlet 21 and 22 outlet, to which the pipelines are connected. For sealing between the fittings and the housing, soft aluminum sealing washers are inserted.

To reduce the flow of fuel along the generating surfaces of the bores and parts of the pumping unit, rectangular grooves are machined on the fixed endplates 8 and 19 and the rear cages of the roller bearings, into which the rubber sealing rings 12 are installed. The fuel leaked along the end surfaces of the pumping unit enters through the housing channels to the inlet cavity.

Movable endplates 10 and 17 have the ability to move along the bore of the pump casing under the influence of springs 16 and hydraulic forces arising from fluid pressure on the end area of the endplates during operation.

Endplates are always pressed against the end face of the gears, while eliminating the clearances between the contacting end surfaces of the gears, endplates and front bearing racers. Movable endplates on the cylindrical surfaces have a number of grooves, with the help of which the gears teeth spaces connect with the cavity formed by the end planes of the endplates and rear bearing racers.

In order for the liquid pressure in this area to be balanced with the variable pressure in the cavities between the teeth, the cavity between the ends of the endplates and the rear racers is divided into several compartments. Adjacent compartments are separated from each other by radially arranged brass cylinders 11, and along the flats of the endplates - by a block (pin).

Sealing cylinders and block are placed in rectangular grooves of movable endplates, in the same grooves there are holes where springs 16 are installed. The use of a cast-iron housing allows maintaining the constancy of the hydraulic parameters of the pump at various ambient and working liquid temperatures.

Cemented steel gears of the pumping unit are manufactured in one piece with trunnions (journals), which are supported by single-row roller bearings. The drive gear 9 has a central through hole with involute splines that mate with the drive shaft 1; the other end of the shaft is mated to the engine drive. From axial movement, the spring is fixed with washers 3 and 23, and a split retaining ring 24 installed in the annular groove of the support flange 2. The shaft seal consists of two seal cups 4 with spring rings 25, pressing the cup to the cylindrical part of the shaft, and a drain ring 5, through which leaked fuel (from the pumping cavity) and oil (from the drive cavity) is discharged into the drainage system of the engine through the fitting 28. Washers 3 and 6 also serve as stops preventing the axial movement of the cups along the shaft 1. Ring 5 has supporting cones on both sides, on which, during installation, the cup cones sit, which protects the latter from turning out under pressure of fuel and oil leaked from the pump cavity and the drive cavity, respectively.

Supporting flange 2 is cast from AL-3 aluminum alloy and is attached to the pump casing with 14 pins; two control sleeves carry out mutual exact fixation of the casing and the support flange. For sealing, a lead foil gasket is laid along the joint.

When the gears rotate, the disengaging teeth release the volume of the cavities, which, together with the walls of the bore and the endplates, form a working filling chamber.

The liquid entering under low pressure fills the volume of this chamber and is then carried by the gears to the outlet side, where it is forced into the supply line by the meshing teeth.

Figure 20 shows a drawing of a **high-pressure gear fuel pump** (**unit 934)**. This pump is used in turbofan engine.

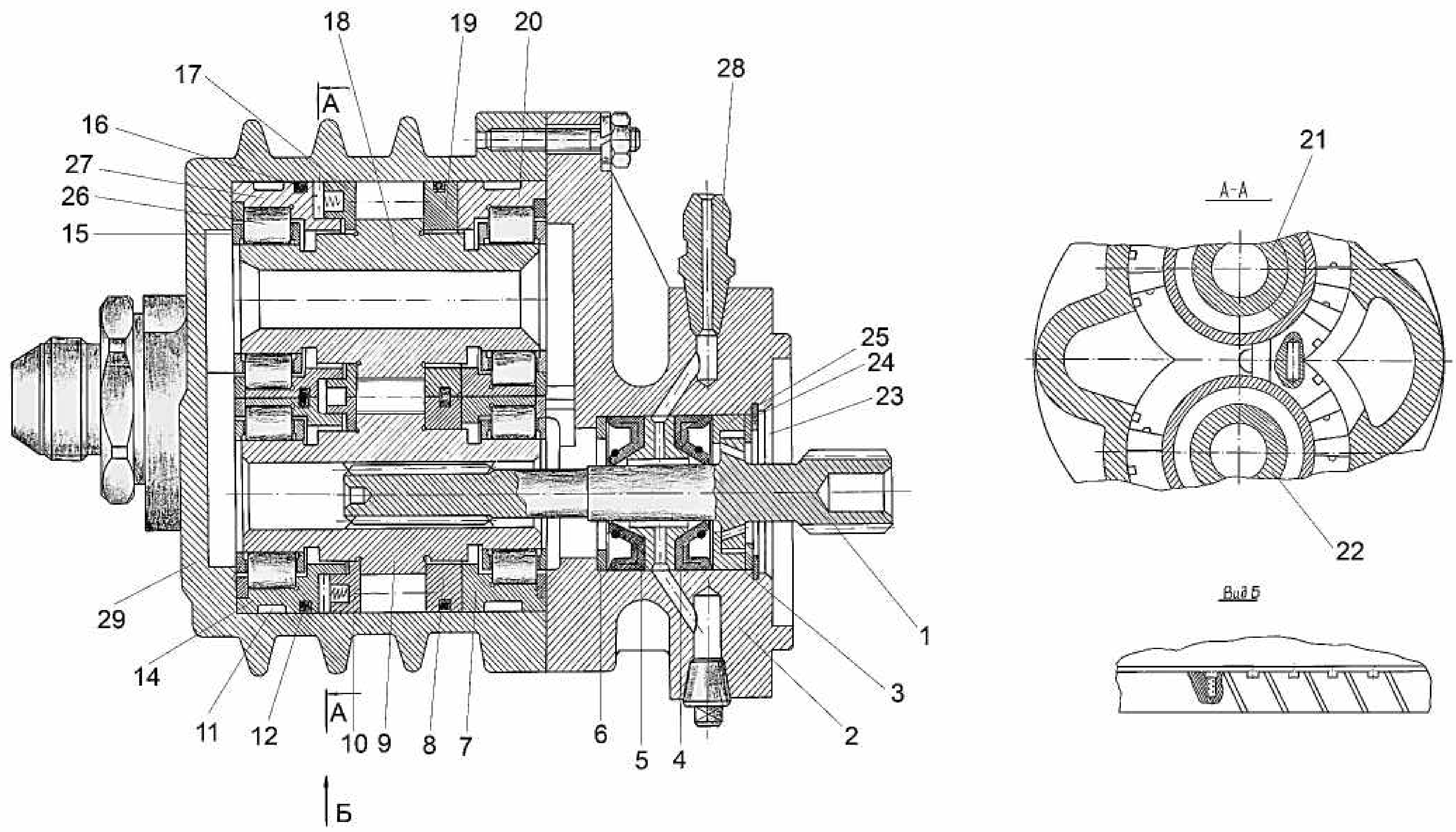


Figure 19 – Gear-type fuel pump of medium pressure 348-И

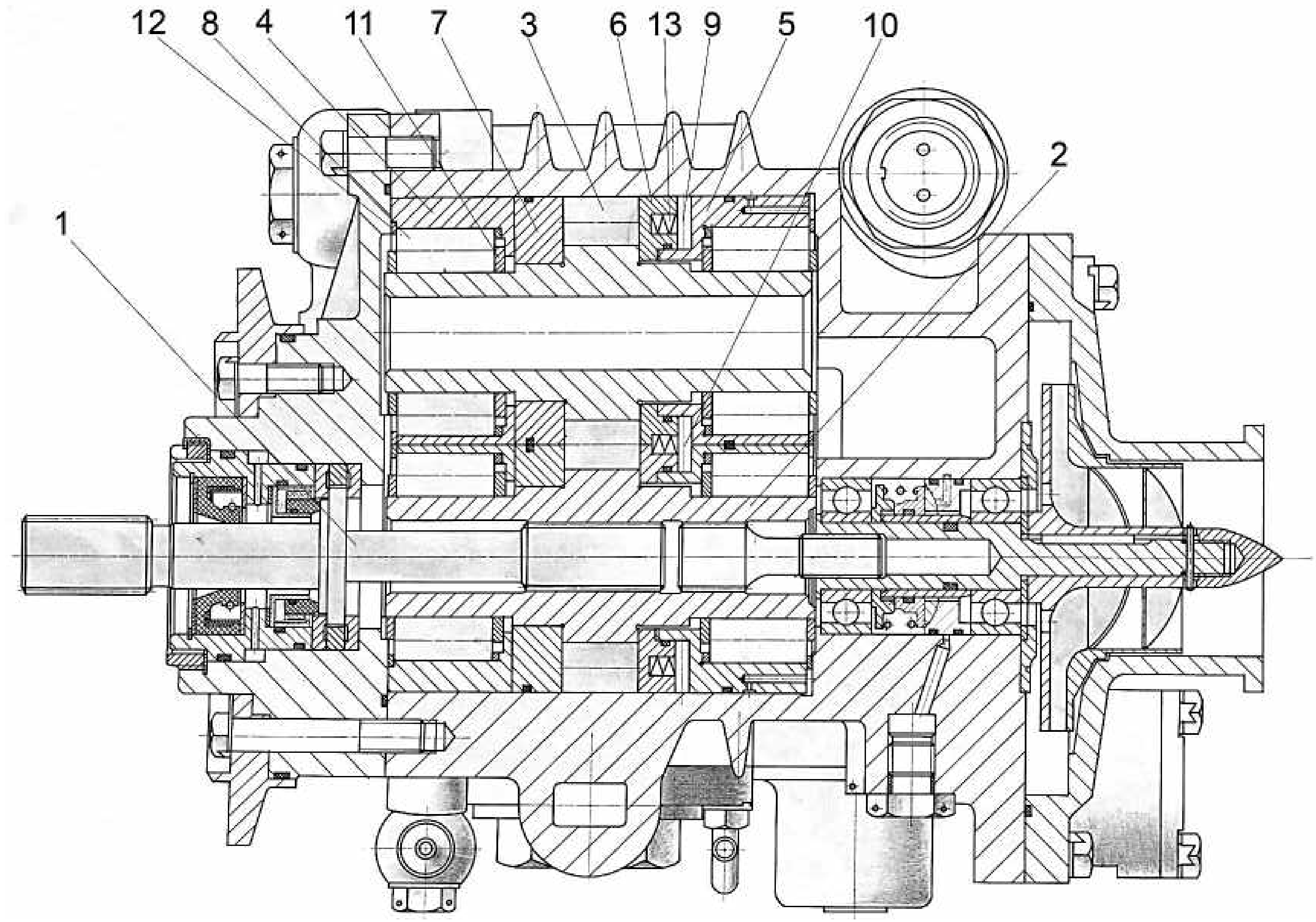
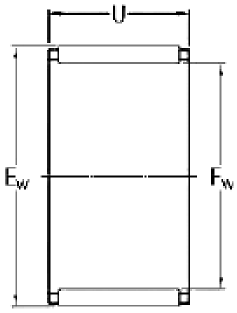


Figure 20 – High-pressure gear fuel pump (unit 934):
1 – drive shaft; 2 – drive gear; 3 – driven gear; 4, 5 – bearing outer races; 6 – movable endplate; 7 – fixed endplate; 8 – roller; 9 – radial pin; 10 – central pin; 11 – bearing cage; 12 – support ring; 13 – spring

APPENDIX

Table A.1 – Needle roller and cage assemblies F_w = 3 – 30 mm

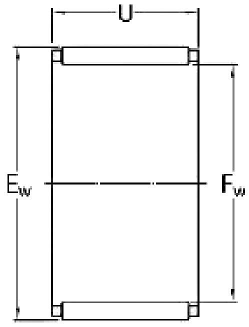


Principal dimensions			Basic load ratings dynamic	static	Fatigue load limit	Speed ratings Reference speed	Limiting speed	Mass	Designation
F_w	E_w	U	C	C_0	P_u				
mm			kN		kN	r/min		g	–
3	5	7	1,51	1,34	0,134	40 000	45 000	0,3	**K 3x5x7 TN**
	5	9	1,68	1,53	0,153	40 000	45 000	0,4	► **K 3x5x9 TN**
4	7	7	1,72	1,32	0,137	36 000	43 000	0,5	**K 4x7x7 TN**
	7	10	2,29	1,9	0,204	36 000	43 000	0,7	**K 4x7x10 TN**
5	8	8	2,29	2	0,212	36 000	40 000	0,7	**K 5x8x8 TN**
	8	10	2,92	2,7	0,29	36 000	40 000	0,9	► **K 5x8x10 TN**
6	9	8	2,55	2,36	0,25	34 000	38 000	0,8	**K 6x9x8 TN**
	9	10	3,3	3,2	0,345	34 000	38 000	1,1	**K 6x9x10 TN**
7	10	8	2,81	2,75	0,29	32 000	36 000	0,9	**K 7x10x8 TN**
	10	10	3,58	3,75	0,415	32 000	36 000	1	**K 7x10x10 TN**
8	11	10	3,8	4,25	0,465	32 000	36 000	1,2	**K 8x11x10 TN**
	11	13	5,01	5,85	0,67	32 000	36 000	1,7	**K 8x11x13 TN**
	12	10	4,84	4,75	0,54	30 000	34 000	2	**K 8x12x10 TN**
9	12	10	4,4	5,2	0,57	30 000	34 000	1,5	**K 9x12x10 TN**
10	13	10	4,57	5,7	0,63	28 000	32 000	1,6	► **K 10x13x10 TN**
	13	13	5,94	8	0,9	28 000	32 000	2,3	**K 10x13x13 TN**
	14	10	5,61	6,1	0,695	28 000	32 000	2,5	**K 10x14x10 TN**
	14	13	7,21	8,5	0,98	28 000	32 000	4,6	**K 10x14x13 TN**
	16	12	7,65	7,2	0,85	28 000	32 000	5,5	**K 10x16x12 TN**
12	15	10	4,73	6,2	0,695	26 000	30 000	2,9	**K 12x15x10 TN**
	15	13	6,16	8,65	0,98	26 000	30 000	2,3	**K 12x15x13 TN**
	16	13	7,65	9,5	1,1	26 000	30 000	3,6	**K 12x16x13 TN**
	17	13	9,13	10,4	1,22	26 000	30 000	4,9	**K 12x17x13 TN**
	18	12	9,52	10	1,18	26 000	30 000	6	**K 12x18x12 TN**
14	18	10	6,93	8,65	1	24 000	28 000	4	**K 14x18x10**
	18	13	7,92	10,2	1,18	24 000	28 000	6,5	**K 14x18x13**
	18	15	9,13	12,5	1,46	24 000	28 000	5	**K 14x18x15 TN**
	18	17	10,5	14,6	1,7	24 000	28 000	8	**K 14x18x17**
15	19	13	8,25	11,2	1,29	24 000	28 000	7	► **K 15x19x13**
	19	17	10,8	15,6	1,86	24 000	28 000	9,5	► **K 15x19x17**
	21	15	13,8	16,3	2	24 000	26 000	11	**K 15x21x15**
	21	21	18,7	24,5	3	24 000	26 000	17	**K 15x21x21**

Continuation of table A.1

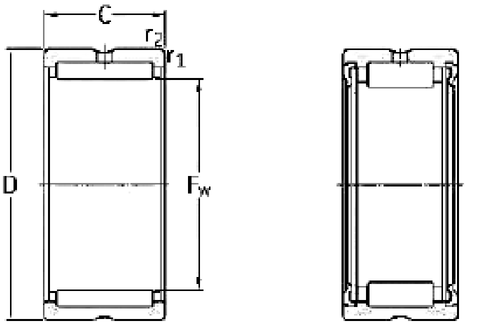
Principal dimensions			Basic load ratings dynamic	static	Fatigue load limit	Speed ratings Reference speed	Limiting speed	Mass	Designation
F_w	E_w	U	C	C_0	P_u				
mm			kN		kN	r/min		g	–
16	20	10	7,48	10	1,16	24 000	26 000	5,5	**K 16x20x10**
	20	13	8,58	12	1,37	24 000	26 000	7,5	**K 16x20x13**
	20	17	11,2	17	2	24 000	26 000	10	**K 16x20x17**
	22	12	11	12,5	1,5	22 000	26 000	10	**K 16x22x12**
	22	16	14,2	17,6	2,12	22 000	26 000	12	**K 16x22x16**
	22	20	17,6	22,8	2,8	22 000	26 000	17	**K 16x22x20**
	24	20	20,5	23,6	2,9	22 000	24 000	22	**K 16x24x20**
17	21	10	7,81	10,8	1,22	22 000	26 000	5,5	**K 17x21x10**
18	24	12	12,1	15	1,8	20 000	24 000	12	**K 18x24x12**
19	23	13	9,13	13,7	1,6	20 000	24 000	8	**K 19x23x13**
20	24	10	8,58	12,9	1,46	20 000	22 000	6,5	**K 20x24x10**
	24	13	9,52	14,6	1,66	20 000	22 000	9	**K 20x24x13**
	24	17	12,5	20,8	2,4	20 000	22 000	12	**K 20x24x17**
	26	17	18,3	26	3,2	19 000	22 000	16	**K 20x26x17**
	26	20	20,1	29	3,6	19 000	22 000	19	▸ **K 20x26x20**
	28	20	22,9	28,5	3,45	18 000	20 000	27	**K 20x28x20**
	28	25	29,2	39	4,9	18 000	20 000	32	▸ **K 20x28x25**
	30	30	34,1	41,5	5,2	17 000	20 000	49	**K 20x30x30**
21	25	13	9,68	15,3	1,76	19 000	22 000	9	**K 21x25x13**
22	26	10	8,8	13,7	1,56	18 000	20 000	7,5	▸ **K 22x26x10**
	26	13	10,1	16,3	1,86	18 000	20 000	9,5	**K 22x26x13**
	26	17	13,2	22,8	2,7	18 000	20 000	12	**K 22x26x17**
	28	17	18,3	27	3,25	17 000	20 000	18	**K 22x28x17**
	29	16	19,4	25,5	3,05	17 000	19 000	16	**K 22x29x16**
	30	15	19	23,6	2,8	17 000	19 000	18	**K 22x30x15 TN**
23	35	16	24,2	23,2	2,9	15 000	17 000	29	**K 23x35x16 TN**
24	28	10	9,35	15	1,73	17 000	19 000	8,5	**K 24x28x10**
	28	13	10,6	18	2,08	17 000	19 000	10	**K 24x28x13**
	30	17	18,7	27,5	3,4	16 000	18 000	19	**K 24x30x17**
25	29	10	9,52	15,6	1,8	16 000	18 000	8,5	**K 25x29x10**
	29	13	10,8	18,6	2,16	16 000	18 000	11	**K 25x29x13**
	30	17	17,9	30,5	3,6	16 000	18 000	16	**K 25x30x17**
	30	20	20,9	36,5	4,4	16 000	18 000	18	**K 25x30x20**
	32	16	19,8	27,5	3,35	15 000	17 000	21	**K 25x32x16**
	33	20	27,5	38	4,65	15 000	17 000	33	**K 25x33x20**
	35	30	44,6	62	7,8	15 000	17 000	65	▸ **K 25x35x30**
26	30	13	11,2	19,6	2,28	16 000	18 000	11	**K 26x30x13**
28	33	13	14,7	24,5	2,85	14 000	16 000	13	**K 28x33x13**
	33	17	19	33,5	4,05	14 000	16 000	17	**K 28x33x17**
30	35	13	15,1	25,5	3	13 000	15 000	14	**K 30x35x13**
	35	17	18,7	34	4,05	13 000	15 000	19	**K 30x35x17**
	35	27	29,2	60	7,35	13 000	15 000	30	**K 30x35x27**
	37	18	25,1	39	4,65	13 000	15 000	30	**K 30x37x18**
	40	30	46,8	69,5	8,65	12 000	14 000	73	**K 30x40x30**

Table A.2 – Needle roller and cage assemblies F_w = 32 – 100 mm



Principal dimensions			Basic load ratings dynamic	static	Fatigue load limit	Speed ratings Reference speed	Limiting speed	Mass	Designation
F_w	E_w	U	C	C_0	P_u				
mm			kN		kN	r/min		g	–
32	37	13	14,7	25,5	3	13 000	14 000	18	**K 32x37x13**
	37	17	19	35,5	4,25	13 000	14 000	19	**K 32x37x17**
	38	20	25,1	45	5,6	12 000	14 000	30	**K 32x38x20**
	40	25	35,8	58,5	7,2	12 000	14 000	49	**K 32x40x25**
35	40	13	15,4	28	3,25	12 000	13 000	19	**K 35x40x13**
	40	17	19,8	39	4,65	12 000	13 000	21	**K 35x40x17**
	40	27	23,8	49	6	12 000	13 000	39	**K 35x40x27 TN**
	42	16	23,3	37,5	4,5	11 000	13 000	34	**K 35x42x16**
	42	18	26,4	44	5,3	11 000	13 000	34	**K 35x42x18**
	45	20	35,2	50	6,2	11 000	12 000	56	**K 35x45x20**
37	42	17	21,6	43	5,2	11 000	13 000	22	**K 37x42x17**
38	43	17	19,8	39	4,65	11 000	12 000	29	**K 38x43x17**
	46	32	52,3	100	12,5	10 000	12 000	76	**K 38x46x32**
40	45	17	20,5	41,5	5	10 000	12 000	31	**K 40x45x17**
	45	27	31,4	73,5	9	10 000	12 000	46	**K 40x45x27**
	48	20	34,7	58,5	7,35	10 000	11 000	49	► **K 40x48x20**
42	47	17	20,9	43	5,2	10 000	11 000	32	**K 42x47x17**
	50	20	33,6	57	7,1	9 500	11 000	53	**K 42x50x20**
43	48	17	20,9	43	5,2	9 500	11 000	30	**K 43x48x17**
45	50	17	21,6	46,5	5,6	9 000	10 000	34	**K 45x50x17**
	50	27	33	81,5	10	9 000	10 000	52	**K 45x50x27**
	53	28	49,5	98	12,2	9 000	10 000	81	**K 45x53x28**
47	52	17	22,4	49	6	9 000	10 000	35	**K 47x52x17**
50	55	20	25,5	60	7,2	8 500	9 500	43	► **K 50x55x20**
	55	30	37,4	98	12	8 500	9 500	65	**K 50x55x30**
	57	18	31,9	64	7,8	8 000	9 000	47	**K 50x57x18**
	58	25	41,8	81,5	10,2	8 000	9 000	90	**K 50x58x25**
55	60	20	27	67	8,15	7 500	8 500	40	**K 55x60x20**
	60	30	39,6	108	13,4	7 500	8 500	71	**K 55x60x30**
	62	18	34,1	71	8,5	7 500	8 500	52	**K 55x62x18**
	63	32	59,4	129	16,3	7 500	8 500	102	**K 55x63x32**

Table A.3 – Needle roller bearings with machined rings with flanges, without an inner ring F_w = 20 – 29 mm

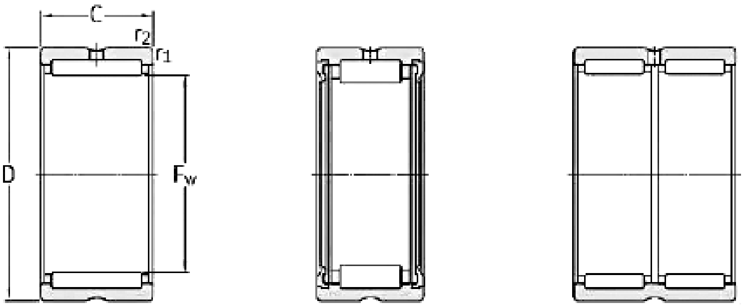


NK(S)
RNA 49
RNA 69

RNA 49 ...2RS

Principal dimensions			Basic load ratings dynamic	static	Fatigue load limit	Speed ratings Reference speed	Limiting speed	Mass	Designation
F_w	D	C	C	C_0	P_u				
mm			kN		kN	r/min		kg	–
20	28	13	9,13	12	1,43	–	9 500	0,022	▸ **RNA 4902.2RS**
	28	13	11,2	15,3	1,83	19 000	22 000	0,022	▸ **RNA 4902**
	28	16	13,2	19,3	2,28	19 000	22 000	0,027	▸ **NK 20/16**
	28	20	16,5	25,5	3,05	19 000	22 000	0,034	▸ **NK 20/20**
	28	23	17,2	27	3,35	19 000	22 000	0,04	▸ **RNA 6902**
	32	20	23,3	27	3,25	18 000	20 000	0,049	▸ **NKS 20**
21	29	16	13,8	20,4	2,45	19 000	22 000	0,028	**NK 21/16**
	29	20	17,2	27	3,35	19 000	22 000	0,035	**NK 21/20**
22	30	13	9,52	12,9	1,53	–	9 000	0,023	**RNA 4903.2RS**
	30	13	11,4	16,3	1,96	18 000	20 000	0,022	▸ **RNA 4903**
	30	16	14,2	21,6	2,6	18 000	20 000	0,03	▸ **NK 22/16**
	30	20	17,9	29	3,55	18 000	20 000	0,037	▸ **NK 22/20**
	30	23	18,7	30,5	3,75	18 000	20 000	0,042	▸ **RNA 6903**
24	32	16	15,4	24,5	2,9	16 000	19 000	0,032	▸ **NK 24/16**
	32	20	19	32,5	4	16 000	19 000	0,04	▸ **NK 24/20**
	37	20	26	33,5	4	15 000	17 000	0,066	**NKS 24**
25	33	16	15,1	24,5	2,9	16 000	18 000	0,033	▸ **NK 25/16**
	33	20	19	32,5	4	16 000	18 000	0,042	▸ **NK 25/20**
	37	17	19,4	22,4	2,65	–	7 500	0,056	▸ **RNA 4904.2RS**
	37	17	21,6	28	3,35	15 000	17 000	0,052	▸ **RNA 4904**
	37	30	35,2	53	6,55	15 000	17 000	0,1	▸ **RNA 6904**
	38	20	27,5	36	4,4	15 000	17 000	0,068	▸ **NKS 25**
26	34	16	15,7	26	3,1	15 000	17 000	0,034	▸ **NK 26/16**
	34	20	19,4	34,5	4,25	15 000	17 000	0,042	▸ **NK 26/20**
28	37	20	22	36,5	4,55	14 000	16 000	0,052	▸ **NK 28/20**
	37	30	31,9	60	7,5	14 000	16 000	0,082	▸ **NK 28/30**
	39	17	23,3	32	3,9	14 000	15 000	0,05	**RNA 49/22**
	39	30	36,9	57	7,2	14 000	15 000	0,098	**RNA 69/22**
	42	20	28,6	39	4,75	13 000	15 000	0,084	**NKS 28**
29	38	20	24,6	42,5	5,2	14 000	15 000	0,05	**NK 29/20 TN**
	38	30	31,9	60	7,5	14 000	15 000	0,084	**NK 29/30**

Table A.4 – Needle roller bearings with machined rings with flanges, without an inner ring F_w = 20 – 43 mm



NK(S)
RNA 49
RNA 69 ($F_w \leq 38$ mm)

RNA 49...2RS

RNA 69 ($F_w \geq 40$ mm)

Principal dimensions			**Basic load ratings** dynamic	static	**Fatigue load limit**	**Speed ratings** Reference speed	Limiting speed	**Mass**	**Designation**
F_w	D	C	C	C_0	P_u				
mm			kN		kN	r/min		kg	–
30	40	20	25,1	44	5,5	13 000	15 000	0,061	► **NK 30/20 TN**
	40	30	36,9	72	9	13 000	15 000	0,092	► **NK 30/30 TN**
	42	17	21,6	27,5	3,25	–	6 300	0,06	► **RNA 4905.2RS**
	42	17	24,2	34,5	4,15	13 000	15 000	0,061	► **RNA 4905**
	42	30	38	62	7,65	13 000	15 000	0,11	► **RNA 6905**
	45	22	31,9	43	5,3	12 000	14 000	0,1	► **NKS 30**
32	42	20	26,4	48	6	12 000	14 000	0,064	► **NK 32/20 TN**
	42	30	34,1	65,5	8,3	12 000	14 000	0,1	**NK 32/30**
	45	17	25,1	36,5	4,4	12 000	14 000	0,073	**RNA 49/28**
	45	30	39,6	65,5	8,3	12 000	14 000	0,14	► **RNA 69/28**
	47	22	34,1	46,5	5,7	12 000	13 000	0,11	**NKS 32**
35	45	20	27,5	52	6,55	11 000	13 000	0,069	► **NK 35/20 TN**
	45	30	40,2	85	10,6	11 000	13 000	0,11	► **NK 35/30 TN**
	47	17	23,3	32	3,8	–	5 600	0,069	► **RNA 4906.2RS**
	47	17	25,5	39	4,65	11 000	13 000	0,069	► **RNA 4906**
	47	30	42,9	75	9,3	11 000	13 000	0,13	► **RNA 6906**
	50	22	35,2	50	6,2	11 000	12 000	0,12	► **NKS 35**
37	47	20	25,1	46,5	5,85	11 000	12 000	0,077	**NK 37/20**
	47	30	36,9	76,5	9,5	11 000	12 000	0,11	**NK 37/30**
	52	22	36,9	54	6,55	10 000	12 000	0,12	**NKS 37**
38	48	20	25,5	49	6,1	11 000	12 000	0,079	► **NK 38/20**
	48	30	37,4	80	10	11 000	12 000	0,12	**NK 38/30**
40	50	20	29,7	60	7,5	10 000	11 000	0,078	► **NK 40/20 TN**
	50	30	38	83	10,4	10 000	11 000	0,13	► **NK 40/30**
	52	20	30,8	51	6,3	10 000	11 000	0,089	► **RNA 49/32**
	52	36	47,3	90	10,8	10 000	11 000	0,16	► **RNA 69/32**
	55	22	38	57	7,1	9 500	11 000	0,13	► **NKS 40**
42	52	20	27	53	6,55	9 500	11 000	0,086	► **NK 42/20**
	52	30	39,1	86,5	10,8	9 500	11 000	0,13	**NK 42/30**
	55	20	27	43	5,3	–	4 800	0,11	**RNA 4907.2RS**
	55	20	31,9	54	6,7	9 500	11 000	0,11	**RNA 4907**
	55	36	48,4	93	11,4	9 500	11 000	0,19	**RNA 6907**
43	53	20	27,5	55	6,8	9 500	11 000	0,086	**NK 43/20**
	53	30	40,2	90	11,2	9 500	11 000	0,13	**NK 43/30**
	58	22	39,1	61	7,5	9 000	10 000	0,14	**NKS 43**

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