teeth), a large portion of gas emerges during breakdown, since in this case fragmentation into separate bubbles does not occur. As a result, a burst of the next tooth takes place immediately. At large amplitudes of the reciprocating motion a multi-tooth MFS behaves as a single-tooth MFS (Fig. 4.24). Moreover, such operation mode is characterized by continuous leakage of the medium to be sealed (Fig. 4.25).

Thus, when designing the reciprocating MFS, it is necessary to take into consideration the fact that their parameters correspond to those of the reciprocating MFS only at small amplitudes (e.g. at a mean velocity of $1 m s^{-1}$ this amount to 5 mm). An increase in the oscillation amplitude decreases the sustained differential pressure up to single-tooth MFS characteristics and there proceeds continuous leakage of the medium to be sealed.

4.2 Magnetofluid lubricants

4.2.1 *Magnetic fluid as a lubricant*

Operation of machines and mechanisms is impossible without using friction units, for example, rollers and sliding bearings, and different pinion drives. Intense use of these units causes rapid wear of their surfaces, leads to essential energy consumption for overcoming the friction forces and even their jamming, and thus to provide the operation of friction units, use is made of different lubricants. However, lubricating fluids may serve as lubricants only when a provision is made for their constant position in a friction zone or their supply to it.

A magnetic fluid is positioned by a constant magnetic field, so it is quite natural to use the fluid as a lubricant. Control of the position of such a medium and its supply to a friction zone presents no difficulties; however, a problem arises concerning its lubricating properties.

Lubricating properties of fluids are specified by two mechanisms. Firstly, when two solid surfaces wetted with a fluid come in contact, the latter is forced out of a gap between them. Removal of a viscous fluid from a thin gap requires large pressure gradients, and therefore a wedging pressure occurring in the fluid prevents the convergence of surfaces. A value of this pressure (and thus the efficiency of a hydrodynamic lubrication mechanism) is proportional to the viscosity of the lubricating fluid.

The second boundary mechanism is attributed to the formation of a thin adsorptive layer composed of lubricating fluid molecules on the surfaces of friction parts that prevents their direct contact. The effectiveness of this mechanism is specified by the thickness and strength of the adsorptive layer.

Magnetic fluids differ from traditional lubricants in the presence of solid particles. However, solid particles do not only not worsen but, in fact, essentially improve the lubricating properties of magnetic fluids, Particle sizes (1 Onm) are much less than a characteristic roughness dimension of friction surfaces, which excludes their abrasive action. Hydrodynamic interaction of particles causes an increase in the fluid viscosity described by the Vand formula (eqn 2.28), that is, the hydrodynamic lubrication mechanism in magnetic fluids is stronger than in their bases. Besides, the material which friction pair components are made of, is, as a rule, steel, which possesses high magnetic properties. Therefore, magnetic particles moving in a fluid flow around microirregularities of the surface are retarded by magnetic forces which causes an additional increase of the viscosity in near-boundary layers. Moreover, magnetic particles form a dense layer on the surface, preventing friction surfaces from direct contact. Thus, both the hydrodynamic and the boundary lubrication mechanisms are stronger in magnetic fluids than in ordinary lubricants.

The universally recognised characteristics of lubricating materials are their viscosity and lubricating power. The viscosity of magnetic fluids has been discussed in detail earlier. Thus, Fig. 2.4 presents rheological characteristics of a magnetic fluid based on turbine oil **Tn-22** widely used as a lubricant. The lubricating power is such that friction losses are decreased and that there is a reduction of friction pair working surfaces under boundary and, to some extent, half-liquid lubrication conditions (when the main lubrication mechanism is either boundary or hydrodynamic mechanism, respectively). This property depends on the strength of a thin film of a liquid lubricant formed during the wetting of a solid surface. At some critical temperature the film tears, the friction surfaces come into direct contact, the temperature increases even more, and the wear of surfaces dramatically enhances seizing or even 'welding' of friction surfaces. For a turbine oil the critical temperature is 130-140 °C.

Quantitative characteristics of the lubricating power comprise critical load P_{cr}, welding load P_w, and scuffing index I_s. These parameters are determined at tests on a four-ball machine, a friction unit of which represents a pyramid of four steel balls in contact with each other. Three lower balls are fixed in a cup of the machine and completely covered with a test fluid. An upper ball rotates with the prescribed speed (1460-70 rpm) at the given load. The tests are carried out for a certain period of time (GOST 9490-75 for 10s for the four-ball machine). After the test a wear spot diameter is determined.

The critical load P_{cr} is determined as follows. Tests are carried out for a certain series of loads P. For each load in this series there exists a limiting wear spot diameter equal to d_r + 0.15 mm, where d_r is the elastic deformation area diameter for the given load. A critical load in this series is that at which the wear spot diameter either exceeds the limiting value or exceeds the wear spot diameter by 0.1 mm at the preceding load in the series. The welding load is determined by the limiting load at which a lubricant is serviceable.

The scuffing index I_s characterizes the ability of the lubricating material to minimize damages of the friction surfaces due to scuffing, and is calculated from the measurement of the wear of balls from initial load up to the welding load in the following manner:

$$I_s = \sum_{i=1}^{n} Q_i / n \tag{4.21}$$

where n is the number of tests, and Q_i is the conditional load during the ith test determined as $Q_i = P_i d_r / d_w$ (P_i is the load and, d_w the mean diameter of a wear sport during the ith tests).

The test results for the lubricating properties of the turbine oil Tn-22-base magnetic fluid, whose rheological and magnetic characteristics have been reported in the above sections, are listed in Table 4.1. From the presented data, it can be seen that the lubricating properties of the magnetic fluid are markedly higher than those of its base: the initial load at which weal appears, the critical load, and the welding load have increased approximately 1.5 times (as compared to the oil base), and the scuffing index has increased three-fold.

Investigations of magnetic fluid properties in magnetic fields [25] show that the applied magnetic field also improves the lubricating properties: the welding load in the presence of a magnetic field has almost doubled. This may be explained by considering that the magnetic field increases both the magnetic fluid viscosity and the interaction of particles with a solid mag-

TABLE 4.1 *Test results for lubricating properties of the turbine oil-base magnetic fluid*

Load (N)	Wear spot mean diameter (mm)	Load (N)	Wear spot mean diameter (mm)
Turbine oil Tn-22		Magnetic fluid MMT-65	
320	0.38	400	0.35
400	0.40	630	0.43
450	1.83	710	0.65
500	2.03	790	0.72
560	2.51	1000	0.9
630	2.84	1260	1.04
890	2.64	1410	1.21
1000	2.78	1590	1.37

Lubricating characteristics: turbine oil Tn-22: P_{cr} = 400 N, P_w = 1120 N, I_s = 13.9; magnetic fluid MMT-65 based on oil Tn-22: P_{cr} = 690 N, P_w = 1780 N, I_s = 46.3

netic surface, that is, enhancing both lubrication mechanisms.

Finally, magnetic fluids possess lubricating properties on par with the properties of their bases and very often compare favourably with them. This indicates that their use as lubricants may be promising.

4.2.2 *Lubrication systems*

The main advantage of a magnetic fluid as a lubricant lies in the possibility of supplying a lubricating medium only to the friction zone and its positioning in this zone with the aid of a magnetic field. This would require a specific design of a magnetic system.

The simplest lubrication system for rolling bearings is suggested in [26) where rolling bodies are made of a magnetosolid material possessing residual magnetization. When such bearing are used, a magnetic fluid is held on the surface of rolling bodies (balls or rollers), thus providing their lubrication.

In order to hold a thin fluid layer on the surface of a quickly rotating rolling body it is necessary to use magnetosolid materials with high residual magnetization for manufacturing the rolling bodies. But such materials, as a rule, are very brittle and are unsuitable for manufacturing the rolling bodies. The problem may be solved by using magnetic systems similar to those used in MFS (Fig. 4.26). A magnetic fluid fills a rolling bearing (not necessary completely) and external magnetic system seals it. Since the magnetic field is concentrated at the places of contact of balls with outer and inner rings then some portion of the magnetic fluid will always be pulled into these regions and the external magnetic system ensures the

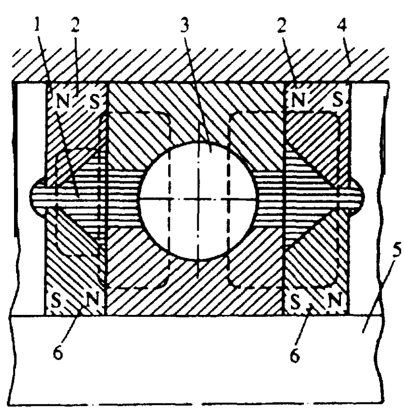

Fig. 4.26. Schematic diagram of the rolling **bearing lubricated by a magnetic fluid [27]: l, magnetic fluid; 2,6, magnets; 3, ball; 4,5, movable and immovable** pieces.

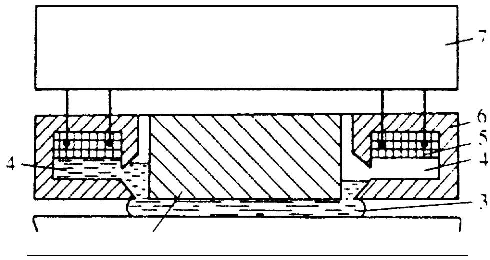

Fig. 4.27. Schematic diagram of magnetofluid lubricant circulation in a sliding bearing [28]: I, bearing; 2, shaft; 3, magnetic fluid; 4, cavities; 5, electromagnets; 6, cores; 7, current source.

availability of a magnetofluid lubricant and simultaneously prevents dirt from getting into a bearing.

Similarly, the problems of lubrication in bearings are solved. However, in these arrangements additional problems arise that are connected with intense heat release and wear of friction surfaces. These problems may be solved by providing permanent circulation of a lubricating medium. Implementation of the circulation with a small volume of the used magnetic fluid is possible at the expense of a periodic change in the direction of fluid motion in a working clearance of bearing. Figure 4.27 shows one of the possible schemes of realizing such an approach. A lubricant moves via the gap, undergoes attraction to a switched electric magnet, and fills an empty cavity. When the cavity is filled with the lubricant, another magnet switches on and the lubricant moves in the opposite direction. Continued motion of the magnetic lubricant provides cooling of the friction surfaces and removal of wear products out of the friction zone, which increases the service life of sliding bearings. Alternatively, the problem may be solved using permanent magnets eccentrically positioned on discs connected to a shaft and rotating with it (Fig. 4.28). A cavity for a magnetic lubricant is also made asymmetrically. Permanent magnets approach, in turn, the cavity from opposite sides, causing circulation of the magnetic lubricant.

Most promising is the use of magnetofluid lubricants in gearings in which the friction zone is local and a magnetofluid lubricant may be supplied to the friction zone by concentrating a magnetic field in the latter. From structural considerations, three main approaches to solution of this problem may be singled out. Magnetic field concentration may be created in a friction zone either at the expense of external systems (Fig. 4.29(a)) or by using gearing components as a magnetic line (Fig. 4.29(b)), or at the expense of location of a magnetic field source inside the gearing components (Fig. 4.29(c)).

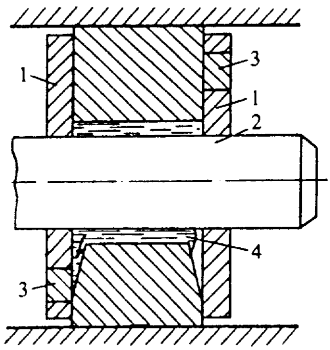

Fig. 4.28. Schematic diagram of magnetofluid lubricant circulation using permanent magnets [29]: 1, non-magnetic discs connected to shaft 2; 3, permanent magnets; 4, lubricant.

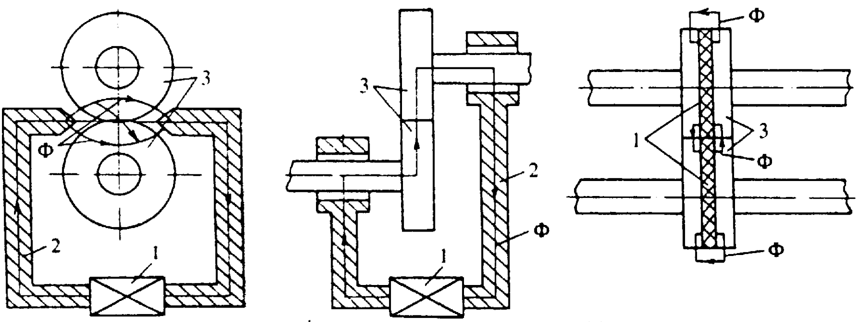

Fig. 4.29, Different methods used to concentrate a magnetic field in the friction zone [30]: 1, field **source;** 2, magnetic pipeline; 3, gearing components; **Φ,** magnetic flux.

Analysis of the gearing operation permits three zones of the surface of gear wheels to be distinguished: the friction zone, the reserve zone containing a magnetic fluid, and on an open gear surface only viscous and adhesion forces act to hold the lubricant and bring it to the friction zone. With such a scheme the reliable holding of the lubricant is ensured only at small speeds of gear wheel rotation (about 10 rps [30]), while at high speeds lubricant drops are thrown off from the gear surfaces. Since a magnetic field is not strong in the friction zone, a variation of a magnetic flux in the system does not in fact, change the lubricating power of the magnetic fluid in the magnetic system of such a scheme. Amongst the merits of this scheme is its simplicity and independence of the gearing system itself. At a small speed of rotary gearing wheels it successfully supplies the friction zone with a lubricant.

The second scheme (Fig. 4.29(b)) is suitable only for friction units made of magnetic materials. A magnetic flux, passing through gearing wheels, is accumulated in a friction zone thus providing the holding of a magnetic lubricant in it. As in the first scheme, the magnetic field is absent on an open surface and only the forces of adhesion and viscosity act. Hence, the capabilities of the first and the second schemes are identical, a difference lies in the fact that with an increase of a magnetic flus the magnitude of the magnetic flux in the friction zone increases to improve appreciably the tribotechnical characteristics of the unit. This effect is particularly pronounced under oil-deficient conditions, that is, with small volumes of a magnetofluid lubricant. An increase of the magnetic field causes its spreading over a magnetized gear surface which excludes the presence of dry friction regions and, therefore, favours operation of the friction unit.

An optimal scheme, from the point of view of holding a magnetic fluid, is one in which magnets are mounted into gearing wheels. More efficient is having the magnetic discs magnetized in opposite directions along their axis, In this case a magnetic lubricant covers the entire surface of a gear wheel. Critical speeds, at which lubricant drops begin to be thrown off the wheel surface, are 1.3-1.5 times higher than in the second scheme [30] although their values are also not high up to 15 rps. At speeds below the critical one continuous lubricant supply to a friction zone proceeds, that is, the wear-free conditions are ensured in practice. However, an excess amount of a lubricant on a gear wheel surface leads to increased power loss due to viscous friction, that is, to decreased gearing efficiency. The complex design of gearing wheels, each of which consists at least of three components (a magnet and two halves of the wheel) should be noted. Serviceability of this scheme, as with the first two schemes, is ensured only at small speeds of the wheels.

For gearing wheels the main mechanism seems to be a boundary lubrication mechanism. Therefore, their operation does not require a large volume of a magnetic fluid to be held in a friction zone; it is only necessary to provide the continued presence of a then lubricant layer on friction surfaces. Moreover, an excess volume of the magnetic lubricant causes excess power losses due to viscous friction, that is, decreases the gearing efficiency. These problems may be settled, if magnetic forces are applied not for positioning of the magnetic fluid in a friction zone but only for its supply to the friction surface where it may be thrown away by centrifugal forces. A portion of the magnetic fluid, which has remained on the surface due to adhesive forces, is sufficient to ensure serviceability of the friction unit. Figure 4.30 shows schematically the design of the friction unit implementing

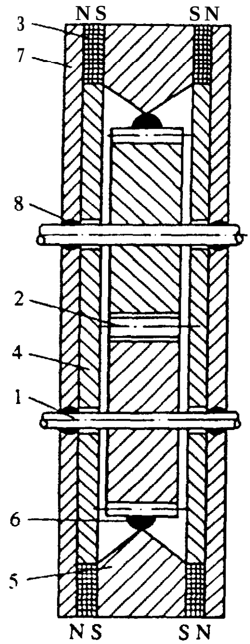

Fig. 4.30. Schematic diagram of the leak-proof friction unit with lubricant circulation under the action of magnetic and centrifugal forces [31]: **1,** shafts; 2, **pair of gearing wheels;** 3, magnets; 4, non-magnetic insert; 5, pole **piece;** 6, magnetofluid **lubricant; 7, end-type covers; 8, sealing magnetic fluid.**

lubrication in the manner discussed above. A **pair** of gearing wheels is embraced round the periphery by an external magnetic system so that a magnetic field is maximal near the surface of the wheels. A magnetic lubricant is thus located along the entire surface of the wheels. When the latter are rotating, centrifugal forces throw the magnetic fluid off the gear surface but falling onto the surface of a pole piece it is again pulled into the region of a maximum magnetic field and finds itself on the friction pair surface, wets it, and thus provides the constant presence of a lubricant film on friction surfaces. A magnetic system may be designed so as not to have too large a magnetic field and to provide the separation condition for the given speed of gear rotation. In practice, a clearance between the pole piece and the gearing wheels must not exceed 0.6-1 .5 mm.

Experiments show that when the volume of a magnetic lubricant is increased, the power losses first decrease (by 10-20 per cent as compared to dry friction) and then begin to grow. This is indicative of the friction unit operation under the above conditions and of the guaranteed lubricant supply to the friction zone. For high-speed friction pairs, designs such as the one depicted in Fig. 4.30 (i.e. embracing the friction pair round the periphery and providing return of a magnetic lubricant to the friction zone) solve the lubrication problem most successfully.

4.3 Supports, bearings, dampers, and shock-absorbers

The operation both of magnetofluid seals and lubrication systems relies upon the capability of magnetic fluids to be pulled into the region of a strong magnetic field. The whole class of arrangements, close in their design to seals and lubrication systems but intended for other, sometimes opposite, purposes, employ another phenomenon typical of only magnetic fluids, namely, pushing non-magnetic bodies immersed into a magnetic fluid out of the region of a strong magnetic field. This effect has been discussed in detail in Section 3.2.1 and we now consider its technical applications.

4.3.1 *Supports*

The simplest application of the effect is the manufacture of magnetofluid supports on its base (i.e. the arrangements capable of holding some load and providing the availability of a fluid layer between movable components). A schematic diagram of such arrangement contains three elements: a magnetic field source, a magnetic fluid, and a non-magnetic support element. A magnetostatic repulsive force acts on the support element submerged into the fluid. If it is larger than the pressure on the support element, then the arrangement causes it to float, (i.e. liquid friction instead of dry friction during its motion).

The main characteristic of the support is the limiting load on to the support element at which the latter is still floating. As follows from Section 3.2.1, the magnetostatic repulsive force arises because of fluid pressure redistribution due to the action of an external magnetic field. For its accurate calculation many factors must be used, such as the fluid magnetization curve, configuration oft the volume occupied with the fluid, and the demagnetizing factor. But its rough estimate is obtained rather simply. If a magnetic fluid volume is sufficiently large, then magnetic intensity on its surface is close to zero. In this case the repulsive force is proportional to magnetostatic pressure $\mu_0 MH$ and cross-sectional area S of the support in the plane perpendicular to the direction of the repulsive force:

$$F = \mu_0 MHS \tag{4.22}$$

where H is the magnetic field on the support area, and M is the fluid