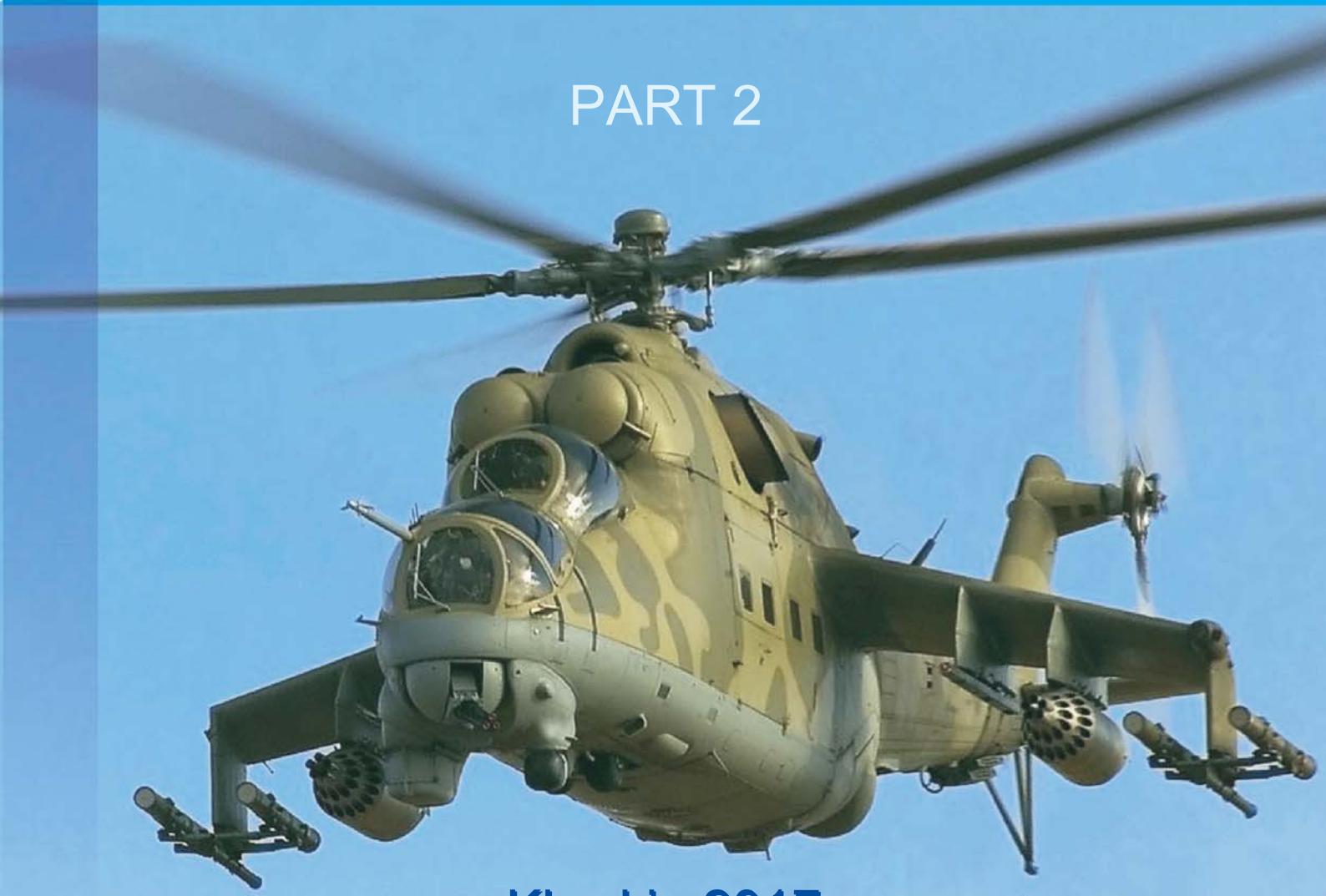


HEAVY SINGLE-ROTOR HELICOPTERS AND THEIR TRANSMISSIONS. DESIGN BACKGROUND

PART 2



Kharkiv-2017

MINISTRY OF EDUCATION AND SCIENCE OF UKRAINE
National Aerospace University "Kharkiv Aviation Institute"

HEAVY SINGLE-ROTOR HELICOPTERS AND THEIR TRANSMISSIONS. DESIGN BACKGROUND

Part 2

Under the editorship of PhD, prof. V.S. Krivtsov

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Розглянуто метод загального проектування трансмісії важкого вертольота, технологію виготовлення і методи забезпечення ресурсу редуктора, особливості технологічної підготовки виробництва для монтажу трансмісії важкого вертольота, методику вибору оптичних і лазерних засобів монтажу і контролю трансмісії, типові операції монтажу і контролю трансмісії на базі оптичних і лазерних систем. Викладено методику розроблення керівництва з технічної експлуатації трансмісії (на прикладі вертольота Мі-26). Описано метод забезпечення надійності трансмісії вертольота при проектуванні і в експлуатації. Розроблено методологію інтегрованого проектування і моделювання вертольотів і їхніх трансмісій.

Для студентів спеціальностей "Літаки і вертольоти", "Технологія літако- і вертольотобудування" і фахівців в області проектування і виробництва вертольотів.

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Heavy Single-Rotor Helicopters and Their Transmissions. Design Background /
H43 A. G. Grebenikov, A. M. Gumenniy, A. I. Dolmatov. – Guide-book. – Kharkov: Nat. Aerospace Univ. "Kharkiv Aviation Inst.", 2017. Part 2. – 413 p.

Presented are the method of designing background of heavy helicopter transmissions, production technology and the methods of gearbox service life provision, features of preproduction planning for installation of heavy helicopter transmissions, the method of choice of optical and laser tools for transmission assembly and control, typical operations of transmission installation and control based on optical and laser systems. The technique for development of transmission maintenance manual (on an example of the Mi-26 helicopter) is given. The method of helicopter transmission reliability in the process of its designing and operation is described. Methodology of integrated design and simulation of helicopters and their transmissions has been developed.

For students of specialty "Planes and Helicopters", "Technology of Plane and Helicopter Manufacturing" and specialists in the design and manufacture of helicopters.

Ill. 260. Tables 18. Bibliography.: 98 references.

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Chapter 11. GENERAL DESIGN OF TRANSMISSION OF HEAVY HELICOPTERS

11.1. GENERAL PRINCIPLES OF DESIGNING

11.1.1. General Requirements for Transmission. Classification. Main Elements and Their Function

The helicopter transmission (from the Latin *transmissio* -. the transition, transfer) – a constructive set of components and assemblies for the power transmission from the engine (s) to the rotors (for main and tail – for the helicopter single-rotor design to carrier – in twin-rotor schemes helicopters) and others power consumers with the necessary speed.

Transmission of helicopter [1, 2] consists usually of the following components and assemblies: gearboxes (main, intermediate, tail, combiner, etc.), transmission shafts, couplings of shafts and bearings; actuators and components necessary for the operation of the various systems of the helicopter; freewheels and clutches; main rotor brake; fan installation.

This chapter deals with calculation and design elements of the various schemes of helicopter transmissions: single-rotor, coaxial, longitudinal. The transmission elements are detailed, such as gearboxes (main, tail, with coaxial rotors), transmission shafts, couplings, shaft bearing assemblies, the sealing devices, main rotor brake. Brief presentation of materials related to the lubrication and cooling of helicopter components and assemblies is given.

11.1.1.1. General Requirements for Transmission and its Elements

Transmission must be reliable in operation and have a long life to first overhaul [2].

In case of refusal of one or more engines (helicopter with the number of engines is more than one) output from running engines must be transmitted to the rotors and other devices to ensure the continuation of the flight and the helicopter control. In case of failure of all the engines necessary to maintain the normal functioning of the rotors in autorotation and the work of other devices necessary for management while reducing and landing the helicopter. The transmission is equipped with devices for the automatic disengagement of any engine in the event of shutdown or failure, for example, with the use of freewheel clutches (FC).

The layout of the transmission must be capable of servicing and replacement of parts in accordance with the "Guidelines" on the technical operation and maintenance.

Auxiliary units must, as a rule, have a "weak link" for transmission protection from excessive torque. The "weakest link" is constructed so that in the event of destruction of its fragments does not cause damage to other transmission assemblies or other actuators helicopter systems.

Static and dynamic (fatigue) stress and strain of the whole range of basic components and transmission components as well as assemblies and vibration places of fastening of transmission units should not exceed certain values. Specifically, allowed is vibration of shaft (torsion and bending) in all operating modes.

These values are set taking into account operating experience and bench test results and test production and repair of transmission units in order to confirm the adequacy of the static and fatigue strength of units (nodes) transmission and durability of their components.

Power train design must be for a certain period of operation (assigned resource) to withstand repeated loads in service without a critical failure of the transmission.

Resources components (e.g. rolling bearings) is set on the basis of tests within the system of units (assemblies) of transmission.

In addition, the transmission should have a high efficiency, i.e. provide low mechanical power costs due to friction in gears, bearings and other components. It should be of simple scheme, comprise parts not complex to manufacture, and have a minimum weight possible.

11.1.1.2. Initial Design Data

1. The transmitted power P, kW.
2. The transmitted torque T_t, N/m.
3. Engine speed n, rpm.
4. Main rotor (screws) speed n_m, rpm.
5. Intermediate shaft rotation speed n_s, rpm.
6. Tail rotor speed n_r,rotation / min.
7. Assigned service life τ, hours.
8. The nature of the change of current load for the assigned service life (transmission parts loading sequence diagram).
9. The loads acting on the shaft of main and tail rotor:
   - drag F, N;
   - rotor weight G, N;
   - gyroscopic moment M_g, N·m;
   - centrifugal force of the rotor unbalance F_c, N.
10. Changes in the ambient temperature during operation Δt_a, ºC.
11. The temperature change of transmission parts when working in different modes Δt, ºC.
12. Allowable oil temperature in the cooling system t_o, ºC.
13. Permitted quantities characterizing the stiffness of the transmission shafts:
   - twist angle φ, degrees;
   - section rotation angle ψ, degrees;
   - maximum shaft deflection y, mm.
14. Permissible skew angle of transmission couplings α, degrees.
15. Probability of failure-free operation of parts.
16. Material of parts and their mechanical characteristics.

11.1.1.3. Classification of Transmissions

The design and layout of the power train are largely determined by helicopter scheme, the number, type, and location of engines on the helicopter [3].

Most often, the helicopters are classified according to the number of MR, their location, method of compensation of MR reactive torque. There are the following basic design helicopters configurations: single-rotor with tail rotor, coaxial dual rotor, tandem dual rotor, side-by-side dual rotor, as well as jet-powered configuration mounted on the blades of main rotor (Fig. 11.1.).

Single-rotor scheme with tail rotor (Fig. 11.1, a) is most prevalent. The reactive moment of the rotor is compensated by torque created by thrust of tail rotor mounted in the rear portion of fuselage.

Coaxial dual rotor scheme (Fig. 11.1, b) involves using two MR installed on the same axis, which rotate in opposite directions. Movement direction of the helicopter of this configuration is controlled by changing the angles of the upper and lower blade of rotors.

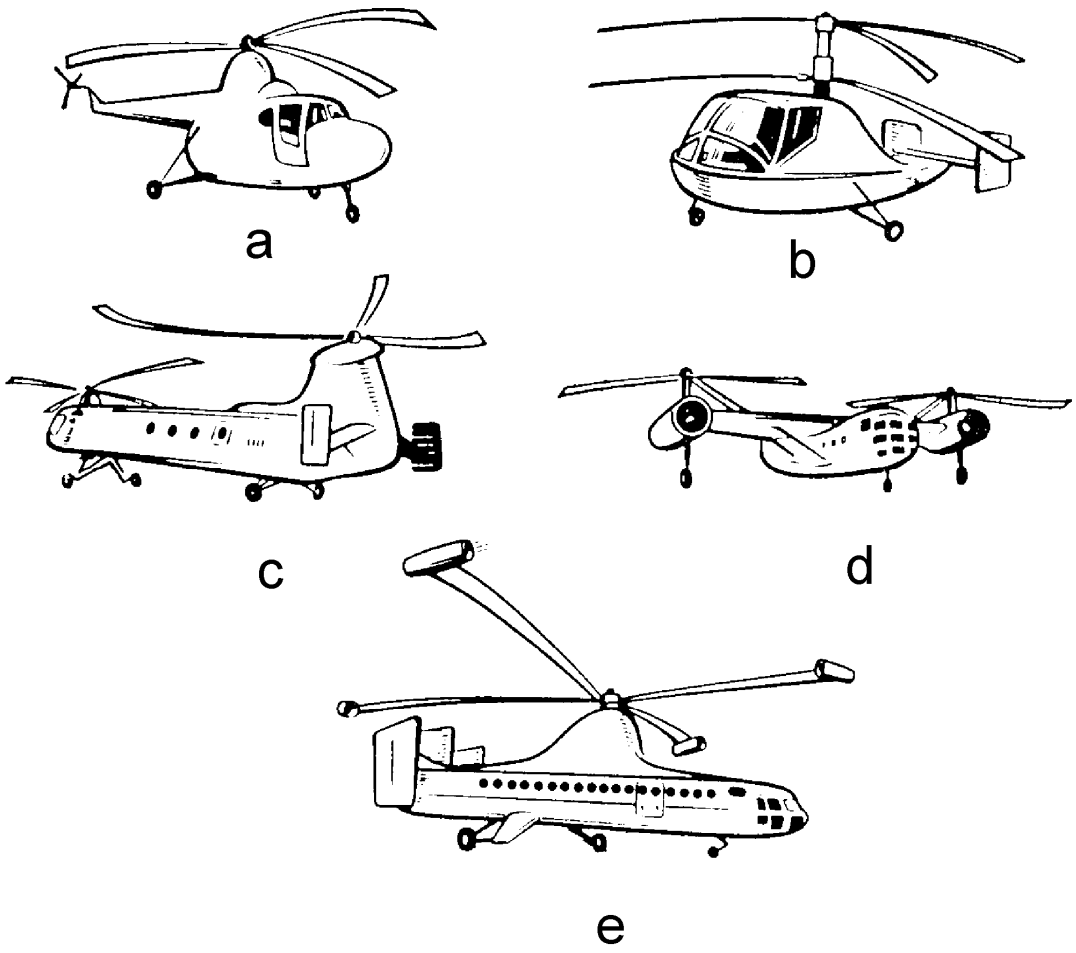


Fig. 11.1. Types of Helicopters

Tandem dual rotor configuration (Fig. 11.1, c) comprises the case with two MR mounted on its ends. The disadvantages of such a scheme are complex control system and the piloting technique.

Side-by-side twin-rotor scheme (Fig. 11.1, d) is based on the same principle as the tandem one. In this scheme, main rotors are located on the sides of the fuselage. To fix the main rotor a wing is generally used.

Possible is helicopter scheme with jet engines mounted on the MR blades (Fig. 11.1, e). In this case, the torque is created directly on the main rotor to rotate freely under the body of the helicopter, so that the reaction torque on the fuselage of the

helicopter is not transmitted. However, such a scheme due to a number of significant shortcomings is widely not received.

11.1.1.4. Primary Elements of Helicopter Transmissions and Their Purpose

By using gearboxes engine speed is reduced to the required for the effective operation of the main rotor or the tail rotor. Gearboxes are intended: to create a high torque required for rotating the rotor; to change the direction of the rotation axis (e.g., horizontal engine rotation axis to the MR vertical rotation axis); for taking up the loads generated by main rotor and tail rotor and transmission then to elements of the fuselage structure; to drive of the various units serving a helicopter. Some types of gearboxes (e.g., main gearbox) can perform all these functions, other types of gearboxes – only certain functions (e.g., intermediate and tail gearboxes).

Transmission shafts are used to transmit rotation from both the engine (engines) to gearboxes and from the gearbox to the gearbox. In some cases, transmission shafts can transmit the rotation from the gearboxes to the MR. It also includes transmission shafts, transmitting rotation to fan installations or any other units.

Wheel cases sometimes are mounted directly to gear units, and in some cases they are independent units, driven in rotation by a special transmission shaft from any gearbox.

Clutches are designed to connect or disconnect the engine (engines) from a helicopter transmission. *When installed on a helicopter gas turbine engine (GTE) with free a turbine clutch is not required.*

Main rotor brake device is intended for fixation of the rotor during the helicopter stand and in some cases freezes MR, which continues to rotate after the landing of the helicopter.

Fan installation serves for cooling (through radiators) gearboxes' oil and in some cases the engine oil.

11.1.2. Transmissions of Helicopters of Single-Rotor with Tail Rotor Scheme

Widespread are helicopters of single-rotor scheme with tail rotor. Below are the basic parameters and design features of transmissions of Russian single-rotor helicopters: Mi-1, Mi-2, Mi-4, Mi-6 and Mi-8, Mi-24 and Mi-26.

Fig. 1.2 shows the layout of helicopter Mi-8 transmission [2, 4]. Power to the helicopter main rotor and tail rotor is transmitted from two turbine engines, installed in front portion of the fuselage. The main elements of the transmission: main gearbox (MGB) 2, intermediate gearbox (IGB) 5, the tail rotor gearbox (TR) 7, intermediate shaft 4, tail rotor shaft 6, fan drive shaft 1, main rotor brake 3. These primary members of the transmission are contained in all other transmissions of single-rotor helicopters.

11.1.3. Overall Layout of Coaxially-Rotor Helicopter Transmission

Coaxial helicopter transmission is designed to transmit power from the engine to the shafts of two coaxial rotors rotating in opposite directions.

Depending on the design features of the helicopter transmission coaxial helicopters may contain either two gears – engine gearbox and main gearbox, or one – the main rotor gearbox.

Table. 11.1 shows the main parameters of the main gearboxes of the coaxial helicopters Ka-15, Ka-25K and Ka-26, which are widely used for various tasks.

The transmission of helicopters Ka-15 and Ka-26 includes engine gearboxes that contain two stages: stage 1 – conical, stage 2 – planetary. The engine gear ratio of the Ka-15K helicopter is 3.326.

11.1.3.1. Transmission of Helicopter Ka-15

The transmission is designed to transmit power from the crankshaft of the piston engine to two coaxial shafts of main rotors rotating in opposite directions, and to lower the engine speed.

Helicopter transmission kinematic diagram is shown in Fig. 11.3, consists of engine gearbox AI-14V (AI-14VF) and main gearbox RV-15.

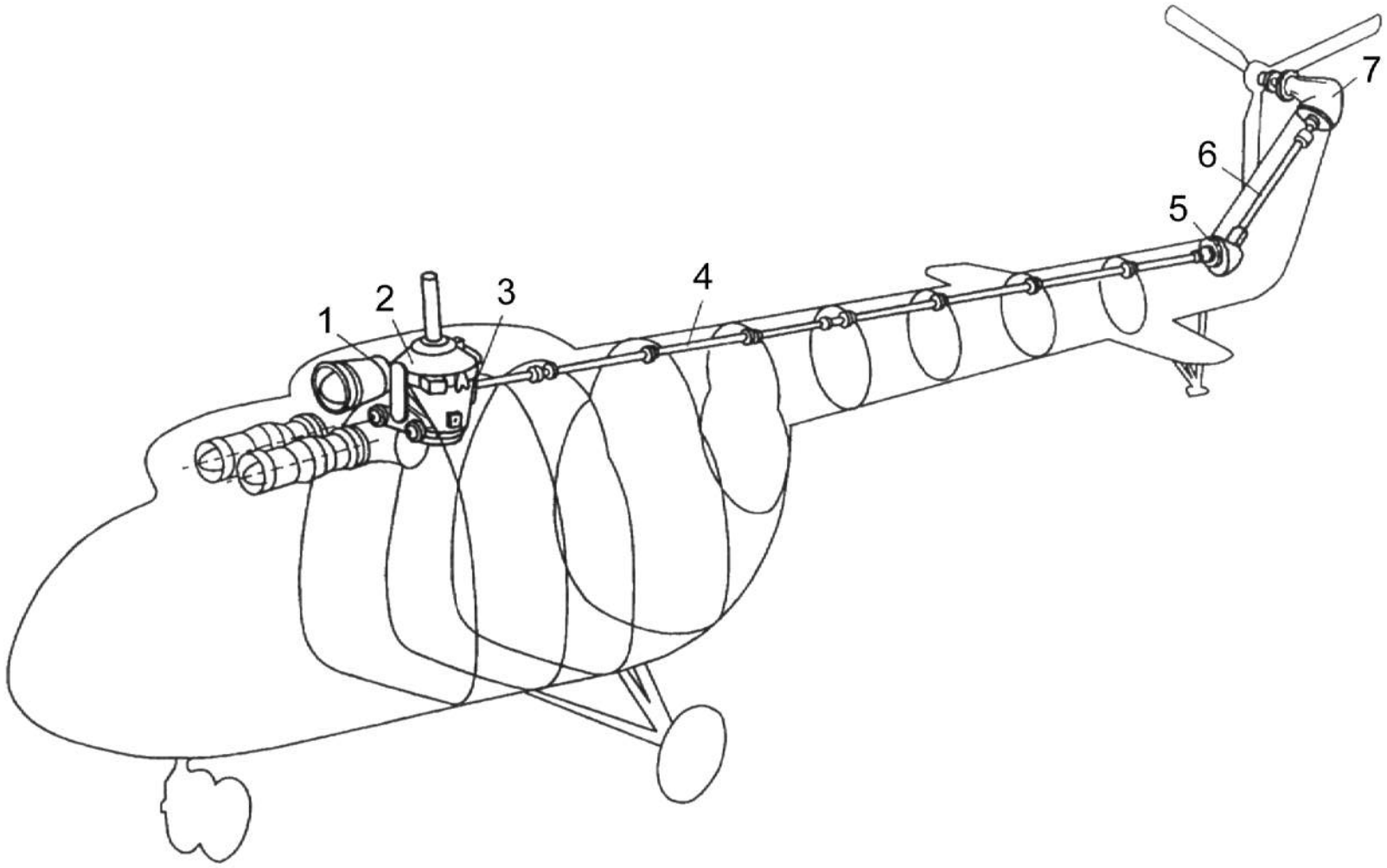


Fig. 11.2. Transmission Configuration of Helicopter Mi-8:
1 – fan drive shaft; 2 – main gearbox; 3 – MR brake; 4 – intermediate shaft; 5 – intermediate gearbox;
6 – tail rotor shaft; 7 – tail rotor gearbox

Table 11.1

Main Parameters of Main Helicopter Gearboxes

Helicopter	**Ka-15**	**Ka-26**	**Ka-25K**
Symbol of gearbox	RV-15	R-26	RV-3F
Feature	Parallel shaft gear: via external engagement – to the internal rotor, through the inner engagement – to the outer rotor	Three-stage with two input shafts: Stage 1 – conical spiral; Stage 2 and 3 – helical planetary	Four-stage: Stage 1 and 3 – cylindrical; Stage 2– conical with circular teeth; Stage 4 – planetary with gear train
Number and type of engines	1, piston	2, piston	2, GTE
Transmitted power, kW	280	239	662
Input shaft speed, rpm	–––	865	–––
Gear ratio	2.0	2.943	80.1
Torque at takeoff, N·m		7920 on one main rotor	
Rotor speed, rpm	377	294	237
Weight of gearbox, kg	80.5	201	560
Specific weight of gearbox		0.0127	

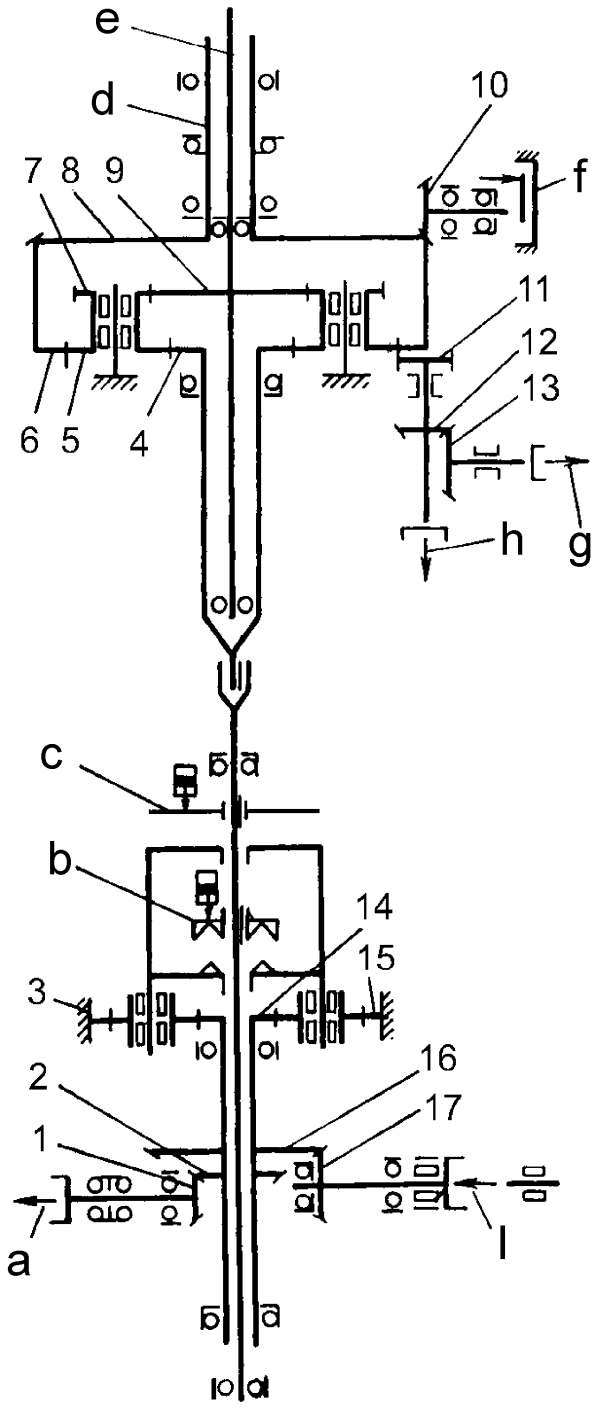


Fig. 11.3. Transmission Kinematic Diagram of Helicopter Ka-15:
a – fan drive; b – cam clutch; c – friction clutch; d – lower main rotor shaft; e – upper main rotor shaft; f – brake of main rotors; g – tachometer sensor; h – oil pump; i – engine crankshaft; 1, 2 – gears driving the fan; 3, 14, 15 – toothed wheel of the second stage engine gearbox; 4, 5, 6, 7, 9 – drive gears; 8, 10 – toothed wheel of drive brake rotors; 11, 12, 13 – gears driving the oil pump and the tachometer sensor; 16, 17 – toothed wheel of the first stage engine gearbox

The engine gearbox is the part of the engine accessory and it transmits power from the horizontal engine crankshaft to the vertical output shaft. The gear ratio of the engine gearbox equals 3.236. The kinematic chain of the gearbox consists of two sequential gear sets – bevel and planetary ones. Rotation is transmitted from the engine crankshaft **j** to the driven bevel gear 16, which is mounted on the axle shaft 14, through the drive bevel gear 17. This shaft transmits rotation through its cylindrical wheel rim to the six satellite gears 15, mounted on the planetary gear shafts in the satellite gear set. The satellite gears roll about the fixed gear 3 and drive the satellite gear set, which transmits rotation to the output shaft through the combination clutch.

The combination clutch comprises the friction clutch ***c*** and the cam clutch ***b***. The friction clutch is designed to disengage the main gearbox from the engine and to spin-up the main rotors smoothly when the engine is started. The cam clutch is engaged by means of the control unit when the rotors are spinned-up. Through this clutch the engine power is transmitted to the vertical shaft of the main gearbox. In the autorotation regime this clutch provides the kinematic disconnection between the engine and the main rotors.

Rotation is also transmitted from the main drive shaft to the fan drive shaft **a** through the bevel gears 1 and 2.

The main gearbox RV-15 is designed to transmit power from the engine gearbox output shaft to the two coaxial shafts of the main rotors of the helicopter. The gear ratio of the main gearbox equals 2. The bottom flange of the main gearbox strut is attached to the tip flange of the engine gearbox, and the output shaft of the engine gearbox is connected to the main drive shaft of the main gearbox RV-15 with splines. Rotation is transmitted from the main drive shaft of the gearbox to the six gear sets through the drive gear 4. Rotation is transmitted from the gears 5 to the shaft **d** of the lower main rotor through the gear 6. It is further transmitted from the gears 7 to the shaft **e** of the upper main rotor through the gear 9. The brake of the main rotors **e** is mounted on the main gearbox. This brake is driven by the bevel gears 8 and 10. The main gearbox lubrication pump **i** is driven by the gears 5 and 11.

The tachometer transmitter **g** is driven by means of the gear 12, mounted on the oil pump drive spindle, and the gear 13.

The schematic drawing of the main gearbox is shown in Fig. 11.4.

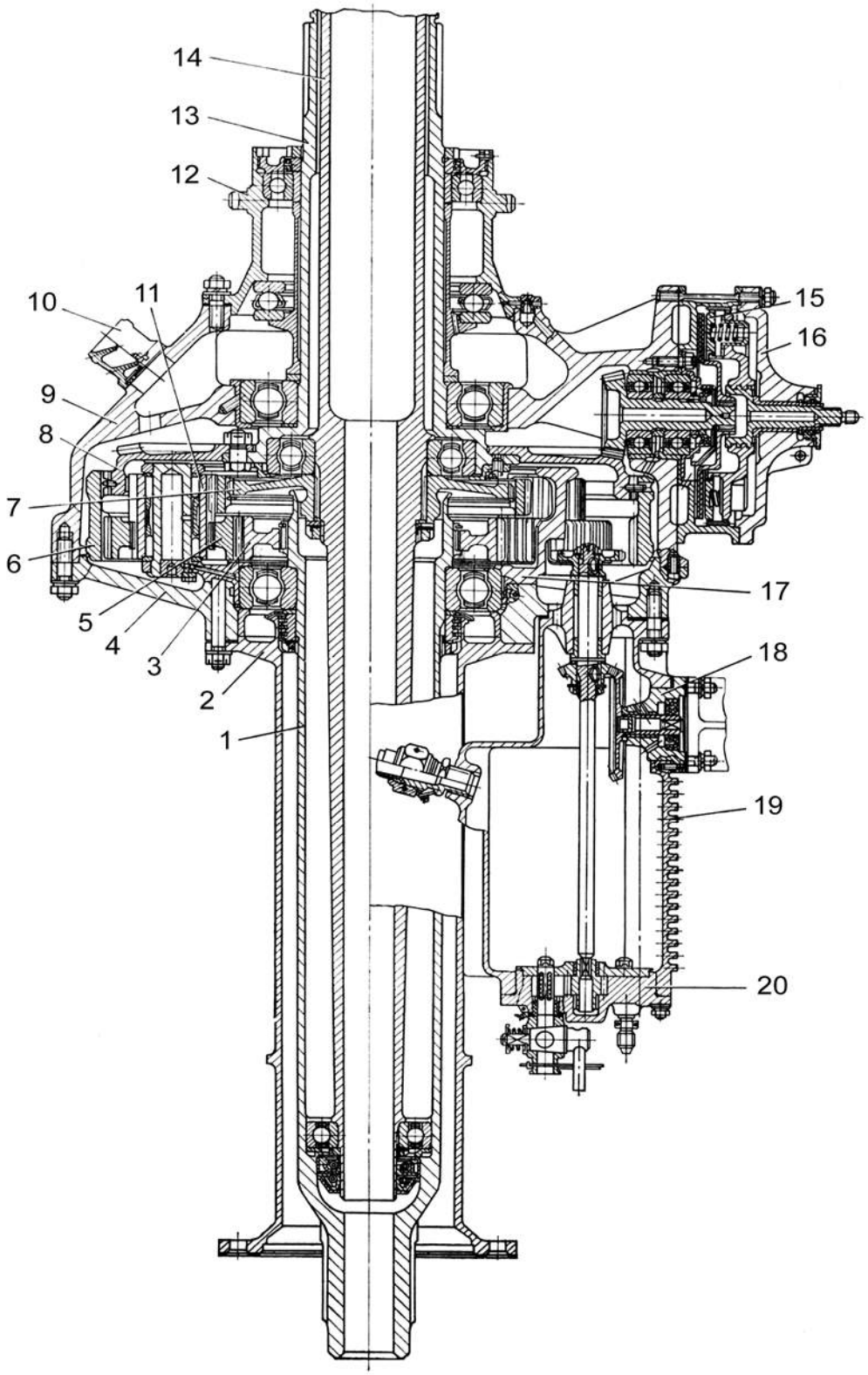


Fig. 11.4. Main Gearbox RV-15:
1 – main drive shaft; 2 – strut; 3 – drive gear; 4 – cover; 5 – pinion gear; 8 – drive block; 9 – main gear casing; 10 – breather; 11 – small gear; 12 – upper casing; 13 – outer shaft; 14 – inner shaft; 15 – disc brake of the main rotors; 16 – cover; 17 – unit casing; 18 – tachometer transmitter drive; 19 – oil sump; 20 – lubrication pump

11.1.3.2. Rotor Drive System of Helicopter Ka-25K

General. The main elements of the Ka-25 helicopter rotor drive system (Fig. 11.5) are the following: main gearbox 3, two main drive shafts 1, fan drive shaft 4, shaft 2 of the SSA-10 servo system actuator, brake of the main rotors 5.

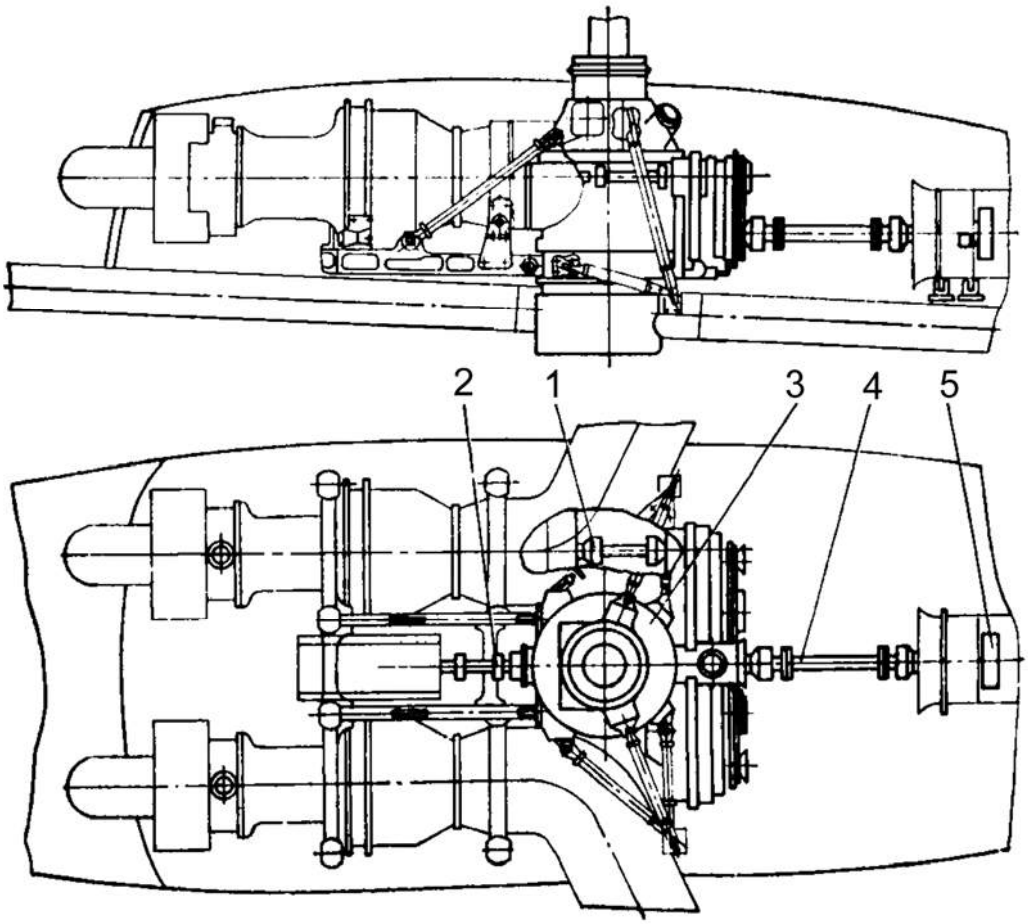


Fig. 11.5. Transmission of Helicopter Ka-25K:
1 – main drive shaft; 3 – main gearbox; 2 – SSA-10 actuator shaft;
4 – fan drive shaft; 5 – brake of the main rotors

The power is transmitted from the two gas-turbine engines GTE-3F to the two coaxial main rotors and helicopter actuators through the main gearbox RV-3F. The coaxial output shafts of the main rotors and the main gearbox are arranged vertically to the engine mounting plane and rotate contrarily at the same rate.

The main drive shaft is designed to transmit the power from the engine to the main gearbox at allowable angular, axial, and radial shifts of the free turbine shaft axis against the main gearbox drive shaft axis.

The fan drive shaft transmits power to the helicopter fan at allowable angular, axial, and radial shifts of the main gearbox drive axis against the fan impeller axis of rotation.

The SSA-10 drive shaft is designed to transmit power from the main gearbox to the accessory at allowable angular, axial, and radial shifts of the main gearbox drive axis against the SSA-10 input shaft.

The brake of the main rotors reduces their spin time when the engines are stopped. Besides, the brakes of the main rotors are designed to prevent the rotors and the rotor drive system elements from spinning up spontaneously under the influence of wind when the helicopter is on the ground.

The power from each engine (the left and right ones) is transmitted to the main gearbox through the respective drive shaft with two elastic clutches.

In the main gearbox the power is transmitted to the two coaxial shafts of the main rotors through the four-stage gear set.

The main gearbox is mounted on the frame in the upper fuselage section together with the engine and is attached to this frame with the braces. The coaxiality of the drive shaft clutches can be adjusted with the engine mounting. The helicopter actuators are connected to the rear box, which also drives the fan.

The fan has a built-in brake of the main rotors, the brake wheel of which rotates together with the fan impeller.

The kinematic diagram of the rotor drive system is shown in Fig. 11.6. The power is transmitted from the free turbines of each engine to the coaxial contra-rotating main rotors and helicopter actuators through the drive shaft and the main gearbox, which has four gear stages.

The first stage (right and left) consists of the spur gears: the drive gear 27 and the driven gear 5.

Between each driven gear 5 and each drive second-stage bevel gear 17 there are freewheel clutches c, which disengage the first stage with the free turbine of the stopped engine from the main rotors **e** and e when the second engine is running or when both engines are stopped during the helicopter's flight in the autorotation regime.

The second stage (right and left) consists of the bevel gears 17, 16 and is designed to turn the horizontal axes of rotation into the vertical ones. Each driven bevel gear 16 is connected to the third-stage drive spur gear 28 in a rigid way. At the third stage the helicopter is powered from the two engines.

The right and left spur gears 28 are engaged with the driven sun gear 15. The power flow from the third-stage driven gear is distributed between the planetary gear set and the fourth-stage counter gear set. The planetary gear set consists of the drive spur gear 11, six satellite gears 10 with the carrier, which is connected to the inner shaft f of the upper main rotor, and the driven internal gear 9.

The drive gear 11 is connected by splines with the third-stage driven gear 15, and the driven gear 9 interacts with the outer shaft e of the main rotor through the splined connection.

The gearing train consists of the drive spur gear 29, six double counter gears 14 and 12, and the driven internal gear 13, which is connected by splines to the outer shaft d of the lower main rotor.

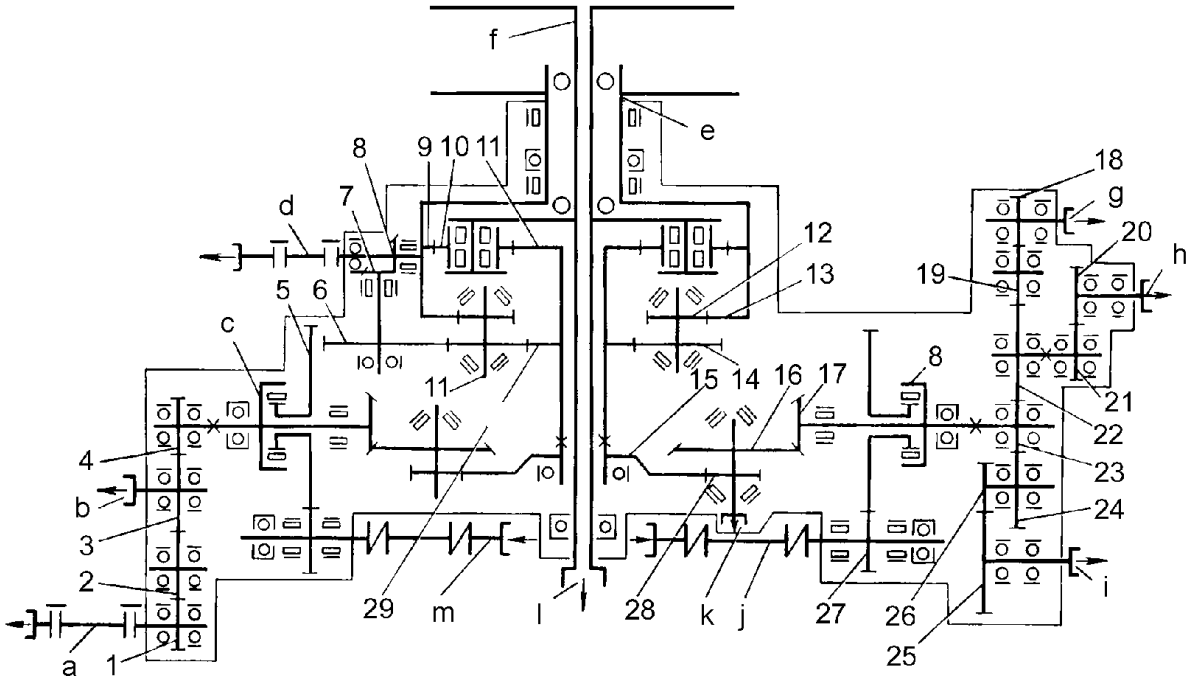


Fig. 11.6. Transmission Kinematic Diagram of Helicopter Ka-25K:
a – brake wheel and fan drive shaft; b – TTD-1 tachometer transmitter drive; c – freewheel clutch; d – SSA-10 servo system actuator shaft; e – lower rotor drive shaft; f – upper rotor drive shaft; g – SGS-4OU generator drive; h – AK-50T compressor drive; i – 435FT hydraulic pump drive; j – main drive shaft of the right engine; k – lubrication pump drive; l – collector drive; m – main drive shaft of the left engine;
1, 2 – fan and brake wheel drive gears; 3 – tachometer transmitter drive gear;
4, 23 – gearbox gears; 5, 27 – starting gears; 6,7, 8 – control unit drive gears;
9 – driven gear of the planetary gear set; 10 – satellite gear; 11 – drive gear of the planetary gear set; 12 – small gearing train gear; 13 – gearing train driven gear; 14 – gear pinion, 16, 17 – second stage bevel gears; 29 – gearing train drive gear; 15, 28 – third stage gears;
18, 19, 22 – generator drive gears; 20, 21 – compressor drive gears;
24, 25, 26 – 435-FT hydraulic pump drive gears

The rotation rates of the main rotors in the main gearbox are equal in value, and the directions of their rotation are opposite to each other.

The schematic drawing of main gearbox RV-3F is shown in Fig. 11.7. The gearbox is mounted on the helicopter with ten mounting brackets, the frame, and the system of brace.

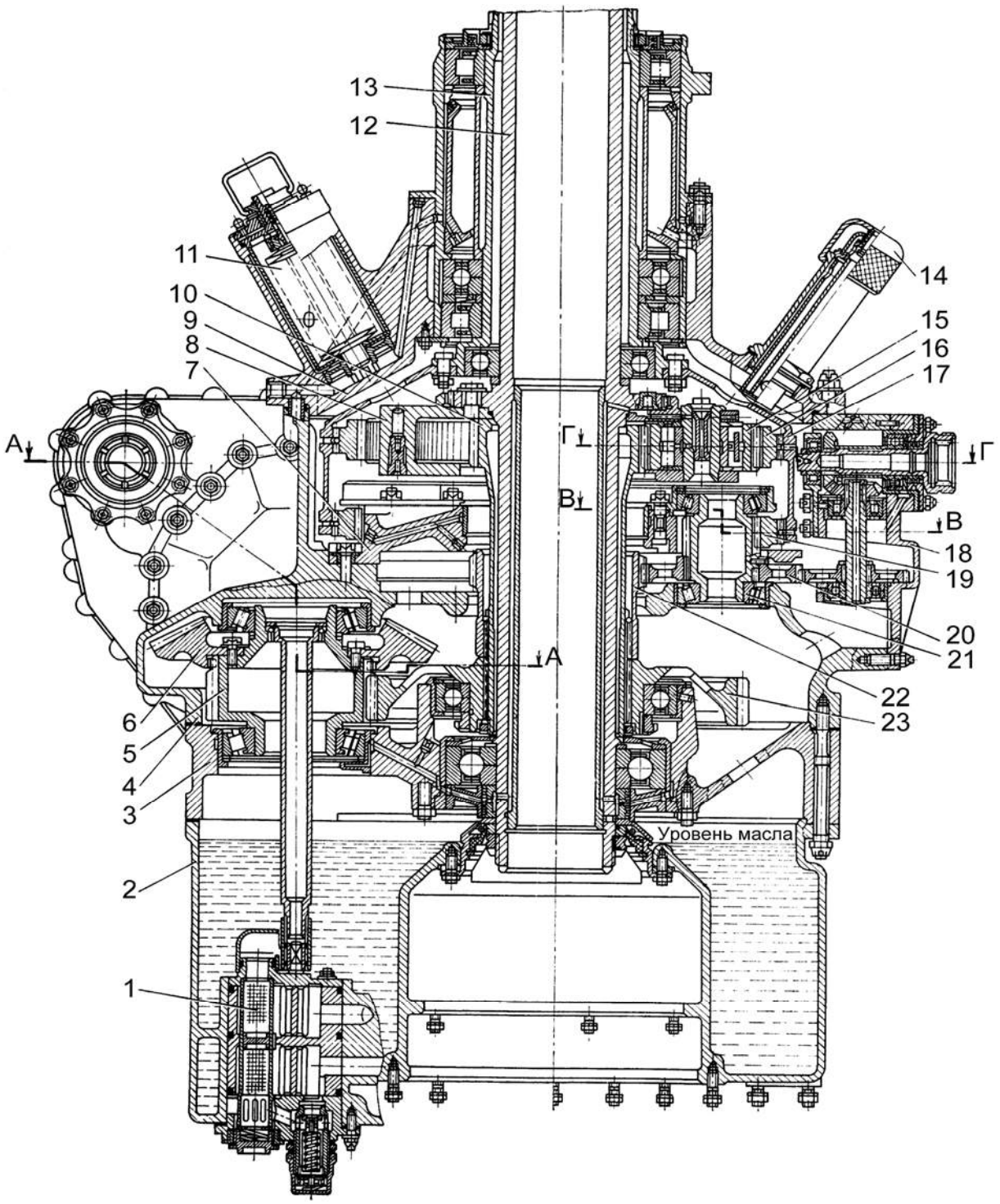


Fig. 11.7. Main Gearbox RV-3F (longitudinal section):
1 – lubrication pump; 2 – pan; 3 – lower casing; 4 – middle casing; 5 – third stage drive gear; 6 – second stage driven gear;
7 – gearing train gear casing; 8 – satellite gear casing; 9 – upper casing; 10 – drive gear of planetary shaft; 11 – main filter; 12 – inner shaft;
13 – outer shaft; 14 – breather; 15 – satellite gear; 16, 19 – internal gears; 17 – splined wheel; 18 – SSA-10 actuator casing; 20 – large gearing train driven gear; 21 – small gearing train drive gear;
22 – gearing train sun gear; 23 – third stage driven gear

11.1.4. Tandem Rotor Helicopter Transmissions

The CH-47 "Chinook" helicopter (Fig. 11.8) rotor drive system is distinctive because of the tandem-rotor configuration of this helicopter [2]. This rotor drive system consists of the gearboxes of the front and rear rotors 1 and 7, the intermediate gearbox 3, two angle gearboxes 5 of the engines 6, the synchronizing shafts 4, which connect the engines with the intermediate gearbox.

The kinematic diagram of the rotor drive system is shown in Fig. 11.9 [4]. In the diagram there are marked the gearboxes of the front and rear rotors I and III, the intermediate gearbox IV, two angle gearboxes II and V of the T55-L-II engines, the synchronizing shafts 8 and the shafts 10, which connect the engines with the intermediate gearbox.

The intermediate gearbox has two bevel gears 9 and 20, engaged with the common driven gear 19, with the gear ratio equal to 1.694.

In the gearbox there is a dephasing device 22, designed to position the main rotor blades in such a way to prevent their crossing.

The gearbox of the front main rotor is a triple-reduction gearbox with the total ratio equal to 30.67. The first gearbox stage consists of the helical bevel gear pair 2; the second 3 and third 4 stages are planetary gear sets. The first stage has the gear ratio equal to 1.754. The gear ratio of the second stage equals to 4.761, and the third stage gear ratio equals to 3.649. The gearbox is designed to transmit power, which equals to 60 % of the power transmitted by the intermediate gearbox.

The gearbox of the rear main rotor is similar to the gearbox of the front main rotor in its kinematic diagram. This gearbox comprises the accessories gearbox. The box is driven by the spur gear pair, which in its turn is driven by the shaft 16.

The engine gearbox is a single-stage angle gearbox with the gear ratio equal to 1.234 and the angle of rotation equal to 90º. It comprises spiral bevel gears.

The synchronizing shafts connect the intermediate gearbox with the main rotor gearboxes. The shaft speed is 7465 rpm. The critical speed is 8900 rpm. The shafts consist of aluminum pipe sections, mounted in the bearings and jointed with elastic clutches. The front shaft comprises six sections, the aft shaft comprises two sections.

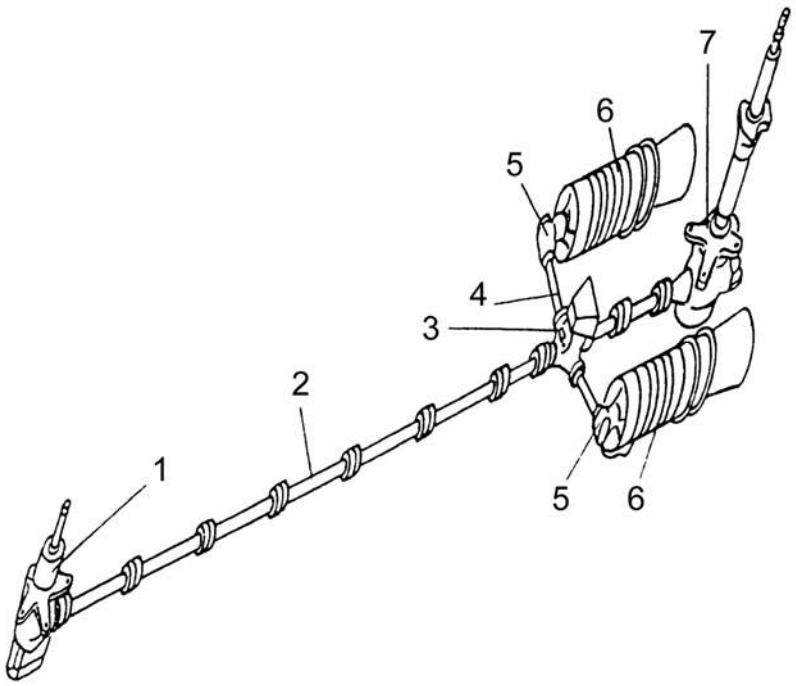


Fig. 11.8. CH-47 "Chinook" Helicopter Transmission Configuration:
1, 7 – main gearboxes of the main rotor; 2 – main rotor drive shaft; 3 – intermediate gearbox; 4 – synchronizing shaft sections; 5 – angle gearboxes; 6 – engines

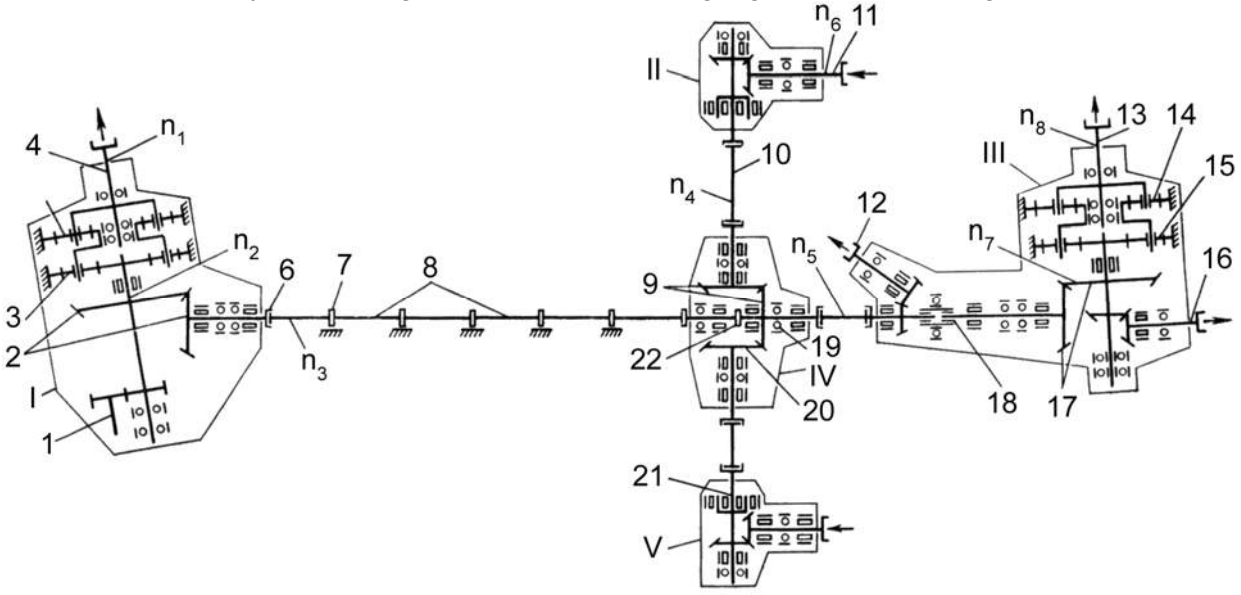


Fig. 11.9. Helicopter "Chinook". Transmission Kinematic Diagram:
1 – gearbox of front main rotor ; 11, V – angle gearboxes of the engine; III – gearbox of rear main rotor ; IV – intermediate gearbox ; 1 – lubrication pump drive; 2, 17 – first bevel stage; 3, 15 – first planetary gear set; 4, 14 – second planetary gear set; 6 – clutch; 7 – bearing support; 8, 10 – synchronizing shaft sections; 9, 20 – drive gears; 11 – engine shaft; 12 – fan drive; 5, 13 – main rotor shaft; 16 – accessories box drive; 18 – splined clutch; 19 – accumulating driven gear; 21 – freewheel clutch ; 22 – dephasing device $n_1 = n_8$ = 243 rpm; $n_2 = n_7$ = 4275 rpm; $n_3 = n_5$ = 7465 rpm; n_4 = 12 690 rpm; n_6 = 7 465 rpm

11.2. DESIGNING OF HELICOPTER MAIN, INTERMEDIATE AND TAIL GEARBOXES

11.2.1. Multi-Power Path Gearboxes. Selection of Teeth Number, Distribution of Forces and Moments, Power Flows

In helicopter transmissions and gearboxes of aircraft engines multi-threaded gearboxes, including planetary and differential gear boxes, are widely used. This is due to wide range of factors. The main ones are:

1) To ensure the coaxiality of the engine shaft and the rotor shaft (by arrangement conditions);

2) To ensure minimal overall dimensions and weight of the gearbox, which is achieved by using the planetary and differential gears which are having multi-power paths – that is power is transferred from the engine to the rotor in several flows;

3) To remove load on the central gears.

Design of planetary and differential gearboxes of helicopter and airplanes begins with the selection of the number of teeth in accordance with the general equations:

1) gear ratio equation,

2) the condition of coaxiality,

3) the condition of the neighborhood,

4) assembly condition.

Then gear wheels strength is calculated by reference to specific features of their work.

Let us consider some common questions regarding the calculation and design of planetary and wave gears. One of the main advantages of these transmissions is central joints (parts) unloaded from the radial and axial forces. These include the central wheels and their bearings and shafts.

Fig. 11.10 shows the strength in engagement of the planetary gear type $\overline{\mathrm{AI}}$, and Fig. 11.11 – in the wave gear.

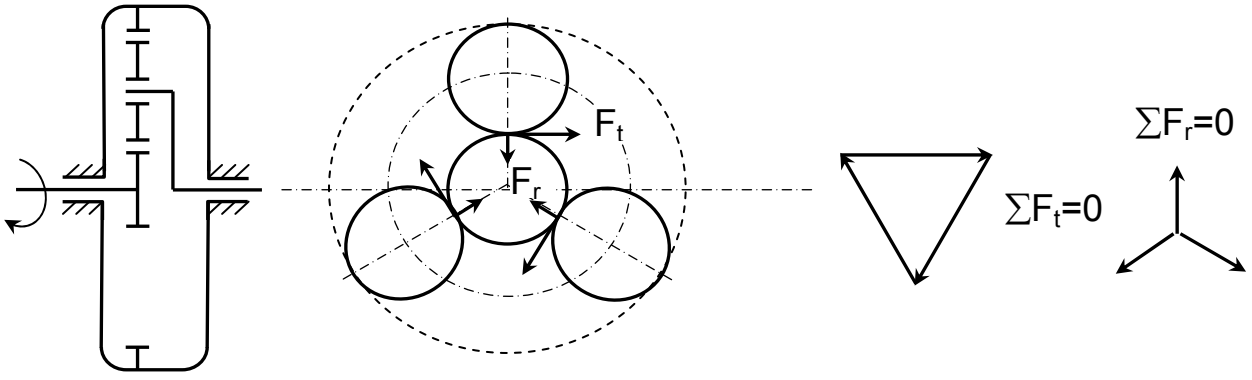


Fig. 11.10. Planetary Gear Driving Forces

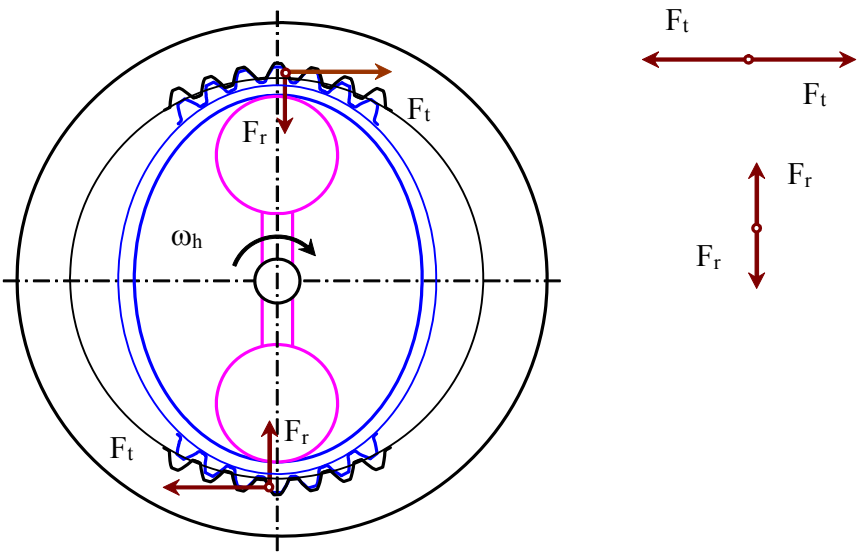


Fig. 11.11. Wave Gear Driving Forces

It follows that in the planetary gear the pinion drive shaft, as well as the carrier and bearings are not loaded, and in the wave gear – a generator shaft and flexible wheels and their bearings are not as well.

Features of Calculation and Design of Planetary Gear.

Consider procedure of **selection the number of teeth** for the planetary gear box with the closing stage (Fig. 11.12).

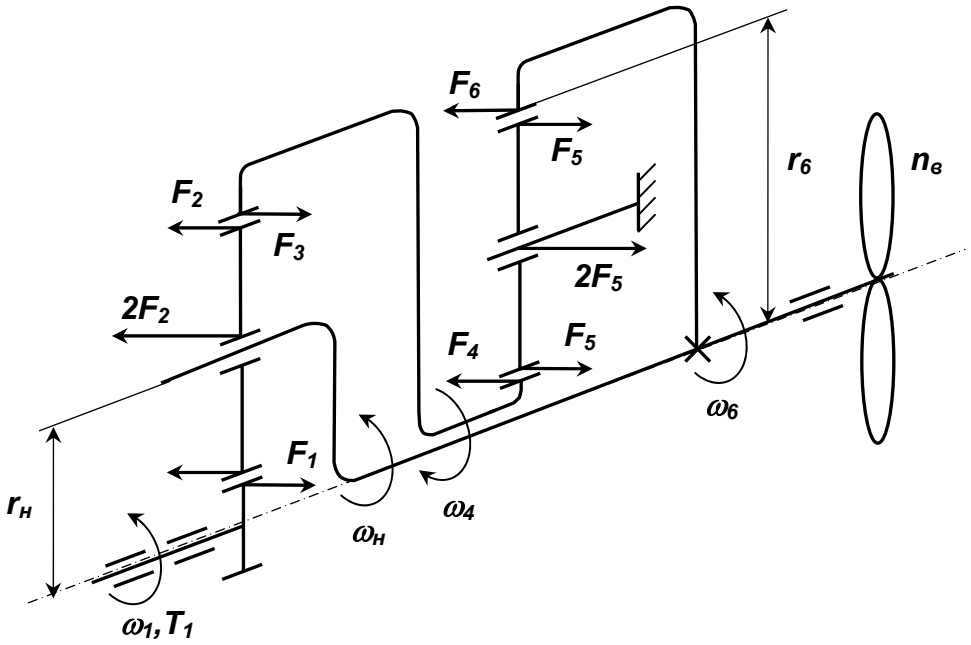


Fig. 11.12. Distribution of Forces and Moments in Closed Planetary Gear

According to this scheme are made the main gearboxes of helicopters Mi-6, Mi-8, Mi-24 (for a similar scheme – the gearbox of helicopter Ka-25K), as well as aircraft engines AI-20, AI-24, TV3-117.

The system of equations defining the required engagement between the teeth of the gear mechanism scheme is [5]:

1) gear ratio equation

$$i_{1H} = i_{16} = 1 + \frac{z_3}{z_1} + \frac{z_3 z_6}{z_1 z_4};$$

2) coaxiality condition

$$\frac{z_1 + z_2}{2} = \frac{z_3 - z_2}{2};$$

3) the condition of the neighborhood

$$(z_1 + z_2)\sin\frac{\pi}{k} \geq z_2 + 2;$$

4) assembly condition

$$\frac{z_1 + z_3}{k} = N,$$

where N - is any integer,

k - number of satellites (gear train).

On the basis of this system are received general equations to determine the number of teeth of planetary mechanism type $\overline{AI}$ (or a similar chart of a simple mechanism with fixed wheels z_5 – gear train)

$$z_1 = k\frac{N}{i_{1H}^3}; \quad z_2 = z_1\frac{i_{1H}^3 - 2}{2}; \quad z_3 = z_1(i_{1H}^3 - 1).$$

Determination of Forces and Moments

In consideration of the closed planetary gear stage 1 is planetary, stage 2 – normal transmission idler gear 5 (gear train). Part of the power is transmitted through the planet carrier to the rotor, and part – through gear 6. They are rigidly connected.

Strength on the drive central gear 1 is determined from the equilibrium conditions of the first link.

$$N_e = T_1\omega_1; \qquad T_1 = \frac{N_e}{\omega_1}; \qquad F_1 = \frac{2T_1}{d_1 k}\Omega \quad .$$

On the equilibrium condition of moments on the satellite $F_2 = F_1 = F_3$.

Under the action of the peripheral forces on the satellite a carrier rotates in the same direction as the drive gear Z_1, under the action of force $2F_2 \cdot k = F_{MR}$, where k - the number of satellites. The relevant moment and the power transmitted via the carrier are determined by the equations

$$T_{MR} = F_{MR} \cdot r_{MR} \quad \text{and} \quad N_{MR} = T_{MR} \cdot \omega_V.$$

Next, consider the balance of the block of gear wheels Z_3 - Z_4

$$F_3 \cdot r_3 \cdot k = F_4 \cdot r_4 \cdot k_1 \quad \text{and} \quad F_4 = \frac{F_3 \cdot r_3 \cdot k}{r_4 \cdot k_1},$$

where k_1 - the number of gear trains.

On gear train $F_4 = F_5 = F_6$. Force $2F_5$ acts on the wheel axle 5 (of gear train).

The total force $F_6 = F_5\ k_1$ acts on the wheel 6 by the wheel 5 and appropriate moment and the power flow is transmitted to the rotor through the wheel 6:

$$T_6 = r_6 \cdot F_6 \cdot k_1 \quad \text{and} \quad N_6 = T_6 \cdot \omega_6 .$$

In normal operation of the gear of the scheme $\omega_6 = \omega_H$.

For a differential mechanism of such a scheme the forces are determined by analog (Fig. 11.13).

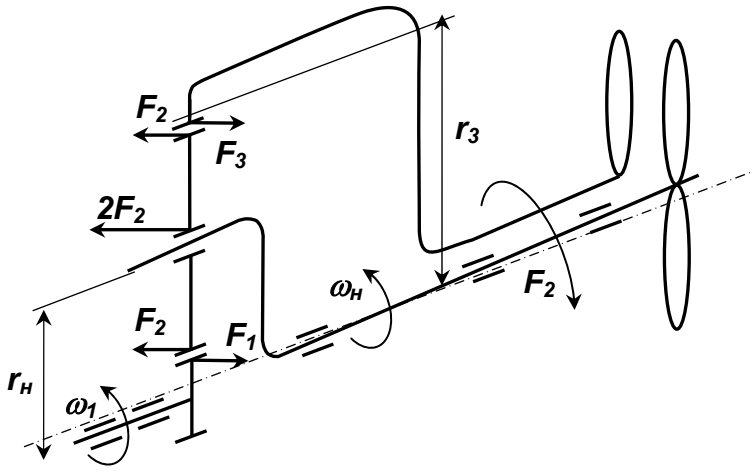


Fig. 11.13. Distribution of Forces and Moments in Differential Gear

This force acts on the central drive wheel

$$F_1 = \frac{2T_1}{d_1 \cdot k}\Omega .$$

This force acts on the satellite

$$F_2 = F_1 = F_3 .$$

The force acts on the carrier $F_N = 2F_2 \cdot k$ and moments $T_N = F_N \cdot r_n$. Force transmitted by the carrier to the front screw is $N_N = T_N \cdot \omega$.

On the back screw the moment $T_3 = F_3 \cdot r_3 \cdot k$ and the force $N_3 = T_3 \cdot \omega_3$ act.

The gear differential gear ratio is determined by the gear ratio of the reduced planetary gear (in our case $\overline{AI}$). As seen from the force analysis, and in case of equality ω_H and ω_3 moment transmitted through the carrier, is greater than the moment on the second rotor. In practice, the equality of the angular velocities of the rotor is achieved by changing the rotor pitch and number of blades.

According to the Willis's formula (with stopped carrier) and condition $n_3 = -n_n$ we get

$$\frac{n_1 - n_n}{n_3 - n_n} = \frac{n_1 - n_n}{-2n_n} = -\frac{1}{2}\left(\frac{n_1}{n_n} - 1\right) = i_{13}^{n} = 1 - i_{1n}^{3} .$$

Determine the conditional gear ratio

$$i_{1n}^{3} = 1 + \frac{1}{2}\left(\frac{n_1}{n_n} - 1\right).$$

By this gear ratio the teeth number of differential gear is selected.

11.2.2. Main Gear Boxes of Single-Rotor Helicopters

In the family of single-rotor helicopters the main gearboxes are the connecting link between the engines, the main rotor and the tail rotor.

Single-engine helicopters, in addition to the main gearbox, have intermediate and tail gearboxes, interconnected via the transmission shafts. There is a main kinematic diagram from the engine to the main and tail rotors. Furthermore, there is a branched chain of drives to various units. On the main body of the main gearbox are closed external forces and moments applied to the helicopter.

Rotational speed of gas turbines of modern jet-prop engines ranges from 6.000 to 17.000 rpm (in low-powered engines can be higher). For the greatest efficiency of MR at the rated mode of helicopter flight the MR rotational speed should be much lower than the frequency of rotation of the gas turbine, which is achieved with the help of the main gearbox.

11.2.2.1. Kinematic Diagrams and Main Parameters

Fig. 11.14 – 11.18 shows five kinematic diagrams of the main sequence of main gearboxes of helicopters Mi-1, Mi-2, Mi-4, Mi-6, Mi-8 [6]. The main parameters of these gears are shown in Table 11.2. The main gearboxes of these two helicopters are three or four-staged, and usually have two engines. Only 2 helicopters Mi-1 and Mi-4 have one engine [6].

According to kinematic diagrams gear mechanisms can be divided into three groups: simple gears; with planetary gears having single and double satellites; mixed gears, mechanisms that have simple and planetary transmission. To prevent high stress in the teeth of simple transmission, it is necessary to install several gear trains, arranging them evenly over the circumference.

The main gear of helicopter Mi-1 has a very simple scheme: it is made by a two-stage coaxial scheme (input shaft coincides with the axis of the MR shaft). In the lower part of the gearbox via bevel gear power takeoff is carried to the tail rotor. Main gearbox MG-2 of helicopter Mi-2 is three-staged. It consists of input bevel stages from two engines and two cylindrical steps which lead on both sides of the screw shaft to rotate.

The last stages of the main gears are usually composed of a coaxial transmission exhaustive search, the differential or planetary types. Their efficiency is over 97 %.

In the above main gears are used spur gears, except the first stage of main gearbox VR-8A, in which the cylindrical helical gears are used (Fig. 11.17). Bevel gears with spiral teeth are used too.

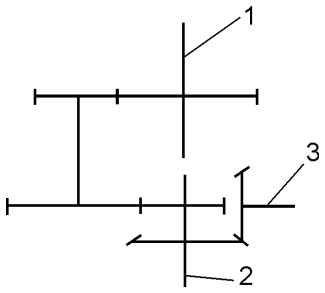


Fig. 11.14. Functional Diagram of Main Gearbox G-1:
1 – main rotor shaft; 2 – input shaft; 3 – tail rotor drive

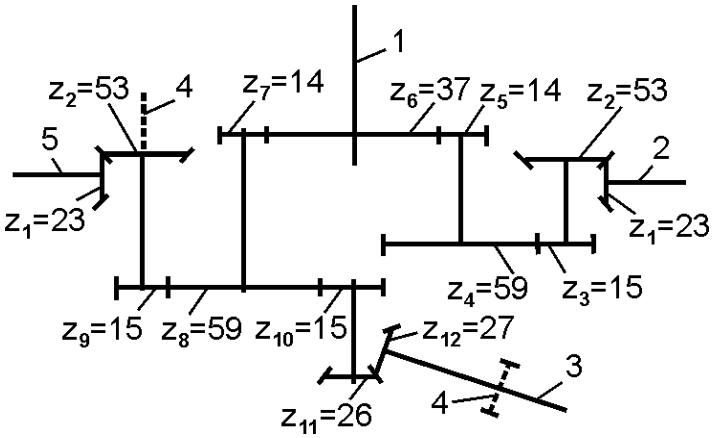


Fig. 11.15. Functional Diagram of Main Gearbox VR-2:
1 – main rotor shaft; 2 – right engine input; 3 – drive of the tail rotor; 4 – drive units;
5 – left engine input

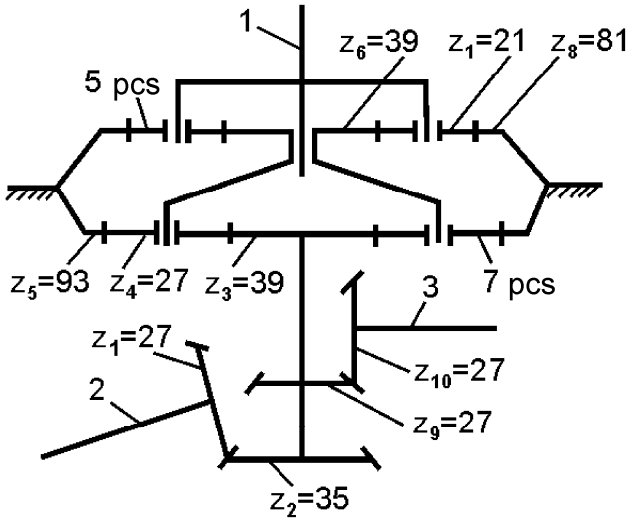


Рис. 11.16. Functional Diagram of Main Gearbox G-5:
1 – main rotor shaft; 2 – input shaft; 3 – tail rotor drive

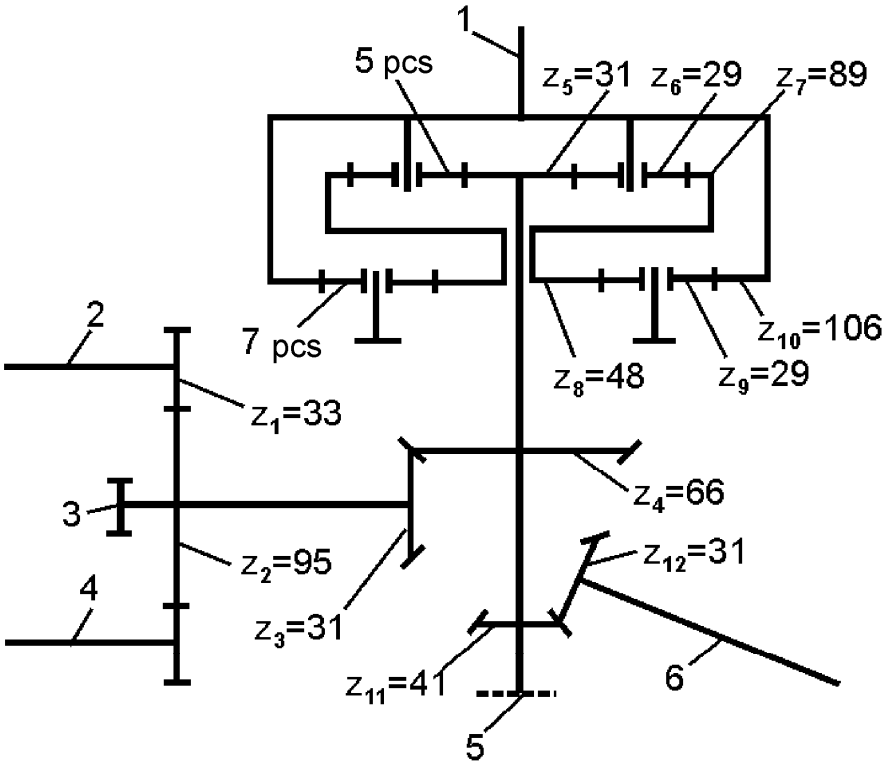


Fig. 11.17. Functional Diagram of Main Gearbox VR-8:
1 – main rotor shaft; 2 – left engine input; 3 – to the fan drive; 4 – right engine input; 5 – to the drive units; 6 – tail rotor drive

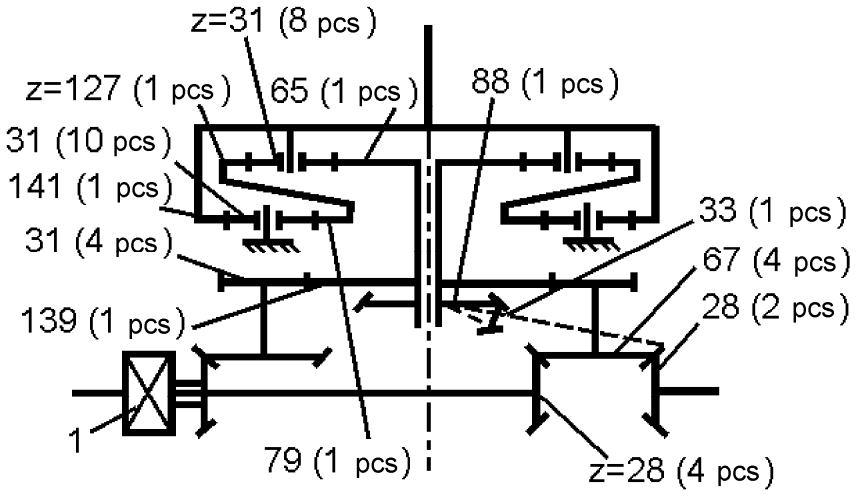


Fig. 11.18. Functional Diagram of Main Gearbox G-7:
1 – equalizing mechanism

Table 11.2

Main Parameters of Main Gearboxes of Helicopters with One Main Rotor

Helicopter		**Mi-1**	**Mi-2**	**Mi-4**	**Mi-6**	**Mi-8**
Symbol		G-1	VR-2B	G-5	G-7	VR-8A
Characteristics		A two-stage cylindrical coaxial with one input shaft	Three-stages with two input shafts-: Stage 1 – conical Single spiral; 2nd, 3rd stage - cylindrical	Three stage with one input shaft: Stage 1 - conical spiral; 2nd, 3rd stage - helical planetary	Four staged with two input shafts: Stage 1 – conical spiral; 2nd stage - cylindrical; 3rd and 4th gearbox transmission or – closed cylindrical planetary	Three-staged with two input shafts: Stage 1 – helical; 2nd stage – conical spiral; 3rd stage – a cylindrical planetary closed
Number of engines and their type		1, reciprocating	2, GTE	1, reciprocating	2, GTE	2, GTE
Frequency of input shafts, rpm		1 580	5 904	2 600	8 300	12 000
Gear ratio	To shaft of MR To shaft of TR	6.82 0.678	23.981 2.392	13.501 1.0	69.204 4.026	62.5 4.634
The torque at takeoff, Nm	On MR shaft		18 900	54 250	540 250	106 600
	Tail rotor drive		175	702	3 819	1 383

Table 11.2 (to be cntd)

Helicopter		Mi-1	Mi-2	Mi-4	Mi-6	Mi-8
Frequency, rpm	of MR shaft	232	246	193	120	192
	of TR drive unit	2 300	2 460	2 600	2 063	2 589
Duration of the gear at takeoff,%			5	5	1.25	5
The number of cycles produced by MR shaft at takeoff per1000 hours			$7.38 \cdot 10^5$	$5.79 \cdot 10^5$	$0.9 \cdot 10^5$	$5.77 \cdot 10^5$
Weight of gearbox, kg		135	300	470	3 200	750
Specific gravity of the gear, kg / N·m			0.0159	0.0087	0.0059	0.0070
Ratio of weight of gear box to the total weight of the engine			1.1111	0,440	1.333	1.136
Gear Box Dimensions						
Length m		—	855	—	1 852	1 055
Width, m		—	876	—	1 551	880
Height, m		—	1 478	1 600	2 795	1760
Diameter, m		—	—	670	—	—

In the last stages the largest module (m = 7 mm) have the gears of the main gearbox R-7. For a number of helicopter gears meshing angle α = 28º. Typically, MR shaft speed of the main gear is 120 rpm $n_{mr} \le 300$ rpm, whereas the tail rotor rotates at a frequency n_{mr} = 2500...3000 rpm or more.

In relation to a high-speed of GTE rotors the gear ratio of the main gearbox to the shaft of main gearbox can reach $\mathbf{u}$ up to 80, to the tail rotor $\mathbf{u}$ to 6. The weight of the main gearbox can be in excess of 3000 kg, and a light helicopter – about 200 kg.

If a specific gravity of the main gearbox is the mass ratio of the gear unit to the torque N·m, MR shaft at take-off mode, this parameter reaches 0.006 kg / (N·m).

The dimensions of the gears, bearings and shafts of the main gearbox are mainly determined by forces dependent of the transmission gear torque. Therefore, the main gearbox weight is calculated by the formula

$$m_{mg} = k_{mg} (M_t)_{mr}.$$

For gear units of the same size, and similar schemes have similar values of gear ratios k_{mg}, the ratio can be regarded as comparable. From the analysis it follows that with a decrease of the transmitted of gear torque weighting factor k_{mg} increases. This is explained by the wall thickness at major parts in small gearboxes are relatively big as a result of technological difficulties in the manufacture of parts with very small wall thicknesses, and for reasons of providing the necessary stiffness and static stability of walls. Because of this, parts mass of small gears is relatively higher. To reduce this effect, it is advisable to make such gearboxes by simpler schemes, in particular with the transmission output torque by a smaller number of engagement points.

It should be iterating to have each coupling or elastic element on each gear, which allows to assemble with guaranteed transmission gap and ensure a uniform load of all gear trains.

In the case of combined use it is rationally to use gearbox planetary gear in a second stage, which reduces the speed of the carrier and centrifugal forces, acting on bearing satellites.

11.2.2.2. Main Gearbox of Helicopter Mi-8 – VR-8A

The main gearbox of helicopter Mi-8 (Fig. 11.19) is designed to transmit torque from the engine to the main rotor of the helicopter, as well as to drive the units mounted on the gear unit [4].

Lowering the speed from the main gearbox is achieved by using three stages of reduction.

The first stage consists of two cylindrical helical driving wheels which are driven by motors and engaged with the third driven gear common to them. The second reduction stage consists of two bevel gears with helical teeth. The third stage of reduction is made by closed differential mechanism scheme, consisting of a differential and a differential closing sequence.

A number of units whose work is also possible in case of failure of the power plant is driven from the main gear drive.

The main gearbox is mounted on the roof panel of the helicopter fuselage and fixed to the assemblies of strong frames using a frame.

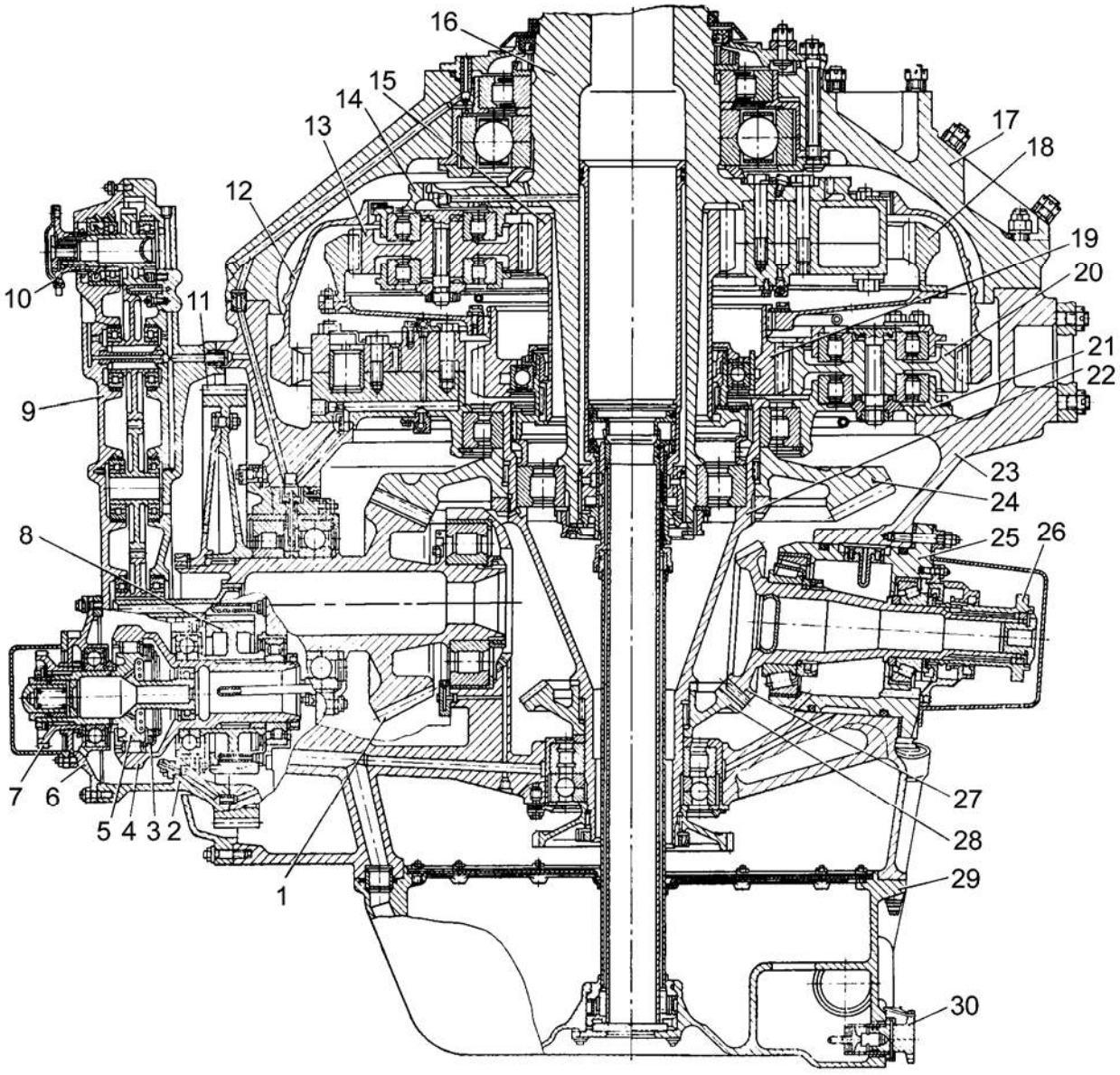


Fig. 11.19. Main Gearbox VR-8A:

1 – driving gear of the second stage; 2 – front cover; 3 – separator;

4 – driven holder; 5 – driving holder; 6 – bearing housing; 7 – bushing;

8 – driving gear of the first stage; 9 – fan drive cover;

10 – flange; 11, 24, 27 – driven gears; 12 – crown gear;

13 – sattelite; 14 – case of satellites; 15, 19 – drive gears; 16 – rotor shaft; 17 – upper body;

18 – crown gear;20 – intermediate gear; 21 – case of intermediate wheels;

22 – hollow shaft; 23 – lower case; 25 – bearing housing; 26 – slotted flange;

28 – drive gear of tail drive shaft; 29 – pan; 30 – magnetic plug

Specification

Input shaft speed, rpm. 12000
Rotor mast speed
(with counter level 95.3 %), rpm . . . . . 192±2
Gear ratio:

to the main rotor drive. 62.5
of antitorque rotor. 4.634
of fan 1.993
of pump NSh-39-1M 4.936
of generator SGO-30URS 1.497
of rotation counter detectors D-1. . . . . 5.04
of volumetric compressor AK-50T1. . . . . 5.984
of oil unit. 4.06

Oil quantity filled to a gear box (without considering filing of gearbox oil system channels), liters. . . . . no more than 32
reducing gearbox residual undrainable oil, liters 5
reducing gearbox oil circulation, l/min . . . . . no more than 140
gear box maintenance weight, kg . . . . . 785± 2%

Gearbox (see Fig. 11.19) consists of the following units: gear case, two freewheel clutches, main rotor drive shaft, rotor mast, anti-torque rotor drive connection and units drive connections.

Gear case is also a load-carrying member that transmits the load from the main rotor to the fuselage.

Reduction gear casing is casted of magnesium alloy. It consists of gear box 23, box 17, rotor mast 16 and gearbox pan 29.

Reducing gearbox 23 has the armored chord with five collars for gearbox bottom frame fixing and studded flange for connection of MR shaft 16 and case 17.

The cylinder counterbore with flange for idle gear bottom box part 21 and the vertical wall with counterbore for bevel gear shaft back roller bearing 1 are made inside the gearbox.

The conic rack wheel 1 shaft bearing boxes of the second gear stage and the freewheel clutches idle shafts roller bearings are mounted in counterbore made in the box front bosses.

The box of the first gear stage rack wheels 8 and 11, the freewheel clutches and the fan covering are fixed to front figured flange. The union for oil pressure indicator mounting is screwed in the box side surface.

The horizontal web is installed in the bottom of gearbox, the vertical shaft 22 roll and ball bearing holder is pressed in central counterbore. In addition, horizontal wall has the unit drives bevel gear shaft bearing bodies installed in its counterbores. The gearbox pan 29 is attached to the flange in the box bottom.

The cylinder counterbore is made in the gearbox bottom for antitorque rotor drive box 25 installation. Both box sides have the flanges for the drives side coverings fixing. The gearbox walls have channels with spraying jets and injectors for oil supply.

The rotor mast box 17 has the cylinder counterbore and the flange in its top. The steel stepped cage is pressed in this counterbore with rotor mast radial roller and combined journal-and-thrust bearings mounted in it.

The box has the drilled holes for the bolt fixing the support flange that fixes rotor mast inboard bearing ring in the cage. The same bolts are used for inside carter part fixing from the top. The conic deflector pressed to the rotor mast compounded with conic upper covering surface forms the moisture-accumulating labyrinth. The covering has the holes for the bolts that fix the swashplate track rail.

The rotor mast box bottom part has flanges for hydraulic boosts brackets and common swashplate lever bracket fixing.

The rotor mast box has the flange in bottom for fixing with gearbox. The breather is screwed to the side hole of box to connect rotor mast cavity with atmosphere.

The gearbox pan 29 that also serves as an oil tank is installed in the gearbox bottom. The pan is formed from magnesium alloy, it has the collar for mounting to the box in the up and inside flange for grid mounting.

The pan has the casted figured web with holes inside, this wall separates cavity with heated oil that comes from gearbox case against cavity with cold oil. The pan has a row of bosses with channels for oil flow and the well for oil filter installation.

The oil return liner pressed to the central hole contains two oil tubes of oil pipeline mounted to it. The holes for oil passing from rounded канавки of the central hole to rounded space of oil pipeline are made in the oil return liner. The collar for fixing is made in the pan bottom.

The pan has the fountain formed in the front of it, equipped by oil micron filter. The collar for filling port fixing is made in the right of pan side surface. The port is equipped by gage filter and the covering for filling port mounting. The filling port is equipped by the gage filter and covering, that holds in lock by crossarm. The oil gage is mounted in the filling port for oil level control. Furthermore, the pan has the collar with hole for mounting the oil supply filler neck that supplies oil from the cooler, and two screw holes for oil temperature sensors mounting. Three magnet plugs for metal fractions catching getting from oil are placed in the side surface holes.

The main rotor drive consists of freewheel clutches, first stage cylinder gear, second stage bevel gear and third stage differentially closed gear.

The main gearbox construction includes two freewheel clutches, both including bushing 7, input shaft 5, separator with roller assembly, driven shaft 4, case 6 and oil seal parts.

Freewheel clutches are assembled in the bores of the front cover 2 that is attached to the body casing. Two cases 6 of ball bearings of coupling drive shafts in front of the cover are attached.

The drive shaft 5 of freewheel clutch is made of steel, with variable cross-section, is based on two bearings: a ball mounted in the housing bore 6 and the roller mounted within the driven shaft 4 of freewheel clutch. In front of the drive shaft 5 splines for mounting the bushing 7 are cut which, together with oil overflow ring and the inner ring of the ball bearing are mounted on the drive shaft. To prevent oil leaks from the cavity of the coupling hub seal assembly is mounted on the splined hub 7. The middle part of the drive shaft is a sprocket with 16 sites, which have a special profile with a hardened surface for steel cylindrical rollers of the separator 3.

Freewheel separator is designed for the simultaneous engagement of all of the rollers. The separator sprocket is mounted on the drive shaft and is axially movable within a small range between the split retaining rings mounted in grooves on the cylindrical surface of the sprocket. In the rear part of the separator protrusions are made that are comprising into grooves on the cylindrical surface of the drive shaft sprocket.

This joint is applied to limit the stroke of the rollers and the separator when the freewheel is disengaged.

Driven steel shaft 4 is hollow, with variable cross-section. It is mounted on two bearings: radial angular contact bearing and radial roller bearing. The front end of the driven shaft is reamed to a holder having an inner cylindrical bore.

The hub 7 is designed for the connection of a motor main drive spring with the drive shaft of the freewheel clutch. It is made of stainless steel and in front is provided with the outer spherical cemented splines, joined at the junction with the mating internal splines of engine springs.

To eliminate longitudinal vibrations of the drive springs inside a cup, pressed into the coupling bore of clutch drive shaft from the outside a spring driving out spring toward the engine is mounted.

Engagement and disengagement of the freewheel clutch occur automatically, depending on the speed of rotation of the driving and driven shafts. Upon rotation of the drive shaft there is jamming the rollers between the working surfaces of the drive shaft sprocket and output shaft holder. At the same time driving and driven clutch shafts start to rotate at the same speed (clutch engages).

When the drive shaft speed begins to decrease (that characterizes the reduction of the engine speed), and the driven shaft, due to the inertia of rotation of the rotor and the transmission, will continue to rotate and pass the drive shaft, the rollers come out of the wedge and will be set in the cavities of the drive shaft sprocket (clutch disengages).

The power of both engines from driven shafts freewheel is transmitted through the drive gears 8 of step I to the common driven gear 11.

The spur gear drive consists of two drive gears 8 and driven gear 11.

The drive gear 8 has a cylindrical crown with external teeth meshing hub and with internal splines for mounting on the driven shaft freewheel. The driven wheel 11 is composed of a gear wheel and the support disc which is pressed on by its hub on the hub of the gear, and is fastened there by bolts. Support plate increases the rigidity of the driven gear with its low mass. The hub has slots for mounting it on the shaft of the leading bevel gear of stage II in the internal bore.

The driven wheel 11 passes the reduction stage I the total torque from the two motors for driving the shaft 1 bevel gear reduction stage II. Stage II of the master gear 1 and driven bevel gears 24, and the vertical shaft 22.

Bevel gear 1 is made as one part with the shaft. It is mounted on three bearings: two radial roller and single angular contact ball. Interior rear roller bearing ring is fixed to the wheel shaft nut and the outer set of the cage, pressed into the bore of the gearbox housing.

In front of the shaft drive gear are mounted the inner rings of the front bearings of the shaft with the adjusting ring between them, while in the slots – the driven gear 11 of stage I, fixed with a nut. The outer rings of the front bearing gear shaft 1 are mounted in a steel cage pressed into a cylindrical bore bearing housings. The outer ring of the roller bearing is pressed into the holder and fixed from the axial movement by a split retaining ring mounted in the holder groove. The outer ring of contact ball bearings is installed in the holder with radial clearance and fixed from axial movement by a thrust flange fixed to the housing bearing studs. This ball bearing installation unloads it from taking up the radial loads and bearing takes up only axial loads acting on the shaft of the gear shaft 1.

In front of the inside of the drive gear shaft 1 is an internal shoulder in the cylindrical bore of which involute splines are cut to install the fan drive spring.

The driven wheel 24 of stage II is made of alloyed steel and has a ring gear with *spiral cemented teeth*. In the wheel hub bore the involute splines are cut, there is *cementing* cylindrical surface for mounting the gear on a vertical shaft 22 which is made of alloyed steel and has a varying diameter lengthwise. The shaft is supported by two rollers and one angular contact ball bearings. The inner rings of roller bearings are mounted on the shaft, and of the ball bearing – on the shank of pinion gear of accessory drives. Outer ring of the upper roller bearing is installed in the central bore of the lower half of the gear housing and is fixed against axial movement by split locking ring. The outer rings of the lower shaft bearings mounted in a common steel cage, pressed into a central bore of horizontal partitions of gear housing 17. From axial movement the outer rings of ball and roller bearings are held by a special flange attached to the partition studs at the bottom.

On the outer surface of the upper part of the shaft 22 there are a shoulder thrust, cylindrical portion, area with involute splines and the threaded portion have place. On the cylindrical part of the shaft the inner ring is pressed onto the upper roller bearing and is installed on the slots of driven bevel gear 24 stage II, the items are clamped on the shaft with a nut, tightened with screws, screwed into the shaft 22 through the nut grooves.

Inside the upper part of the shaft 22 a cylindrical bore and internal involute splines are placed. In the cylinder bore the outer ring of a bottom roller bearing of shaft 16 of main rotor and the transition spline bushing are mounted. Between them a support ring is laid that limits the lowest position of splined sleeve. Moving up the slotted sleeve is limited by circlip mounted in a groove bore of the upper portion of the shaft 22. The lower shaft portion of the shaft has a collar on the outside, and a cylindrical splined area. At the top of the shaft splines is mounted a driving bevel gear 28 of the tail rotor drive, which rests on the shoulder of the shaft through the thumbwheel. The ring provides the necessary adjustment of gearing tail rotor drive. On a shaft cylindrical portion a thrust ring and the inner ring of the lower roller bearing are mounted. On the lower splined zone a drive gear of drive units carrying on its inner rings of angular contact ball-bearing in a cylindrical shank portion is installed. Installed in the bottom of the shaft 22, the parts are tightened by a nut and locked by the lock plate against turning.

Differentially – closed transmission of gear stage III consists of the drive cylindrical gear 15, five satellites 13, double gear 18-19, seven intermediate gears 20 and the bell-shaped gear 12 with internal gearing. The driving element of the differential stage is a gear 15, transmitting torque to five satellites 13 mounted on the housing 14, rigidly connected to the shaft 16 of the main rotor. From satellites 13 the power is partially transmitted directly to the main rotor shaft 16, the other part of the power is transferred to them through the double gear to seven intermediate gears putting into rotation the bell-shaped gear 12 connected to the shaft 16 of the main rotor.

The driving gear 15 of the differential gear stage is a steel hollow shaft with a bottom exterior involute splines and a top - the cylindrical toothing with cemented teeth. Splines connect it to the gearbox shaft 22 through the spline hub and is held against axial movement on one hand by internal sleeve shoulder, on the other one – by a bearing sleeve mounted in the bore of the splined bushing and secured with a special nut.

Differential satellite 13 is a steel spur gear with cemented teeth with external engagement. Satellites are mounted in housing 14 in two radial roller bearings each.

The ends of the satellite hub have splines for holding of the special bolt, which is secured to the satellite body, against rotation.

Satellites housing consists of two halves – the top and bottom, interconnected. Each half of the satellite housing is a steel disc with five cylindrical sockets for mounting the outer rings of roller bearings satellites. The upper half of the body has a crown with internal splines for connection with complementary splines of the shaft of the rotor and the outer splined crown to set the bell-shaped gear 12. Besides splined connection, the satellites body 14 are connected with the shaft 16 of the main rotor by two rows of special bolts.

Satellites are installed in the housing 14 during assembly of its halves. The inner rings of roller bearings of satellites are pressed onto a hub of the satellites and fixed with bolts, under nuts of which the supporting steel washers with the end slots are enclosed. On the inner end surface of the satellites bolts heads end slots are also located. This allows to fix the inner bearing rings of satellites on their hubs.

The double gear is composed of upper 18 and lower 19 toothed wheels, that are connected by a cover plate. The upper gear 18 is made of alloyed steel and has an internal gearing tooth ring and a flange with which the gear is connected with the wheel support flange. The cover plate is a disk in the upper part of which a connection flange for mounting the toothed upper wheel 18 is provided, in the lower one – a cylindrical hub bore, wherein the slots are cut. Lower gear is hollow, has a splined outer circumferential groove in the upper part, and the bottom – a cylindrical outer engagement toothing. In the inner cylindrical bore of the gear the outer ring of the radial ball bearing, which is a support for this gear, is installed. The inner ring of the bearing is clamped on the outer cylindrical surface of the spline bushing with a nut.

Double gear wheel is set on the lower support gear splines which are fixed with plates against axial movement, incorporated in the annular groove and fixed by screws to the inner bearing flange.

Supports are connected with the upper gear 18 on their flanges with bolts.

Lower gear drives seven intermediate gears 20, which are similar to the satellites by design.

The housing of intermediate gears 21 consists of the upper and lower halves. In the bottom half of the housing seven slots for radial outer rings of roller bearings lower intermediate gears 20 of the differential circuit closing are placed. The outer rings of the upper roller bearing of the gears are in the bores of the upper half body of gears. Both halves of the body gears are interconnected by bolts fastened to the gear housing and take up the part of the reactive torque of the main rotor, which occurs in the closing of the differential circuit.

Intermediate gears 20 are engaged with the bell-shaped gear 12 of the differential circuit closing. Gear 12 is made of steel, bell-shaped, the bottom of the ring gear has internal toothing, and the top has an internal hub with involute splines and an annular groove. Bell gear is mounted on the upper half of the body slots 14 of satellites and recorded by the axial movement of the lock. This setting gear 12 allows his centering when the gear is used.

The main rotor shaft 16 is made of steel, hollow, in the middle part is reamed in a disc with outer splines for mounting and fixing the satellite body 14. The upper part of the shaft adjacent to the disc, the collar has an outer cylindrical zone and thread. The adjusting ring, the inner ring of the composite angular contact ball bearing, the adjusting ring, the inner ring of the radial roller bearing are consistently set there and a nut screwed onto the shaft thread.

Angular contact ball bearing takes up axial and radial loads on the rotor and roller bearing takes radial loads, partially unloading from these load ball bearings.

On the shaft 16 resistant collar, involute splines and threads, designed for mounting and securing a helicopter rotor are placed.

The conical bottom ends with the shaft shank on the outer surface of which is a shoulder abutment, a cylindrical portion and a thread required for mounting and fixing the inner ring of the roller bearing of the main rotor shaft.

In the cylindrical bore of the lower part of the main rotor shaft 16 oil overflow tube is pressed and fixed which is centered in the cylindrical cavity shaft with two belts positioned at its ends. Seal of oil chamber is formed by the tube walls and the rotor shaft is provided with rubber rings mounted in circumferential grooves of the cylindrical tube belts. Inside the lower end of the pipe reinforced rubber sleeve and oil seal sleeve fixed locking ring are installed. In the underside of the steel oil overflow sleeve tube is shafted and pressed by stopper lock, on the inner surface of which six iron rings oil seal sleeve bypass oil line operate. The bypass oil line of oil seal sleeve and the pipe sleeve oil overflow rotor shaft provides radial and axial holes for oil passage.

A tail rotor drive consists of driving bevel gear 28, driven bevel gear 27, actuator housing 25, tapered roller bearings, toothed flange 26 and seal assembly.

Driving bevel gear 28 has internal splines on the hub, with which is connected to the splined shaft 22 with the gear of the gear pre-heating prior to coupling.

The driven bevel gear 27 is made together with a hollow drive roller. In front of the wheel a conical outer ring gear engagement is provided. In the front part the gear is provided with external conical toothing. On cylindrical part of the roller, adjacent to the drive gear teeth, is pressed onto the inner ring of tapered roller bearing and clamped on the roller nut locked in plate lock. Driven bevel gear on the shank roller has an outer shoulder, a cylindrical portion and a splined portion. The adjusting ring, the inner ring of the tapered roller bearing, the oil slinger and slotted flange 26 are installed there and clamped on the shank with a nut screwed into the internal thread of the shank and locked in plate lock. Duralumin plug is mounted in the inner cavity of the shaft.

Spline flange 26 has a square flange with four holes for attachment of the tail shaft of the transmission and threaded with outstripping shank. On the inner side of the flange the splines are cut.

The housing 25 of the tail rotor drive is casted from magnesium alloy, the outer flange is attached to its flange pins of back portion of the housing 23 of the gearbox. Between the flange's housings 23 and 25 adjusting ring is set. The outer flange of housing 25 of drive enclosure has mounting holes and holes with press-fit pins with bronze bushings for screw puller. On the rear end of the drive housing screwed are pins for securing the body of the labyrinth seal.

On the surface of the actuator body are handled two cylindrical belts, which centers the bore of the gearbox housing. Between the drive housing and inner surface the gearbox housing bore the annular oil body cavity installed in the injector drive housing is formed.

In the bores of the front and rear parts of the drive enclosure steel cages are pressed, in which outer rings of the front and rear tapered roller bearings are mounted and fixed. Behind the housing of the labyrinth seal is attached to the drive housing by using pins. Cylindrical bores are made in this case: the slinger is installed in the front bore and two rear, combined with a thread oil outstrip splined flange 26, form a two-stage labyrinth seal.

The fan is driven by four series external connected spur gear engagement set forth in the cavity formed by the front cover of the crankcase and the cover 9.

All the gears of fan drive are made of alloyed steel and have the ring gears and trunnions with axial holes for relief. Each gear is mounted on two radial roller bearings. Inner bearing rings are pressed onto the treated portions of the cylindrical pins of gear wheels, mounted in the outer cage of steel pressed into covers groove bore.

The trunnion drive gear has internal splines. A shaft for mounting flange 10 is splined on the driven gear, which together with the slinger and the inner ring of the front roller bearing is fixed to the shank special nut. Output of splined flange 10 of the fan is sealed by an oil deflector and two-stage labyrinth seal.

Drive gear assemblies are used to provide operation of basic helicopter systems in case of engines failure.

On the cover of the drive units disposed on the left side gear, are equipped: two D-2 encoder turns counters and hydraulic pump NSh-39M of backup hydraulic system. Alternator CGO-ZOU-4 is mounted in a housing separate flange.

On the cover of the drive units, located on the right side of the gearbox, are installed: air compressor AK-50TZ, hydraulic pump NSh-39M of main hydraulic system and auxiliary drive.

The units, arranged on the gear housing, are driven by the central gear wheel with external gear mounted on gear vertical shaft splines.

11.2.2.3. Primary Members of Mi-24 Helicopter Transmission

The Mi-24 helicopter transmission and the VR-24 main gearbox structure (Fig.11.20) do not differ much from the transmission and the main gearbox of the Mi-8 helicopter [7].

The transmission gear consists of the main gearbox, wheelcase, tail drive shaft, accessory gearbox and tail gearbox, fan drive cardan and the main rotor brake. The unique feature of the transmission gear is a wheelcase drive shaft located between the fore and mid parts of the tail rotor drive shaft and mounted with two AC generators.

The main gearbox is intended for the rotation rate reduction and torque transmission from engines to main rotor shaft, tail rotor shaft and fan system. It has three hydraulic pump drives: an air generator actuator and two revs counter actuators, and freewheel clutch to cut out an inoperative engine from the transmission gear. The complete reduction to the main rotor shaft is conducted sequentially on the three stages of the main gearbox. The main gearbox is mounted on the frame attached to the fuselage subassembly by the pins. The frame consists of eight supporting struts jointed pairwise into 4 V-shaped forks, where each has a main strut and an articulated strut.

Wheelcase is intended for the two AC generators drive and the tail drive shaft rotation rate reduction. The wheelcase is provided with a fan and pipes to cool the generators and the wheelcase with a conditioned air that blows through the ribbing case and deflector. The exhaust air goes to the atmosphere.

The accessory gearbox is intended for the torque transfer and for the tail drive shaft deflection on 45° angle. This is provided by two spiral bevels, when the rotational rate remains unchanged. The gearbox is mounted by 4 screws to the main keel beam askew rib.

The tail gearbox is intended for the anti-torque rotor mounting. The gearbox is attached by nine screws to the main keel beam rib. It has a pair of spiral bevels inclined on 90° and an anti-torque rotor blade pitch-change mechanism.

The tail drive shaft is intended for torque transfer from the main gearbox through the wheelcase, accessory gearbox and tail gearbox to the anti-torque rotor. The tail drive shaft consists of the fort part, that connects the main gearbox with a wheelcase; the mid part (from the wheelcase to the accessory gearbox) and the tip, connecting the accessory gearbox with the tail gearbox. The tail shaft is mounted on five trestles, first of which is on the fuselage, and the others are mounted to the tail beam bulkhead.

The main rotor brake is intended to cut the propeller shutdown time and to break the transmission when at parking and when performing installation and scheduled works. The break is mounted to the main gearbox structure at the tail drive shaft end, and the break drum is mounted to the main gearbox shaft disk.

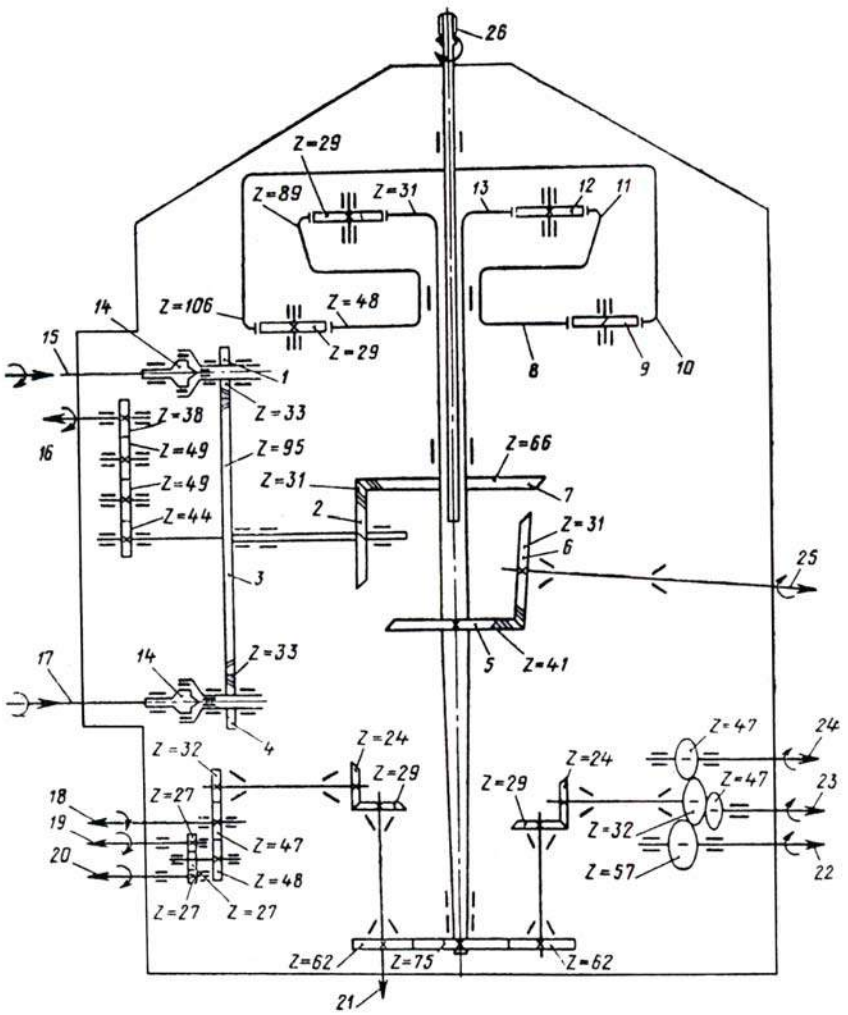


Fig. 11.20. Main Gearbox VR-24. Gearing Diagram:
1,3,4 – cylinder gear with scewed teeth;2, 5, 6, 7 – spiral bevel gears;
8, 9, 10, 11, 12, 13 – orbital differential gear train; 14 – free-wheel clutch; 15, 17 – engine inlet (n = 15000 r/min); 16 – fan actuator (n = 6033 r/min); 18 – hydraulic pump actuator (n = 2436 r/min); 19, 20 – revmeter actuator (n= 2384 r/min); 21 – oil pump actuator (n= 2960 r/min); 22 – air generator actuator AK-50T1 (n = 2008 r/min); 23, 24 – hydraulic pump actuator (n = 2436 r/min); 25 – gear box and tail shaft actuator; (n =3237 r/min);
26 – main rotor shaft (n = 240 r/min).

11.2.2.4. Mi-6 P-7 Helicopter Main Gearbox Structure

In Fig. 11.21 is the R-7 main gearbox section side view across the fan drive and the tail shaft drive connection. [4] The R-7 gear case is a primary member of the gearbox construction that takes up and transmits the load from the main rotor to the helicopter center body.

For the oil cooling the main gearbox is provided with outer oiling system, that consists of the oil scavenge section MNR-23, pipes and heat dissipator.

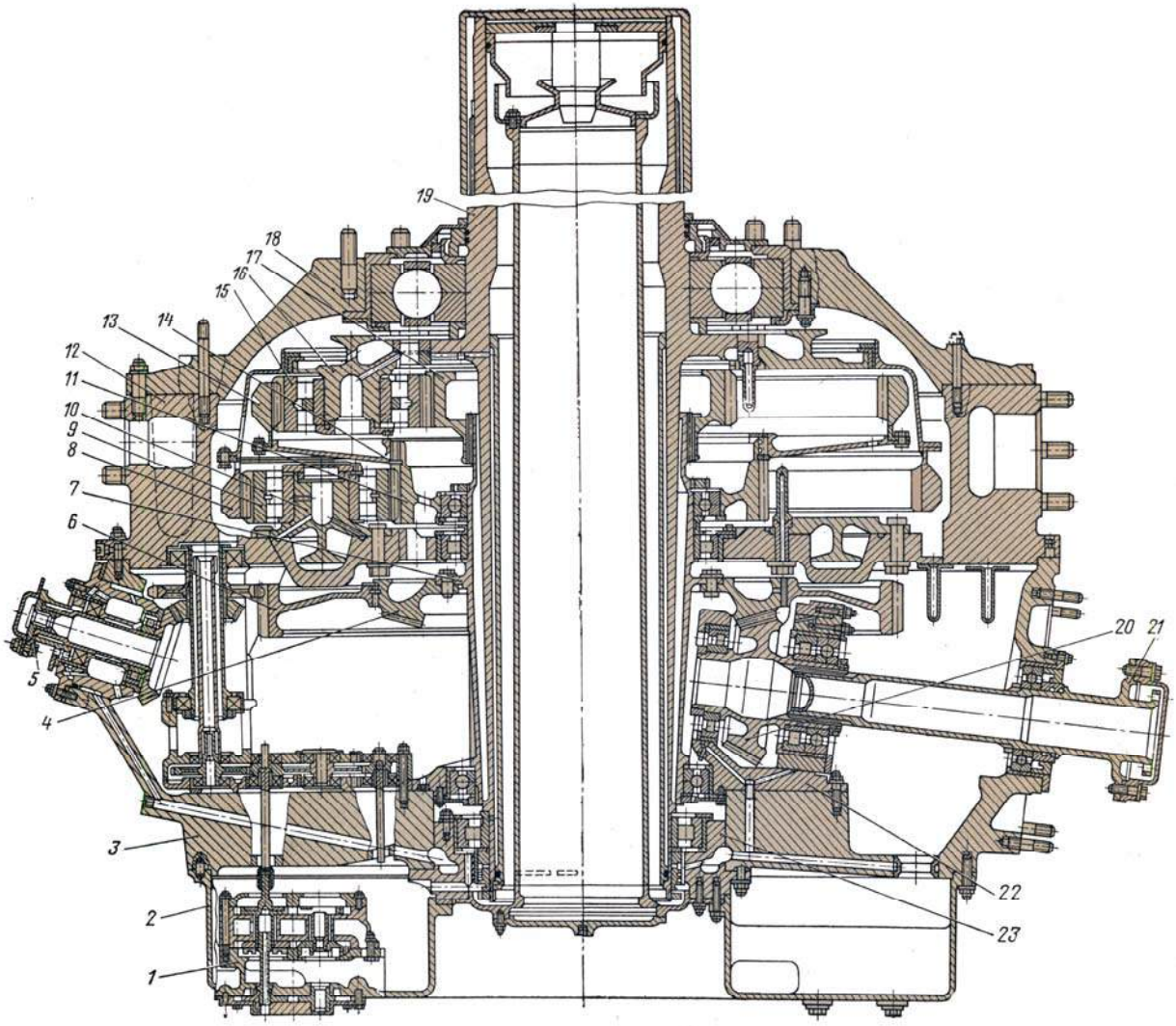


Fig. 11.21. Main Gearbox R-7 (cut away side view):
1 – oil pump unit MNR-23; 2 – oil well ; 3 – underbody section; 4 – tail shaft drive gear spiral bevel; 5 – flange adaptor ; 6 – second stage central spiral bevel; 7 – support spindle; 8 – driven gear with internal teeth; 9 – idler gear; 10 – carrier; 11 – train driver gear; 12 – center housing; 13 – planetary gear set gear rim with internal teeth;14 – interstitial rim; 15 – mate pinion; 16 – planetary gear set carrier; 17 – planetary gear set main spiral bevel; 18 – dorsal section; 19 – main rotor shaft; 20 – tail shaft driven spiral bevel; 21 – shaft; 22 – tail shaft drive body; 23 – ground base.

11.2.2.5. Main Gearbox of Mi-26 VR-26 Helicopter

Let's consider the Mi-26 helicopter transmission in details. Mi-26 is a considerably new helicopter of the single-rotor configuration [2]. Fig. 11.22 shows a gearing diagram of this helicopter main gearbox intended for load transmission from two turbo-propellers to the main rotor, equal to 22000 h.p.

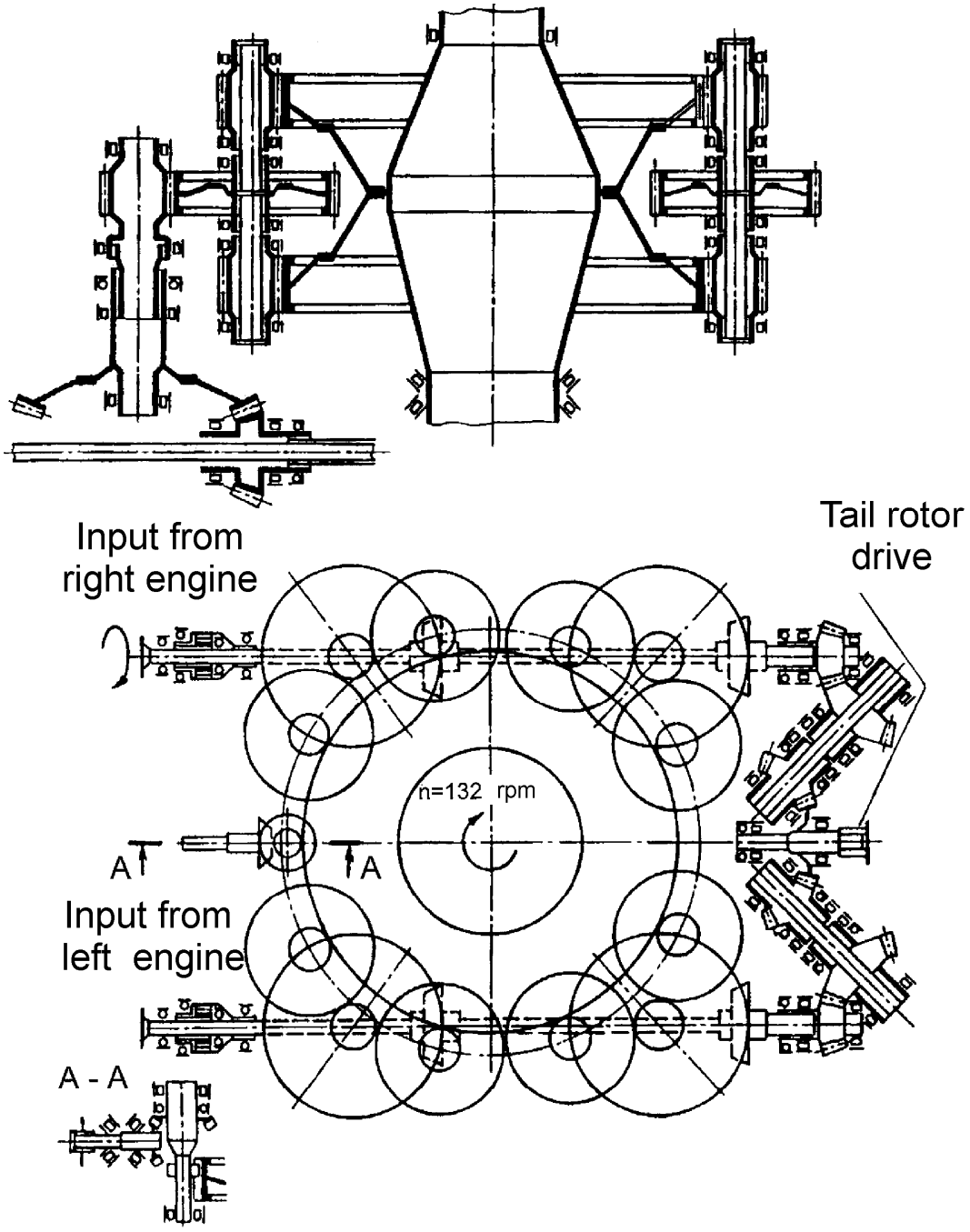


Fig. 11.22 Main Gearbox of Helicopter Mi-26. Gearing Diagram

The feature of the main VR-26 gearbox construction is a high reduction ratio in the last stage of reduction. For the first time there was used an ordinary involute gear with a high reduction ratio (i = 8.76) as a last reduction stage. The gearbox is of the unitized construction. The separate units of the gearbox, those are engine ball and socket bearing, laminar free wheel slip couplings, fore and tail bevel gear speed gearboxes, tail rotor drives, upper gear (two last stages of reduction of main kinematic chain), oil well and oil pump unit have a form of independent units in body frames. They are connected by flange adaptors and splined shafts. The construction modularity of the gearbox shortens the mass and makes production and maturation simpler.

The upper gearbox consists of the frame body with a main rotor shaft on two bearing assemblies. Just on this shaft two driven helicals are mounted with two axle-journals. Each driven helical is meshed to eight entry wheels. The screw wheels of the upper and lower rows have teeth skewed in opposite directions. Every entry wheel is boarded on two rolling-contact bearings with no thrust lands on inner races. Axial forces that appear from the last stage on entry wheels are of opposite directions and taken up by tube ties. In the issue, there appears a kind of v-gear, with two halves of the entry wheel mounted on its bearings. The possibility of the screw wheel groups of free axial deflection provides an opportunity to divide equally the power between the last stage upper and lower entry wheels. The main rotor shaft is thin-walled and barrel-contoured in the lower part, that makes it durable and rigid when the mass is minimum.

The upper gearbox body frame takes all the loads from the main rotor including the torque effect and sends them to the helicopter fuselage through the eight-rod undergear frame (Fig. 11.23). The frame body has a girder with 6 flange adaptors, to which the frame flange adaptors are mounted.

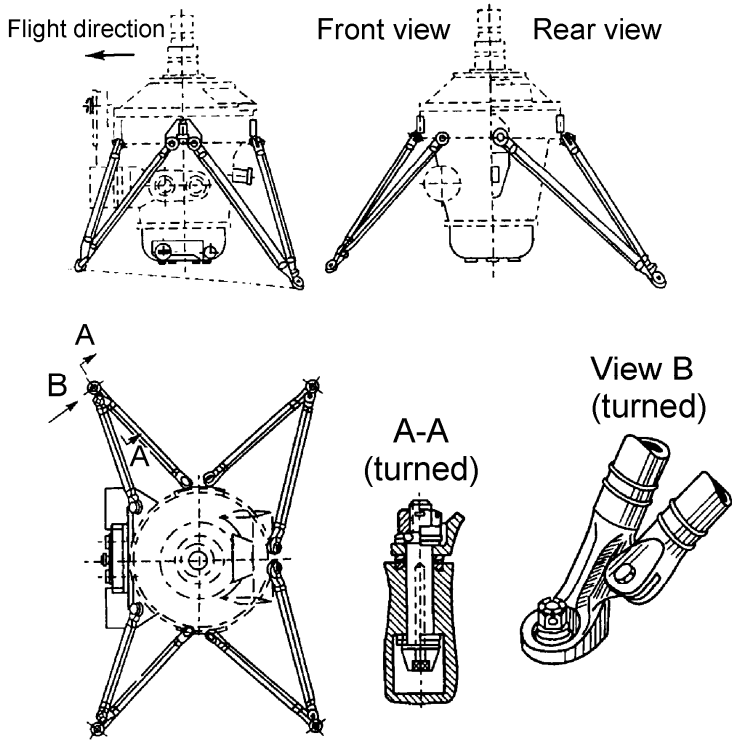


Fig. 11.23. Helicopter Gearbox Mooring

11.2.3. Helicopter Intermediate and Tail Gearboxes. Seal Assemblies and Lubrication

11.2.3.1. Intermediate Gearboxes. Tail Gearboxes

Intermediate gearboxes (Fig. 11.24) are intended for direction change of power transmission shaft axis of rotation [2] according to design of tail beam (for example, in the Mi-1 and Mi-2 helicopters a total angle between axes is $\Sigma = 150^{\circ}$). The gearboxes gearing ratio is equal to the unit.

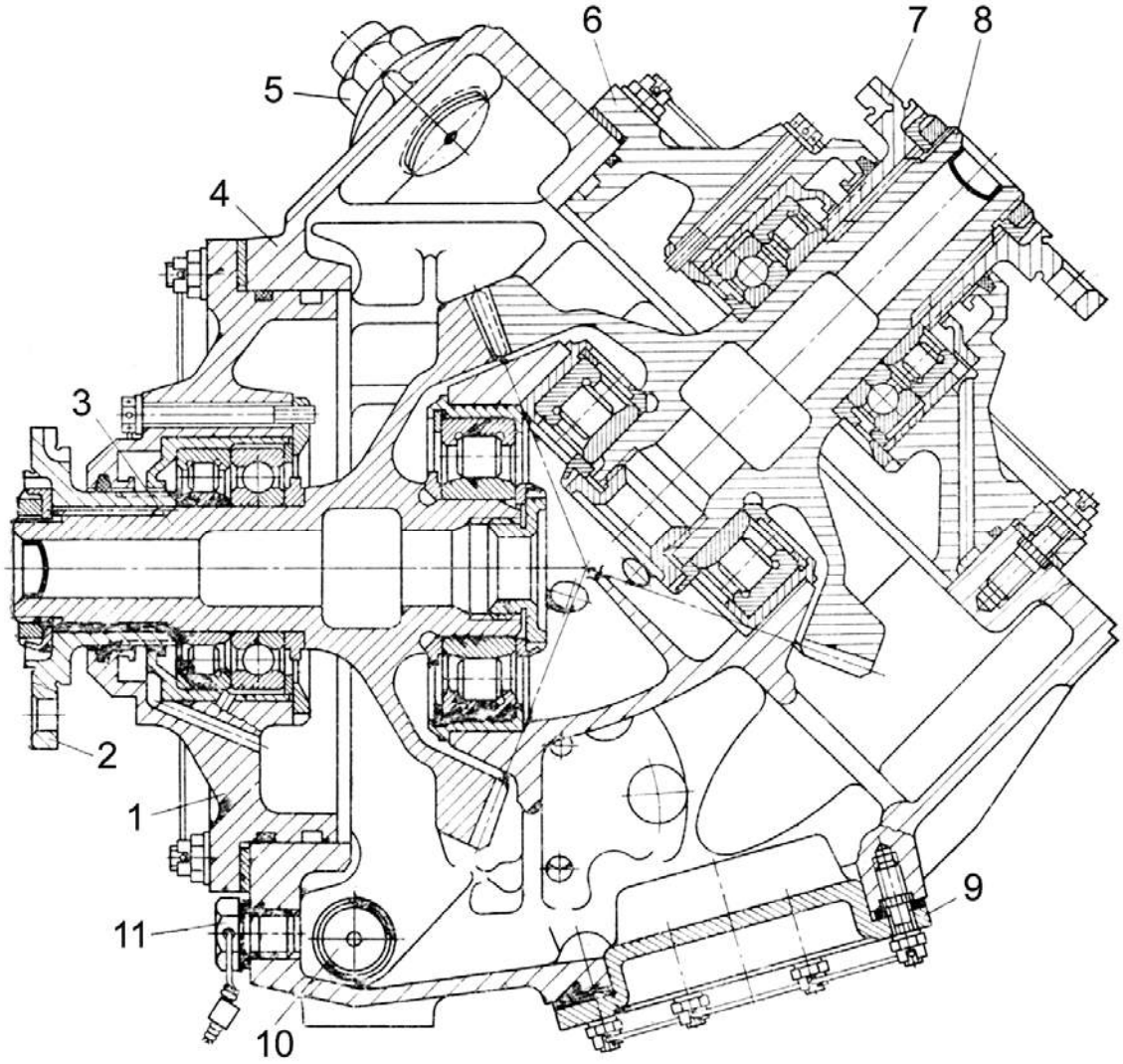


Fig. 11.24. Intermediate Gearbox of Helicopter Mi-8:
1 – pinion carrier, 2 – splined flange, 3 – pinion, 4 – pinion housing, 5 – breather, 6 – driven gear carrier, 7 – splined flange, 8 – driven gear, 9 – access cutout, 10 – magnetic sample, 11 – cover plug (the installation place of the oil temperature sensing element)

The tail rotor gearboxes are intended for power transfer to the anti-torque rotor, gearing down and change of rotation axis on 90 degrees (Fig. 11.25). Also the actuator of collective pitch control of anti-torque rotor consisting of chain wheel gear and screw gear is placed in this gearbox.

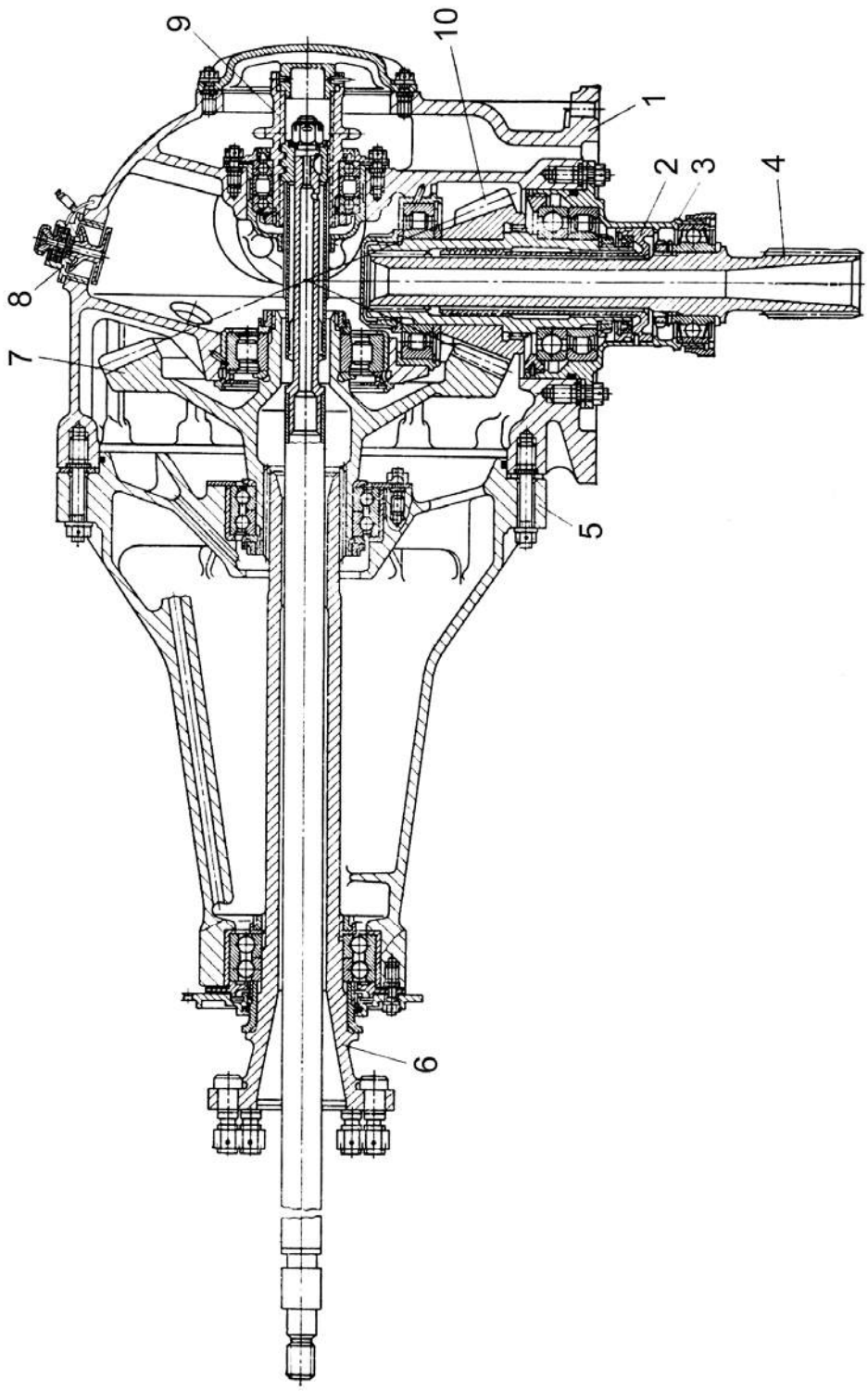


Fig. 11.25. Anti-Torque Rotor Gearbox of Mi-8 Helicopter:
1 – housing, 2 – sealing bush ring, 3 – pinion bush ring, 4 – drive shaft,
5 – housing cover, 6 –driven shaft, 7 – driven gear, 8 – breather, 9 – actuator of anti-torque rotor, 10 – pinion

Basic parameters of a number of intermediate gearboxes and tail gearboxes of the single-rotor helicopters developed by the Mil's Design Bureau, are provided in Table 11.2.

Table 11.2

Anti-torque rotor drive gearboxes

Heli-copter	Transmitted power, kW	Output shaft speed, rpm		Gearing		Weight, kg	
		Intermediate gearbox	Tail gearbox	Intermediate gearbox	Tail gearbox	Intermediate gearbox	Tail gearbox
Mi-1	31	2300	1348	1	1.7062	13.5	17.5
Mi-2	44	2460	1445	1	1.7067	12.4	17
Mi-4	86	2400	1040	1	2.3917	18.5	43
Mi-6	2096	2063	688	1,033	2.9069	108	271

The drive shaft speed from the main gearbox to the anti-torque gearbox and size of reduction in its conical pair is chosen in the process of mass minimization of the anti-torque rotor drive elements. Anti-torque rotor speed is determined for aerodynamic reasons.

The force diagram of the anti-torque rotor housing is obtained when forming mass of its gearbox and anti-torque rotor shaft. As a rule, the anti-torque rotor shaft is a shaft that transfers only a torque effect. In this case side loads and bushed rotor thrust are transferred through bearing blocks directly to the gear box. For light helicopters it is possible to accept other force diagram, and then parameters of a projective shaft of anti-torque rotor will be defined by all loads acting on it – torque moment and bending moment and anti-torque rotor thrust.

The conical gears with circular and spiral (circular) teeth (gearboxes of Mi-2, Mi-6) both orthogonal, and not orthogonal are applied as power trains of intermediate gearboxes and tail gearboxes. As a rule, these cogwheels are made with shaft (stem gear) at the same time. Hollow thin-walled with the lightened hub shafts are used.

The transmission parameters [6] (gear face, module) are chosen taking into account the expected size of structural members' deformation. The choice of size of the gearing module is influenced also by the cementation coating level, in turn, depending on allowance for grinding. Toothed wheel rim width of conic orthogonal gearing is $0.25\mathrm{Re} \le \mathrm{b} \le 0.3\mathrm{Re}$. Due to the features of gear cutting the restriction $\mathrm{b} \le 10\mathrm{m}_{\mathrm{te}}$ is also admissible.

Bending resistance of the drive wheel and driven wheel when cutting teeth is leveled by high-rise and tangential displacement of the tool.

For oblique teeth their direction should be chosen so that axial forces push out both wheels from gearing.

For the intermediate and tail rotor gearboxes that are transferring small capacities, splash lubrication is provided, i.e. one of gear wheels is partially shipped in oil and during rotation greases another gear wheel at the time of teeth contact. Besides, in the housing oil fog appears which when settling on teeth and bearings greases them. The splash lubrication system demands existence of the special pockets, catching oil, drillings in a housing and glasses through which oil moves from pockets to all bearings of the housing.

Otherwise (for example, in the Mi-6 helicopter gearboxes) force feed lubrication which stream goes to a place of contact of the teeth entering gearing is required. In intermediate gearboxes of heavy helicopters gear wheels are greased generally by

means of force feed lubrication.

Internal cavities of gearboxes are sealed by means of labyrinth devices, a oil thrower thread, baffles and special gland rings [4].

Depending on the size of the power transferred to anti-torque rotor oil can be cooled by blowing of finned external surface of the housing of its gearbox. If thus heat can not be transferred from hot oil to atmospheric air, then oil is cooled in a special air-oil radiator or by blowing of the gearbox housing with the fan.

The housings of intermediate gearboxes and anti-torque rotor gearboxes are usually made of aluminum and magnesium cast alloys.

The oil measuring with the help of oil dip stick, temperature control and breathing of gearbox cavities are provided in gearboxes.

The calculations of bevel gears can be found in special literature, for example, in work [8].

11.2.3.2. Sealing Devices

Input and output shafts of assemblies of helicopter transmissions are supplied with sealings of bearing blocks [2]. Most often sealing caps, elastic metal rings' sealings, face sealers, labyrinth two-level sealings with an oil thrower thread are used. Besides, to avoid leak of oil from gearboxes the great value is attached to sealing of joints.

As peripheral velocity of the sealed shaft of helicopter transmissions is changed from 100 m/s and more (on input shafts) to 2…3 m/s (on main rotor shafts), it is possible to apply all known types of contact and noncontact seals.

The applied types of seals, without having fundamental differences from the same seals used in various transport vehicles are characterized by smaller overall dimensions and are exposed to more careful control in manufacturing and testing.

Contact Seals

Sealing caps are applied at the shaft peripheral velocity of 10…15 m/s. Depending on in-service environment and quality of lubricant they are made of rubber and polymeric materials of various grades. Seals can be reinforced and not reinforced.

Seal consists of a cup and the bracelet spring creating certain tightness between shaft and working surface of a cup (usually preliminary tightness is created depending on the difference between diameters of shaft and opening in a cup and from action of spring).

The tightness size significantly influences the quality of seal, value of heating and wear factor of the contacting surfaces. It at the same time serves as a factor of indefinability of quality of seal assembly and dispersion of the condensing properties of a seal.

The rubber cup-type seals applied at peripheral velocities to 15 m/s require roughness of shaft surface $R_a \leq 0.63$.

Labyrinth seals with a oil thrower thread [6] work well in case if good drain from interlabyrinth cavities is organized and these cavities have a certain volume. In certain

cases to improve oil drain a special breather opening is made in the top part of a cavity. Both drain and the breather opening have to be directed to the zones protected from streams of oil, foam, etc.

Seal by elastic metal rings [9] is applied at rather big peripheral velocities (to 80…100 m/s). Seal consists of one or several spring split rings made of special grades of cast iron or bronze which are located in cavities of a shaft and due to elastic forces of the compressed ring its external surface contacts a housing face.

Seal should be treated carefully of its elements and complex technology of applying covering to rings, cavities (leading or chromium plating of rings, nitriding of cavities, etc.).

Seals with split rings are applied, for example, to input shafts of gearbox R-7 at $\upsilon_c \leq 50$ m/s. Rings are made of antifrictional cast iron. Mounting surfaces of mating steel parts are nitrated to the hardness of HRC 60 and polished. The roughness of surface is $R_a \leq 0.63$. Rings are installed so that their cuts are displaced on 120° relative to each other. Special attention should be paid to a gap in a cutting place during compression of a ring to diameter of the mating collar. During the work this gap has to provide free expansion of a ring when heating and at the same time to be minimum for leak prevention.

Joints are sealed by means of the rubber rings of round section or sealing parts which are cut out from a paronite and put in flange connectors of housings. All elements of joints sealed by rubber rings are normalized, however good work of seal depends on workmanship of elements: exact observance of allowances, roughness of surfaces, polishing the leads in chamfers of housing parts.

Grades of rubber and paronite have to correspond to quality of the applied lubricant oil (to avoid change of their properties under the influence of oil).

Face sealers seal the rotating parts with a washer which is pressed to the sealed surface on a body or to a shaft by face surfaces.

These seals differ in the following merits:

– Long service life;

– Small friction losses (by 5-10 times less, than in the cup seals);

– Less, than for other types of seals, sensitivity to radial and, if the bellow valve is available, to butt beats of a shaft;

– High tightness;

– Possibility of reliable work in any working environment at the increased temperatures in a zone of contact of sealing parts;

– Possibility of long work without greasing with the corresponding selection of sealing materials.

Seals are applied at peripheral velocities to 25 to 30 m/s.

Face sealers are more difficult and expensive type of seals, than a cup-type one, and their application has to be reasonable.

Noncontact seals

In helicopter transmissions both static and dynamic contactless seals are used.

Static contactless seals (slot-hole, labyrinth ones) reduce a gap between the sealed parts and by that interfere with big leaks.

Dynamic contactless seals (spiral, impeller ones) also do not eliminate a gap completely but the parts are sealed by "throwing away" the liquid from a gap, that requires certain power consumption.

As the problem of oil leak from internal volumes of units of helicopter transmissions is particularly important to specifics of operation of the helicopter and use of expensive grades of oils, the preference is given to contactless seals of dynamic type or their combination with other types of seals.

In Figs 11.26 and 11.27 examples of sealing assemblies of helicopter transmissions for a number of helicopters are given. The nomograph which is represented in Fig. 11.28 is convenient to choose the seals.

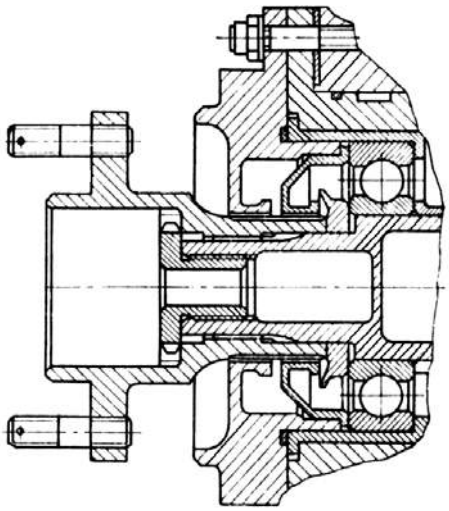


Fig. 11.26. Dynamic Contactless Seal of Fan Drive Shaft

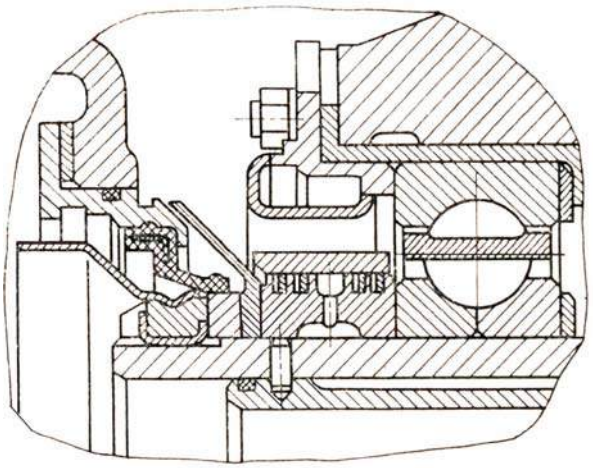


Fig. 11.27. Combined Seal of Rotor Shaft

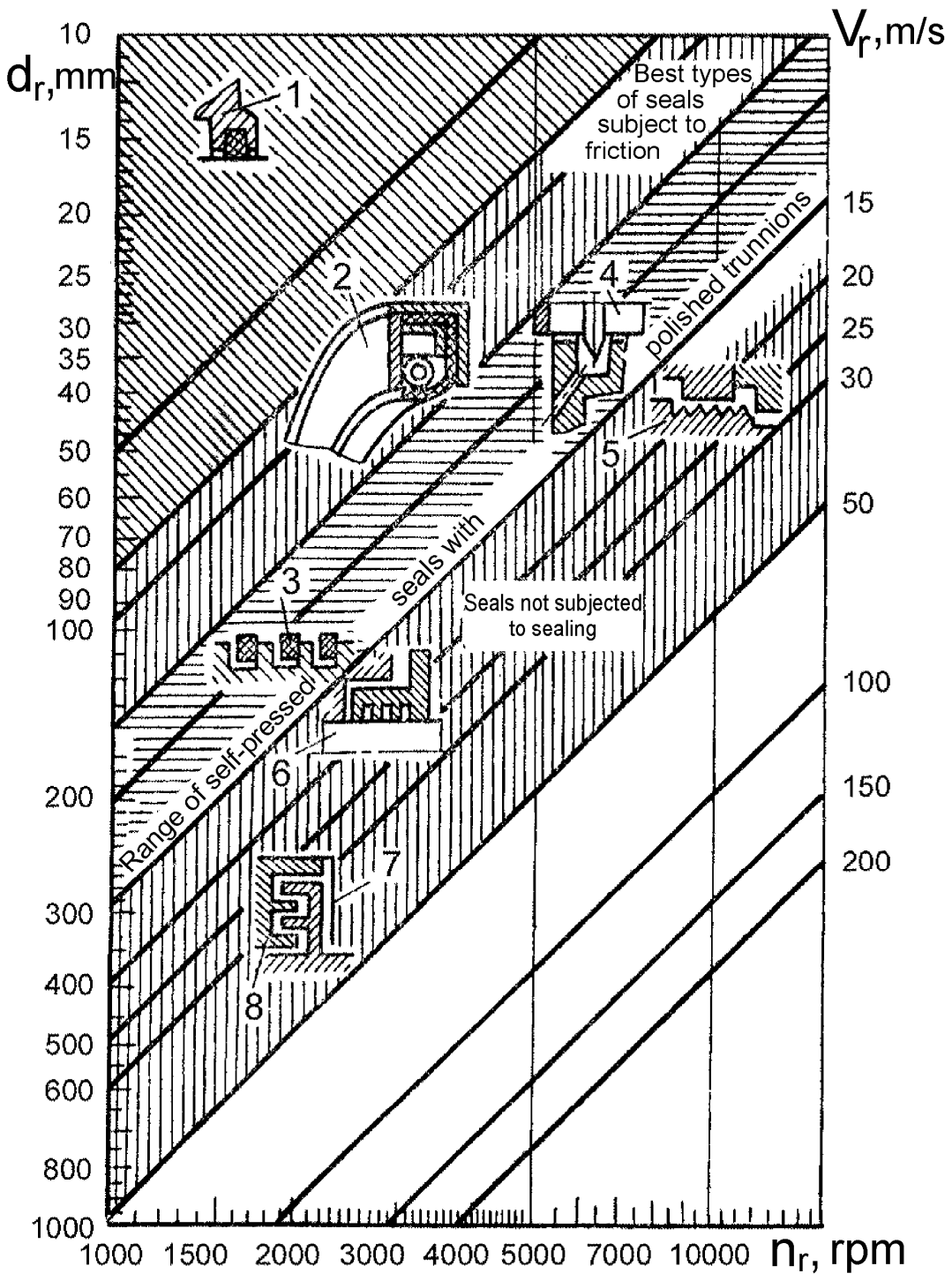


Fig. 11.28. Nomogram for Selection of Seals:
1 – felt seals of all types, including adjustable; 2 – spring-loaded seals of leather and plastics; 3 – seals of piston rings type; 4 – oil distributors; 5 – oil thrower thread; 6 – labyrinths and grooves; 7 – cover; 8 – radial labyrinth

11.2.3.3. Greasing of Transmission Elements

In helicopter transmissions parts of gearboxes, and clutches and additional supports of transmission shaft are greased.

The function of gearbox lubrication system is to supply lubricant to all rubbing parts – cogwheels, bearing supports, spline connections. Lubricants in gearboxes are intended for reduction of friction and wear of the rubbing surfaces of gearboxes' parts, rejection heat from them, prevention of corrosion and scuffs, and for removal of products of wear. Anticorrosive additives are a part of many lubricants.

The gearbox lubrication system should ensure reliable functioning of mobile elements at all operating modes of engines and flight of the helicopter in various climatic conditions. It should be tight, has low lubricant consumption and hydraulic resistance, be convenient in operation.

The lubrication system of transmission is usually made autonomous for each gearbox (main, intermediate, anti-torque).

Main Gearbox Lubrication System

Open single-circuit or double-circuit circulating closed lubrication systems are most widespread in the main gearboxes [2, 4].

In the single-circuit closed system oil circulates in circuit "tank – gearbox – tank". In double-circuit lubrication systems in one circuit oil is supplied to greased elements of the gearbox according to the scheme "tank – gearbox", and in the second one – elements are cooled according to the scheme "tank – gearbox – tank". The lubrication systems of the RV-ZF, R-7, VR-8, etc. gearboxes are made according to the double-circuit scheme.

We will consider the single – circuit closed system. In gearbox R-26 (Fig. 11.29) oil from the drip pan of gearbox 1 which at the same time is a tank of system, having passed the filter detector 19, comes to the oil pump 4 which supplies oil for greasing of gear wheels and bearings of the gearbox.

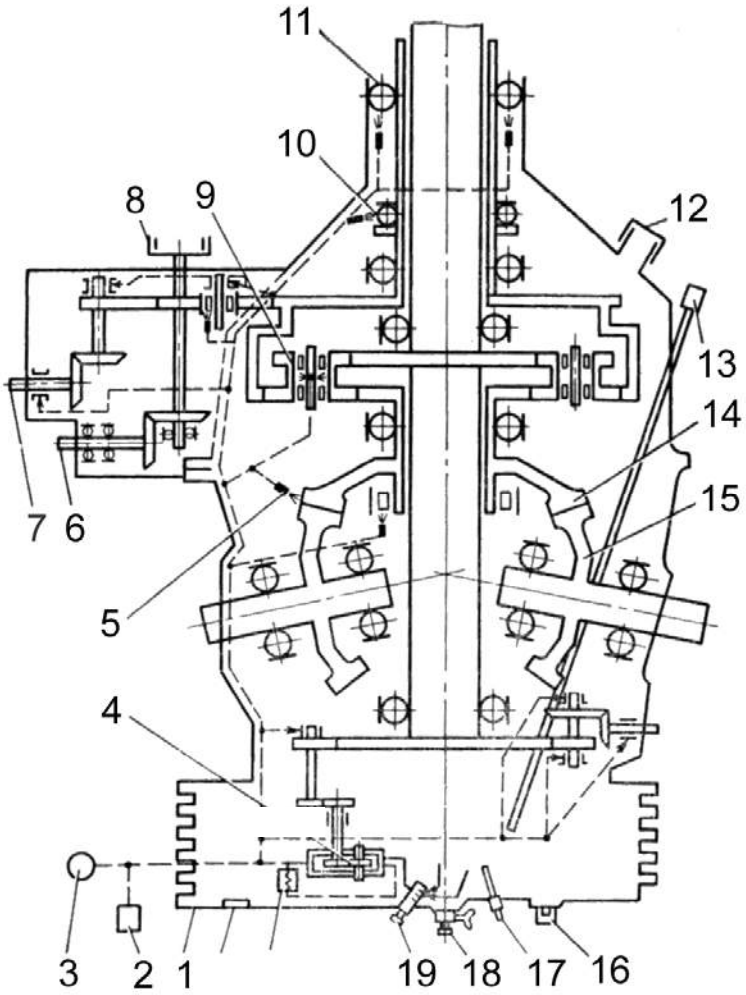


Fig. 11.29. P-26 Gearbox Lubrication system:
1 – gear drip pan; 2 – minimum pressure sensor alarm; 3 – oil pressure pointer;
4 – oil pump; 5 – nozzle; 6 – drive generator;
7 – speed sensor drive; 8 – brake; 9 – slug; 10, 11 – bearings; 12 – breather;
13 – dipstick; 14 – driven bevel gear; 15 – driving bevel gear; 16 – magnetic tube; 17 – oil temperature sensor; 18 – drain valve;
19 – filter indicator; 20 – reducing valve; 21 – maximum temperature sensor alarm

Oil under pressure is supplied to the teeth of the drive and driven bevel gears 14, 15, gear unit 9, two upper bearings 10 and 11, needle bearings of gear unit, roller bearing of the driven gear 14 and all friction bearings, and the rollers of the drive box and pinion gear of speed sensor actuator. Other parts of the gear are lubricated by oil spraying.

Hot oil is drained into the oil sump, which housing is provided with ribs, that improves heat exchange.

Oil is poured into the gearbox through a breather 12, the level of the oil is measured with a dipstick 13. Oil is drained through the valve 18 and the bottom of the gearbox drip pan.

During operation gearbox components heating and foaming of the hot oil leads to increased pressure inside the crankcase, and possible leak in planes of housing connectors. The pressure increase inside the gearbox casing deteriorates the work of

oil pumps. To avoid this, the internal cavities of helicopter gearboxes crankcase are connected with the atmosphere through the breathers made as cast or welded connections. Inside the tubes are labyrinths allowing the air come from the crankcase, but preventing the release of the oil droplets into the atmosphere.

Lubrication system operation is controlled by a temperature gauge pointer 17, a maximum temperature alarm sensor 21, a pressure sensor pointer 3 and a minimum oil pressure sensor 2. To detect the chips in the gearbox drip pan filter indicator and the magnetic tube are installed.

Fig. 11.30 is a schematic diagram of the lubrication system of the main gearbox VR-8, which presents all the elements inherent in the lubrication system of helicopter gear boxes [4].

This lubrication system has two ducts: inner duct of lubrication supply into the gearing and bearings (the so-called injection system) and outer cooling duct. The oil sump is divided into two compartments - a hot one, in which oil is drained after passing through a gearbox pump and from which is supplied to the radiators, and a cold one, after which oil is supplied from the radiator and is taken for supply to the pump injection system.

Oil supply to all points requiring lubrication and cooling is provided by gear pumps with given capacity and special ramified channel system. Expiration of oil going through calibrated (on consumption) and strictly oriented in space nozzles or jets.

The nozzle typically provides several lubrication points, for example, supplies lubricant to a ring gear. In this case the uniform spacing between the feed points in the engagement zone is provided. Nozzle is a small piece with a hole arranged axially for feeding oil. The diameter of the holes in jets or nozzles may be from 0.8 to 1.5 mm.

Oil into high-speed transmissions is usually supplied at the outlet of engaging gears, in transmissions with periphery speed of 50 m/s – at the inlet into engagement.

While lubricating in some cases the jet direction of current oil supply is of great importance, especially when oil supply into the gap between the cage carrier and the bearing ring or the jet at a considerable distance from the place where the oil should be supplied. In these cases, in manufacturing the jet sprayer the jet direction is carefully controlled.

When assigning the number and diameter of holes for oil flow, the experimental data on the costs, dependence between the pressure and temperature of the selected oil grade and the nozzles of different diameters are used.

Jets made in the form of hollow cantilever rods are usually variable section increasing towards the bottom. This is done to increase the frequency of natural oscillations of the jet and reduce stress in the dangerous section. Since the gearbox is the source of a wide range of vibrations, each such jet needs to be checked for natural frequency to avoid resonance. Because of the high sound activity of gearboxes the oil lines are made in the form of channels drilled into housing parts.

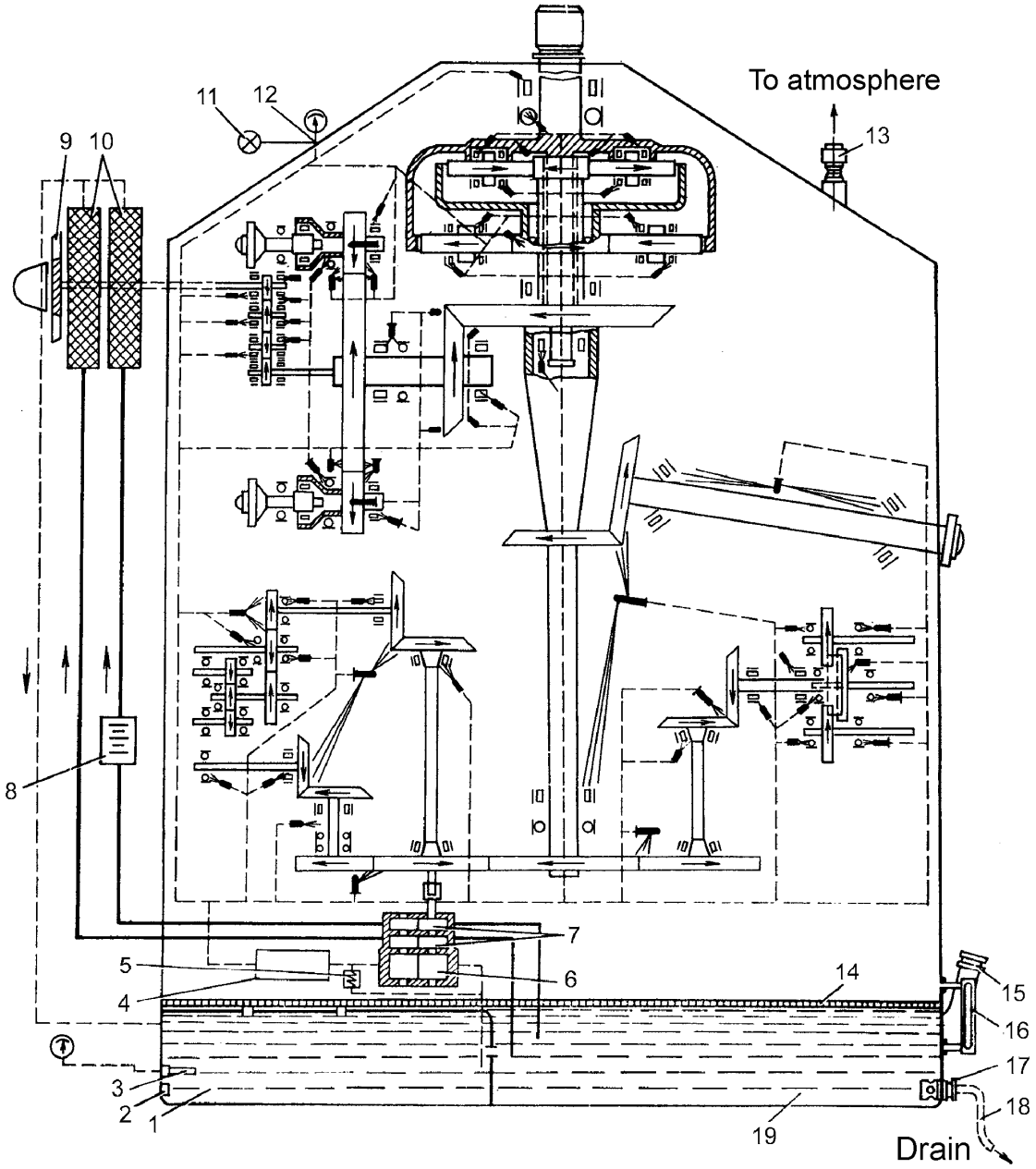


Fig. 11.30. Scheme of Lubrication System of the VR-8 Main Gearbox:
1 – oil cooled compartment; 2 – maximum temperature alarm sensor;
3 – temperature sensor; 4 – fine filter; 5 – reducing valve; 6 – oil supply pump;
7 – scavenge oil pumps; 8 – filter indicator; 9 – fan; 10 – air-oil coolers; 11 – minimum pressure sensor; 12 – pressure gauge; 13 – breather; 14 – grid; 15 – neck;
16 – oil-level gage; 17 – magnetic tube; 18 – hose; 19 – drip pan

In the internal duct after a pump the oil passes through the fine filter. The bypass valve filter is used to discharge oil into the system, bypassing the filter elements when the filter is clogged or high viscosity of oil exists in the case of start-up of gear unit at low temperatures.

Adjustable pressure reducing valve is used to maintain constant pressure, for example 350 ± 50 kPa in the system discharge at all modes of operation of gearbox over the entire temperature range. Maintaining of constant pressure in the system

allows to have accurate data on the oil flow through the jets.

Gearbox is filled with oil through the filler neck having an easily removable cover and filter. The number of filled oil is controlled by the oil-level gage on which the specified minimum and maximum oil levels are indicated.

Before the injection system intake the sensor, which controls the temperature of the oil in this system, is installed. Oil pressure in the injection system is controlled by a pressure sensor installed in a remote discharge point of the pump system, which provides a measure of pressure in the system, taking into account the hydraulic losses.

Helicopter gearboxes lubrication systems are arranged in such a way that the oil level, going to the radiators is located above the oil level coming into the inner duct. If the outer duct in the gearbox is failed, sufficient amount of oil for emergency operation is left.

Design features of the main gearbox lubrication systems can also include duplication of oil supply to the most critical units and parts. For example, oil is supplied to each load-carrying bearing at least in two points. In some gears pumps of pumping and cooling systems are duplicated, and thus they are driven separately.

Identification of the required oil pumping. When the main gearbox operates it is accepted that all the friction power in gears and bearings, is converted into heat and heat generated is rejected by oil. From this position the required oil pumping through the gearbox can be determined by the formula

$$W_g = \frac{Q_g}{\Delta t \cdot C_o \cdot \rho},$$

where Q_g – heat-to-oil characteristic; Δt – oil temperature difference at the inlet and outlet of the gearbox; C_o – oil heat capacity; ρ – oil density.

The heat amount, released in the gearbox, is determined by the formula

$$Q_g = N(1-\eta_g),$$

where N – gearbox capacity; η_g – mechanical efficiency of the transmission.

It is evident that to reduce pumping it is desirable to have a larger temperature difference Δt by reducing inlet temperatures and increasing at the output of the gearbox.

However, it is not desirable to provide the low temperature at the inlet due to increased viscosity, and high temperature should not be allowed due to overheating of gears and bearings, oil degradation and reduction of its lubricating properties.

To ensure normal conditions at steady-state operating modes temperature of the gearbox oil is maintained not higher than 120°C at outlet and inlet – in the range of 60...80°C. This temperature drop of oil Δt is usually around 40º. In these circumstances, for the applicable aviation oils C_o = 2.1 J / (kg ºC) and = 890 kg / m^3.

The total amount of oil W_g pumping includes costs ensuring reliable operation of each bearing and gear. The quantity of required pumping for each wheel and bearing is set generally through the use of statistical data.

The helicopter gearboxes oil temperature supplied to the engagement rarely exceeds 90...95°C. It should be noted that the decrease in operating temperature of the oil has beneficial effect on the operation of the unit due to the reduction of thermal deformations that modify the internal bearing clearances and violate the correct

engagement.

Lubrication ability and starting properties of the oil are largely dependent on its viscosity.

Due to the fact that the helicopter gearboxes are multistage, viscosity of the oil is chosen based on the mean operating conditions of high-speed and low-speed stages or of the operating conditions of the most critical stage. For the lubrication of the main helicopter gearboxes oil and synthetic oil with high viscosity – 4…13•10^{-6} m^2/s at 100°S or about 15...70•10^{-6} m^2/s at 50°C are used.

Some physicochemical properties of the lubricants for gearboxes are shown in Table. 11.3.

Table 11.3

Lubricant Properties

Physico-chemical properties	Lubricant		
	75%МС-20 +25% МК-8	50%МС-20 +50% МК-8	Synthetic oil Б-3В
Density at 20°C,kg/m^3	887…897	886…892	990…997
Kinematic viscosity, $m^2/s•10^{-6}$, at 100°C	11…12.6	6.5…7.5	5.0
Flash point temperature in closed melting pot, °C, not lower	165	155	235
Chilling point, °C, not higher	-22	-30	-60

Lubrication of Intermediate Gearboxes and Tail Gearboxes

Lubrication system of transmission intermediate and tail gearboxes of single-rotor helicopters, depending on the transmission power is made either of splash or forced types.

During splash lubrication the gears partially submerged in oil, in case of rotation, create oil mist in the crankcase, providing adequate lubrication of teeth of the wheels and bearings.

During forced lubrication oil is supplied to gearbox friction parts by oil pump, mounted in the housing of the oil unit. Usually, oil filters and a safety valve are mounted in one housing with a pump. In the event of failure of the oil pump back-up splash lubrication system is provided.

In relatively simple types of gearboxes (like intermediate and tail gearboxes of the Mi-8 helicopter) where periphery speeds in engagement do not exceed 20 m/s, the lubrication by spraying, or splash lubrication is applied. In this case, the oil level in the gearbox is selected in such a way that one of the wheels is immersed in oil for 15...20 mm. When rotating the gears the oil is sprayed and gets on the gear wheels, as well as special pockets to trap oil spray. Each pocket is located higher than bearing. Therefore, the oil collected by gravity therein flows to the bearing located under it.

The oil is cooled by heat exchange between the oil and the walls of the housing, which may have special cooling fins. During the helicopter operation gearboxes are blown by air flow from operating rotors.

To improve oil cooling the gearbox housings are provided with fins both outside

and inside. This particularly applies to intermediate gearboxes, external air flow blowing of which is difficult. On heavy helicopters, where more power is transferred through the gearboxes, the intermediate gears are cooled by air supplied by fan, which is mounted on the gearbox as this is done on the Mi-6 helicopter. On this gearbox air cooling fan is fed through the device between the gearbox and deflector wall. Fan wheel, molded from magnesium alloy, is bolted on the driven gear output flange. Fan diffuser is also cast from magnesium alloy, it has a straightening vanes and mounted on the gearbox housing.

For lubrication of intermediate gearboxes hypoid oil is commonly used. In some gearboxes in winter a mixture of hypoid oil (2/3 of volume) with the AMG-10 oil (1/3 of volume) is used.

As additional bearings of gear shafts the enclosed bearings, which are filled with plastic (grease) lubricating composition such as TSIATIM-201-203-221 are usually installed.

11.3. DESIGNING OF TRANSMISSION SHAFT AND MAIN ROTOR SHAFT

11.3.1. Transmission Shafts and Springs

Shafts and springs used in helicopters' transmissions vary in purpose, transmitted torque, speed, dimensions and designs. Different are the loading conditions.

Operating conditions of gearboxes' shafts, in particular helicopter main rotor shaft and, consequently, their calculation and designing differ from operation of transmission shafts, which connect the engine and main gearbox, transmit power to the intermediate gearbox and tail gearbox.

The gearbox shafts take up rotating and bending moments, shear forces and axial loads – from forces in engagements, from aerodynamic loads on helicopter tail rotor and main rotor, as well as from angular and linear accelerations during evolutions of helicopters [6]. The shafts on which the gears are mounted, at the same time carry torsion, tension, compression, and bending forces with alternating loads.

In addition to shafts loaded by torque moment, bending moment, and axial forces in gear boxes of helicopters the so-called springs loaded by torque moment only are used. The springs are torsion shafts that have splines on its ends and they are centered on these splines. Springs connect coaxial shafts together and they are used for compensation of errors of their mutual disposition.

Transmission shafts are conventionally divided into three groups [4]:

main shafts that transmit power from the engine to the gear box;

shafts that transmit power to the drive shaft, and a synchronizing shafts;

shafts for auxiliary units drive of the power unit of the helicopter (fans, accessory drives of electrical-, oil-, and hydraulic systems etc.).

Main shafts, which are usually made without intermediate bearings, consist of two bearings and a spring (Fig. 11.31). While operating the helicopter the cock (swash) at deformation of fastener assembly of the main gear box from external loads (aerodynamic forces of main rotors, stress loads at helicopter evolutions) is added to assembly cocking in the couplings of these shafts. One of the couplings of these shafts works in the area of the heated parts of the engine that is necessary to be taken into account when designing the couplings. The connection of main shafts with the main gearbox is provided with free wheel clutch.

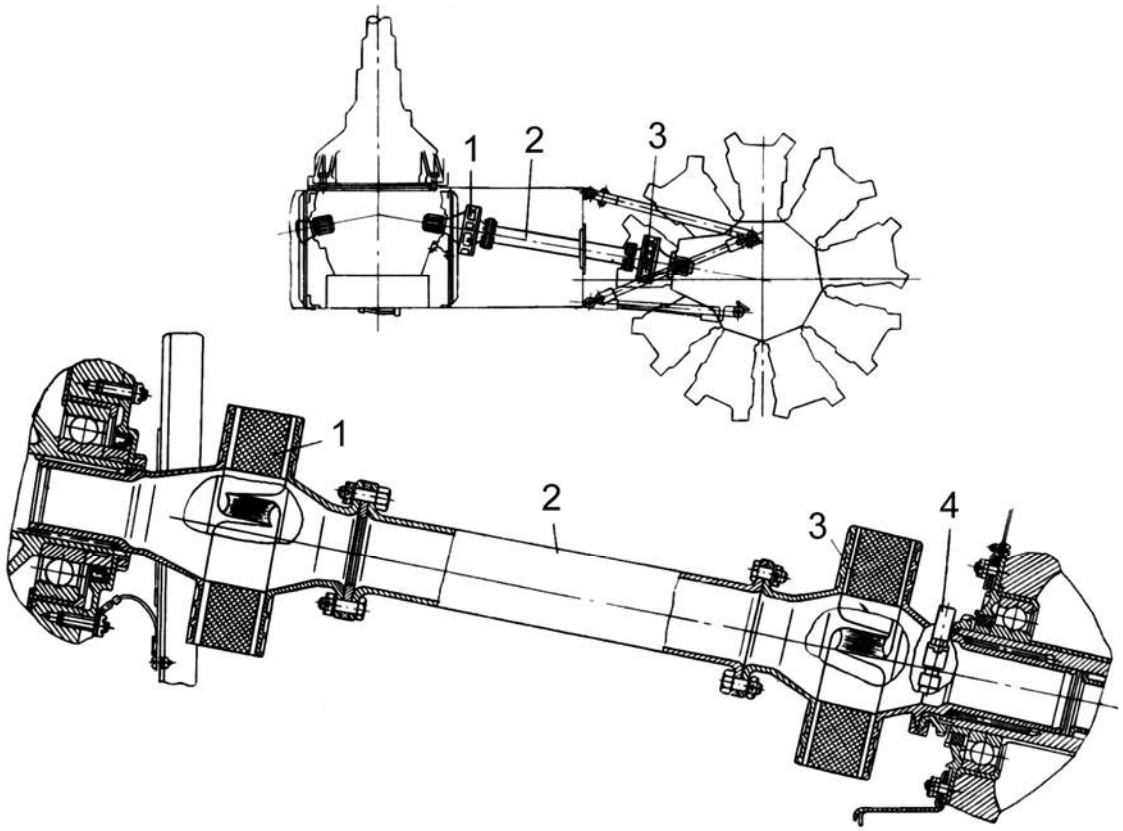


Fig. 11.31. Design and Arrangement Diagram of Main Shafts of Helicopter Ka-26:
1 – clutch of the main gearbox; 2 – spring; 3 – engine clutch;
4 – fixing strap of the engine clutch-to-output shaft attachment

Transmission shafts of anti-torque propeller transmit part of the engine power and usually have several intermediate bearings. Sectors of the transmission shaft with intermediate bearing, articulated coupling and spring are shown in Fig. 11.32, a, b.

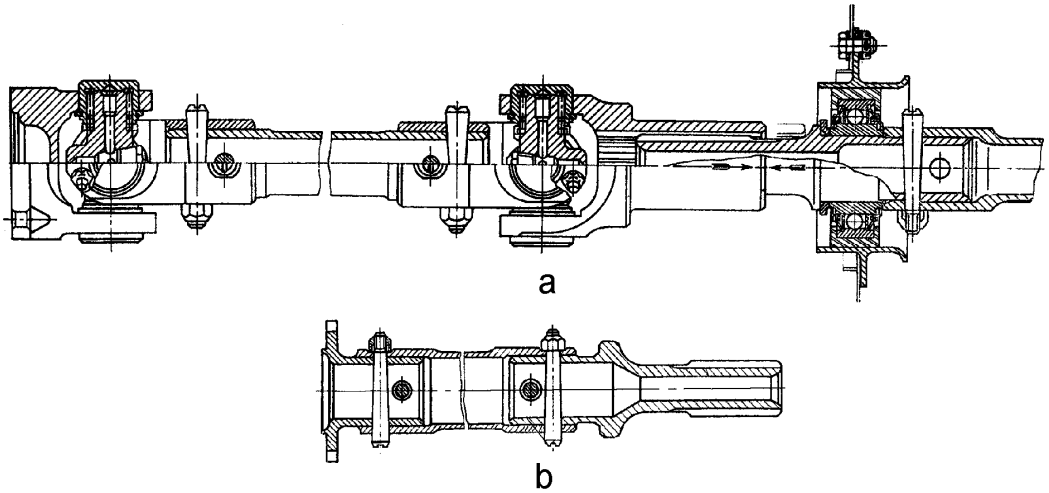


Fig. 11.32. Elements of Tail Drive Shaft:
a – front hinged part; b – end part

Such shafts are operated at a significant shift of the fastener assemblies of intermediate bearings in comparison with their installation position.

The number of bearings of the adapter couplings can be chosen according to the maximum permissible operating cocking for the selected type of couplings. Knowing the value and direction of shifts of individual sections of the shaft during operation, assembly cocking in adapter couplings opposite in sign to the cock received during operation is specially brought.

The drive shafts of auxiliary units transmit relatively small capacity and are usually made without bearings.

In general, the following loads act on the transmission shafts (TS) [4]:

- torsional (constant and variable);
- bending (constant and variable);
- compressive (tensile);
- vibration;
- temperature.

Torsional load is the main one. It defines the geometric parameters of the shafts. The constant part of the torsional load is determined by the transmitted torque moment M_t; variable part is determined by torsional vibrations arising from engines operating, gearboxes or lifting rotors and anti-torque propellers of the helicopter.

Bending stresses in the TS arise from the operation of the couplings with cocking. This operation of the adapter couplings requires the expenditure of power on friction force breaking during the movements of separate elements of the coupling relative to each other or on a periodic deformation of the resilient member of the coupling. Simultaneously, the presence of a certain bending stiffness of the coupling leads to rising of a bending moment in TS. It reduces the resource parts of the shafts brings vibrations in the connected transmission units. It is possible lower values at the expense of decrease stiffness of the couplings, the operating angle of the couplings cock, as well as rational couplings installation.

Compressive (tensile) force in TS occurs during operation when the mounting distance is changing between the joined units at the absence or unavailability of assembly that compensate changes of shaft length.

Axial force at compression of the shaft in some cases can have high values and lead to the loss of its stability.

Vibrating loads cause a general reduction of resource of TS elements, in particular the destruction of intermediate support bearings, the damaging of the splines, etc.

The usage of resilient couplings, the use of shock absorbers, resilient dampers, in intermediate supports reduces the negative impact of vibration loads.

The temperature change of TS elements at the operation may cause additional loads. Knowledge of the temperature field in the area of operation of the TV allows to predict the thermal deformation of the shafts coupling in the whole.

11.3.2. Main Rotor Shaft

The shaft of LP is loaded with forces and moments of its hub and torque moment, made at the output of main gearbox [2]. Length of LP shaft is defined on the assumption of configuration, aerodynamic and operational considerations.

By the terms of configuration the least allowable height of LP hub location over fuselage accounting blade deflection is chosen; aerodynamics defines excess of back LP on the front LP of the tandem-rotor helicopter; under operation conditions the safe altitude of LR surface is fixed when maintenance staff is in the area of the rotating propeller.

The space between coaxial propellers $\Delta y_R \approx 0,2R$ is chosen for support of necessary spare for approach, by which impact of top propeller and low propeller at all flight operation and while implementation of necessary evolution is excluded.

Forces and moments cause bend fluctuating stresses changing with propeller speed in rotating shaft. Force and moments values are defined from aerodynamic calculation of helicopter. Helicopter centering changing of which affects direction of propeller tractive effort relatively shaft axis of MR considerably has an influence on force and moments values. Value of transverse force acting on shaft is changing at the same time.

The shaft usually has stepped shape (Fig. 11.33). Such shape can be explained by irregularity of load distribution on its length and necessity at assembly of part movement to slot without sticking.

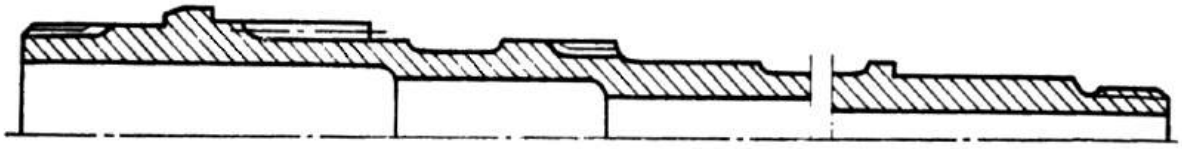


Fig. 11.33. Output Shaft of Main Gearbox of Single-Rotor Helicopter

For improvement of shaft flexural hardness besides choice of rational configuration (smooth transitions from one diameter to another, large radius of rounding) appropriate manufacturing technological techniques (see subsection 11.3.5 below).

As a rule, the shaft of MR is made in the form of single part that takes all the forces and moments. In some helicopters (for instance helicopter UAH-84) [2]) overhung shaft is divided into free-standing parts. Internal shaft (spring) is for transmission torque moment from gearbox on hub of MR. The thrust and lateral force of MR is taken by fixed relatively fuselage external wingtip shaft.

11.3.3. Strength Calculation

Gearbox shafts are loaded with forces and moments set up at the operation of transmission gears. Besides bending and torque moments set up at the operation of transmission overloads, inertial forces and gyroscopic moments can function on LP

gearbox shafts at helicopter evolutions, inertial forces and moments at instability of shaft unit.

Scheme of force impacts loading intermediate shaft of double gearbox with spur gear and helical cylinder escape wheels; with bearings like radial thrust bearing is considered as example (Fig. 11.34). Forces acting in gearing make bending moments in two mutually perpendicular surfaces xoy и xoz. Total bending moment in section

$$M_\Sigma = \sqrt{M_y^2 + M_z^2},$$

where M_y and M_z – bending moments in horizontal and vertical surfaces.

Torque moments from forces F_{t1} и F_{t2} will be respectively equal

$$T_{k1} = 0{,}5 \cdot F_{t1} \cdot d_{w1};\ T_{k2} = 0{,}5 \cdot F_{t2} \cdot d_{w2}. \qquad (11.1)$$

Extra loads acting on the shaft is estimated by the following dependencies:

$$F_i = m\rho\Omega_e^2;\ F_u = m\omega^2 e;\ M_o = k F_g l;$$
$$M_g = a J_p \omega\Omega_e;\ M_u = m e \omega^2 l. \qquad (11.2)$$

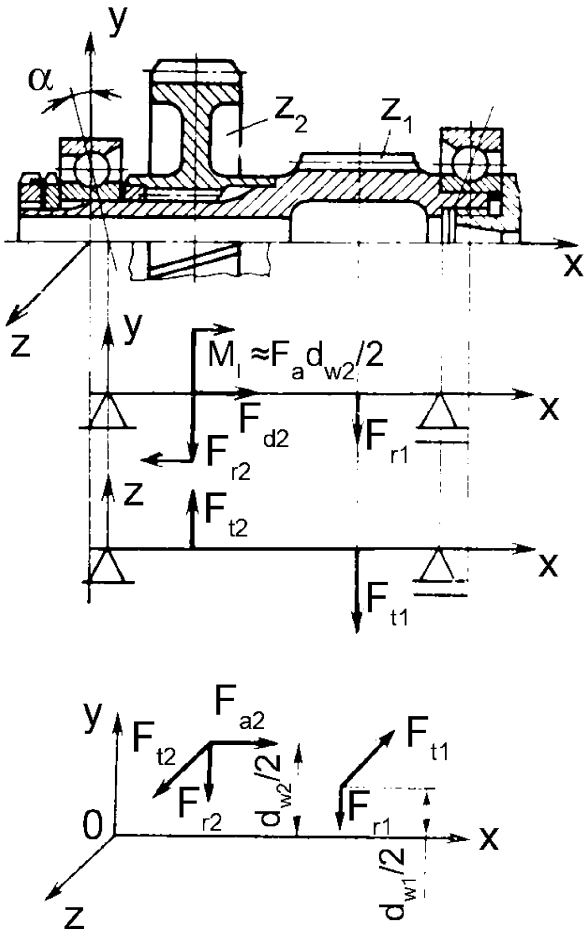


Fig. 11.34. Shaft Design Diagram

Here F_i is inertial force set up at helicopter evolution and acting on radius of curvature in the direction from center; m – shaft unit mass; ρ – radius of curvature of helicopter trajectory; Ω_e – angular velocity of evolution; F_u – inertial force from rotor unbalance ; ω – angular velocity of shaft; $å$ – eccentricity (mass center shift of rotor from the shaft axis); M_b – bending moment from overload; k – overload coefficient (for helicopters $k=2$); F_g – gravity force of shaft unit; I – arm of force; M_g – gyroscopic moment set up at helicopter evolution; $à$ – coefficient including influence of rotation angle of lifting propeller on M_g ($a=2$ – for output shafts with two-bladed propeller, $a=1$ – for propeller with three or more blades, and also for shafts with massive toothed wheels and dicks of any aircrafts transmission); J_p – polar inertial moment of propeller, wheel or dick; M_u – inertial moment from rotor unbalance.

For operability assurance of the shafts it is necessary first to observe strength and stiffness conditions. If the shaft transmits only torque moment $Ì_t$ the diameter can be found by formula

$$d = k_1 \sqrt[3]{\frac{M_t}{\sigma_{-1}(1-c^4)}}, \qquad (11.3)$$

where σ_{-1} – endurance limit of the sample from the material of the shaft, tested in conditions of simple shear with rotation; $c = d/d_0$ – the ratio of external and internal diameters of a hollow shaft; k_1 – coefficient taking into account shaft speed.

If the shaft is impacted by torque moment M_t and bending moment M_b, from the condition of strength the diameter is defined by formula

$$d = \sqrt{\frac{M_b + (\alpha \cdot M_t)^2}{0.1[\sigma]}}, \qquad (11.4)$$

where $[\sigma]$ – allowable stress of the material of the shaft in the process of stretching; α – coefficient taking into account steadiness of torque moment.

Corrected calculation of the shafts is carried out on fatigue strength [9]. Under the action of cyclic and vibration loads, inertial forces and moments, short-time overload non-reparable micro damages leading to generation of microscopic cracks in superficial layer accumulate in material of shafts and axles. Their development leads after a certain number of cycles of load variation to the appearance of macro cracks of a critical size and the subsequent fatigue destruction.

For shafts and axles at calculation for fatigue strength the following assumptions are taken: shaft is substituted by hinged beam; static and fatigue characteristics of material standard are used for calculation; distributed loads are substituted by concentrated ones; the influence of gravity force of shafts (axles) and the fitted parts are neglected; the calculation is carried out accounting constructive and technological

factors, affecting strength and endurance of section (the influence coefficient of size, shapes, surface conditions, strengthening technologies, etc.); gyroscopic effect and overload is taken into account.

The calculation for fatigue strength is carried out in this sequence:

1. Make up calculation model of the shaft (axles) (see. Fig. 11.34).
2. Define support reaction, using equilibrium equation of a beam.
3. Draw diagram of bending and torque moments.
4. Explore sections, where least reserves of strength are expected, taking into account the value of loads in accordance with diagrams and existence of stress raiser (fillets, holes, slots, spines, grooves, thread etc.).
5. Define amplitude and average value of bending stress and torsion stress, using dependencies given in course of strength of materials. At simultaneous action in rated section of bending moment and longitudinal force N total maximum stress is defined as

$$\sigma_{\Sigma max} = \sigma_a + \sigma_m = M_i / W_i + N / S, \qquad (11.5)$$

where for shafts the moment of resistance of round section of bending

$$W_i \approx 0.1 d^3 (1 - c^4). \qquad (11.6)$$

Here $c = d_0 / d$.

If stress raisers are available and reduce sectional plane the moment of the shaft resistance to bend $W_è$ is carried out by the following formulas:

a) in case of the solid shaft that has cross holes with diameter d_0:

$$W_i = \frac{\pi d^3}{32}(1 - 1.54 d_0 / d);\ W_p = \frac{\pi d^3}{16}(1 - d_0 / d); \qquad (11.7)$$

b) in case of section of key-slotted shaft in breadth b and depth t:

$$W_i = \pi d^3 / 32 - b t (d - t) / 2d;\ W_p = \pi d^3 / 16 - b t (d - t) / 2d. \qquad (11.8)$$

6. To define safety bend n_σ and torsion n_τ strength coefficient, using dependencies given in work [9].

7. Under combined action of bend and torsion total safety coefficient is found and compared with the permissible value

$$n = n_\sigma n_\tau / \sqrt{n_\sigma^2 + n_\tau^2} \geq [n], \qquad (11.9)$$

Where $[n]$ is chosen according to recommendation [9], in dependence of calculation accuracy and completeness of shaft strength check by full-scale test.

Working capacity of the shaft is defined from conditions of not only strength $\sigma \le [\sigma]$ but stiffness:

$$\varphi_{max} \le [\varphi],\ \psi_{max} \le [\psi] \quad \text{and} \quad y_{max} \le [y]. \tag{11.10}$$

Here φ_{max}, ψ_{max} and y_{max} – is maximum torsion angle, angular deflection and shaft bending or checking of the shaft stiffness by torsion angle it is necessary to use formula

$$d = k\sqrt{\frac{N}{n[\varphi]}}, \tag{11.11}$$

where k – coefficient taking into account the irregularity all loads acting on the shaft; N – transmitted capacity; n – rotary speed; $[\varphi]$ – available torsion.

11.3.4. Design and Calculation of Transmission Shafts Springs

The springs of helicopter transmissions are tubular shafts [4]. The mass of transmission shafts 60...70 % is calculated by mass of spring. Besides the mass the structure of transmission shaft, especially structure of joint assembly of spring with couplings, arrangement and structure of intermediate bearings has influence on choice of optimal ratio of external diameter of spring and wall thickness.

Since the most operational destructions of springs are fatigue ones when selecting available torsion stress and bending stress particular focus is paid attention to fatigue resistance.

The most responsible places of spring are its joints. Two types of connections of springs joints with couplings are widely spread, they are flanged (pinned) and splined. The shaft section is chosen taking into account loss of stability that can happen at destructive torque moment [1, 2].

For transmission shafts of single-rotor helicopters, considering by repeated flight tests with turning at ground, moment $Ì_t$ did not increase more than 2.2–2.6 times in comparison with moment at hovering mode.

For helicopters of single-rotor scheme the moment transmitted by tail transmission on anti-torque propeller:

$$M_{t.b} = \frac{N_{tr}}{n_s} \cdot n_l \cdot f, \tag{11.12}$$

where N_{tr} – maximum capacity transmitted to anti-torque; n_s – rotary speed of shaft, c^{-1}; f – safety coefficient, f=1,5; n_l – torque moment operational overload.

It is very important for receiving of minimum mass of shaft tube to choice optimal value $\bar{d} = d/\delta$, where d and δ – diameter and tube thickness. The thickness of tube δ, calculated by the formula satisfies this requirement

$$\delta = \sqrt[3]{\frac{2(M_t)_b}{\pi \bar{d}^2 \tau_a}}. \qquad (11.13)$$

The value of torsion available stresses τ_a for tubes made from duralumin is calculated by formula

$$\tau_a = 26.4 - \frac{0.06 \cdot \ell}{d} - 0.035(\bar{d})^2, \qquad (11.14)$$

where ℓ – distance between shaft supports.

The definition of critical rotary speed is important calculation for springs of transmission shafts of helicopters.

The frequency of natural bending vibration of shafts is found at calculations and compared with forcing frequency. The coincidence of pointed frequencies and their multiplicity means existence of resonance phenomenon that is not desired and available in operation of the shaft.

Determination of frequency of natural vibrations reduces to calculation of differential equations and finding of infinite series of natural frequencies. The practical importance has critical rotary speed that is the first frequency of natural vibrations [2].

Besides spring hardness the supports resilience and their construction, existence of axial (longitudinal) forces acting on shaft in operational condition and a number of other factors has influence critical rotary speed.

In the design process according to the calculation scheme of fixing of shaft mounted on supports some of its critical speeds are set, then in the process of the calculation for the helicopter transmission for torsional oscillations they determine the probability of resonance occurrence.

As a rule the value of first critical speed exceeds the max operational shaft speed at least 1.5-2 times. When the shaft of constant cross section fixed on two supports rotates the forms of its resilient line are rather various ones [2]. Every form of resilient line of shaft has its own value of critical angular velocity ω_t. The least value ω_t is received at first form of resilient line of the shaft. The other critical velocities are bigger than the first one in k^2 times, where k - value of half-wave resilient line, located between supports of the shaft.

The critical angular velocities at different forms of resilient lines that the rotating shaft can take are found by formulae

$$\omega_t = \sqrt{\frac{c}{m}},\ c = \frac{k_1 \cdot E \cdot I}{\ell^3}, \tag{11.15}$$

where m – mass of the shaft; c – shaft rigidity depending on its size, the location of load application, condition of shaft placement on supports and shaft material; k_1 – coefficient defined influence of shaft supports, the place of application and the character of the load; E - module of longitudinal resilience; I - inertia moment of cross section of the shaft; ℓ – distance between shaft supports (it is the length of the shaft part that correspond to half-wave resilient line with stiffness c). So for the second form of resilient lines it is necessary put 1/2, for third 1/3 etc.

For the first form of resilient line of hollow shaft with internal diameter d the critical speed is equal

$$\omega_t = 2{,}47\frac{D}{\ell^2}\sqrt{\frac{E}{d}\left[1+\left(\frac{d}{D}\right)^2\right]}. \tag{11.16}$$

Margin at critical speed, at airworthiness requirements of helicopters cannot be less than 1.2.

The shafts of big length it is reasonable to have high speed to reduce transferred torque moment ($M_t \approx N_{tr}/n$).

If the operational angle velocity of shaft rotation is less than the first critical speed $\omega_o < \omega_t$ that the shaft is called stiff (subcritical); at $\omega_o > \omega_t$ the shaft is called flexible (postcritical).

The diameter and distance between supports of subcritical shaft is chosen on the assumption of necessity of the shift ω_t out of operating range.

High critical velocities are provided by using rigid tubular shaft of big diameter at short distances between bearing supports. Similar short rigid segments of the shafts in many cases are had to joint resilient couplings in order to exclude the problem of on-center alignment of supports and shafts (if in the process of work the support construction can shift). As a result shafts system will complicate and mass increases. In order to prevent the occurrence of critical regime, an engineer reduces operational angular speed. In this case it is necessary to increase the transfer torque moment $Ì_t$ in order to save the capacity invariable, it means that it is necessary to have the shaft of larger size and mass.

When using flexible (postcritical) shaft its operational angular speed must exceed critical velocities with some margin.

The crossing through ω_t should occur as fast as it possible at condition of excess capacity. For shaft damping it is necessary to make its good balance. Please be aware that the shaft can be balanced only for one angular velocity; for other velocities as a rule the shaft is provided to be unbalanced.

Special devices limiting shaft bend at crossing through ω_t [2] are provided for construction of transmission shafts.

11.3.5. Shafts' Materials, Thermal Treatment, Design Features and Manufacturing Technique

The shafts are made from high-strength alloy steels type12X2H4A, 18XH2BA, 4OXH2MA besides electroslag remelting heat-treated up to HRC 32…38.

The main materials that springs are made are steels 40X2H2MA (40XHMA) and 18X2H4MA (18BA) as well as titanium alloy BT3-1 and aluminum alloys.

Maximum torsion stresses in springs made from steel do not exceed 200…250 MPa.

Shafts mass is significant part of gear assembly mass. That is why without exceptions all gear assemblies shafts of helicopters are produced hollow, thin-walled and stepped (see Fig. 11.33). Usage hollow shafts are due to fact that at equal moments of bending and torsion resistance in solid shaft and equal rigidity the hollow shafts give significant advantages in mass.

Where it is possible the shafts are produced together with other elements of construction (see Fig. 11.35): screw-wheels, shrouds, chain wheel of free running coupling, drive flanges.

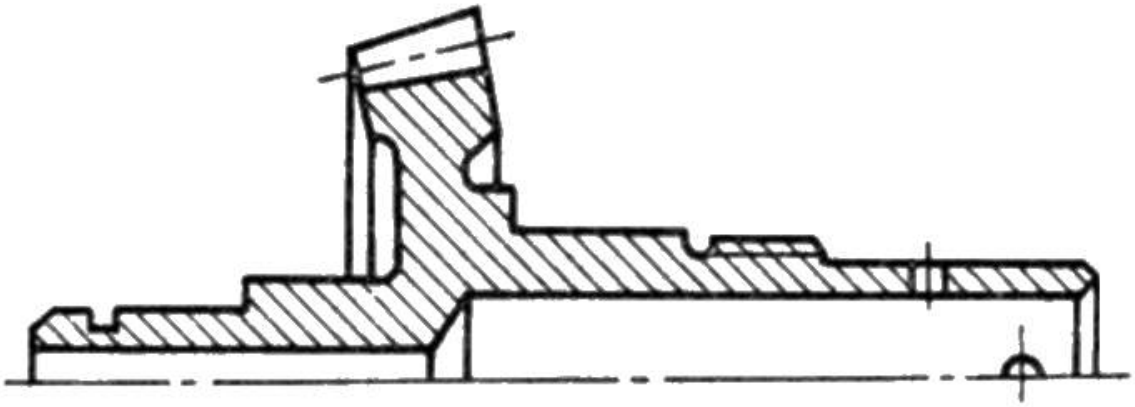


Fig. 11.35. Shaft of Angular Gear Assembly

The construction of separate part of the shaft is defined by their appointment. Next functional parts can be marked out: supporting (under the bearings), for connection of different elements, for centering, under sealing of bearing assemblies.

The shafts and axis are made from rolled metal particularly major shafts are made from forged piece. The shafts and axis seats after lathe machining are necessarily filed. Particularly major shafts are recommended to file along whole surface, and highly – stressed areas to file even in disconnected surfaces. In the process of cyclic alternating load of the shaft the roughness as microconcentrator of stresses are the source fatigue cracks. Filing, polishing and head treatment reduce the value of roughness and increase shafts live. The roughness of the shaft surfaces must not be less than R_a = 2.5 mkm and in the place of installation of rolling bearing – bellow R_a = 0.63 mkm.

To improve bending endurance of the shafts except choice of rational configuration (smooth transitions from one diameter to another, progressive changing of

section thickness, big spherical radii, etc.) appropriated technological measures of treatment are applied.

To reduce possible damages of surface layer after mechanical treatment the shafts used in aeronautical engineering are exposed to surface plastic deformation (SPD). These methods are cloudbursting, treatment by balls on vibration test (vibration strengthening and vibrogrinding), rolling the surface after finishing treatment, pneumatic peering, etc.). These methods are high-performance to increase purity of surface, to create squeezing residual stresses, to increase surface hardness and back-to-back endurance and enduring quality.

Rollers running, chasing fillets can strengthen the part of the shaft and to improve its bearing capacity into 1.5–2 times. Vibration treatment (by balls) of shafts spline sections considerably improves fatigue strength. Some particularly responsible shafts are exposed to diamond smoothing.

To reduce stress concentration the transition radius between cylinder surface and ends is made maximum one on the assumption of spherical radius of bearing ring accounting tolerance on manufacturing. To avoid defects precipitation on landing surfaces of the shafts and thrust faces their parts are cemented and azotized. In this case groove on the shaft is made. Landing surface is treated with thermochemical, but does not polish to avoid defects in zone of stress concentration.

The element of shaft can be thread on which screw nuts of bearings fixation, gear rims, flanges are screwed. As a rule the metric thread at a pitch of 1.5 mm and the angle of thread 60º is applied. The length of threaded part is within 8 to 12 mm. For the shafts with big diameters (100 mm and more) thread pitch can be increased. The threads in the spaces shafts had alternating bending stresses are avoided. Where it is impossible to avoid the shaft thickness under the thread is increased accounting stress concentration. The thread in such unsafe places is made with round hollow that can be hardened with knurl.

The upcoming trend is thin-walled shafts of a big diameter made from ductile mark of steel with the help of rolling of thick-walled rough part. The length of the shafts is limited by ability of technological equipment.

The titanium alloys are often applied for shaft manufacture. The shafts made from such alloys are light, corrosion-resistant, and reliable in operation.

At present when composition materials became widely implemented the springs made from hybrid composition materials (spiral winding of the threads of carbon fiber and organic fiber) are tried to manufacture. It is necessary to exclude the possibility of percussive defects of the shafts made from composite at installation work and at operation.

11.4. DESIGN AND ANALYSIS OF CRITICAL PARTS AND ASSEMBLIES

11.4.1. Design and Strength Calculation of Gears

11.4.1.1. Nature of Work and Types of Failure to the Gear Wheels

In the preparation of this chapter materials contained in the Interstate Standards draft "GEAR WHEELS. Types of Damage. Classification and Description" were used [10].

Failures classification of gear wheels are shown in Table. 11.4. All failures are divided into six types, which in its turn are divided into certain groups of the nature, the general appearance and degree of failure.

Table 11.4

Classification of Types of Failures

Group of Failure	Nature of Failure	General Type of Failure	Degree of Failure
Runout	Mechanical	Adhesive	Initial
			Minor
		Abrasive	Initial
			Minor
			Maximum
		Runout with Mutual Intrusion of Gears	
		Smooth Finish	Initial
			Minor
			Intensive
		Fatigue damage	Initial
			Minor
			Maximum
			Frettage
		During Load Start	
	Corrosion-Mechanical	Chemical (Oxidative)	Scaling
			Oxidation at Overheating
		At Fretting Corrosion	
	Erosional	Cavitational Hydroerosive Gas Erosion Hydroabrasive Gas-abrasive Radiation-erosive Electrospark	
Jamming	Jamming	Jamming	Initial (Localized)
			Minor
			Catastrophic

Table 11.4. (cont'd)

Group of Failure	Nature of Failure	General Type of Failure	Degree of Failure
Permanent Distortion	Implementation	Implementation	
	Structural Yielding	When Cold Flow In Hot Strength In Rolling On Impact with Gears	
	Ripple Dislocation	Ripple Dislocation	
	Formation of rigs and dents	Formation of rigs and dents	
	Formation of nicks	Formation of nicks	
	Structural Yielding at the Gear Bottom	Structural Yielding at the Gear Bottom	
	Interference Effect	Interference Effect	
Contact Fatigue	Superficial Fatigue Flaking	Macroflaking	Running-In
			Progressive
		Microflaking	
		Scouring	
		Flaking-off	
	Deep Fatigue Flaking	Deep Fatigue Flaking	
Initiation And Progression of Cracks	Heat Treatment Cracks	Heat Treatment Cracks	
	Grinding Cracks	Grinding Cracks	
	Cracks of Rim and Disc	Cracks of Rim And Disc	
	Cracks in Hardened Case	Cracks in Hardened Case	
	Fatigue Fractures	Fatigue Fractures	
Failure of Gear	Fracture of Gear when Overwork	Fragile Fracture	
		Plastic Fracture	
		Mix Fracture	
	Shear of Gear	Shear of Gear	
	Fracture of Gear after Structural Yielding	Fracture Of Gear after Structural Yielding(Continuous Destruction)	
	Fatigue Fracture of Gear	Low-Cycle Fatigue Fracture	
		Multicycle Fatigue Fracture	Fracture at Crack at the Base of Tooth
			Fracture at Crack on the Side Surface of Tooth
			Fracture at Crack at Tooth End

From the table failures of gear work of helicopter transmissions and aircraft accessory the following failures to the surface or surface layer of the gear are most common: fatigue flaking, excessive wear of gears, jam (caused by cyclically operated contact stresses and frictional forces) and broken gear- in case of overload or fatigue character.

Therefore examine in further details processes associated with these types of failure.

CONTACT FATIGUE

Represents failures, developing under the influence of surface and subsurface stress repeatedly arise in the process of gears contact. Characterized by the formation and evolution of the metal bubbles on the surfaces of gears. These failures fatigue fracture should be distinguished from failures during wear. There are the following types of contact fatigue:

- Superficial fatigue flaking;
- Scouring;
- Flaking-off;
- Deep fatigue flaking.

Superficial fatigue flaking - the phenomenon of surface fatigue, which manifests itself in gear mesh of gear wheels. During repeated impacts of contact stresses generated each time when gear enters in gear. It takes initiation and growth of fatigue microcracks on the surface or at shallow depth beneath the surface. Cracking caused insufficient thickness of the lubricant layer and metal contact of singular surface peaks of gears and also stress concentrators in the form of individual traces of the cutting tool. Cracks may arise from existing and non-metallic inclusions. Cracks are usually spread over a considerable distance in a direction parallel to the initial surface the gear wheel, and then begin to turn or to branch. With a significant increase in individual fractures and further consolidation separation occurs from the surface of the damaged layer of material and the formation of micro-pockets are usually tapered appearance. Near the borders of the bubble can be observed fatigue cracks. Usually some initial pockets spalling are arranged along a straight line on the fillet of the gear. Then there are individual pockets along another line that is separated from the first by a certain amount. Further growth is accompanied by pockets merging and the increase in the area of the damaged surface covering, ultimately, the entire space between these load lines. To improve the resistance to fatigue flaking gear teeth must have a high surface hardness and a relatively soft core. The greatest resistance to fatigue flaking have cemented gears, because after carburizing and hardening as a result of structural transformations in the metal gears emerge residual compressive stress, reducing the impact of existing tensile stresses transmitted from the disposable load.

The use of high purity surface gears, and also an abundant supply of clean cooled grease with high viscosity provides a lubricating film of sufficient thickness, it slows down the development of fatigue flaking. Improving the quality of steel by vacuum refining reduces the possible number of inclusions and improves resistance to fatigue flaking. Deposition of thin porous covers of tungsten carbide improves fatigue behavior, wear properties and corrosive resistance of the gears working surfaces. The presence of water or abrasive particles in the grease promotes fatigue flaking. Photo illustrating flaking is shown in Fig. 11.36.

To prevent fatigue flaking it is recommended to:

- reduce the contact stresses in the gears by the choice of rational geometry of gearwheels;
- use high-quality steel and cemented surfaces gears to ensure the hardness of not less than 58 HRC;
- increase the surface conditions gears;

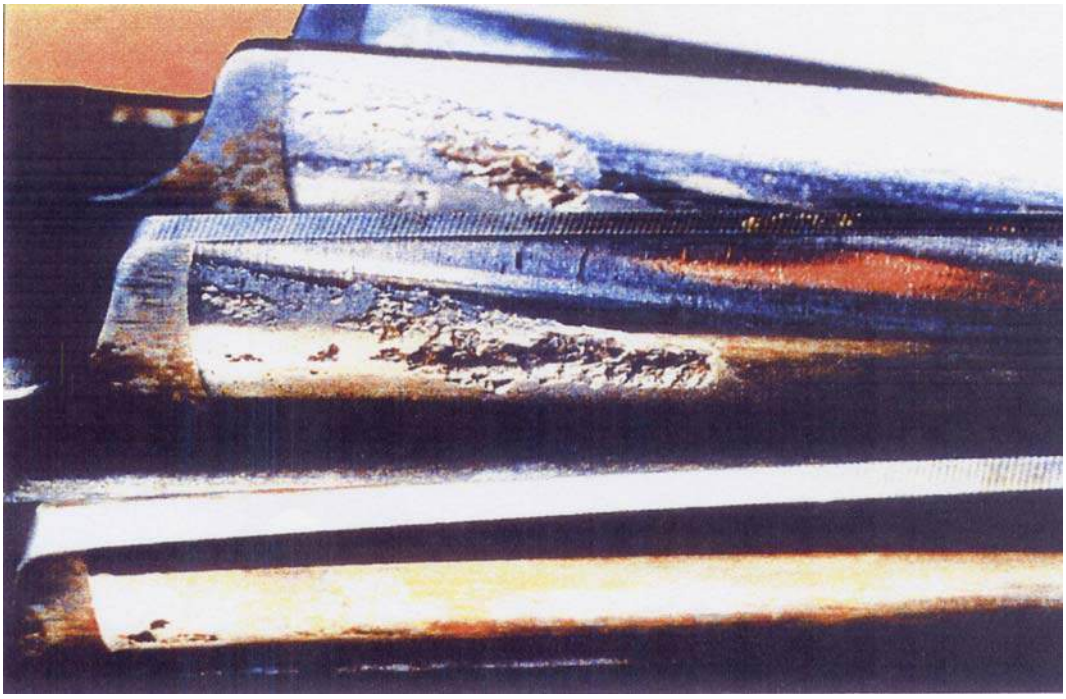


Fig. 11.36. Fatigue Flaking. Cracking and Flaking of the Surface of the Fragments Resulting from the Action of Cyclic Contact Stresses

- ensure the effective supply of engagement of sufficient quantity of clean, cooled appropriate viscosity grease.

According to the degree of fatigue flaking they are the following:

- initial flaking;
- destructive flaking;
- microflaking.

Initial flaking. These are a fairly small pockets less than 1 mm in diameter, formed in a non-uniform along the length of the contact overhead line. Usually some initial pockets of flaking are arranged along a straight line in a localized area on the gear root. Often, in the case of elimination of uneven contact and more uniform distribution of the load, no further growth of flaking occurs. As shown in Fig. 11.37, initial flaking on gear teeth may appear, and at the beginning of operation, even during running on alleviated load.

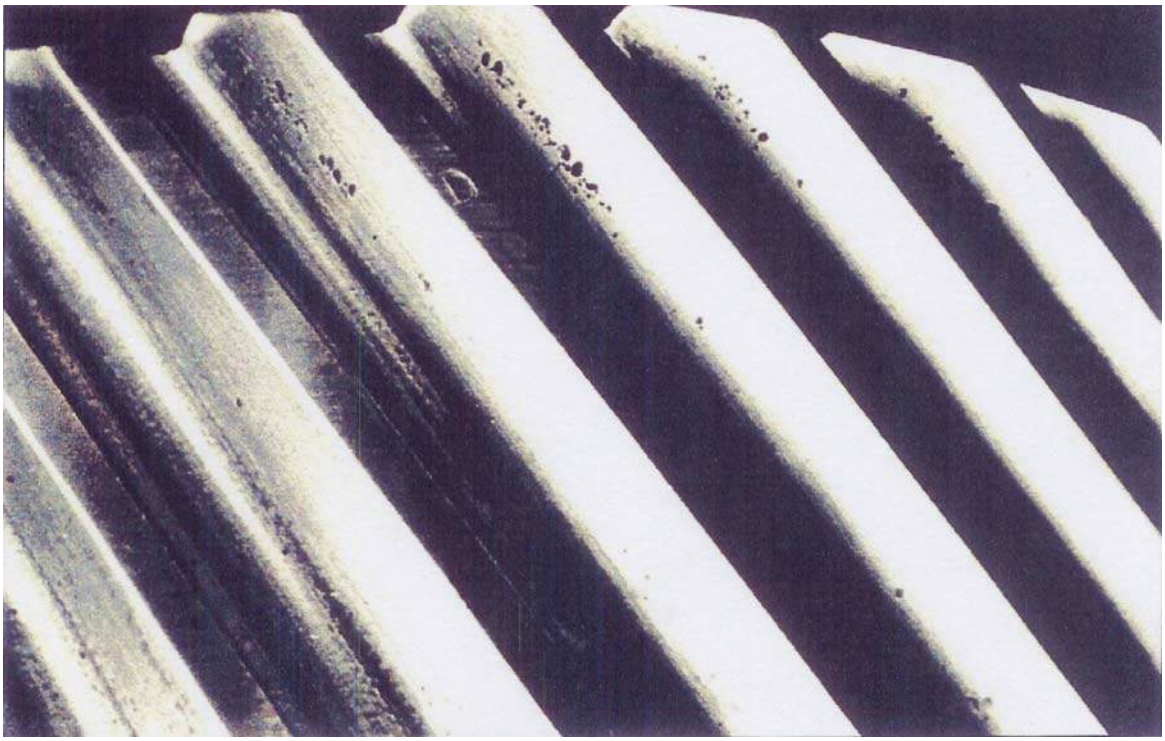
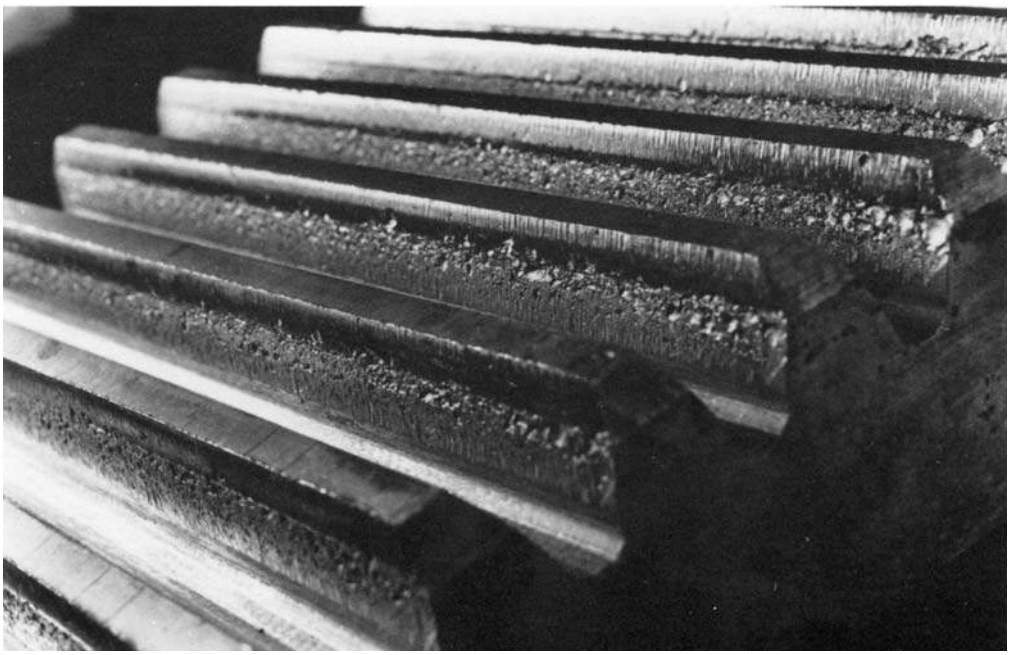


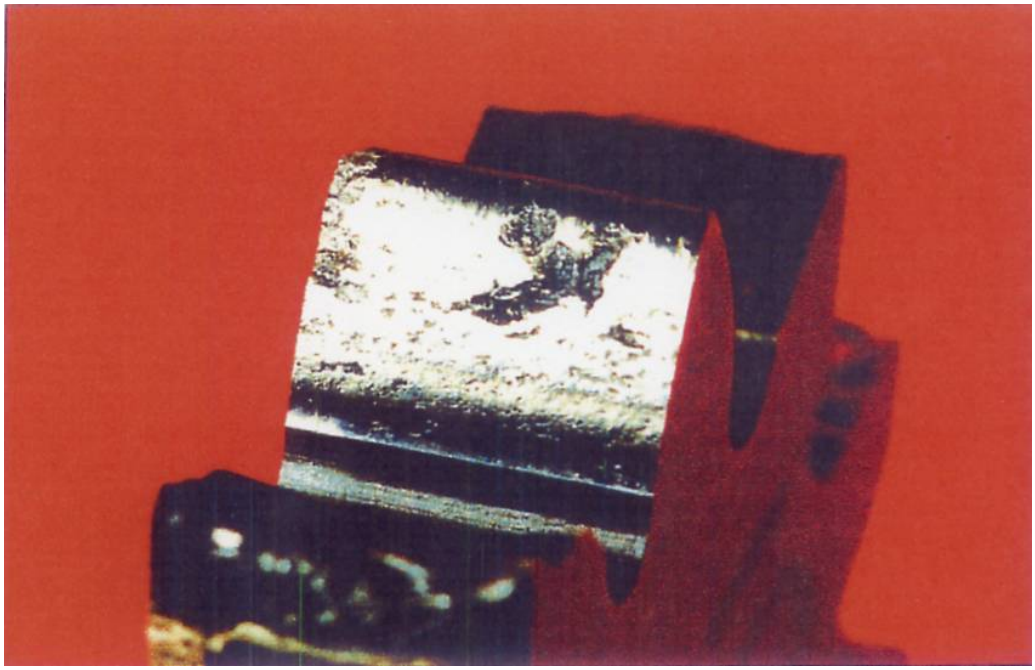
Fig. 11.37. Examples of Initial Fatigue Flaking Pits, Development of Which Was Suspended After Smoothing the Tops of Ridges Roughness

Destructive flaking is flaking, which continues to evolve with mechanical life of transmission age. Characterized by the fact that much of the surface of the gear root flaking damaged bubbles. Bubbles reach a significant size larger than 1 mm, both on the surface and in depth. The progressive flaking characterized by fusion of individual bubbles and form a continuous zone of local damage. According to the standard ISO-6336-5 [11] flaking is considered progressive and gear must be replaced if the total area of the zone with the flaking constitutes 2% of the working surface of the hardened gear gears volume, or 0.5 % of the working surface of gear teeth surface hardening, or 4% of the working surface of a gear. NASA for the bevel gear of the helicopter with the module 10.9 mm abroad destructive chipping size of pockets taken more than 0.4 mm and damage them more than 25 % of the working surface gear. In cases where the pockets depth exceeds the depth of the hardened surface layer and may cause the development of fatigue cracks leading to breakage gear. In cases where the bubble depth exceeds the depth of hardened skin may cause the development of fatigue cracks leading to breakage of gears. At NASA for the bevel gear of the helicopter with the module 10.9 mm abroad destructive chipping size of shells taken more than 0.4 mm and damage more than 25 % of the working surface of the gear. In cases where the pockets depth exceeds the depth of the hardened surface layer and may cause development of fatigue cracks leading to breakage of gears. Therefore, for such gear is necessary to introduce culling gear when the depth of the at least one shell is greater than 0.7...0.8 of the effective depth of the hardened layer. Sometimes there may be interim periods of

the restriction of flaking development, which are accompanied by periods of its development, as shown in Fig. 11.38.



a



b

Fig.11.38. Examples of Progressive Surface Flaking

WEAR

The wear process is any change in the surface of gears including the removal or displacement of mating surfaces' material by mechanical, chemical or other environment. There are such types of wear:

- mechanical wear;
- corrosion-mechanical wear;
- erosive wear.

The process of gears deterioration is observed in each working transmission. The numerical measure of wear is a wear determined in units of length, volume or weight. The magnitude of the wear limit is set depending on the destination of transmission, durability requirements, noise, and vibration. Wear is considered to be excessive in case of a high noise level, significant dynamic loads or substantially reduce of the gear thickness, which can lead to its fracture. The most wear susceptible details are high-load low-speed gear trains, operating under the boundary lubrication or its virtual non-existence. For the surface and thermochemical heat treated gears wear limit is determined by the remaining value of the hardened layer, capable of transmitting the design load.

Mechanical wear is wear due to mechanical impact of the mutually moving gears' surfaces on each other as well as the abrasive impact on the last odd particulate materials or separated wear debris.

Due to the physical nature of the process wear is distinguished:

- Adhesive wear;
- Abrasive wear;
- Polishing;
- Fatigue wear;
- Wear during start-up under load;
- Cavitation wear.

The first two kinds of wear are considered further. The remaining material can be found in the GOST full text [10].

The adhesive wear is the process that occurs in the interaction of microroughnesses that contact with surfaces' gears. As a result of high local pressures and the interatomic cohesion forces occurs the connection of microroughnesses, the so-called "cold welding" of contacting lobes subsequent plastic deformation upon relative movement of the lobes, the destruction of local clutches and, ultimately, removal or transfer of the metal. The surface condition of the lobes with traces of adhesive wear is shown in Fig. 11.39.

According to the degree of adhesive wear is distinguished as:

- Initial adhesive wear;
- Moderate adhesive wear;
- Extremely adhesive wear.

The ultimate adhesive wear, leading to gear failure is classified as seizure.

The adhesive wear is beneficial to some extent in the development of the gear, as by eliminating minor microroughnesses working surfaces of the teeth increases the actual contact patch and improves the lubrication conditions of transmission.

The lubrication quality is of significant impact on the amount of adhesive wear. If the lubricant is properly selected (its viscosity, cleanliness, operating temperature, lack of water, etc.) the adhesive wear should be limited only on the initial stage.

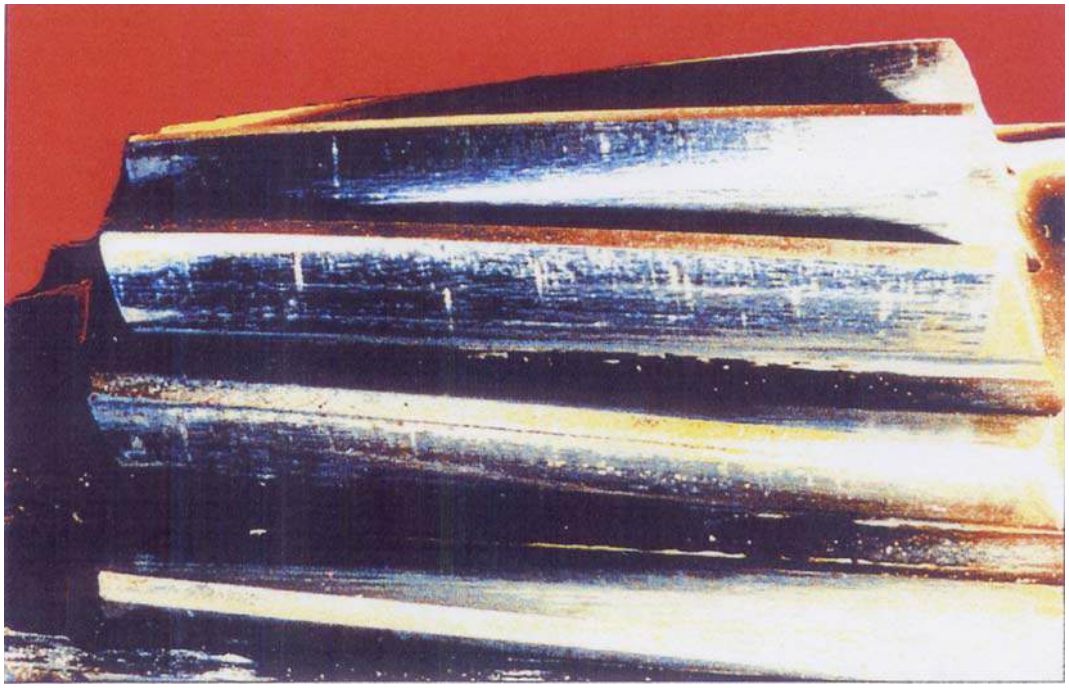


Fig. 11.39. General View of Teeth with Traces of Adhesive Wear

In order to prevent the development of adhesive wear it is recommended to:

- increase the frequency of treatment of the lobes' working surfaces;
- carry out a running-gear at partial load;
- use nitrided steel instead of cemented or hardened bulk quenching for heavy duty transmissions.

The initial adhesive wear typically occurs during the running of the gears and usually ends with the removal of minor errors of gears' surfaces by local wear. The naked eye can see the working surface of the gear intact, with traces of machining clearly distinguishable. A microscopic examination can be observed between the individual flat microportions be handled by gear tool. Flat flakes of 10...15 mkm are separated at low adhesive wear.

The moderate adhesive wear. Adhesive wear is classified as moderate, even if some or all traces of machining removed from the working surfaces of the gears. Under certain conditions, the adhesive wear leads to the continuous removal of surface oxide coatings and layers from the surface of gears, causing considerable wear.

The abrasive wear. Is the removal or displacement of the material because of the presence of the gear contact with hard particles (such as pieces of metal debris, rust, sand, hard abrasive particles, etc.) existing in the lubricant or made contact between the working surfaces of the gears.

The abrasive wear is based on the process of separating from microcutting chips and scratches process using plastic deformation of the grain, which is accompanied by the formation of pushing ahead of the bulk of the moving particle. These piles are composed of durable material with a large number of cracks and tears. Newly applied scratch cause destruction to the bulk metal particles separation. Externally, the abrasion

is characterized by the presence of scratches or grooves oriented in the direction of sliding. The nature of the abrasive wear is shown in Fig. 11.40.

The lubricated abrasive particles that are larger than the thickness of the oil film in the process of engagement of the gears can fall there between. Under influence of large contact pressure fragile particles disintegrate into smaller ones, some of them penetrate into the working surface of the teeth. Viscous particles are deformed in the teeth contact, a portion of them are also embedded in the working surface of the teeth. Potholes caused brittle particles are relatively small in size, but fairly deep potholes and sharp edges are different. In contrast, the bumps caused by plastic particles are large relative to the depth and at the bottom planar surface. The edges of the rounded potholes and raised above the surface of the teeth, forming rolls around potholes.

The size of abrasive wear is proportional to the hardness of the working surfaces of the gears. Abrasive wear is a typical type of clear transmission gear damage. The presence of abrasive wear of the enclosed gear lubricant indicates contamination or damaged sealant. To minimize abrasive wear it is recommended to use closed transmission filter elements that can delay the particle size of 3 microns. Application of cemented gears, high purity tooth surfaces and the use of more viscous lubricants reduce the intensity of abrasive wear.

According to the degree of abrasive wear it is distinguished:

- initial abrasion;
- moderate abrasion;
- limit abrasive wear;
- wear in the mutual implementation

The initial and ultimate abrasions are shown in Fig 11.41 and 11.42.

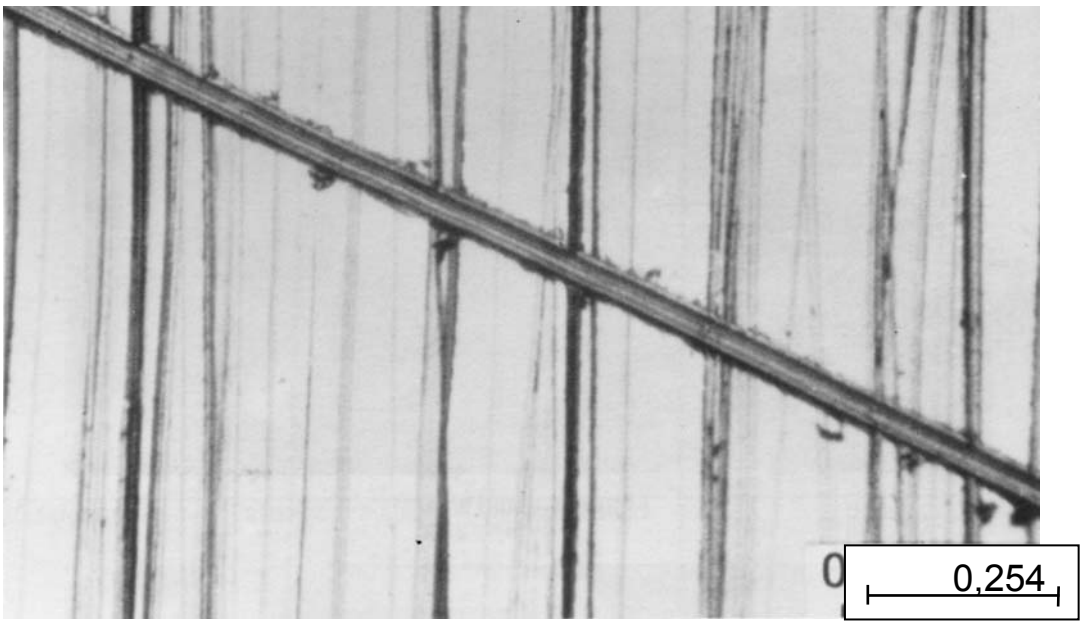


Fig. 11.40. Abrasive Wear:

A diagonal line - abrasive furrow plow solid particle that looks smooth. Vertical lines - original grinding marks

Fatigue fracture of gears - the gradual accumulation of damage of the material under the influence of a cyclic stress, which leads to a change in its structure and properties, the formation and development of cracks and ultimately to tooth fracture is called fatigue breakage of the tooth. Unlike the damage in case of overload, when fracture occurs most often several teeth at once, fatigue failure usually occurs only on one tooth.

Fatigue fracture of the teeth is a gradual process consisting of three separate stages:

- The origin of the crack. Plastic deformation of the developing stress concentrators, leading to the emergence of microscopic cracks.
- The growth of cracks. It is perpendicular to the vector of maximum tensile stress. Fatigue cracks occur and propagate under the influence of repeated variables or cyclic stress, which may be significantly lower than the tensile strength of a material

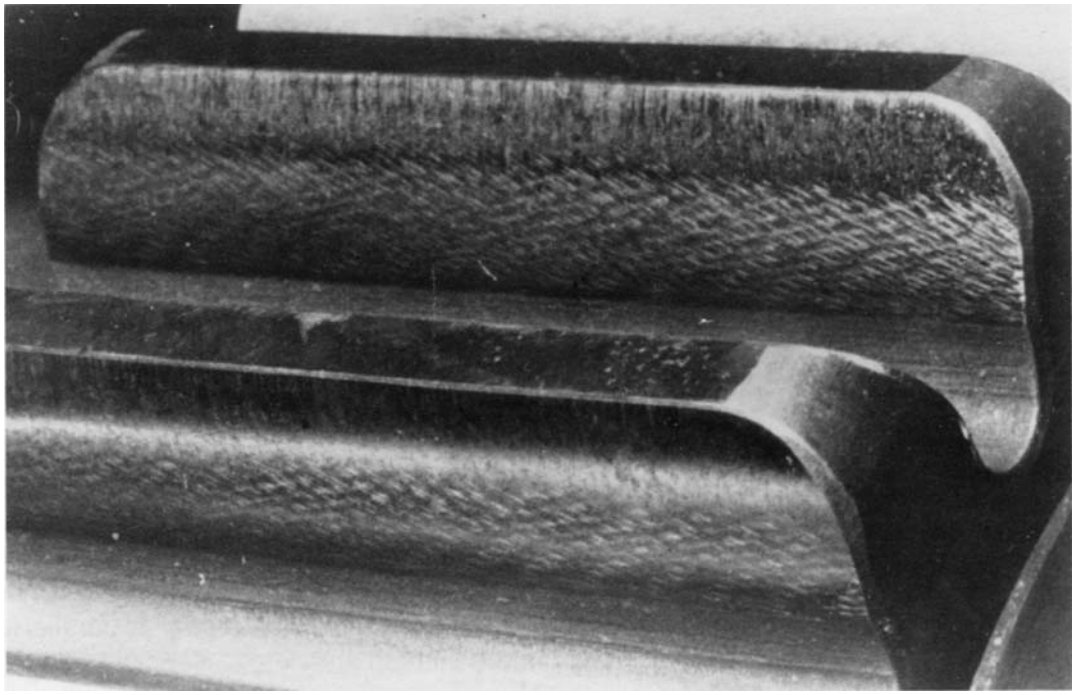


Fig. 11.41. Initial Abrasion on the Polished Side at the Apex of the Tooth Surface

under tension.

- Breaking. When the crack gradually during repetitive loading cycles reaches a critical size, there is a sudden failure of the tooth.

Most of the life of the gear stage hold nucleation and growth of a fatigue crack to a critical size.

One should distinguish between:

- high-cycle fatigue fracture;
- low-cycle fatigue fracture.

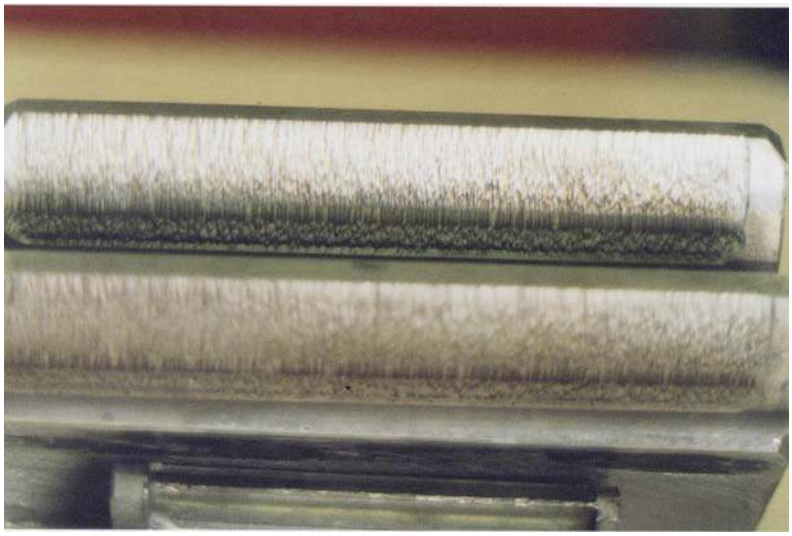


Fig. 11.42. Limit Abrasive Wear

Cycle fatigue fracture of the teeth is the most characteristic type of breaking damage teeth of gears. It occurs under cyclic stresses do not exceed the yield strength of the material. The number of cycles operating voltage exceeds 10 000, and much of life is the period of origin of fatigue cracks, but not their development.

Fracture surface consists of two distinct zones of fatigue damage and ultimate failure of the tooth. In the zone of plastic deformation fatigue symptoms are virtually absent. The surface is smooth and matte. The line of damage can be observed, as well as a gradual transition between the stages of crack propagation. The surface of the final breakdown of the tooth corresponds to one of the types of breaks when overloaded (fragile, viscous or viscoelastic).

Fatigue cracks appear, usually on the operating side of the tooth fillets, which are tensile stresses (Fig. 11.43).

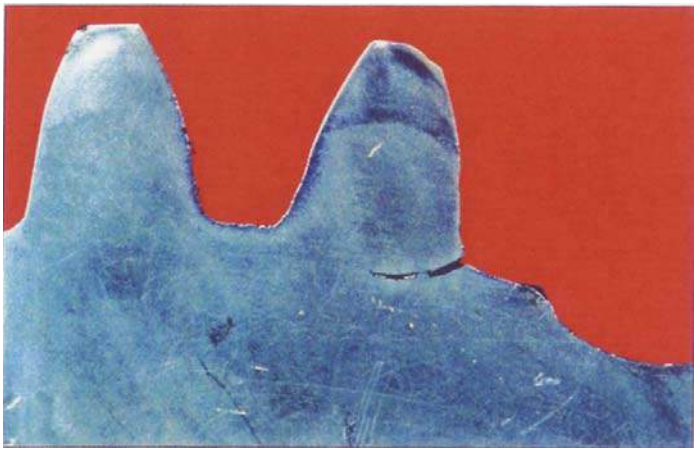


Fig. 11.43. Crack breakage at Base of Tooth

To extend the life-cycle of the gear bending endurance necessary to increase the strength of the material in tension and to optimize the microstructure of the surface and subsurface layers of the teeth.

Low-cycle fatigue fracture of the tooth occurs during the development of plastic deformation in each loading cycle and the total number of loading cycles of less than 10 000 (Fig. 11.44). Such a break less typical for well-designed gears and could be due to a substantial overload the transmission or significant misalignment arrangement of gears. With low-cycle fatigue, superficial concentrator does not play such an important role, because the presence of plastic deformation reduces the effect of the concentration of existing and residual stresses. Fatigue cracks may occur on the surface or below the surface of the tooth, and the time of crack is relatively small compared with the time of their development.

In order to increase the life of the gears on the low-cycle fatigue bending teeth appropriate to provide increase in ductility and toughness of the material gears. Thus, in particular, for cementitious gears it is recommended to:

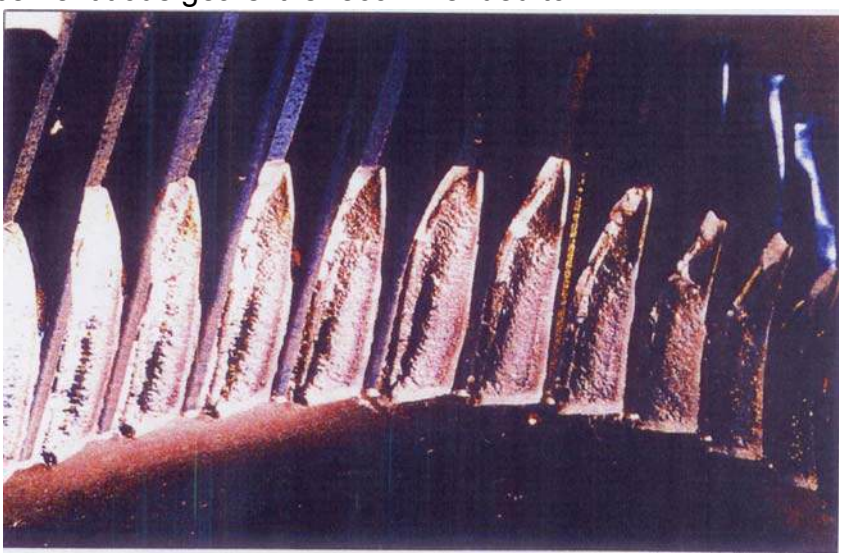


Fig. 11.44. Fatigue Fracture of a Tooth in the Low-Cycle Fatigue:

- use steel with a nickel content of more than 1 %;
- provide quenching of residual austenite in the hardened layer within the range of 15 to 30 %;
- reduce the hardness of the hardened layer from 58 to 55 HRC.

It should be noted that the adoption of the recommended measures to improve the low cycle bending endurance of teeth reduces their flexural-cycle endurance, so the best way to increase low-cycle endurance is to reduce the operating voltage.

11.4.1.2. Bases for Tooth Gears

Gear mechanisms in aircraft (helicopter transmissions, gearboxes of turboprop and turbofan engines) in most cases are heavily loaded with high-speed transmission. It is known from the statistics that the circumferential speed of only 20 % transmission of

air is 15 to 20 m/s, greater than 50 % transmission have a peripheral speed of 20 to 100 m/s or more.

Increasing the bearing capacity, reduction of weight and dimensions, increased lifetime and efficiency of aviation gears is an urgent and complex problem. It is resolved by providing such transmissions by the design and processing properties, as well as the use of high quality materials, efficient tool offset methods and lubrication. The geometry of the aircraft of the involute gear has its own characteristics in comparison with the general machine. Aircraft gears are made by special source with generating circuit. The main parameters of the gearing that meet accepted the original contour are the pressure angle α, the mechanical factor and the total overlap, as well as the parameters and forms the base of the tooth.

General engineering uses the original path with $\alpha = 20^{\circ}$, for aero power gears in accordance with aviation rules uses $\alpha = 25^{\circ}$ and $\alpha = 28^{\circ}$.

The teeth of the gear made with tool with an angle $\alpha = 25^{\circ}$, have undercut (protuberance) at the base of the tooth, which simplifies the grinding technology of the tooth surface. Gear teeth have higher contact strength at an angle $\alpha = 28^{\circ}$, but their manufacture is more complex.

The efficiency's increase of the transmission is achieved by modifying (correcting) tooth profiles, were the standard tool of the initial contour is shifted. Such a modification is used mainly for involute spur gear.

At high altitude modification positive tool offset for negative offset equal to the gear to the wheels. This increases the flexural strength of the gear teeth due to the strength of the gear teeth.

How to Calculate Teeth Strength

When there is engagement between the teeth, the oil film occurs, in which the hydrodynamic effects are manifested. This is ensured by the V-slits between the teeth and the considerable pressure of the lubricating oil therein. The pressure depends on the external load, geometric dimensions and elastic properties of the contacting surfaces. When the lubrication process is considered it is important to take in consideration a hydrodynamic effect and the effect of elastic deformation of the contact surfaces. The phenomenon of interdependence appears which is considered in a contact-hydrodynamic lubrication theory.

Due to the high hydrodynamic pressure of the lubricating oil and the variability of the construction of the calculation on this basis it is a very difficult task. This led to the use of the conditional approach to the contact strength of teeth. Assumptions that do not account for the effect of hydrodynamic effect of the interaction of the teeth are made. This corresponds to the Hertz problem of compression of two cylinders. The diagram of the compression in this case differs from the diagram, which is derived from contact-hydrodynamic theory.

When two cylinders are compressed, the normal stresses are given by Hertz's formula.

$$\sigma_n = \sqrt{\frac{w_n}{\rho} \cdot \frac{1}{\pi\left(\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2}\right)}} \le [\sigma]_n, \qquad (11.17)$$

where $w_n = \frac{K_H F_n}{l_\Sigma}$ is a normal contact load per unit length of the line is specific load; $\frac{1}{\rho} = \frac{1}{\rho_1} + \frac{1}{\rho_2}$ is reduced curvature ("+" sign – for an external contact and "-" is for domestic); E_1 и E_2 are the module's material elastic of gears and wheel; ν_1 и ν_2 are Pousson's ratios (for steel is ν= 0.3; for iron is ν = 0.25); K_l – design load's ratio.

For steel wheels (with ν= 0.3 and $E_1 = E_2$) we get

$$\sigma_n = \sqrt{\frac{w_n}{\rho}\frac{E}{2\pi\left(1-\nu^2\right)}} \le [\sigma]_n . \qquad (11.18)$$

when calculating the teeth it is considered that the contact of the teeth is similar to the contact of the two cylinders with radii ρ_1 and ρ_2, where ρ_1 and ρ_2 are the given radii of curvature of the profile of involute gear teeth and the wheel at the point of contact.

Behavior in Teeth Bending

When calculating the teeth bending a short cantilever beam with clamped at the base is considered. Calculation of beams based on the hypothesis of flat sections (Bernoulli hypothesis) can lead to significant quantitative errors.

Considering this the bending teeth' calculation is performed by local stresses, which is allowed for a very short beam. Local voltage at the base of the tooth is determined by the dependence

$$\sigma_F = \left(\frac{w_F \cos\gamma h_p}{\cos\alpha_t w_s} - \frac{w_F \sin\gamma}{\cos\alpha_t A_s} \right) \alpha_\sigma ; \qquad (11.19)$$

$$w_F = \frac{2T_1 \cdot 10^3 \cdot K_F}{t \cdot d_1} = \frac{F_t K_t}{t}; \qquad w_s = \frac{S^2}{\hat{a}} \quad \text{(resisting moment);}$$

T_1 – gear's moment (the master element);

S – tooth's thickness in the dangerous section;

As - dangerous platitude section of unit length;

hp - shoulder's design of the force;

α_σ - theoretical concentration of the stress factor;

γ – the angle characterizing the position of the tooth at the output of the engagement.

If the edge of the tooth is taken as a place the strength application, this case should be referred to the two-pair engagement zone, which is also an assumption.

Design Load

The gear teeth are loaded with not only nominal, but also with the additional forces, the latter takes into account the load factor calculated K_l and K_F. Taking into account additional loads circumferential force on a pitch cylinder

$$F_t \cdot K = \frac{2T_1 \cdot 10^3 K}{d_1}, \qquad (11.20)$$

where $K = K\alpha \cdot K_\upsilon \cdot K_\beta \cdot K_A$;

K_A – coefficient of the external dynamic load factor;

K_y – coefficient of the internal dynamic load;

K_β – coefficient of the stress concentration factor;

K_α – coefficient taking into account the load distribution between the teeth pairs, which are in engagement.

The numerical values of K_A for a uniform loading of the engine operation and hosting machines - 1, with a small unevenness - 1.25; average - 1.5; large - 1.75.

In preliminary calculations K is taken as 1.3 ... 1.5.

In the calculations for the K value input such indexes: K_l refers to the contact strength, K_F - to bending. Then, for example, for contact strength is $K_l = K_{lv} \cdot K_{l\beta} \cdot K_{l\alpha}$.

Additional internal dynamic loads arise due to errors in the main district and steps away and the profile of the active surface of the teeth, and are caused by the deformation of the wheels and associated parts.

When intermating the teeth takes place due to the direct impact of their load, and inertial forces and active vibration.

The action on the gear substantial largest time-varying torsion moment of the internal dynamic load increases due to torsional vibrations. This applies gearbox connected to the internal combustion engine and the like transmissions. Modern calculations of internal dynamic loads are indicative. In their estimation, there are two approaches that meet one or the other theory.

The impact theory is considered a blow to the tooth point surge in the vibration - load due to kinematic errors and changes in tooth stiffness. In engineering of Ukraine and the CIS common standard for cylindrical gears [12], which operate at speeds up to 25 m / s. It is based on the theory of shock.

Selection of Materials for Gears

The materials for gears should be selected according to the conditions to ensure the strength of the teeth of endurance under the influence of contact and bending stresses. This is achieved by using high-strength steel alloys that are heat and chemical heat treatment (hardening, carburizing, nitriding, carbonitriding, etc.). Thus, it provides a surface hardness of steels used teeth 60...65 HRC.

For the manufacture of helicopters' gears and other aircraft, high-alloy steels (12HN3A, 12Kh2N4A, 18H2N4MA, 38HMYUA etc.) are used.

The use of low-carbon low-alloy steels after surface treatment allows to obtain a high hardness of the surface of the teeth and soft, viscous core, which is needed to improve their contact and flexural strength, respectively.

Allowable Stresses

In determining the permissible contact voltage for the base, the ultimate stress in the horizontal section of the curve of fatigue is taken (Fig. 11.45).

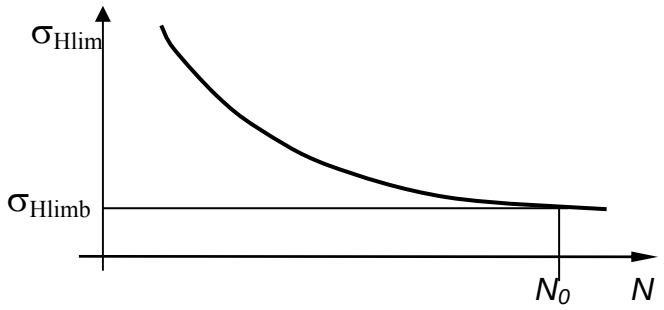


Fig.11.45. Fatigue curve

In this case, the allowable contact pressure of the wheels for high speed determined depending on

$$[\sigma]_s = \frac{\sigma_{s\lim b} \cdot Z_R \cdot Z_v}{S_s} \cdot Z_N, \qquad (11.21)$$

where $\sigma_{s\lim b}$ – contact fatigue limit of the base;

S_s = 1.2...1…1.3 – coefficient of assurance factor;

Z_R – factor which takes into account the surface roughness of the tooth,

ZR = 1 when Ra = 1.25…0.63; ZR=0,95 when Ra = 2.5…1.25;

Z_v – factor which takes into account the influence of the speed, and is determined by the dependencies

Zy = 0.35 V $^{0.1}$ when MR ≤ 350,

Z_y = 0.925 V $^{0.05}$ when MR > 350;

Z_N - the durability ratio.

Permissible voltage when the timing of operation that meet the area of the inclined portion of the curve of fatigue, should take into account the number of loading cycles. Slope exponential function is approximated curve (see. Fig. 11.45)

$$\sigma_H^m N_i = \sigma_{H\lim b}^m N_{H0} = \text{const}. \qquad (11.22)$$

It follows:

$$[\sigma]_H = [\sigma]_{H\lim b} \sqrt[m]{\frac{N_{íî}}{N_i}} = [\sigma]_{H\lim b} \cdot Z_N.$$

The helical gear contact line is placed at an angle to the axial lines. When it is engaged on a tooth end (on the leg), and the head, the length of the contact line increases. This explains the increased load bearing capacity of helical gears and improving the smoothness of their work.

Formulas for determining the base of the contact fatigue strength, depending on the hardness of the tooth surface are shown in Table. 11.5.

Table 11.5

Characteristics of Materials
Used For Manufacture Of Toothed Wheels

Wheels' materials	Thermal or Thermochemical Treatment	Hardness of Surface	σ_{Hlimb}, MPa
Carbon and low alloy steels	Improvement, normalization	≤ 350 HB	2 HB + 70
	Through hardening	38…50HRC	17 HRC+100
	Case hardening	40…56HRC	17 HRC+200
Alloy steels	Carbonization, dry cyaniding, tempering	56… 65HRC	23 HRC
	Azotization	550...750HV	1050

Allowable Bending Stresses

When the term of gear's operation meets the area of the horizontal section of the fatigue's curve with a one-way (non-reversing), the transfer of the basic allowable stress

$$[\sigma]_{F0} = \frac{\sigma_{F\lim} Y_R Y_x Y_\delta}{S_F}, \qquad (11.23)$$

where σ_{Flim} – long fatigue limit at a bend to zero cycle;

S_F – coefficient of the safety factor;

Y_R – factor which takes into account the roughness of the surface of the tooth transition. When milling and grinding to $R_Z \le 40$ microns Y_R = 1, when polishing Y_R = 1.05…1.2 (higher values - after improvement and hardening of high frequency current (HFC); Y_x - is a scale factor at $d \le 400$ mm Y_x = 1;

Y_δ – factor which takes into account the sensitivity of the material to stress concentration.

In general terms (including the possibility of transferring operation when the number of cycles is less than the basic and reverse operation) permissible bending stress

$$[\sigma]_F = [\sigma]_{Fo} Y_A Y_N. \qquad (11.24)$$

Here Y_A - factor that accounts for two-sided application of force;

$$Y_N = \sqrt[m]{\frac{N_{F0}}{N_i}} \geq 1,$$

where N_{F0} = $4 \cdot 10^6$ – basic number of cycles for steel teeth; N_i - number of cycles for a period of work or an equivalent number of cycles (when $N_i \geq N_{F0}$ Y_N = 1);

m = 9 – for nitrided and cemented teeth without grinding, m = 6 - for the other cases.

11.4.1.3. Example of Main Gearbox Gear Calculation of Single-Rotor Helicopter

Consider the example of the calculation of the gear main gearbox of helicopter Mi-2 – the last stage associated with MR.

Initial Data for Calculation

Maximum power (takeoff), transmitted from the engine to the main rotor, N_{TO} = 233 kW.

The maximum torque acting on the rotor T_{max} = 18 900 N·m.

2. Shaft rotational speed of the rotor n_s =246 rev/min.

3. The total gear ratio of the main gearbox U_{1-6} = 23.9305.

4. The frequency of the third stage gear rotation n_5 = 651 rev/min.

5. The numbers of the third stage teeth z_5 =17, z_6 =45.

6. Straight teeth (β =0), the relative teeth width $\psi_{bd} = b_w / d_w$ = 0.68.

7. Gears' materials are alloy structural steel with a hardness number 12X2H4A with 320HB gears, 270HB wheels, σ_u =1180 MPa tensile strength, σ_y =950 MPa yield strength. After heat treatment, the hardness of the surface of the gear teeth is 60 HRC, wheels - 57 HRC.

8. The temperature operating conditions: from 60 ° C to + -60º C.

9. The specified resource **t**=5000 h.

Calculation of the gear strength were conducted in accordance with GOST 2175-87 (for involute cylindrical gear transmission external link) [12] and the data [8].

The calculation is carried out to prevent fatigue spalling workers (side) surfaces of the teeth and fatigue fracture at the base of the teeth.

It is known that during operation of the helicopter loads, the acting on the parts and components of the helicopter, the flight cycle model determined that one of the helicopter being developed by ZMKB "Progress", shown in Fig. 11.46. As it can be seen

from this figure, the flight cycle duration for 60 minutes contains about 20 loading modes, which vary in steps.

When a stepped mode is on, the load calculation of gears on the fatigue strength is performed on an equivalent number of cycles, which is determined on the basis of the hypothesis of linear summation of damage [8].

Most long-acting loads as the estimated is taken. In our case, the maximum torque acting on the shaft of the screw– 18 900 N·m for the design load is taken.

Since some modes of engine operation (start-up, warm-up, taxi, stop, etc.) make a minor contribution to the total fatigue of gears, the calculation will be carried out under the simplified flight cycles, which is illustrated in Fig. 11.47.

Here, the maximum power modes (takeoff and landing) account for 5% of the total cycle time of flight, the nominal power mode (90% N_{max}) - 35% of the flight cycle mode and cruising modes (80% N_{max}) - 60% of the flight cycle. Also assume that the shaft speed remains constant in all modes. These simplification of the flight cycle are in a margin of safety.

Taking into account the efficiency and power supply from two engines torque applied to the gear of the third stage, is given by

$$T_5 = \frac{T_{max}}{2u_{5-6}} \cdot \eta_g^2 \cdot \eta_b^2, \qquad (11.25)$$

where η_g – efficiency gear; η_b – a pair of bearings efficiency.

Having, in accordance with the data of [8] η_g=0.95, η_b=0.997, we obtain from (11.25)

$$T_{5\,max} = (9450 / 2.647)\cdot 0.95^2 \cdot 0.997^2 = 3968 \text{ N·m}.$$

To calculate the contact resistance equivalent number of cycles is determined by the dependence

$$N_{HE} = 60\tilde{N}\sum_{i-1}^{n} n_i t_i \left(\frac{T_i}{T_{max}} \right)^3, \qquad (11.26)$$

where C – a number of teeth contacts per revolution; n_i – a toothed wheel speed at the i mode; t_i – Number of hours of work gear on i-mode;

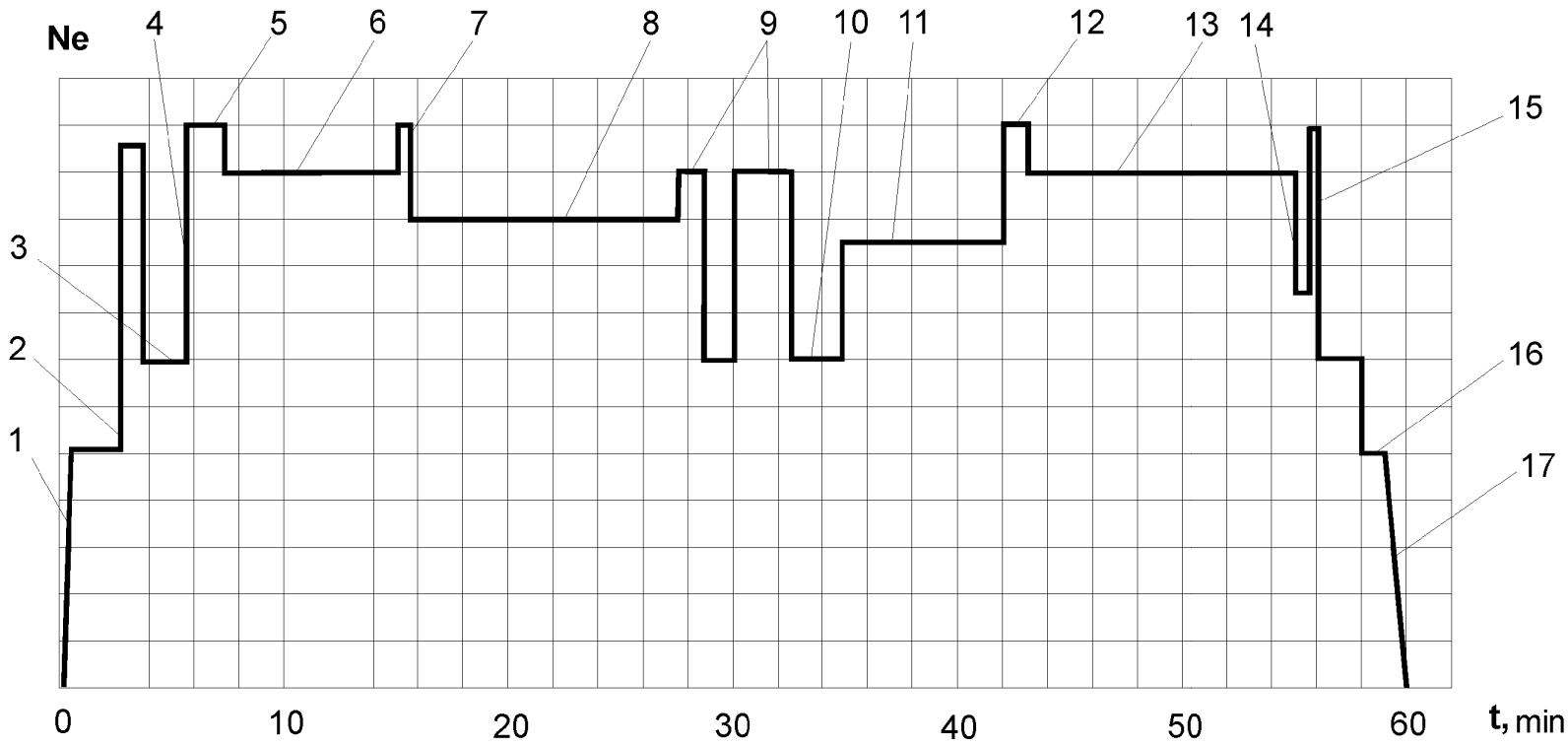


Fig. 11.46. Generalized Helicopter Flight Cycle:
1 – start; 2 – warm-up; 3 – taxiing; 4 – takeoff; 5 – hovering, height gain; 6, 8, 9, 11, 13 – horizontal flight;
7, 12 – horizontal flight V_{max}; 10, 14 – motor planning; 15 – braking and landing; 16 – taxiing;
17 – stop

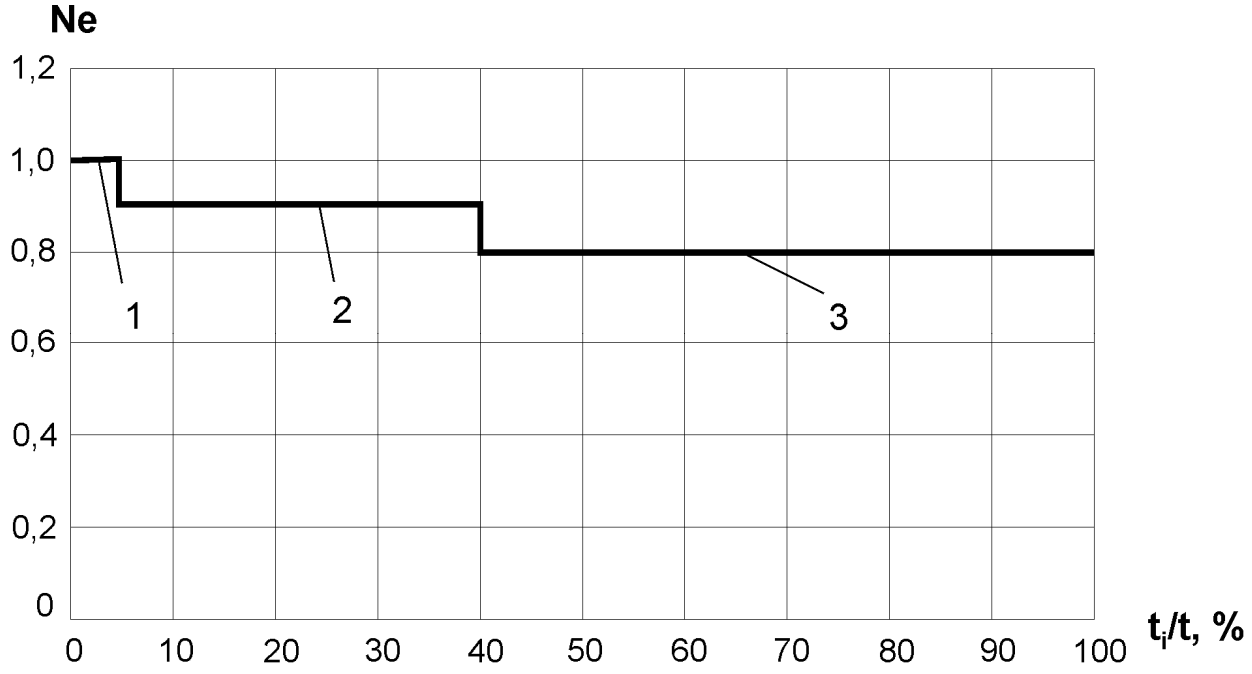


Fig. 11.47. Simplified Sequence Diagram of Loading:
1 – maximum power mode; 2 – nominal conditions;
3 – cruise mode

T_i – gear's torque acting on i mode; T_{max} – maximum torque, appropriate for takeoff; n – number of calculated modes.

We determine the equivalent number of cycles for gear. Since gears comes once per revolution into contact with the wheel, then C_5=1.

When the resource is t=5000 hours of work's duration, the corresponding to the different modes are equal:

$t_1 = t_{max}$ = 0.05·5000 = 250 h;

$t_2 = t_n$ = 0.35·5000 = 1750 h;

$t_3 = t_t$ = 0.6·5000 = 3000 h.

We define the torques on the pinion and wheel corresponding to different loading modes:

$T_{5\,n} = 0.9 \cdot T_{5\,max}$ = 3571 N·m;

$T_{5\,t} = 0.8 \cdot T_{5\,max}$ = 3174 N·m.

Since the gear speed is assumed constant, then in accordance with formula (11.26)

$$N_{HE\,5} = 60 C_5 n_5 \sum_{i=1}^{3} \left(\frac{T_i}{T_{max}} \right)^3;$$

$$N_{HE\,5} = 60 \cdot 1 \cdot 6.51 \cdot (1 \cdot 250 + (0.9)^3 \cdot 1750 + (0.8)^3 \cdot 3000) = 119.6 \cdot 10^6.$$

An equivalent number of cycles for the gears is approximately equal to the base $N_{H\,lim}$=120·10^6 so durability coefficient Z_N is assumed to be one.

For the wheel $N_{HE\,6} = 60C_6 \cdot n_6 \sum_{i=1}^{4} \left(\frac{T_i}{T_{max}} \right)^3$;

$N_{HE\,6}$ = 60·2·246·(1·250+0,9^3·1750+0,8^3·3000) = 90,36·10^6.

Here C_6 equals two, since the wheel contacts with two gears.

Since the number is less than the equivalent of basic cycles, the durability factor Z_N for the wheel is

$$Z_N = \sqrt[6]{\frac{N_{H0}}{N_{H6}}} = \sqrt[6]{\frac{120 \cdot 10^6}{90{,}36 \cdot 10^6}} = 1{,}048,$$

where N_{H0} – base number of cycles equals to 120·10^6.

Allowable contact voltage is determined separately from the formula for the gears and wheels

$$[\sigma]_H = \frac{\sigma_{H\,lim} Z_N}{S_H} Z_R Z_V, \qquad (11.27)$$

where $\sigma_{H\,lim}$ – fatigue limit of the base; Z_N – durability factor; S_H – safety factor; Z_R – coefficient that takes into account the roughness of tooth surfaces; Z_V – coefficient that takes into account the environmental speed.

In accordance with the recommendations contained in the work [8], Z_R=1, Z_V=1.085, S_H=1.35.

Basic contact fatigue limit for the accepted values of hardness of surfaces is determined by the dependence $\sigma_{H\,lim}$=23 HRC.

Then $\sigma_{H\,lim\,5}$ = 23·60 = 1380 MPa; $\sigma_{H\,lim\,6}$ = 23·57 = 1311 MPa.

Accordingly, the allowable contact stress is calculated from the formula (11.27):

for gears $[\sigma]_{H\,5} = \frac{1380}{1.35} \cdot 1 \cdot 1.085 = 1109$ MPa

and wheels $[\sigma]_{H\,6} = \frac{1311 \cdot 1.048}{1.35} \cdot 1 \cdot 1.085 = 1104$ MPa.

To calculate, the lesser of the calculated allowable stress is taken σ_{HP}=1104 MPa.

In designing calculating the approximate value of the initial gear circle diameter (d_{w5}) mm, is determined by the dependence

$$d_{w5} = K_d \sqrt[3]{\frac{T_5 \cdot K_{H\beta}}{\psi_{bd} \cdot \sigma_{HP}^2} \cdot \frac{u+1}{u}}, \qquad (11.28)$$

where K_d – auxiliary coefficient, for spur gears K_d=770; T_5 – torque on the sprocket, Н·м; $K_{H\beta}$ – stress concentration factor; ψ_{bd} – the relative width of the sprocket.

The values ψ_{bd} and u are taken as input values and definitions σ_{HP} and T_5 are defined above. Value $K_{H\beta}$ is calculated from the data [12], depending on the gear unit and the value of the scheme ψ_{bd}: $K_{H\beta}$=1,1.

Then $d_{w5} = 770 \sqrt[3]{\frac{3968 \cdot 1{,}1}{0{,}8 \cdot 1104^2} \cdot \frac{2{,}647+1}{2{,}647}} = 141$ mm. Hence we find the estimated value of the module $m = 141/17$=8.30 mm and rounding it up to the standard value m = 8 mm. The diameters of the gears and wheels, respectively, will be equal to $d_{w5} = m \cdot z_5$ = 8·17 =136 mm, $d_{w6} = m \cdot z_6$ = 8·45=360 mm.

Contact Strength Verification Calculation

Rated contact voltage is given by

$$\sigma_H = Z_E \cdot Z_H \cdot Z_\varepsilon \sqrt{\frac{F_{tH} \cdot K_H}{b_w \cdot d_w}} \cdot \frac{U \pm 1}{U} \geq [\sigma_H], \qquad (11.29)$$

where F_{tH} – engaged circumferential force; b_w – the ring gear's width; K_H – the estimated load factor: $K_H = K_{HA} \cdot K_{HV} \cdot K_{H\beta} \cdot K_{H\alpha}$.

Defining the load factor. Assume the coefficient K_{HA}, taking into account the external dynamic load equal to one [8]. The value $K_{H\beta}$ is in the preliminary calculation is assumed to be 1.1. The coefficient's $K_{H\alpha}$ value, which takes into account the load distribution between the teeth for spur gears accepts $K_{H\alpha}$=1.

The coefficient K_{HV}, which takes into account the dynamic load which occurs in engagement, is determined by the dependence

$$K_{HV} = 1 + \frac{W_{HV} \cdot b_w}{F_{tH} \cdot K_{HA}}. \qquad (11.30)$$

Specific circumferential dynamic force

$$W_{HV} = \delta_H g_0 v \sqrt{\frac{a_w}{u}}, \qquad (11.31)$$

where a_w – center distance, equal to

$$a_w = \frac{d_{w1} + d_{w2}}{2} = \frac{136 + 360}{2} = 248 \text{ mm},$$

$$v = \frac{\pi n_5}{30} \cdot \frac{d_{w5}}{2} = \frac{3.14 \cdot 651}{30} \cdot \frac{0.136}{2} = 4.63 \text{ m/s}.$$

For an example of the case we take δ_H=0.1; g_0=3.1.

Circumferential force in engagement

$$F_{tH} = \frac{2T_{5H}}{d_{w5}} = \frac{3968 \cdot 2}{0{,}136} = 58\,353 \text{ N}.$$

Then $W_{HV} = 0.1 \cdot 3.1 \cdot 4.63 \sqrt{\frac{248}{2.647}} = 13.89$ N/m.

We define the coefficient value of K_{HV}:

$$K_{HV} = 1 + \frac{13{,}89 \cdot 92}{58353 \cdot 1} = 1.022.$$

Then K_H=1.1·1.022=1.124.

In accordance with the recommendations [8] Z_E=190. We define the value Z_ε:

$$Z_\varepsilon = \sqrt{\frac{4 - \varepsilon_\alpha}{3}}, \qquad (11.32)$$

where $\varepsilon_\alpha = 1.88 - 3.2\left(\frac{1}{z_1} + \frac{1}{z_2}\right) = 1.88 - 3.2\left(\frac{1}{17} + \frac{1}{45}\right) = 1.62$. Then,

$$Z_\varepsilon = \sqrt{\frac{4 - 1{,}62}{3}} = 0.89;$$

$$Z_н = \frac{1}{\cos\alpha_{tw}} \sqrt{\frac{2}{tg\alpha_{tw}}} = \frac{1}{0.94} \sqrt{\frac{2}{0.364}} = 2.43.$$

Estimated value of the contact voltage equals

$$\sigma_H = 190 \cdot 2.43 \cdot 0{,}89 \sqrt{\frac{58353 \cdot 1.124}{92 \cdot 136} \cdot \frac{2.647 + 1}{2.647}} = 76.5\sqrt{6.71} = 1107 \text{ MPa}.$$

The strength condition is satisfied: σ_H=1107 MPa ≈ 1104 MPa.

Overloading is 0.27 %.

Verification Calculation of Teeth on Flexural Endurance

Define the permitted voltage

$$[\sigma_F] = \frac{\sigma_{F\lim b}}{S_F} Y_N \cdot Y_\delta \cdot Y_R \cdot Y_x . \qquad (11.33)$$

Here

$$\sigma_{F\lim b} = \sigma^0_{F\lim b} \cdot Y_T \cdot Y_Z \cdot Y_g \cdot Y_d \cdot Y_A . \qquad (11.34)$$

According to the recommendations [8] we take $Y_T = Y_Z = Y_A = 1$; $Y_g = 0.75$; $Y_d = 1.1$.

For steel 12X2H4A the basic limit of bending endurance $\sigma^0_{F\lim b}$ = 820 MPa, safety coefficient $S_F = 1.55$.

Then $\sigma^0_{F\lim b}$ = 820·1·1·0.75·1.1·1·0.927 = 627.1 MPa.

Substitute $\sigma^0_{F\lim b}$ in the expression for the allowable stresses, pre-determining Y_N - factor of durability for an equivalent number of alternating stress cycles N_{FE}:

$$N_{FE} = 60 \cdot C \cdot n \sum_{i-1}^{n} \left(\frac{T_i}{T_{1F}} \right)^m t_i . \qquad (11.35)$$

Since the base number in bending cycles $N_{F\lim} = 4 \cdot 10^6$ is significantly less, than for contact strength, and N_{FE} in this case will be bigger than N_{HE}, calculated previously, then to the pinion and to the wheel equivalent number of cycles will be more base, the durability factor $Y_N = 1$. The coefficient which takes into account the roughness of transitional surfaces for grinding and hobbing take $Y_R = 1$.

Coefficient which takes into account the size of the wheels:

– for gears: Y_{X5} = 1.05-0.000125 d_3 =1.05-0.000125·136=1.035;

– for wheel: Y_{X6} = 1.05-0.000125 d_4 =1.05-0.000125·360=1.005.

Substituting these values of coefficients, we obtain the permissible bending stress

$$[\sigma_F] = \frac{\sigma_{F\lim b}}{1,55} \cdot 1 \cdot 45 \cdot 1 \cdot 1,035 = 627,1 \cdot 1,035 = 418.7 \text{ MPa}.$$

Coefficient which takes into account the stress gradient:

$$Y_\delta = 1.082 - 0.172 \cdot \lg m = 1.082 - 0.172 \cdot \lg 9 = 0.927.$$

Then $W_{FV} = \delta_F \cdot g_0 \cdot v \sqrt{\frac{a_w}{4}} = 0.11 \cdot 3.1 \cdot 4.63 \cdot \sqrt{\frac{248}{2.647}} = 15.28$.

To spur gears with modified heads we take δ_F=0.11.

Calculated bending stress is determined by the dependence

$$\sigma_F = \frac{F_{tF}}{b \cdot m} K_F \cdot Y_{FS} \cdot Y_\beta \cdot Y_\varepsilon \le [\sigma_F]. \quad (11.36)$$

Circumferential force $F_{tF} = F_{tH}$ = 58353H, tooth width b = 92 mm, m =8 mm.

To spur gears coefficient taking into account the slope of the teeth Y_β, and the coefficient, taking into account the overlap of the teeth Y_ε, can be equal 1.

Coefficients that take into account the shape of the teeth, independent of the number of teeth, and therefore they are different for gears and wheels:

– for gear Y_{FS} = 4,25 (Z=17);

– for wheel Y_{FS} = 3,67 (Z=45).

Load coefficient $K_F = K_A \cdot K_{FV} \cdot K_{F\beta} \cdot K_{F\alpha} \cdot$

For spur gears can be approximately taken K_A, $K_{F\alpha}$=1.

The coefficient which takes into account uneven load distribution along the length of the contact lines, define the graphs [12]:

with $K_{H\beta}$=1.1 and b/m = $92/8$ =11.5 $K_{F\beta}$=1.1.

Coefficient which takes into account the dynamic load of engagement K_{FV} can be calculated by the formula

$$K_{FV} = 1 + \frac{W_{FV} \cdot b_w}{F_{tF} \cdot K_H} = 1 + \frac{15,28 \cdot 92}{58353 \cdot 1} = 1.024,$$

where b_w =92 mm, F_{tF} =58353 H, K_A =1.

Then

$$K_F = K_{FV} \cdot K_{F\beta} = 1.1 \cdot 1.024 = 1.126.$$

Estimated value of the bending stress for pinion:

$$\sigma_{F5} = \frac{58353}{92 \cdot 8} \cdot 1.126 \cdot 4.25 \cdot 1 \cdot 1 = 379.4 \text{ MPa}.$$

For the wheel σ_{F6} will be less, as the tooth shape factor Y_{FS} is less for it.

Therefore, the condition of the flexural strength is satisfied:

$$\sigma_{Fd} \le [\sigma_F].$$

11.4.2. Clutches

11.4.2.1. Functions, Main Types and Arrangement of Clutches as Part of Transmissions of Helicopters

Clutches work for connection of the ends of shafts for transfer of the rotating moment without change of angular speed [9]. Besides, clutches can compensate errors of shaft axles arrangement, can be dampers of torsion fluctuations, provide smooth start-up and a stop of the mechanism, to protect mechanisms from an overload, etc. Clutches are widely applied in drives of aviation mechanisms, including in transmissions of helicopters.

Some common requirements should be met by the helicopter clutches, the general for clutches of aircraft: the minimum deformations and tension in the shafts and supports connected by them, the minimum losses of power, the increased reliability, the minimum overall dimensions and weight.

On design, functions and the principle of operation the clutches are quite various. On the basis of controllability all mechanical clutches are divided into three main types: the uncontrollable – don't allow disengagement of shafts in the course of work; the operated – allow to engage and disengage forcibly driving and driven shaft in the course of work; the self-operated – automatically disengage shaft at change of the set operating mode.

Uncontrollable clutches on nature of shaft engagement are divided into following: rigid clutches – for continuous rigid connection of strictly coaxial shafts; the compensating – for connection of shafts with the insignificant mutual shifts and distortions of axes caused by errors of production and installation; the elastic – for protection from dynamic loadings.

The operated clutches can be of cam and gear type, they are intended for engagement and disengagement of shafts at almost equal angular speeds, and frictional – for smooth connection of the forcibly driving and driven shaft at any difference of their angular speeds.

Self-operated clutches are divided into the following: the centrifugal – for automatic engagement and disengagement of shafts at the set angular speed of the driving shaft; the overrunning – for transfer of the moment only in the set direction and only at equality of angular speeds; the safe – for protection of parts of machines from overloads.

The main characteristic of the clutch is the design torque $T_d = K_1 \cdot K_2 \cdot T_n$, where T_d and T_n – are respectively design and nominal torques; K_1 – safety coefficient; K_2 – operating mode coefficient.

In aviation mechanisms both standard, and special non-standard clutches are used.

Depending on what assemblies and units of the helicopter the clutches used in transmissions of helicopters connect, they can conditionally be divided into two groups [4]:

1) the clutches located between the engine and the main gearbox of helicopter (gearbox of the main rotor);

2) the clutches providing the transfer of power to the tail rotor, and also the clutches connecting the synchronizing shafts, etc.

The main types and characteristics of clutches of the first group are given in Table 11.6.

Table 11.6

Clutches of the First Group

Name of the Helicopter	Maximum Torque Moment N·m	Rotation Frequency rev/min	Type of Clutches
Mi-1	2620	1581	Elastic with rubber bushes
Mi-2	450	5904	Universal hinges (kardans)
Mi-4	4850	2400	Elastic with rubber bushes
Mi-6	—	8304	Gear clutches
Mi-8	710	15000	Gear clutches
Ka-15	—	754	Cam and frictional clutch
Ka-25K	340	19000	Elastic with rubber elements
Ka-26	2700	865	Elastic with rubber elements
V-44	—	—	Elastic plate clutch

The transmission shafts, connecting an engine and a gearbox of the main rotor, usually consist of two clutches and a spring. Intermediate supports are not available. One of the clutches often works in a zone of heated parts of the engine that it is necessary to consider at calculation, designing and test of clutches. The main shafts-to-main gearbox connection is carried out by the free wheel clutch.

The main types and characteristics of clutches of the second group are given in Table 11.7.

Table 11.7

The Clutches of the Second Group

Name of Helicopter	Maximum Torque Moment, N·m	Rotation Frequency rev/min	Type of Clutches
Mi-1	143	2263	Cardan clutches
Mi-2	215	2467	Cardan clutches
Mi-4	486	2400	Cardan clutches
Mi-6	4300	2065	Gear clutches
Mi-8	790	2596	Gear clutches
S-58	284	3150	Elastic rubber
V-44	215	4300	Elastic plate clutches
SN-47C	2520	7065	Elastic plate clutches

11.4.2.2. Design and Bases of Calculation of Clutches of Various Types

11.4.2.2.1. Elastic Clutches

For mitigation of dynamic loadings and decrease in intensity of fluctuations shafts are engaged by means of elastic clutches. Elastic clutches allow, some compensation of inaccuracy of mutual position of shafts.

The elastic clutch consists of two semi-clutches and elastic elements which are made metallic (plate disks or steel springs) or nonmetallic (rubber or plastic).

When applying dynamic loadings elastic clutches accumulate and partially dissipate energy. By introduction of elastic clutches it is possible to leave a resonance and to prevent emergence of resonant fluctuations. Energy in clutches dissipates under the influence of external and internal friction in material of elastic elements. In clutches with steel springs which crucial importance has external friction, in clutches with nonmetallic elastic elements – internal friction.

A. Clutches with Rubber Elastic Elements

Clutches with rubber elastic elements possess the following advantages:

– ability to accumulate a large amount of energy per unit mass – in 10 times more, than clutches made from spring steel;

– the considerable damping ability;

– the electroisolating ability.

Clutches with rubber elastic elements are highly elastic, compact, reliable and easy-to-work, have good depreciation properties. In the technological relation they are simpler, than clutches with steel elastic elements. The considered clutches have some limitations: structural changes under external influences and, as a result, gradual change of elastic properties; insignificant angular shift 20…30′ (for high-speed clutches 10…15′); big unevenness of loading of rubber blocks at compression and stretching. Thereof there is rubber destruction – cracking of the stretched zones of rubber blocks. Therefore the durability of rubber elastic elements is less, than steel.

In clutches [2] elastic rubber elements, carrying the uniform and uneven compression, difficult resistance with prevalence of compression, bend, shift, shift with compression and bend, and torsion, are applied.

In Fig. 11.48 the elastic clutch consisting of two metal semi-clutches and rubber blocks pasted to blade-type semi-clutches when curing rubber is shown.

When developing a design of the elastic clutch the geometrical sizes of metal and rubber elements (Fig. 11.49) and the characteristic of rigidity of elastic clutches are determined. Main recommendations about a choice of parameters of elements of the elastic clutch:

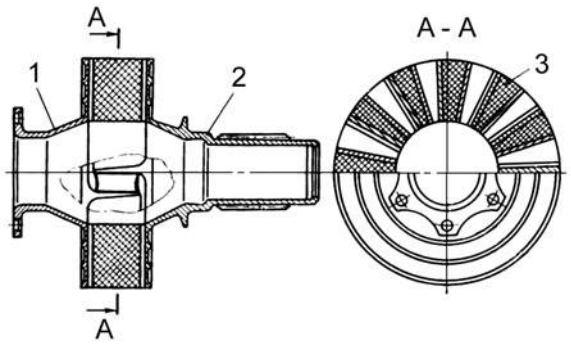


Fig. 11.48. Flexible Clutch:
1, 2 – semi-clutches; 3 – rubber blocks

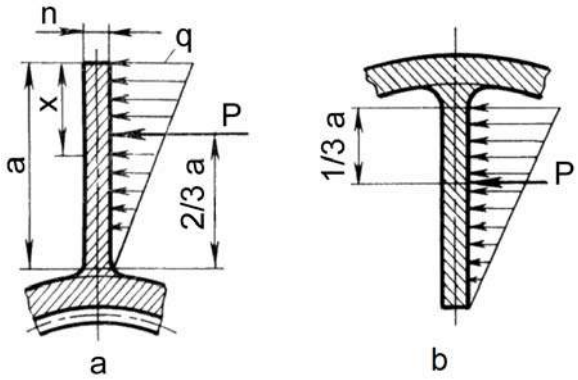


Fig. 11.49. Scheme of Loading of Semi-Clutches:
a – internal semi-clutch;
b – external semi-clutch

– the ratio of outer and internal diameter of blocks shouldn't exceed 1.8 to 2.0;

– with an accuracy, sufficient for practice, it is possible to assume that all rubber blocks take up identical forces from a torque:

$$P = \frac{M_t}{zR_a}, \qquad (11.37)$$

where M_t – the torque effect on the clutch; z – number of blocks; $R_a = (R + r)/2$ (R and r – external and internal radii of the block respectively);

– width of the block b is defined by the size of admissible tension of shift of rubber (b/R =0.2…0.4);

– compression tension in blocks 1.5 to 2.5 MPa (smaller value for high-speed clutches), shift tension τ – 0.08 to 0.14 MPa.

Calculated values of radial and flexural rigidity [4] of elastic clutches allow to estimate own frequency of flexural fluctuations of connecting shaft of helicopter transmission.

Semi-clutches produce from alloyed steel of types 40X2H2MA, 40XHMA or from aluminum alloys; elastic elements – from special grades of rubbers (for example, in the clutch of the main shaft of the Ka-26 helicopter grade 2462 rubber is used).

B. Clutches with Metal Elastic Elements

In clutches [1, 2] metal disks (lamellar clutches), radial and torsional rigidity of which is commensurable with radial and torsional rigidity of the connected shaft are applied. Clutches are used for transfer of big torques at high circumferential speeds, providing reliability and necessary resource of transmission with minimum costs of the operational period.

Clutches (Fig. 11.50) consist of two semi-clutches and the elastic link gathered from the thin steel lamellar disks bolted with semi-clutches.

Lamellar elastic clutches are made of steel 23X15H 5AM3-Ш-H1 with a fluoroplastic covering.

When using the elastic lamellar clutch of a transmission shaft the maximum angle of axes distortion no more than 30′ (an assembly angle – 15′) is allowed.

When applying dynamic loadings elastic clutches accumulate and partially disseminate energy thanks to external friction between plates and to internal friction in material of elastic elements. In clutches with steel springs crucial importance has external friction. By introduction of elastic clutches it is possible to leave resonance area.

The main difficulty at design of these clutches is strength analysis of elastic disk and definition of characteristics of the clutch rigidity. The method of clutch calculation is given in work [4].

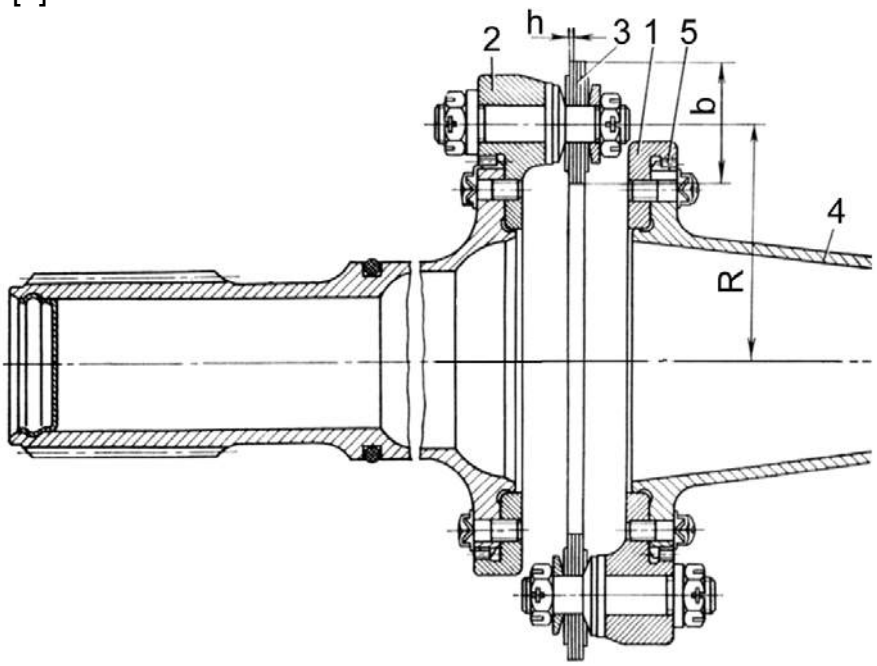


Fig. 11.50. Clutch With Elastic Element in Form of Steel Disks Set:
1, 2 – semi-clutches; 3 – set of steel disks; 4 – spring; 5 – spline joint

11.4.2.2.2. Slip Clutches

These clutches are applied to connect shafts with inexact combined axes and the deformable basis [2]. They compensate the small radial, axial, angular or combined shifts. Inaccuracy of installation, deformation of fastening assemblies of the connected units can be the cause of shifts during operation of the helicopter, and in certain cases shifts are regulated at assembly of transmission shaft by helicopter in advance.

Operation of the slip clutches is connected with power consumption on overcoming of friction forces when moving their separate elements relatively each other or with deformation of elastic element of the clutch. Emergence of the bending moment M_b in installation site of clutches reduces a resource of parts of shafts, causes vibrations in the connected transmission units.

The design of the slip clutch and its parameters are defined by a torque M_t, angular speed ω and required angle of distortion α.

In helicopter transmissions such types of the slip clutches are applied:

- gear;
- hinged (cardan) with spline joint.

A. Gear Clutches

Gear clutches (Fig. 11.51) are compact, hi-tech, have stable operational data. Their shortcomings are: need of greasing and periodic control for operation, an insignificant admissible distortion angle. Clutches can transfer M_t from 800 to $1 \cdot 10^6$ N·m (M_t are actually regulated from 400 to 7000 N·m at n =3000 rpm).

Teeth of clutches should have the high contact durability, not less than 58 HRC, which is provided with their high superficial hardness. In changing of angle of distortion from 0 to 1°30′ bearing stress σ_b increases more than three times. Teeth of plugs and holders have a usual involute profile. Now in helicopter transmissions gear clutches with angles of engagement α_e =20° and α_e =30° are used.

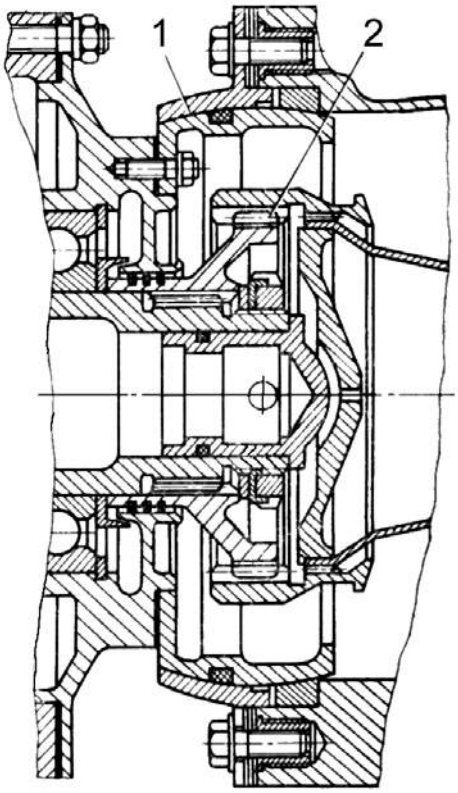


Fig. 11.51. Connecting Couplings of Engine Shaft
with Main Gearbox Shaft of Helicopter Mi-8:
1 – spherical aligning surfaces; 2 – gear clutch

Centering of semi-clutches is carried out on the outer diameter of spherical teeth or on the special aligning surfaces providing a distortion of semi-clutches. For uniform strength of teeth of sleeves and holders gearing is corrected, as a rule. Number of teeth of each wreath is of 30 … 80 depending on the size.

It is necessary for possibility of compensation of linear and angular errors of arrangement of the connected shafts:

- to carry out gear connection with a side gap;
- to grind blank of the sleeve for cutting of teeth on a spherical surface;
- to give tooth the barrel-shaped form.

During the operation of the clutch the contacting teeth for each turn slip from each other in the axial direction therefore the main criterion of operability of clutches is

wear resistance of teeth. The sizes of clutches depend on the ultimate moment M_{ut}, which is determined by the greatest acting torque M_t transferred by the clutch:

$$M_{ut} \le k_1 k_2 M_t.$$

Here k_1 – coefficient of safety (k_1=1.8); k_2 – coefficient of operating conditions of the clutch (k_2=1.3).

After that calculation check strength condition on the greatest quickly acting torque:

$$M_{t.\max} \le 2 M_t.$$

Moment M_t for clutches is chosen according to calculation for the contact tensions proportional to width of teeth b and a square of diameter of a pitch circle d_p:

$$M_t = G \cdot b \cdot d_p^2,$$

Where G – proportion coefficient, G=11.25.

The method of calculation of gear clutches with full-height teeth by bearing stress and gear clutches with barrel-shaped teeth for contact and flexural durability is given in works [4, 9]. The durability of gear clutches is, as a rule, limited to teeth wear.

B. Hinged (Cardan) Clutches

The hinged clutch (with use of the hinge of Guk) is represented in Fig. 11.52. The hinged clutch [1, 2] can transfer the moment M_t to $3 \cdot 10^6$ N·m with angular shift to 45º. Semi-clutches of hinged clutches are made in the form of the forks turned from each other at right angle and hinged to the intermediate body which usually has a crosspiece form with the hinge. The friction bearings or antifriction bearing which are periodically demanding controlled greasing are installed in hinges.

Hinged clutches are technologically more complex than gear ones, and their application is expedient only in the presence of big angular shift between shafts (assembly and operational). Rotation of the driven shaft of the single hinged clutch if it isn't coaxial with the driving shaft, happens unevenly at uniform rotation of the driving shaft.

Ratio between angular speeds of the driven ω and driving shaft ω_0 (i.e. the instant transfer relation):

$$\frac{\omega}{\omega_0} = \frac{\cos\gamma}{1 - \sin^2\gamma \cdot \cos^2\alpha'}, \qquad (11.38)$$

where γ – an angle between axes of shaft; α' – angle of rotation of the driving shaft from some starting position.

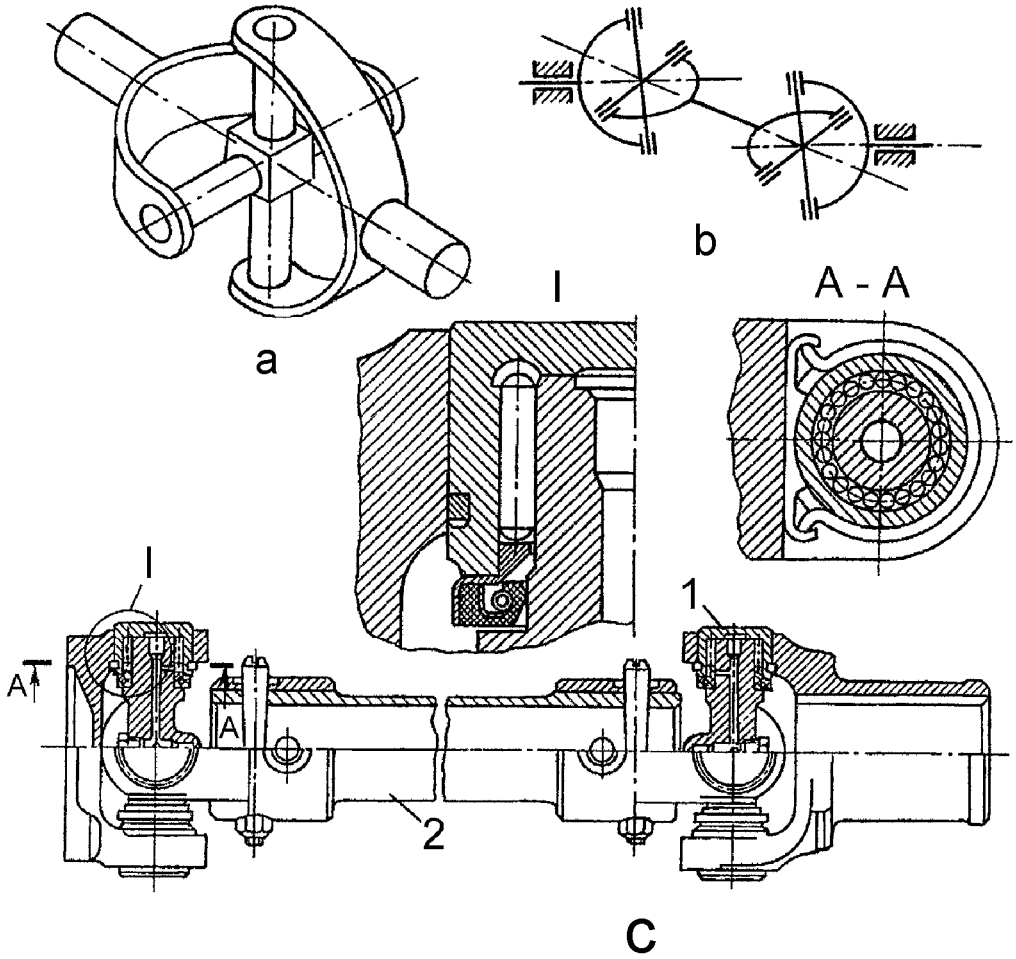


Fig. 11.52. Hinged (Cardan) Clutch:
a – simple hinge; b – dual hinge; c – shaft section with Guk's hinges:
1 – hinge crosspiece; 2 – spring

Coefficient of unevenness of rotation of the driven shaft

$$k = \frac{\omega_{max} - \omega_{min}}{\omega_0} = tg\,\gamma \cdot \sin\,\gamma\,. \qquad (11.39)$$

At small angles γ

$$k \approx \gamma^2 \ (\gamma \text{ – in radians}). \qquad (11.40)$$

The variable instant transfer relation of the hinged clutch causes additional dynamic loadings; also blows at selection of gaps are possible in different directions. Synchronism of rotation of the driven shaft of the clutch can be provided with installation of two consistently connected hinged clutches with an intermediate shaft. Axes of the driving and driven shafts have to make identical angles with an intermediate shaft, and forks on both ends of an intermediate shaft have to be in one plane. Synchronism can be also reached by use of special synchronous hinged or dual clutches [4].

At operation of hinged clutches at big angular speeds of rotation and the considerable transferred torques $\grave{I}_t$ there are torsional and flexural fluctuations which can significantly affect a shaft resource.

Hinged clutches are calculated on restriction of specific pressure [9] by pressure size in hinges p = 35…40 MPa.

Recently in helicopter transmissions hinged clutches have been mainly used for the drive of auxiliary units of the power plant (fans, boxes of drives, etc.).

11.4.2.2.3. Clutches

In need of a smooth torque transmission from the working piston engine on a shaft of the main rotor clutches are used. In transmissions of helicopters frictional and cam clutches are used as coupling joint.

Operation of frictional clutches is based on creation of friction forces between elements of the clutch [2]. Friction forces are regulated by force of compression of the rubbing surfaces. Therefore frictional clutches allow smooth clutch at any speed with adjustment of actuation time. In an initial stage of the frictional clutch actuation the sliding takes place. At the established movement sliding is absent.

The greatest application was found by multidisc clutches, for which flat side surfaces of disks are working surfaces. The number of disks shouldn't exceed 8 to 12 that is related to unevenness of forces of compression because of friction on splines and with a bad disengaging capability at a large number of disks.

Frictional clutches are:

– normally opened which at the released (idle) control mechanism are opened;

– normally closed which at release of the control mechanism are linked under the influence of a spring of the control mechanism.

Normally opened clutches have the main application.

As an example we will consider the combination clutch of the engine reducer of the Ka-15 helicopter [4]. The combination clutch of a reducer consists of frictional and cam clutches (Fig. 11.53).

The frictional clutch is intended for a detachment of the main gearbox from the engine at its start, and also for smooth promotion of the main rotors after start of the engine, it consists of the case of satellites 1, nine driving and ten driven disks, a hub of the combination clutch 2 and the piston 3 with a piston ring.

The cam clutch consists of a mobile flange 4 with four face teeth and the cam flange made as a single whole with the case of satellites 1. Eight recess openings from the top side of a flange are intended for installation of the springs which are constantly wringing out a mobile flange to cams on the case of satellites. The mobile flange is installed by internal evolving splines on output shaft splines 5 along which it can move. The mechanism of control of clutches is intended for consecutive engagement of frictional and cam clutches.

Engagement and disengagement of clutches is made by turn of the roller 6. At turn of the roller fork levers 7 turn.

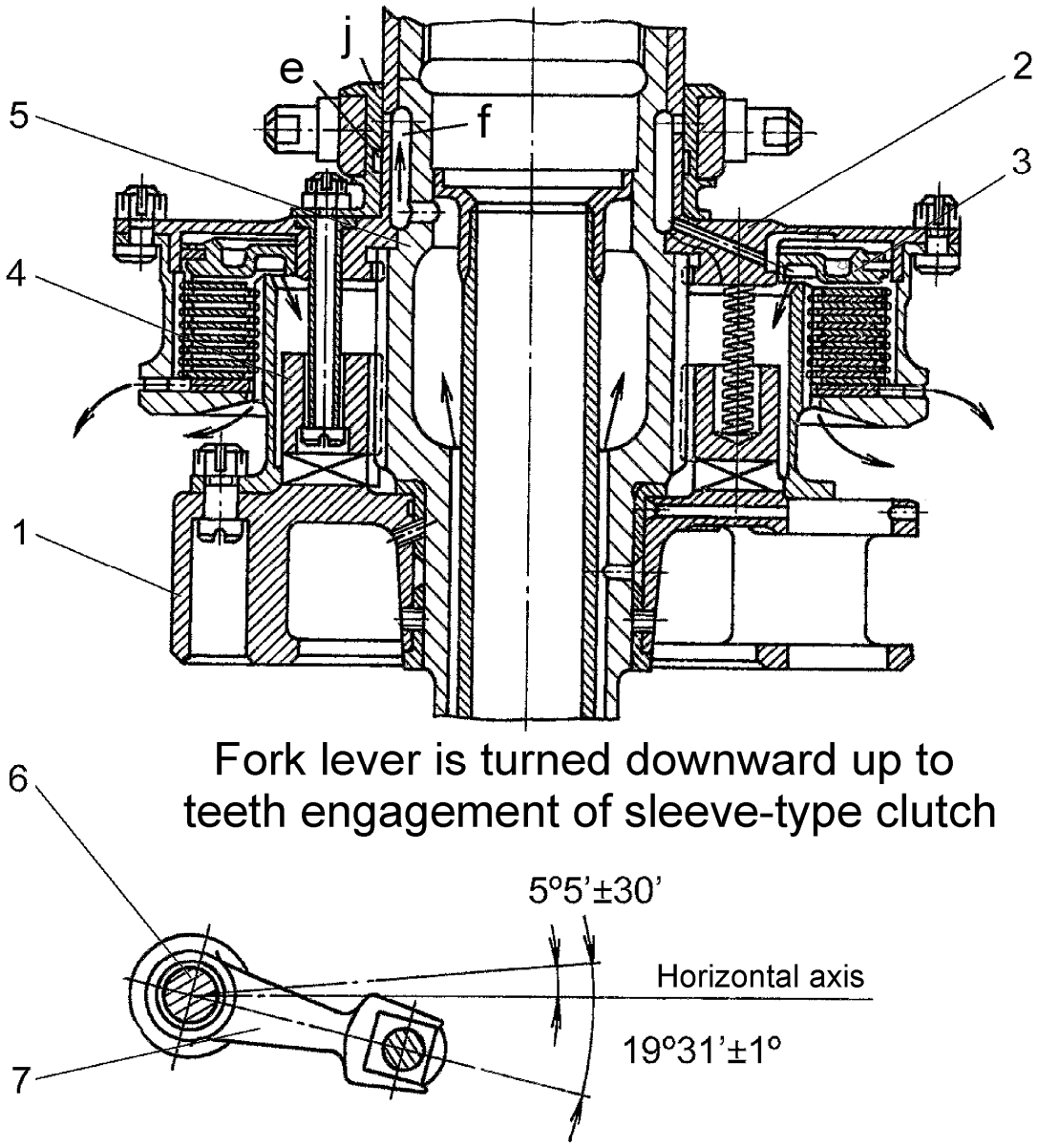


Fig. 11.53. Combination Clutch (Frictional and Cam)

At the initial moment at start of the engine both clutches are disengaged. In this case fork levers 7 are in the top position, the mobile flange 4 is lifted up to a stop in a hub of the combination clutch. Springs thus are squeezed.

Then in process of gaining of turns by the engine the frictional clutch engages. At turn of fork levers down on an angle of 6°47′-30′ oil moves from extreme top position in a cavity over the piston. Oil, arriving from an oil pump, presses on the piston, the piston squeezes the driving and driven disks and thus turns on the frictional clutch. In this position the cam clutch isn't engaged, and the torque is transmitted from the case of satellites to a vertical output shaft through the frictional clutch. Thereof there is a smooth spin up of the helicopter main rotor.

The cam clutch is engaged by turn of the fork lever on an angle of 9º31′+1º from extreme top position. In this position the frictional clutch isn't switched off as oil still moves in a cavity over the piston.

Turn the fork lever on an angle 19º31′ + 1º from extreme top position and engage fully the cam clutch. At this position oil supply stops, and the frictional clutch will be disengaged. Springs completely enter into gearing cams of a mobile flange 4 in mating cams on the case of satellites 1, and the cam clutch via which transfer of engine capacity to a vertical shaft of the main gearbox is carried out and further by the helicopter main rotor, is engaged.

On the autorotation modes when the engine works at the low frequency of rotation or in general it is stopped, the mobile flange 4 is uncoupled on teeth with the case of satellites and there is a rupture of kinematic communication between the engine and the main rotor.

Frictional clutches are calculated by the nominal moment with admissible overloads.

At the chosen radial sizes of the disk clutch the required number of surfaces of friction is determine by a formula

$$i \geq \frac{\beta M_t}{\pi(R^2 - r^2)R_a \cdot [p] \cdot f}, \qquad (11.41)$$

where β – engagement safety coefficient (β=1.25…1.5); $\grave{I}_t$ – torsional moment; R and r – external and internal radii of a ring surface of friction (frictional disks), the recommended ratio r/R=0.5…0.7; $R_a = (R + r)/2$ – average radius of a surface of friction; f – friction coefficient (f=0.3 – pressed material on a basis "asbestos-steel", f=0.4 – "metal ceramics – the tempered steel"); p – allowed pressure: p=2…3·10^2 KPa – pressed material on a basis "asbestos-steel", p= 2…4·10^2 KPa – "metal ceramics – the tempered steel"); i – number of couples of surface of friction.

Required compression force of disks

$$Q = \frac{\beta \cdot M_t}{R_a \cdot i \cdot f}. \qquad (11.42)$$

11.4.2.2.4. Freewheel Clutches

Freewheel clutches are mandatory elements of helicopters transmission. They typically are the part of the main gear and connect high speed input shafts with the first stage of reduction mechanism [6]. The main purpose of helicopter's freewheel clutches is automatic disconnection of the failed engine shaft from the gearbox and the flight safety in autorotation or with one operated engine. In addition, freewheel clutches allow the engines work in different modes during their alternate test on the ground.

Depending on the mode of functioning there exist ratchet and friction freewheel clutches. Ratchet freewheel clutches work on the principle of engagement. Friction freewheel clutches operate due to jamming between driving and driven links of auxiliary elements (balls, rollers, wedges, etc.).

In aeronautical engineering the friction roller clutch is widely used to transfer moments from 500 to 5000 N·m and n=2600...11.700 r/min. It becomes possible due to a number of roller clutches advantages, the main ones are:

– the ability to transmit heavy loads at relatively small overall dimensions;

– high relative speed of links and small moments of frictional forces during the overrunning;

– jamming with a minimum angle of relative rotation of links, ease of wedging;

– manufacturability;

– ease of operation and maintenance.

The main structural elements of the freewheel roller clutch are sprocket, holder, rut rollers and holding down device.

Purpose of the holding down device is to provide the rut rollers contact with a ferrule and a chain wheel. During this the elimination of "backlash" while jamming the freewheel clutch is achieved, i.e., the readiness of a clutch for instant jamming and simultaneous jamming of all the rollers is provided. Furthermore, the holding down device facilitates equal load distribution between both the rollers and lengthwise of each.

Holding down devices can be divided into individual and group (separator).

Separator holding down devices allow much greater rate of relative motion of the links during free stroke than individual clamps, and allow to place a large number of rollers in the freewheel clutch. It provides high coupling load capacity of clutches and their widespread use in helicopter transmissions.

By design features helicopter freewheel clutches of gearboxes are divided into two main types: 1) with spring pressers; 2) with friction clamping devices.

They may also be combined with the design of the coupling clamping devices containing spring and friction elements. The first of these types includes, for example, the freewheel clutches of gearboxes VR-2, VR-8, and the second is freewheel clutches of gearboxes R-5 and R-7.

As an example, Fig. 11.54 shows the design of gearbox F-5 freewheel clutch.

In the process of freewheel clutch operation in the relative displacement of sprocket 2 and holder 1 due to frictional engagement with the roller end faces 3 with the pressure ring 6 and thrust ring 5, the separator 4 moves with the rollers relative to the sprocket. Spring 7 tightens pressure and thrust rings to the end faces of rollers. By moving the sprocket with respect to the holder in the torque transmission direction the rollers rolled in the narrowest part of the wedge space that allows the engagement of the freewheel clutch. During free wheeling when the holder rotational speed is above

the sprocket speed, frictional forces move separator with rollers in the opposite direction up to the limit stops made on the sprocket and the separator.

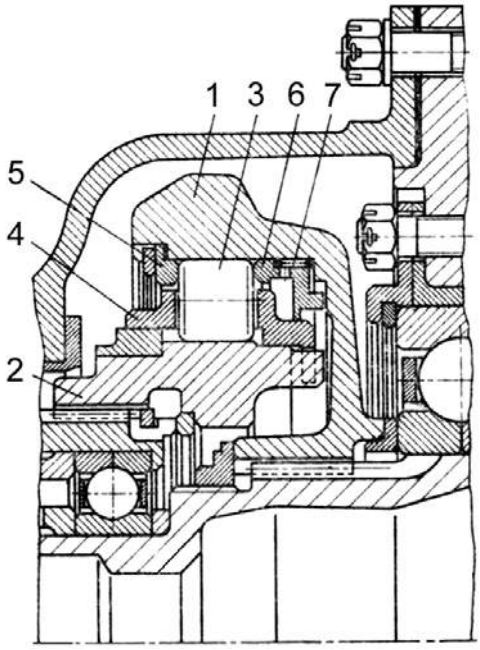


Fig. 11.54. Construction of Gearbox R-5 Freewheel Clutch:
1 – holder; 2 – sprocket; 3 – rollers; 4 – separator; 5 – thrust ring;
6 – press ring; 7 – spring

During this process the rollers come out of the interaction with the working sites of sprocket. This embodiment of the pressure device ensures minimum friction loss and prevents wear of the sprocket working areas during free running of the freewheel clutch.

Separators of gearboxes R-5 and R-7 freewheel clutches of the first series are made of aluminum alloys D1T and D1; separators of gearboxes R-7 freewheel clutches of the second series is made of bronze AZHN 10-1-1.

The Choice of Design Parameters of Freewheel Clutch Parts. In order to reduce the dimensions of the freewheel clutches it is necessary to apply the greatest possible number of rollers and provide more uniform contact with a holder and a sprocket. The uniformity of the pressure distribution along the length of the rollers can be enhanced by making the rollers slightly convex (with cambering) or slightly tapered rollers at the ends.

A great influence on the uniformity of load distribution between the rollers and along the length of each of them has a manufacturing accuracy of separator and sprockets, and during prolonged operation - the wear resistance of the separator material. The use of aluminum alloys for separators showed that they had unsatisfactory durability. Bronze and steel separators have higher durability. Operating experience and research results of high-speed helicopter freewheel clutches indicate the feasibility of using lightweight separators and the need to improve the quality of their dynamic balancing. Pairing a separator with a sprocket must ensure easy and smooth running of the separator, since its seizure inevitably leads to a violation of the clutch efficiency.

Particular attention in the freewheel clutch design should be paid to the centering accuracy of a sprocket, holder and a separator. In case of eccentric location of sprocket, relative to the holder, sprocket rollers wedge at different angles α, which can lead to considerable overloading of some rollers, while others may not transmit torque.

The recommended number of rollers in the separator for a helicopter freewheel clutches is z_p = 8…20 pcs. The values of parameters $K_1 = D/d$ and $K_2 = \ell_p/d$ (where ℓ_p is the roller length, d is the roller diameter, D is the diameter of the holder hole) are advisable to choose within K_1 = 7…10 and K_2 = 1.5…4.

Holders and sprockets of helicopter gearboxes freewheel clutches are made of hardened steel 12X2H4A (carburizing depth is 1.0…1.7 mm, HRC>60).

For the manufacture of rollers it is recommended to use steel SHKH15 (HRC 59…63), U8A (HRC 60…62), U10 (HRC 59…62), HVG (HRC≥62).

The roughness of the contact surfaces of high-speed freewheel clutches basic parts must comply with the 7-9th grade. Operating experience of the freewheel clutch with the 9th grade roughness of cage and sprocket working surfaces, indicate a high level of performance and reliability during the period of free running as well as during jamming.

Great influence on the effective work of the freewheel clutch has a wedge angle α - the angle between the tangent drawn through the point of contact between the roller with a holder and a sprocket. To reduce the effect of manufacturing inaccuracies and wear of parts working surface by an amount of wedge angle of rollers it is not recommended to make sprockets not planar but cylindrical with the guide made as a circle, Archimedean or logarithmic spiral.

The choice of roller wedge angle when designing roller freewheel clutches is usually made on the basis of experimental data of their construction and operation. For self-wedging freewheel clutches wedge angles are selected within 3º< α < 5º30′, and tolerance on the value of the wedge angle is ±15′.

Computation of Freewheel Clutches. The complete movement cycle of freewheel clutch members includes four periods: free stroke, seizure, jammed state and wedging.

The main criteria of the freewheel clutch performance are [2]:

– the ability of clutch to wedge without slipping;

– contact fatigue strength and endurance of the most loaded elements;

– the absence of increased wear of the working surfaces of the parts at high speed relative angular movement of the members between during free wheeling.

Calculation of contact strength. The maximum torque transmitted by the freewheel clutch, is defined as the sum of the nominal and dynamic moments $M_{t.\,max} = M_n + M_d$ or $M_{t.\,max} = k_d \cdot M_n$, where k_d is dynamic coefficient depending on the type of engine, gearbox, transmission design features. For the

freewheel clutch propulsion of helicopters the dynamic factor is selected according to the experimental studies results of torsional vibrations of shafts, which connect the engine to the gearbox. For example, the most dynamic factor for the freewheel clutch of gearbox R-7 of helicopter Mi-6 is 1.2. For approximate calculations k_d=1.1…1.2.

Errors in the manufacture of parts, resulting in uneven load distribution between the rollers, are taken into account due to the accuracy coefficient k_m. For freewheel clutches with flat working sites of sprockets this coefficient equals k_m=1.1…1.5; for freewheel clutches with the curved profile of sprocket working surfaces k_m=1.0…1.25.

Ultimate moment

$$M_{t.\max} = k_d \cdot k_m \cdot M_n. \qquad (11.43)$$

Freewheel clutches of helicopters are mechanisms with a small number of engagement during life cycle (three to five engagements per hour of operating time). They are a long time working in a jammed state with a smooth change of the transmitted load and the relatively low intensity of torsional vibrations in the drive train.

By the nature of the load of contact surfaces of parts in a jammed state freewheel clutches are the closest to the rolling bearings operating under the rocking (oscillating) motion.

Verification calculation of the roller clutch is carried out by contact stresses in the roller-to-sprocket contact zone [9]:

$$\sigma_n = 0.418\sqrt{\frac{N}{\ell}\frac{2E_1E_2}{E_1+E_2}\left(\frac{1}{r_p}-\frac{1}{r_0}\right)} \le [\sigma_n], \qquad (11.44)$$

where N is normal interacting force of roller with a sprocket; E_1 and E_2 – modulus of elasticity of roller and sprocket; r_p – roller's radius; r_0 – the radius of curvature of the sprocket working surface; ℓ – roller length.

11.4.3. Selection of Helicopter Transmissions' Bearing Blocks

The bearing units of helicopter transmissions use different types of rolling bearings [2,4,6,9]. Plain bearings are used very infrequently (e.g., satellites bearings of helicopter Mi-4 main gearbox), in recent constructions they are used just in assistive devices (low-power drives, control devices etc.). Types of bearings and their varieties depend on the size and nature of the acting loads, technological and operational requirements.

Construction of bearing assemblies is carried out simultaneously with the construction of other transmission components. It is determined by the general requirements for a specific assembly of transmission.

After development of general layout of the transmission unit kinematic, power and performance characteristics of the bearing assembly and lubrication system are determined. One can calculate the resource of assembly bearing, rigidity of the crankcase, the required quantity of lubricant, and so on. The quality structural features is checked by the long bench and field tests, at which shortcomings in the work of bearing assemblies can be identified and eliminated.

Let's consider the features of the bearing assemblies used in helicopter transmissions, examples of embodiments of such units, as well as some basic recommendations for their settlement.

11.4.3.1. Features of Rolling Bearings Used in Helicopter Transmissions

Features of rolling bearings used in helicopter transmissions, can be defined by the area of application and characterized by limited dimensions, low weight, high dynamic load-carrying capacity and increased reliability within the specified resource.

The bearings used in helicopter transmissions are usually heavily loaded units, often specially designed for specific operating conditions.

In comparison with the bearings used in the transmissions of other transport vehicles, helicopters bearings are usually made in view of the special technical conditions and have a relatively limited lifetime.

The required durability of bearing units and helicopters mechanisms ranging from a few hours for short-term response units to several thousand hours in helicopter gearboxes.

In catalogs and reference books the dynamic load values for a normal class accuracy and uptime probability is given as $P=0.9$. Units and mechanisms for helicopters generally require higher reliability. Therefore helicopters bearings should have reliability higher than standard bearings of zero class. So, for a number of units and the helicopter mechanisms reliability of bearings must not be less than 0.99.

For helicopters bearings the high-quality materials are used (electroslag remelting steel or vacuum melting steel), massive separators made by taking into account the work of the bearing in the assembly are used; these materials perform correction form of rolling elements ("cambering" of rollers). These bearings are characterized by increased precision components and a higher volume of monitored parameters, both geometrical and technological.

Furthermore, special bearings are widely used (with spliced rings, double balls bearings, etc.).

11.4.3.2. Calculation of Helicopter Transmissions Bearing Blocks

Rolling bearings are almost exclusive transmission blocks for which there is an established and regulated method of choice and calculation.

Currently methods of selecting and calculating of rolling bearings, given in the international standard, is accepted ISO [2]. Rolling bearings are calculated for durability (service life) by dynamic load and static load capacity. Bearing life is determined by flaking on one of the bearing parts. The relationship between the design life L and reduced dynamic load P (combined load is reduced to either radial or axial, which is equivalent to its destructive action) is determined by empirical dependencies,

$$L = \left[\frac{C}{P}\right]^{\alpha} \quad \text{or} \quad L_h = 10^6 \frac{L}{60n} \ , \tag{11.45}$$

where L is the number of revolutions in millions during the design life L_h, h; C is dynamic load capacity of a bearing; α is an exponent of 3 for ball bearings and 10/3 for roller bearings; n is rate of rotation.

Due to static load they select or check the friction bearing, taking up external load at rest or during rotation at a frequency of no more than 1 r/min.

The values of dynamic and static load rating, N, are indicated in catalogues for each bearing size and type. [47]

The reduced load is determined by the formula

$$P = (X \cdot K_R \cdot R + Y \cdot A) \cdot K_s \cdot K_T ,$$

where R and A are radial and axial loads; X and Y are coefficients of radial and axial loads (dimensionless); K_K is dimensionless coefficient of rotation (during rotation of the inner ring K_K=1, and the outer one K_K=1.2); K_s is safety coefficient; K_T is temperature coefficient.

Values of the loading coefficients X and Y are specified in the catalogue [13].

The reduced load for the roller bearings with short cylindrical rollers (no axial load)

$$P = R \cdot K_s \cdot K_T \cdot K_R . \tag{11.46}$$

The reduced load for thrust ball bearings and roller bearings

$$P = A \cdot K_s \cdot K_T . \tag{11.47}$$

Equivalent loading at intermittent operation if it varies linearly from P_{min} to P_{max} is defined by the formula

$$P_{eq} = \frac{P_{min} \pm 3P_{max}}{3} . \tag{11.48}$$

At step change of loads and the frequency of rotation the equivalent load can be calculated by the formula

$$P = \sqrt[3]{\frac{P_1^3 L_1 + P_2^3 L_2 + \ldots + P_n^3 L_n}{L}} , \tag{11.49}$$

where P_1, P_2, P_n are constant loads, which operate during L_1, L_2, ..., L_n of rotations respectively; $L = L_1 + L_2 + \ldots + L_n$ are the total number of revolutions during which the load acts on the bearing P_1, P_2, ..., P_n.

The concept of bearing rapidity is very important for helicopter bearing units. Reserve specific speed of rolling bearings n_l is the greatest for the bearing speed above which the calculated bearing life is not guaranteed.

High speed is estimated due to $(D_a n)$ = const, where $D_a = (D + d)/2$.

Values $(D_a n)$ for some types of bearings with stamped steel separators are given in Table 11.8.

Table 11.8

Range of Specific Speed Settings

Bearing type	$(D_a n)$, mm·r/min
Single radial and angular contact ball	$5 \cdot 10^5$
Double row self-aligning ball	$4.5 \cdot 10^5$
Thrust single ball	$1.5 \cdot 10^5$
Roller bearing with short cylindrical rollers	$4.5 \cdot 10^5$
Conic single-row	$3.0 \cdot 10^5$

11.4.3.3. Designing and Manufacturing of Bearing Assemblies

The construction of helicopters bearings takes into account the nature of the load and the design features of the transmission elements. The 5th and 6th classes precision bearings depends on the rotational speed and the unit function are used. High speed bearings generally have one-piece bronze separators and less speed bearings – the unsplit separators made of aluminum alloy.

Roller bearings are often used for gear work with spur gear wheels in bearing units [4].

In gears with bevel wheels (in the early stages of the main gearboxes, in the intermediate and tail gearboxes) in the design of bearing assemblies for taking up the axial forces and axial fixation of bevel gears an ordinary or special double angular contact ball bearings with a split inner ring are used.

Bearing blocks of output shafts of helicopters main gearboxes take up considerable axial and radial forces and operate at relatively low peripheral speeds. Either angular contact roller bearings or combination of roller radial and angular contact ball bearings with a split inner ring are used.

Fig. 11.55 shows a typical design of the bearing of the drive wheel of bevel gear units. Its feature is the installation of roller bearings to take up radial loads and angular contact ball bearing with a split inner ring only to take up the axial load. These bearings take up axial loads from both sides. Under identical outer diameters, they can withstand much greater axial loads at high speeds than other bearings. In fact, they are thrust and radial or thrust bearings.

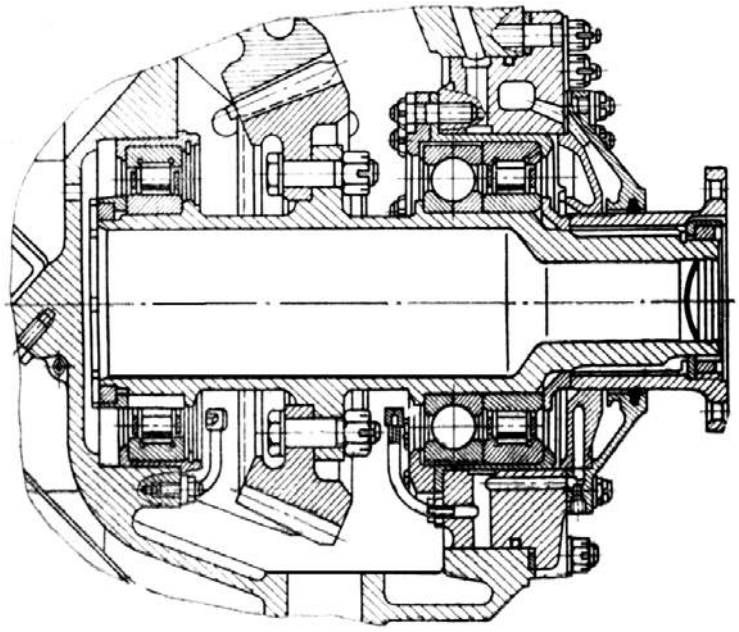


Fig. 11.55. Conic Tooth Wheel Supports of Helicopter Mi-6 Intermediate Gearbox

For operating temperatures up to 200°C, such bearings are made of steel ShKh15 with special heat treatment. At the operating temperatures from 250 to 450ºC rolling elements and rings are made of heat-resistant steel type EhE347 at higher temperatures special heat-resistant alloys are used. This type of bearings is used in helicopter main gearbox.

To exclude the possibility of the thrust bearing by radial force it is mounted in the housing with radial clearance guaranteed from 0.5 to 1 mm. This constructive solution allows to reduce significantly the bearing unit diameter.

The inner rings of bearings are mounted on the shaft at great tension. Reducing the tolerance and, therefore, the impact of fitting on the internal clearance in the bearing is achieved by using a small tolerance of fit field. The most commonly used for power transmission are the inner rings fit onto the shaft – $n5$, $n6$, $m5$ and $m6$; the roughness of the shaft seating surfaces is not more than 0.63; fits of bearings into housing assembly are $Js6$ and $K6$; surface roughness of seating surfaces is not less than R_Z=2,5.

One of the main defects in the power bearing assemblies is cranking of bearings inner rings relative to the shaft and fretting corrosion in the connection "roller-bearing". To prevent these defects they increase tightness and a thickness of shaft under the bearing, i.e. stiffness of the shaft. Typically, the thickness of the shaft under the roller bearing is not less than the thickness of the bearing rings. The configuration of part as the whole (the presence of thrust shoulders, hubs, thread, which will be affected by tightening the nuts) and loading of part or assembly torque, bending, etc, are also taken into account. The seating surfaces for bearings are often cemented, nitrided or chromed.

Tapered bearings are adjusted during the assembly, so there is much stronger acceptable fitting of rings on the shaft and in the housing. When fitting on the shaft fittings n6 and m6 are used, while fitting in housing – Js6 is used or special fit R7 (for thin-walled cases).

Bearing cylindrical wheel assemblies have their own characteristics. Spur wheels do not generate an axial load on the shaft, therefore they don’t require axial bearings.

Bearings can be mounted on the shaft by nuts or simply pressed onto the shaft, depending on the load. Recommended fits on the shaft are k6 and m6, in housing - Js6 and H6. The roughness of the shaft surface is not more than R_a=0.63, inside the housing it equals R_Z=2.5. In such assemblies the axial clearance must be provided.

Requirements for bearing internal clearances for cylindrical gears are less stringent. Axial retention of spur power transmission is carried out with the help of thrust shoulder of cylindrical roller bearings, in particular, the use of bearing with thrust washer on the inner ring.

In the bearing blocks of the satellites, the radial clearance influences radically on the load distribution between the satellites and should be strictly regulated within a single planetary gear set. In this sense, it is better to design special bearing assemblies of satellites, in which bearing is loaded without roller rings and the separator. Bearing tracks of inner and outer rings, treated in a single pass, facilitate the selective assembly of bearing blocks. With this construction, a two-way load application to the satellite causes its distribution on a large number of rollers, resulting in a significant increase in bearing durability compared with the durability of the bearing assemblies of conventional type.

All highly loaded bearings are usually fixed to the shaft in the axial direction by means of nuts. Bearing rings are heated before installation in the oil in a temperature range from 110 to 120°C, while the shaft may be pre-cooled. Then the ring is fitted onto the shaft under the pressure. Nuts are tightened by calibrated moments. Before tightening the thread is lubricated with mineral oils. Tightening is carried out in several cycles: tightening-loosening, etc. Compression of thread at this initial tightening is produced by a torque, which exceeds the assigned one in 1.2 – 1.3 times.

The direction of nut thread is selected in the way that in the event of locking and turning of the bearing ring the nut is tended to screwdriving. Every nut is locked on the shaft by the special safety devices. Some locking variants are shown in Fig. 11.56 (on the right figure: a – place of the sleeve extension deformation in the nut slot).

In the places where the power bearings are mounted into housing made of light alloys, steel molded cups are pressed in. The outer rings of the roller bearings are fixed in the axial direction by the flat split spring rings, light clamping rings or individual strips by means of pins.

In the bevel gear the roller and ball angular contact bearings, arranged side by side, are usually mounted first in a separate cup and then, together with the cup the entire assembly is mounted in the housing.

Sometimes for multisupport helicopter transmission shafts the self-aligning bearings are used. They have considerable advantages in shaft deflection and misalignment of supports. These bearings allow the mutual axis rotation of rings up to 2 ... 3º.

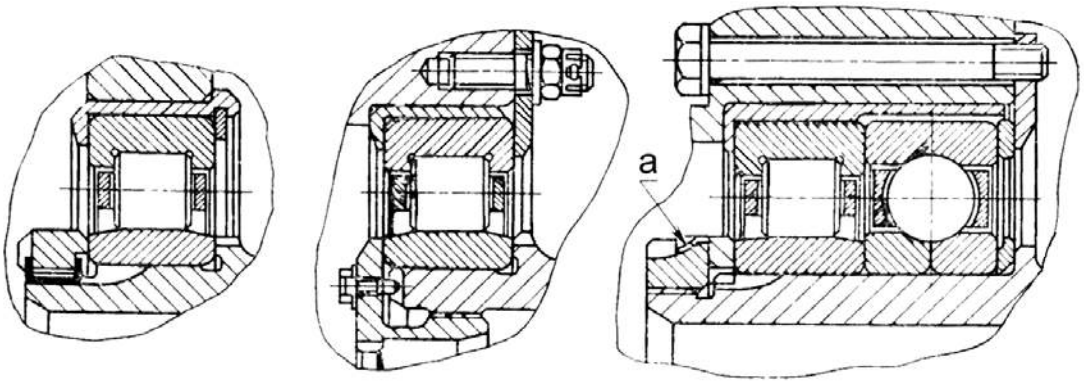


Fig. 11.56. Mounting of Bearings on Shafts and in Housings

Moreover, bearings the structure of which may include shields or seals are used in transmission shafts as additional supports. These parts serve to protect the bearings from dirt and dust, they keep grease from flowing at the temperature of +60°C and higher, and from its extruding within the bearing under dynamic loads action.

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Chapter 12. MANUFACTURING TECHNOLOGY AND METHODS FOR PROVISION OF SERVICE LIFE OF GEARBOXES

12.1. MANUFACTURING TECHNOLOGY OF TOOTH GEARS

12.1.1. Basic Concepts and Definitions

The teeth are widely used in modern technology for the transmission of rotational movements, and for converting rotational to translational motion.

Gears have a number of advantages over transmission of other types (belt, clutch, chain). They provide high power transmission; gear ratios remain constant, have a high efficiency (up to 0.99). The gears are also distinguished by high durability, reliability and relatively small size.

Gears are used in a simple single-stage transmission and the transmission of several different combinations of built into the machine or made as separate units. Gears are widely used to reduce the angular velocity and torque increase in gearboxes. Reducers operate normally in separate buildings of one-, two- and three-stage gear ratios, respectively, with 1.6 ... 6.3; 8 ... 40; 45 ... 200 [1, 2]. The most common are two-stage gearboxes (about 95 %). To obtain different speeds of the output shaft at a constant speed of drive motor gearboxes are used. Possibilities of gear mechanisms are extended with planetary gears that are used as gears and differential mechanisms. Small dimensions and weight of the planetary gear are determined by load balancing across multiple commit planetary motion gears (satellites) and the use of internal gearing, which has a higher load-bearing capacity. In the transition from the simple to the transmission planetary weight reduction is achieved by 1.5–5 times [1]. The lowest relative dimensions are the wave gears, providing transmission of large loads at high kinematic accuracy and rigidity.

Gears are divided into the power ones, to transmit much effort at low speed (transmission of various hoisting and transport mechanisms); speed gears running at speeds up to 150 m / s (for example, gears turboprop engines); the reference to ensure accurate alignment angles of rotation of the driven and driving wheels (tracking system devices, resolver mechanisms and others) and general purpose gears operating at low peripheral speeds and loads.

Profile of lateral tooth surfaces is described by various curve. The most widely used in the industry got gears with involute profile (involute engagement). In some areas of technology, particularly in the instrument, is used a wheel with teeth profiles, rounded on the cycloid. In recent years, variety of heavy-duty mechanisms are increasingly using transmission with Novikov gearing (Fig. 12.1). Working areas of tooth profiles, in this case, are made by arcs.

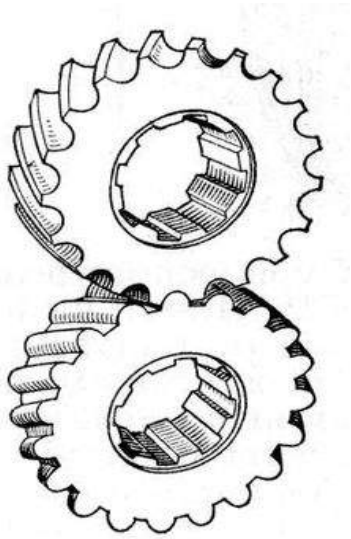


Fig. 12.1. . Novikov's Gear

Gears vary by mutual arrangement on the transmission axes (Fig. 12.2.) With parallel axes – cylindrical; with intersection-repentant axes - bevel and rarely used cylinder-conic and flat face; with intersecting axes – rack and screw (worm, hypoid and spiral) [3, 4].

A particular case is a gear rack and rack gear transforming rotary motion into forward or reverse.

In the majority of machines and mechanisms gears with external gearing are used, i.e. with gear wheels having teeth on the outer surface, at least with internal gearing in which the teeth of the wheel are cut in the inner surface.

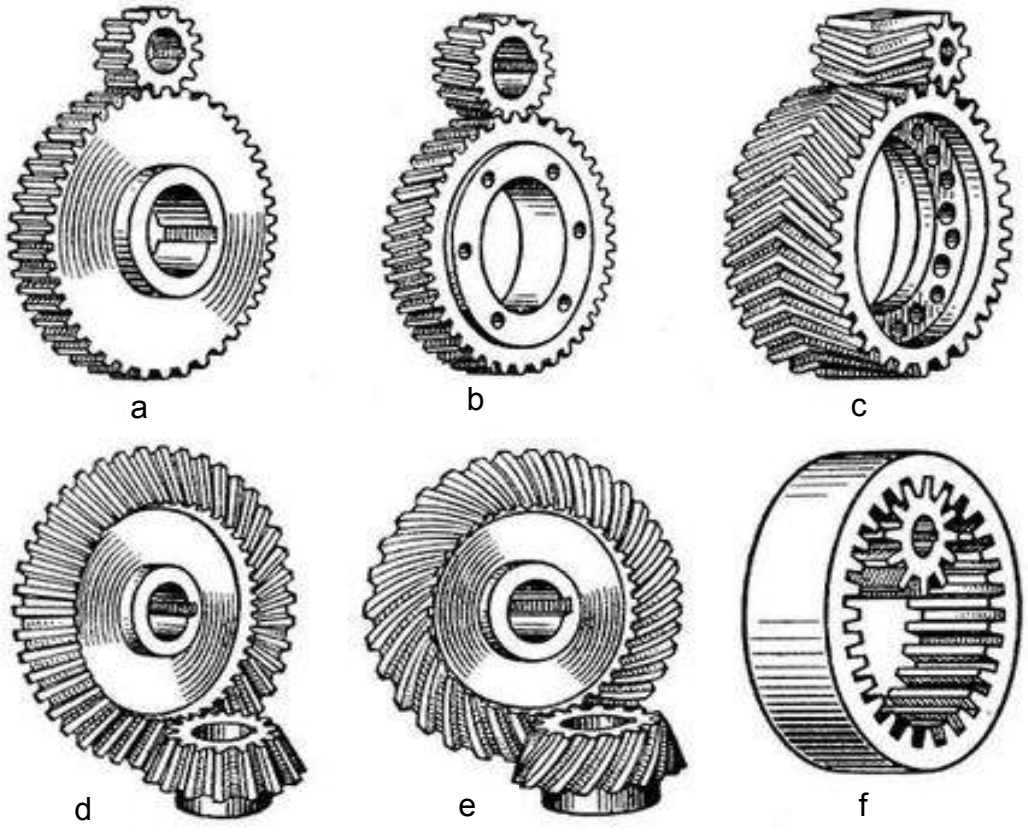


Fig. 12.2. Gear with Cylindrical Wheels: a – spur gear; b – helical; c – chevron; d – conical; e – with circular tooth; f – with inner engagement

Gears can have straight teeth to work at low and medium speeds in open gears and gear boxes; helical teeth for use in the transfer of responsibility at medium

and high speeds (over 30 % of the cylindrical gears); herringbone teeth for the transmission of large torque and power in heavy vehicles; circular teeth – all critical bevel gears.

In order to simplify the shape and size of cut teeth and tools introduced the concept of the rack tooth profile (Fig. 12.3). Standardized parameter values of the initial profile are given in Table 21.1. The same table shows the recommended settings of the tooth profile with a modification (Fig. 12.3, b) used to improve the efficiency of cylindrical wheels. Parameters of the tooth profile are set by GOST 13755 – 81 [3].

Table 12.1

Rack Tooth Profile Data (in accordance with GOST 13755 – 81)

Parameter	The numerical value or formula ($m \geq 1$ mm)
The angle of the main profile α	20°
Factors:	
height of the tooth head h_a^*	1
radial clearance in a pair	0.25
initial contours c^*	
dedendum h_f^*	$h_a^* + c^* = 1.25$
limiting height h_l^*	2
the teeth depth in the pair	$2h_a^* = 2$
initial contours h_w^*	
radius of curvature of the transition curve	0.38
curve ρ^*_f	
modification height h^*_g	0.45 and less
modification depth Δ^*	0.02 and less

Notes: 1. The increase is allowed ρ_f *, if it does not violate the correct wheel engagement.

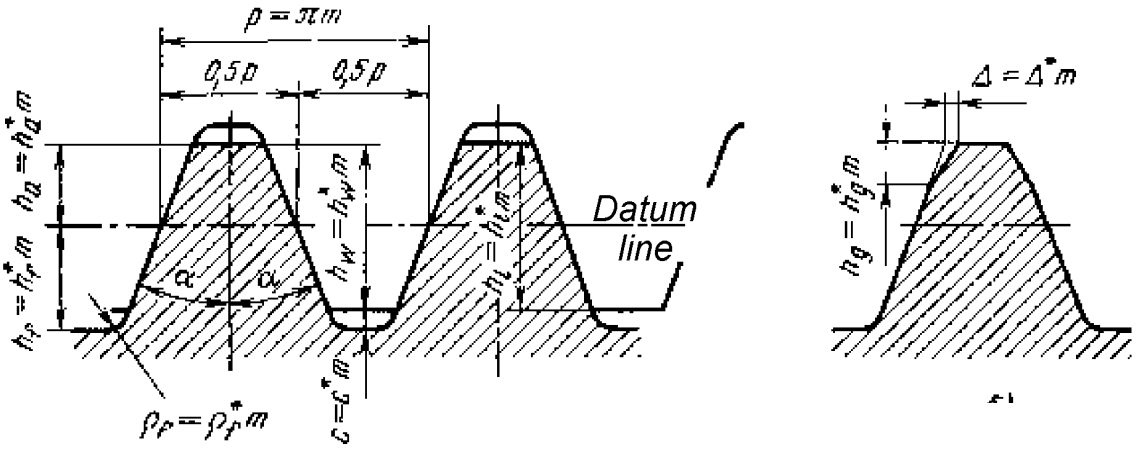
2. For m> 1 is allowed to increase the coefficient of radial clearance from c* to 0.35 in the processing of cutter teeth and to 0.4 in the processing of grinding teeth beneath.

3. 0.1 <m <1 form cavity height hl below the cut can be made or a single arc, or two arcs of a curve and the conjugate or in the form of any transition curve.

4. The values of transmission parameters, which have special requirements, may differ from those set by GOSTs.

5. For m> 1, the value Δ^* must be chosen depending on the module and the degree of accuracy.

Modules values (over 1 mm) for cylindrical and conical gears are regulated by GOST 9563 – 60, selected from the series, mm: 1.0; (1.125); 1.25; (1,375); 1.5; (1.75); 2; (2.25); 2.5; (2.75); 3; (3,5); 4; (4,5); 5; (5.5); 6; (7); 8; (9); 10; (eleven); 12; (14); 16; (18); 20; (22); 25; (28); 32; (36); 40; (45); 50; (55); 60; (70); 80; (90); 100 [3]. Priority should be given to values not in parentheses. For some industries, other modules can be applied.



a b

Fig. 12.3. Rack Tooth Profile: a – without modification; b – with modification

Standards of accuracy and performance of involute cylindrical gear transmission ne-internal and external engagement with the tooth module 1...56 mm are regulated by GOST 1643-81 (see Table 12.2). In accordance with GOST standards four groups of accuracy are established:

- kinematic accuracy;
- smooth operation;
- contact wheel teeth;
- backlash among their non-working surfaces.

The first three groups of norms determine the accuracy of the transmission of rotation and separated by 12 degrees of accuracy, designated in descending order of their accuracy – 1, 2, ..., 12, and 1st and 2nd degree tolerances are not regulated. The most common are 7 – 9th degrees of accuracy.

Thus, the tolerances for the cylindrical wheels 10 are mounted on the degrees (from 3rd to 12th). The first two degrees of accuracy are promising. The most widely used in engineering transmissions are cylindrical wheel of the 5 - 9th degree of accuracy. Manufacturing the accurate bevel gears is more labor intensive than cylindrical gears. Therefore, tolerances are set for bevel wheels and tolerances only for nine degrees of accuracy (from 4th to 12th) [3, 4].

Standards of gear backlash value determine values of backlash for different types of coupling gears A, B, C, D, E, H (backlash value decreases from A to H). Backlash tolerances are indicated in ascending h, d, c, b, a, z, y, x.

To fulfill the operating conditions in the gears manufacture minimize the kinematic error F_{ir}, radial run-out of the ring gear F_{rr} and cyclic error tooth f_{zzor} transmission frequency. The speed gears main operational requirement is to ensure the smooth operation of the wheels that depends on the tooth profile error f_{fz} and deviation of gear pitch f_{pbz}.

Table 12.2

Definitions and Designations of Accuracy Rate of Internal and External Engagement Gears (according to GOST 1643–56 / GOST 1643-81) [5]

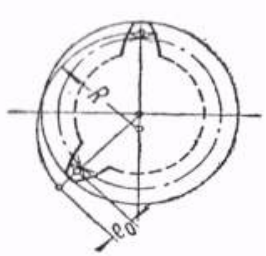
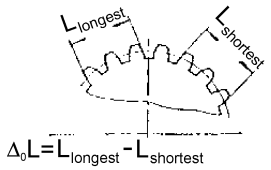
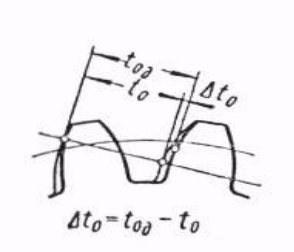
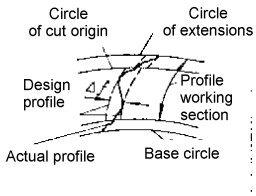
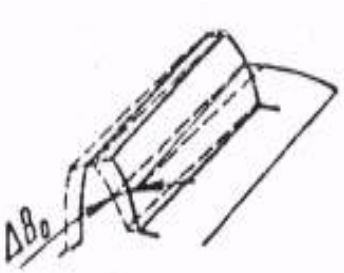
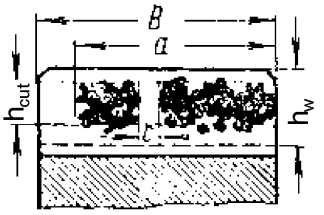
Accuracy figure	Legend: tolerance value of l	Determining the accuracy indicator
Radial run-out of the ring gear 	e_0 / E (by GOST 1643–56) F_{rr} / F_r (by GOST 1643-81)	The greatest fluctuation of distances from the permanent teeth chords to the wheel axis of rotation
Fluctuation of common normal length 	Δ_0L / δ_0L (by GOST 1643–56) F_{vwr} / F_{vw} (by GOST 1643-81)	The difference between the highest L_h and the lowest L_l common normal lengths in one and the same wheel
Deviation of the main pitch 	Δt_0 (by GOST 1643–56) f_{ptr} (by GOST 1643-81)	The difference between the actual and the nominal distance between the two mutually parallel tangents to the two adjacent profiles of the gear teeth of the same name. Defined in section perpendicular to the tooth in a plane tangential to the main cylinder.
Profile error 	Δf / δf (by GOST 1643–56) ffr / ff (by GOST 1643-81)	A distance along the normal between the two theoretical gear tooth profiles that limit the actual profile within its operating area. Relative to the wheel axis is determined in the cross section perpendicular thereto. Tolerance to the profile within a portion having the cut, located in the body gear tooth from the intersection point of the actual beginning of the profile with the circumference of the cutoff. For wheels without cut-offs the error in the profile heads and flank is allowed only in the body of the tooth.

Table 12.2. (to be continued)

Accuracy figure	Legend: tolerance value of l	Determining the accuracy indicator
Accuracy of the tooth direction 	$\Delta B_0 / \delta B_0$ (by GOST 1643–56) $F_{\beta r} / F_\beta$ (by GOST 1643-81)	The distance between the two straight lines or screw nominal direction, lying on the cylinder, passing about the middle of the height of tooth, and is limited to the actual direction of the tooth along its entire length (over-given to spur and narrow helical gears)
Deviations of axial pitches	ΔB_Σ	The difference between the actual nominal distance between any two any surfaces of the teeth of one element of the cylinder
Tooth bearing 		Part of the side surface of the tooth, on which teeth traces fit to the pair wheel teeth after rotation of the assembled gear with gentle deceleration are arranged. It determines the relative time-measures of tooth bearings (in percentage): on the tooth length – the ratio of the distance between the outermost points of abutment traces, minus breaks exceeding the value of a module, mm, to the total length of the tooth $\frac{a-c}{b}\cdot 100\%$; the height of the tooth – the ratio of the average height of abutment traces along the entire length tooth for tooth working height: $\frac{h_{av}}{h_t}\cdot 100\%$

The power gears need to ensure full contact of the teeth in the pairings, i.e. the required total contact area with the allowable error of the tooth direction $F_{\beta r}$ and parallelism of axes f_{xr}.

The roughness of the teeth surfaces in accordance with GOST 2789-73 is assessed in the direction that gives the greatest value: after hobbling – in the transverse direction of the tooth (tooth adjustment); after shaving – in a longitudinal (along the tooth). Surface precision gear teeth have the average roughness R_a = 0.8 mm; wheels used in engineering industry are generally m_a, - R_a = 1.6 ... 6.3 mm.

12.1.2. Tooth Gears' Operating Conditions and Failure Causes

The gear classification according to their functions is the following:

1. Spur and bevel gears of the planetary and direct gearboxes, which transmit high torques. These gears are made of high-strength alloy steel at a high level of accuracy and subjected to different methods of heat, chemical heat, and final treatment.
2. Drive gears: spur, bevel, worm gears (shaped in a simple or complex way as hubs, roller gears, uni- and bilateral cluster gears).
3. Oil pump and oil pump block gears that require producing at a high level of accuracy to provide the necessary air-tightness of engaged surfaces.

Aircraft engine gears operate at high loads, often reaching 700...800 N per 1 mm of the tooth length in a reverse mode at the wheel rim speed of 70 mps maximum. That is why they must be high tensile, durable, highly resistible to the contact fatigue breakdown, light and reliable in operation.

The main reasons for gear sets failure *are the following*:

- teeth defects;
- fatigue spalling of teeth surface layers;
- abrasive wear;
- teeth binding (which occurs when the oil film is destroyed by high pressure or temperature).

12.1.3. Tooth Gear Material

Gears are made mainly of carbon steel (40, 45, 50), alloy steel (50Г, 20Х, 35Х, 40Х, 45ХН, 30ХМ, 30ХГС, 3ОХН3А, 5ХНМ, 38ХС, 18ХГТ, 38Х2МЮА) containing 0,1…0,6 % of carbon, heat treatable alloy steel 12Х2Н4А, 18Х2Н4ВА, 12Х2НФА, 14ХГСН2МА (ДИ-3А), 20Х3ВМФ (ЭИ-415) and subjected to the heat and chemical heat treatment. It is recommended to use the 16Х3НВФМ5 (ДИ-39) steel when producing gears operating at t<450°C [2, 3].

Gears of less important gear sets are made of gray iron; gears operating at high speed (under 50 mps) and low loads are made of nonmetallic materials, such as fabric-reinforced laminate, nylon, etc.; worm gears are made of bronze and brass.

Gears made of steel are subjected to the heat treatment before the final teeth treatment:

– the heavy loaded ones are subjected to cementation;

– the low loaded ones are subjected to carbonitriding or cyanide hardening.

The depth of the cemented layer is specified in the ОСТ 1.41467-73 standard with regard to the modulus value. Its absolute values are given in the Table 12.3.

The hardness of the cemented tooth surface is equal to HRC58...62 after the final treatment.

The nitralloys 38ХМUА and 30Х2Н2ВА are widely used steel types.

The depth of the nitride and carbonitrided layers is specified in the OST 1.90005-74 standard and is equal to 0.6 ... 0.9 mm; their hardness is within the range of HRC62 ... 65 after the final teeth treatment. In most cases, spur gear teeth are corrected to strengthen them and to equalize pressure and slip.

Table 12.3

Depth of the Cemented Layer Depending on Modulus Value

Modulus value, mm	Cemented layer depth, mm
1.5 ...2.0	0.4... 0.7
2.0... 2.5	0.6... 1.0
2.5... 5.0	1.2... 1.6
Over 5.0	Over 1.6

12.1.4. Tooth Gear Production

The main methods of gear processing are the template method and the generating method.

The template method (Fig. 12.4, a) lies in the fact that the gear tooth space is made with the cutting tool with the cutting edges profile identical to the gear cut tooth space profile. Being processed, the part does not rotate. After processing of each tooth it rotates at an angle equal to $2\pi/z_1$, where z_1 is the gear teeth number.

The generating method (Fig. 12.4, б) lies in the fact that the tool and the rough part is driven to simulate the gear engagement with a rack or another gear, or the engagement of a worm and a worm gear.

There are also used other processing methods such as metal powder industry, high- and low-temperature gear teeth generation, etc.

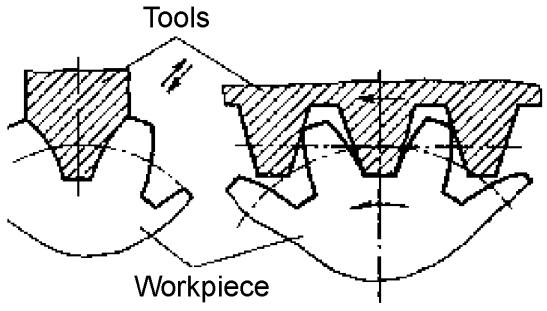


Fig. 12.4. Gear Teeth Processing Methods:
a – template method; b – generating method [3]

The metal powder industry lies in the fact that powder blends are sintered in closed dies at the pressing force equal to 50…60 kN and the high forging temperature. Gears produced in such a way are used to operate at high speed and low loads; they are wearproof and their accuracy grade is 8 or 9.

Gear teeth rough cutting is made by hot stamping method. This method is rather effective as it saves metal significantly. The roughness of the teeth side surface is R_a = 1.6…10 μm.

The method of high-temperature gear teeth generation is effective, but it does not provide the appropriate level of gear accuracy. There are methods with axial and radial feed used to process spur and bevel gears.

The method of teeth cold rolling is used as the final treatment instead of gear-tooth shaving.

Table 12.4 shows the characteristics of the main gear production methods.

Table 12.4

Characteristics of Main Gear Production Methods [3]

Processing method	The way of processing and the tools used	Cutting machine model	Gear types	Gears	
				Accuracy grade and roughness, μm	Modulus, mm / Diameter, mm
Spur gears					
Generating method	Milling with worm milling cutters	5303П; 5К310; 5327; 5348; 5342; 5355	Plain and helical gears, including internal double helical gears	4…8; от Rz = 10…40 до Ra = 0.8… 2.0	Under 30 / Under 12000
	Cutting: with shaper cutters	5107; 5150; 514; 5В15О; Е3-15В; 5171	Plain and helical gears, including internal herringbone gears	6…7; от Rz = 15…30 до Ra= 0.8…1.6	Under 40 / -
	with racks	SH-75K; SH-1200	Plain and helical gears, including herringbone gears	6…8; Rz = 7…20	Under 70 / Under 12000
	Grinding with one or two disks	586В; 5850В	Plain and helical gears, including internal gears	4…7; Ra = =0.32…1.25	Under 20 / -
	Grinding with a helically formed wheel	5А830; 5835	Plain gears	4-6; Ra= 0.32-1.25	Under 10 / Under 700
	Gear turning with generating cutters	–	Plain and helical gears, including internal gears	7…8; Rz= 7-20	Under 6
	Shaving with shavers	5701; 5М14; 5А703; 5717; 5А714; 5717С		4…6; Ra= 1-2.5	Under 14
	Gear-tooth honing (abrasive shaving)	5913; 5А913	Plain and helical gears	4…6; Ra= 1-0.25	Under 10 / Under 600

Table 12.4. (to be continued)

Processing method	The way of processing and the tool used	Cutting machine model	Gear types	Gears	
				Accuracy grade and roughness, μm	Modulus, mm Diameter, mm
Template method	Milling with interlocking disk and involute end milling cutters	E3-11; E3-20; КУ-306; E3-40C-1; 5342; 5355M	Plain and helical gears, including herringbone gears and internal	8…10; Rz=10…40	Under 24 Under 12500
	Gear-tooth shaping with rack-type tools	SH-75K; SH-100/140	Plain and helical gears, including herringbone gears and internal gears	5…7; Rz=10…40	Under 30
	Shear-speed	5110; 5A110; 5120; 5130	Plain and helical gears	6…7; Rz = 10…40	Under 8 Under 250
	Gear-tooth grinding with form wheels	596; 5861; 5Б861; 586B; Б861B	Plain gears, including internal gears	6…7; Ra=0.32…1.25	Under 12 -
	Broaching with cylindrical broaching	7A520; 7720B	Internally toothed spur gears	7…8; Ra =2.5…1.25	-
Bevel gears					
Generating method	Double straight edge cutter planing	5326П; 526; 5A250; 5E283 (gear-shaping)	Plain gears	7…8; Rz = 7…20	1.5…30 Under 1600
	Milling: with double side straight edge milling cutters	5C237; 5220; 5230; 5C277П	Plain gears	8…9; Rz = 10…40	2.5…12 Under 500
	with tool heads	5П23A; 525; 528C; 5240; 5A27C4	Spiral gears (spiral angle is 0…60°)	7…8; Rz = 7…20	Under 20 Under 1600
	One straight edge cutter planing with continuous indexing	5284	Plain and helical gears	7…8; Rz = 7…20	

Table 12.4. (end)

Processing method	The way of processing and the tool used	Cutting machine model	Gear types	Gears	
				Accuracy grade and roughness, μm	Modulus, mm / Diameter, mm
Template method	Contour planing with one or two cutters	5283; 52ТМ-2	Plain gears of any profile	8…9; Rz =15…40	Under 40 / Under 3200
	Circular broaching: with a continuous ring-type gear cutter	5245; 5С268	Plain gears (arc tooth profile)	7…8; Ra =2.5…1.0	Under 5 / Under 175
	with a face-mill-type cutter head	5Б232; 5281Б	Spiral gears (thread tooth profile)	7…8; Ra = 2.5…1.25	Under 15 / Under 800
	Cutting with a face-mill-type cutter head by plunging	525Б; 5Б231		7…8; Rz = 7…20	Under 15 / Under 800
	Milling: with an involute end milling cutter	–	–	9…10; Rz = 40…80	Under 30 /
	with a side milling cutter	Е3-1	Plain and helical gears	9…10; Rz = 40…80	Under 10 / –

12.1.4.1. Gear-Tooth Cutting by Template Method

Depending on accuracy, production program, and equipment, the template gear teeth are cut by:

1) milling – with gear side and end milling cutters;
2) broaching – with modular broaches on broaching machines;
3) multiple cutting head planing on gear-shaping machines.

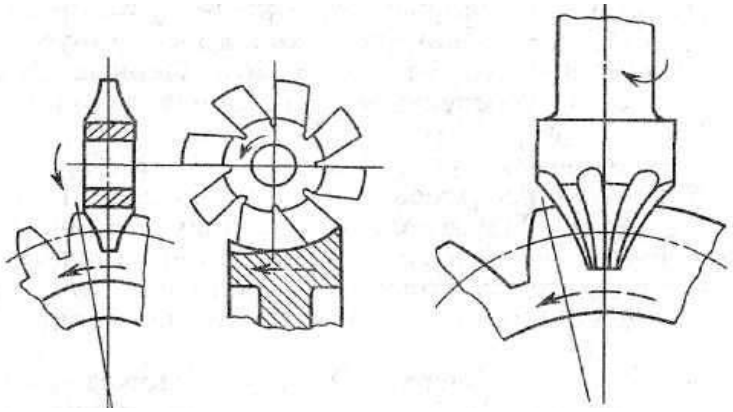
12.1.4.1.1. Cutting with Gear Milling Cutters

Gear teeth are cut with hobs in single-piece production. This method can be used to repair in the absence of special tooth cutting machines, or when the necessary parts are too large.

The cutting tools (interlocking disk and end milling cutters) (Fig. 12.5) having the same modulus for a certain teeth range (5-6) decreases the profile error [1].

Side milling cutters at horizontal milling machines are used to cut outer plain gear teeth; end milling cutters are used to cut plain, spiral, and double helical gear teeth on vertical milling machines. Each tooth space is cut separately. The wheel is reindexed with

the indexing attachment to process another tooth space. Fig. 12.6 shows the scheme of processing.



a *b*

Fig. 12.5. Milling Processes:
a – with a side milling cutter; *b* – with an end milling cutter

It is preferable to use the facility design, pre-processed with the accuracy degree not lower than 7, as the *processing base*. Run out of indexing attachment fitting element must not exceed 0.01…0.015 mm. The workpiece is set onto the fitting element with the clearance of 0.02...0.03 mm, which size influences the pitch cylinder and facility design concentricity. The indexing error is usually within the range of 4...7 min. It can be reduced by increasing the indexing plate diameter.

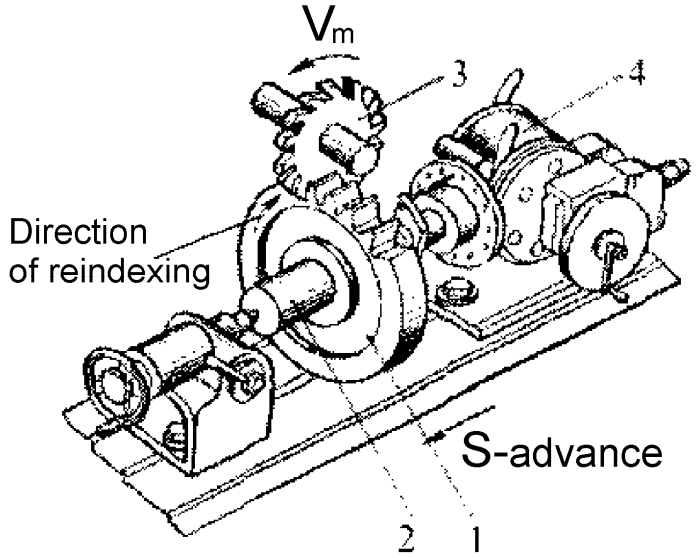


Fig. 12.6. Spur Gear Teeth Cutting by Template Method: 1 – workpiece;
2 – mandrel; 3 – interlocking disk milling cutter; 4 – indexing attachment

Treatment schedule. The depth of cutting depends on the modulus value and the number of threading cuts during the tooth space cutting, the cutting depth after a threading cut is equal to the tooth space depth. When the number of threading cuts increases, the cutting depth is equal to the side allowance value for each next threading cut.

The feed is set according to the router bit hardness, as during the usual parts milling of alloy steel.

For milling cutters made of high-speed steel and hard alloys S_z=0.02...0.03 mm/tooth.

The recommended cutting speed during the rough milling is within the range of 25...30 m/min; and within the range of 30...35 m/min during the final milling.

In the former case it is limited by the machine power and the feed mechanism durability; in the latter case it is limited by the surface roughness and the tool durability (see Table 12.5).

Table 12.5

Cutting Modes of Spur Gear Teeth Cutting
with High-Speed Interlocking Disk Milling Cutters [3]

Material processed (*HB* 156-207)	Feed, mm/min, at the modulus, mm							Cutting speed	
	2	3	4	5	6	7	8	Rough processing	Final processing
Steel types:									
45; 50	190	155	134	120	100	90	90	32	40
40Х; 50Г	130	106	92	82	75	70	65	30	37
30ХГС; 12ХН3А; 38Х2МЮА	76	62	54	48	44	41	38	23	27
Gray iron	285	230	200	180	165	150	140	25	32

The cooling lubricants are sulfofresol, 5 % emulsol solution, or Ucrinol-1.

The processing efficiency is low because of idle running, i.e. the workpiece returning to its original position to cut another tooth space. Using the cut down milling in this case increases efficiency almost twice.

12.1.4.1.2. Pulling the Gears of Cylinder Escape Wheels

It is expedient to apply pulling gear cutting of the fine-grained wheels with $m<2,5$ external and internal gearings with a straight tooth.

It is pulled by two methods:

a – each hollow is pulled individually by one broach during pass;

b – all wreath at the same time complex broach is pulled through all gears for one pass.

The schemes of pulling are in Fig. 12.7 and 12.8.

Processing accuracy is high (the 7th and even the 6th degree under grinding, shaving or its final, depending on the set degree). The surface roughness is R_a = 0.8 microns.

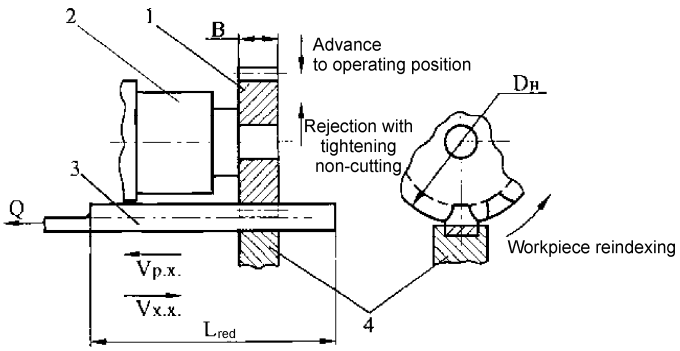


Fig. 12.7. Scheme of Teeth Pulling of Cylindrical Wheels: on Horizontal Broach: 1 – work part; 2 – automatic divider; 3 – modular broach; 4 – adapter [2, 3]



Fig 12.8. Shear-Speed Cutter

Individual process of dimples is carried out on horizontal broach of models 7B55U, 7B56, 7B57, etc. (see Fig. 12.7); on the vertical models 7B74, 7B75 and 7B76 with the divider working on an automatic cycle:

– part supply in a working zone after a broach return to a starting position for pulling of the next dimple;

– branch it and reindexing on the next dimple at the reverse (single) motion of a broach;

– supply for performance of the working path.

Processing is performed before wreath full cutting, i.e. performance of one pass.

The **broaches** are modular combined, consisting of several sections. Material of the cutting part is the quick cutting steels that are widely applied at slot broaching for blade retention in disks of the compressor and turbine: R12F5M, R12F2K5MZ, P9M4K8, etc. It is expedient to use combined broaches with hard-alloy teeth from BK6, BK8, T15K6.

Broach length is teeth quantity, depending on the pulling depth of dimples between teeth and thickness of chip load a_z by one tooth. At **m**<2,0...2,5 mm teeth cutting are carried out by one broach; at **m** > 2,5 - the number of passes, and according to broaches increases to 2... 3 and more.

The **part installation** is desirable on the design bases that are previously processed at a draft stage with accuracy not below the 7th accuracy degree, i.e. grinding.

The beating of adjusting elements of the adaptation can not exceed 0.015... 0.020 mm; a landing gap – 0.02... 0.05 mm. The division error has to be within 4... 7'.

The **mode of processing** is defined by thickness of the chip to be taken a_z and speed of cutting of V_{px}. Thickness of the chip to be taken is established usually within 0.02... 0.04 mm proceeding from broach tooth durability.

Cutting speed at elements pulling of a parts design from alloy steel accept equal 4... 6 m/min.

The advantage of the method is high precision of processing and rather high efficiency; special equipment isn't required. Broaches can be applied after a repoint and dividers for slot broaching in disks of the turbine and compressor for blade retention.

12.1.4.1.3. Pulling of External Teeth of Cylindrical Wheels by Group Broaches

External teeth of the spline wheels of $m < 1.5$, small length, are pulled on vertical broach in the presence of a broach exit by means of the special divider (see Fig. 12.9).

The tool is a complex broach, consists of several sections – the broaches equal to quantity of the dimples located in the rigid cylindrical case.

A blank is mounted on a setting element of the device with the gap providing the set size of a beating of the initial cylinder.

The broach is connected with the ram of the machine and moves on directing, rigidly fixed with the basis of the adjusting adaptation.

Cutting of all wreath is carried out for one working path at the end of which the wheel is removed and the broach comes back to the top starting position.

In the pulling process in a cutting zone cooling lubricant is intensively moved. Chip load is thickness of the removing chip of 0.02...0.04 mm, and is also accepted from a tooth durability condition. The quantity of the cutting teeth and length of a broach depend on the size of the wheel module and tooth length.

Speed of cutting is set, as at usual slot broaching, – 4...10 m/min.

The method is used widely at plants of 'Iveco', and at other enterprises of a mass and large-lot production.

The advantage of this method is high efficiency. It provides the 7...8th degree of accuracy [2, 4]. A shortcoming is complexity of a group broach production.

12.1.4.1.4. Teeth Cutting by Copying Method on Machines with Multiple Cutting Head

In Fig. 12.10 is shown the scheme of direct teeth processing of a cylindrical wheel on the machine of model 5120 of multiple cutting head which cutters have a dimples between teeth. Their quantity is equal to number of teeth. Metal removing is carried out by shaping.

When processing the head isn't mobile, and work part makes the alternate heave mode.

With each working path of working part cutters automatically move to the center in the radial direction at feed 0.05...0.1 mm before full cutting of a dimple. At the return movement cutters are taken away on 0.1 mm in avoidance of friction of the processed surface about a back side.

The method differs in high efficiency, but provides only the 9...10th degrees of accuracy. With increase in quantity of teeth and the module the total force of cutting is increasing that is one of the reasons of accuracy decrease. High precision at cutters production and their installation into working position is required. Therefore, it is expedient to apply it in a mass and large-lot production as preliminary processing, and teeth cutting at wheels of average degree of accuracy.

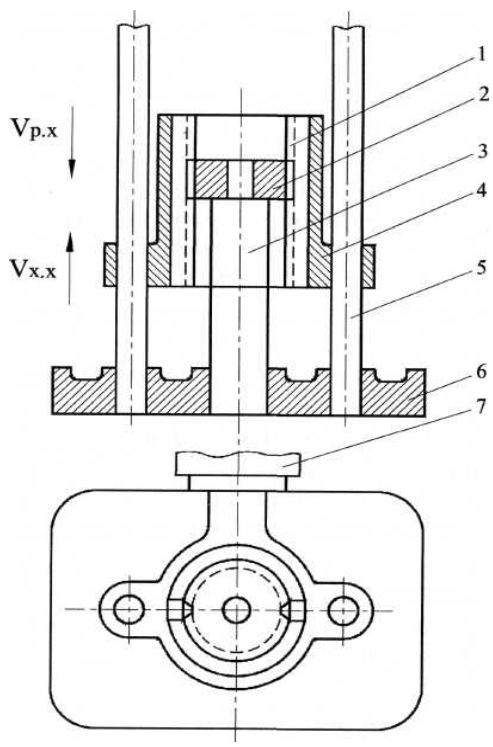


Fig. 12.9. Scheme of all external teeth pulling of spline wheels simultaneously by a block of broaches: 1 – one of broaches; 2 – work par; 3 – column; 4 – case of broaches; 5 – the directing; 6 – device basis; 7 – ram of the vertical machine [2]

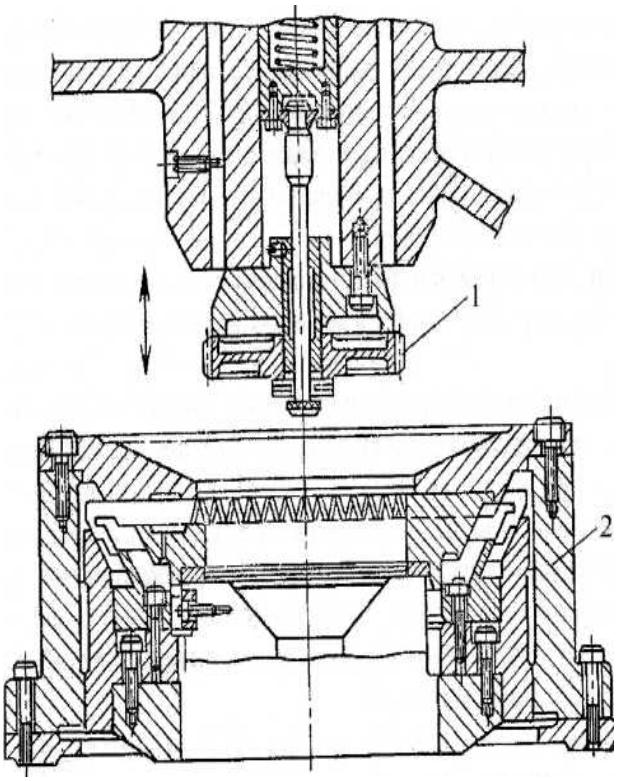


Fig. 12.10. Scheme of teeth processing of a gear wheel on the machine with a multiple cutting head: 1 – gear wheel; 2 – multiple cutting head

12.1.4.2. Teeth Cutting of Cylindrical Wheels by Generating Method

This method is the basic in an aircraft and motor engineering. Depending on a part design – forms of sizes, modules, and from the cogwheels production release program: gear hobbing, gear turning, gear-tooth shaping and gear-tooth planning are applied.

12.1.4.2.1. Gear Hobbing

Straight and spiral external teeth of cylindrical wheels are cut of various diametrical sizes and modules on the gear hobbing machines with gear hobs. The working zone of the gear hobbing machine is shown in Fig. 12.11, and the scheme of milling – in Fig. 12.12.

The mill in axial section has an appearance of the gear rack which is well combined with an involute surface of teeth. All cutting part of a gear hob represents a set of gear racks with a screw arrangement of teeth in the axial direction. The similar design of the tool allows to use it for cutting of any number of teeth at wheels of one module with an identical gearing angle.

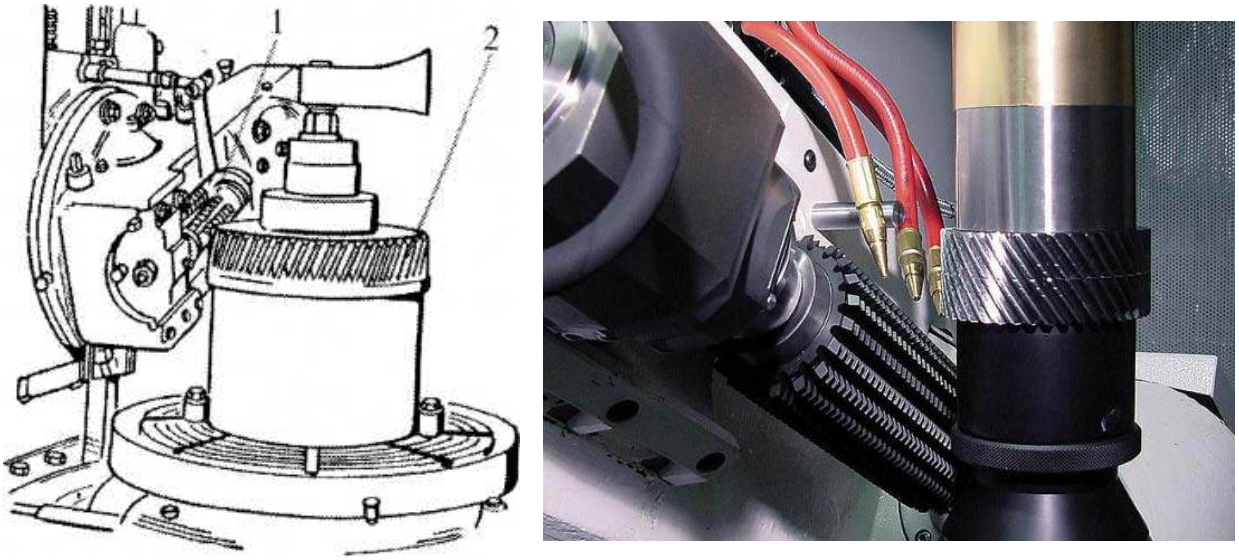
The essence of generating method is that the gear hob and a cogwheel are in process of cutting in relative gearing with transfer number

$$i = \frac{n_c}{n_p} = \frac{z_p}{k},$$

where n_c and n_p are the frequency of rotation of a mill and part respectively, rpm;

Z_p – is teeth quantity of the cut wheel, piece;

κ – is number of entries of gear hob.



a b

Fig. 12.11. Working Zone of Gear Cutter: 1 – worm gear cutter; 2 – blank [2]

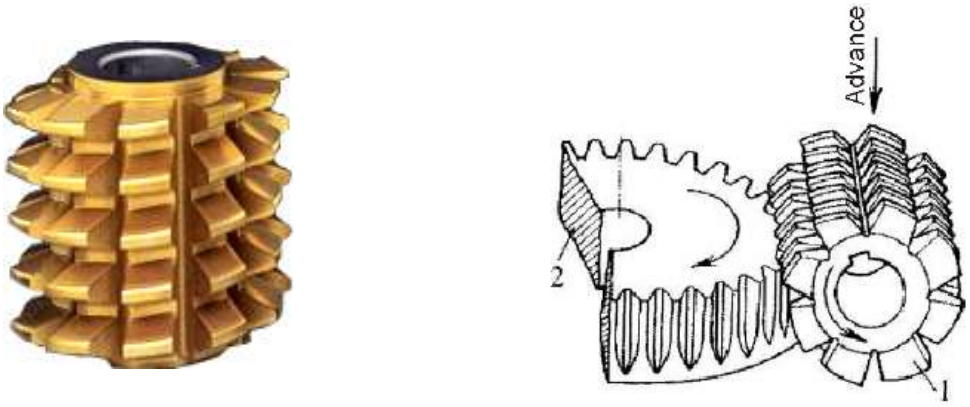
As you can see from Fig. 12.13, the axis of a gear hob is mounted at an angle by β to the basic plane to an end face of the cut wheel.

When cutting direct teeth (see Fig. 12.13, a) an installation angle is

$$\beta = \alpha_c,$$

where α_c is an angle of lead of gear hob string on the pitch cylinder. When cutting spiral wheels: $\beta = \alpha_m \pm \alpha_s$, where α_s is a spiral angle of lead (a tooth deviation from the vertical plane) of the cut wheel.

Thus angles are subtracted if the direction of a spiral wheel and a string of gear hob coincide (see Fig. 12.13, b), and summarized if they have an opposite direction (see Fig. 12.13, c).



a b

Fig. 12.12. Scheme of gear hob processing: a – is worm gear cutter;
b – is the scheme of teeth milling;
1 - worm gear cutter; 2 - work part of the cut wheel

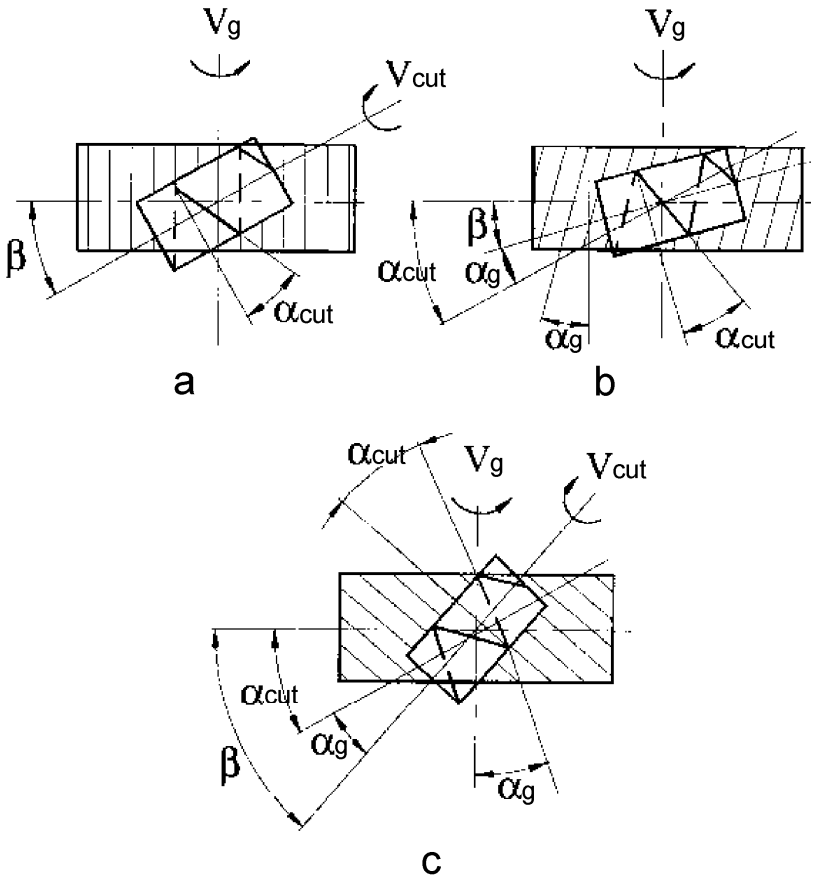


Fig. 12.13. Installation Angle of Mill:

a – $\beta = \alpha_m$ – are spur wheels; b – $\beta = \alpha_m - \alpha_s$ coincidence of the directions of wheel spiral and gear hob string; c – $\beta = \alpha_m - \alpha_s$ is opposite directions of wheel spiral and gear hob string; [2].

The kinematic scheme of processing is shown in Fig. 12.14.

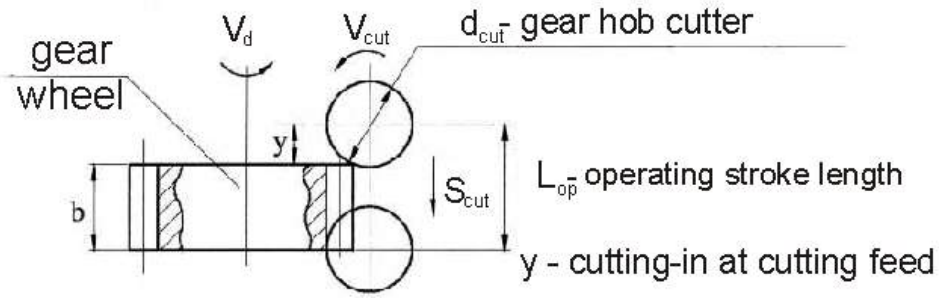


Fig. 12.14. Kinematic Hobbing by Generation Method

Machine tools – hobbing, high rigidity "Komsomolets" mod. 532, 5320, 5D32,5A325, 5342.5330, 535B, as well as foreign companies, "Pfauter," "Reyneker", "Gould-Hebert" and others.

The high rigidity of the machines allows the teeth to cut feed S = 15 mm / rev, parts and the cutting speed V = 300 m / min.

Machines have a mechanism for conventional milling "against advance direction" as well as for milling "in advance direction" in the performance of the second pass, and the mechanism of smooth movement along the axis of the cutter, which substantially exceeds its on-resistance (see Fig. 12.15).



a b

Fig. 12.15. Working Zone of Machine: a – vertical; b – horizontal

The **tools** are worm gear cutters: the single flute are for fair processing and multiple thread (two - three and even four) are for preliminary teeth cutting of big modules. Thus productivity increases on 60... 70% in comparison with single flute cutter.

Mills material is quick cutting steels: R12F5M, P6M5K5, P9M4K8, etc., and plates from firm alloys of BK8, BK6, T5K10

Geometry of the cutting teeth. Cutter angle is $\gamma = 0°$; clearance angle is $\alpha = (1.5...5.0)°$.

Number of passes depends on the module size. When m <2 teeth cut in a single pass (clean). When m = 2.5 ... 6 two passes are performed: first with depth of cut-

ting tooth height of about 0.6; second – 0.4, i.e. the remaining allowance is removed. When m = 6...10 teeth are cut in three passes. In all cases, the final finishing allowance of 0.5 ... 1.0 mm per the tooth thickness is done.

The accuracy and surface quality. In the presence of finishing for grinding by a well sharpened cutter and with minimum error of setting the hobbing allows to get the 8th and even the 7th degree of precision wheels medium modules and tooth surface roughness R_a = 6.3 ... 1.6 mm in accordance with GOST 2789 -73.

Relatively low accuracy of the cutting teeth is caused by:

1) distortion of the profile rail in the axial section of the worm by lifting the yarn;
2) inaccurate sharpening of the cutting teeth;
3) variations in elastic elements wrings of MDTP technological system.

To improve processing accuracy design bases should be used as technological bases, preparation of which is generally performed at the instrument-draft stage of the process.

Treatment mode. When rough cutting it is recommended: the part carrying-in S = 1.5 ... 3.0 mm / rev, cutting speed V = 25 ... 30 m / min. When part finishing cutting S = 0.5 ...2 mm / rev.; V = 30...35 m/min for the wheels of steel of medium-firmness. As a coolant it is advisable to apply sulfofrezol.

Treatment performance is high due to:

1) continuous contact of the cutting tool and the part to be treated;
2) use of multiple-cutters;
3) ability to handle several parts simultaneously;
4) reliable basing of a blank in finely treated design bases without reconciliation;
5) high-speed fastening elements.

Single flute cutter wheel speed is doubled when using double-threaded cutters on the cutting mode, respectively, increasing the performance. However, the number of "cutters" needed to form the tooth profile is divided in two that leads to the accuracy decrease and surface roughness increase. Therefore, usage of multiple-cutters is only advisable to precutting of teeth of large modules.

Besides usual cutting of cylindrical gears, there is a diagonal one, the cutter has two advances: one – in the direction of wheel axis; second – tangential along the cutter axis. In this case, when the operation involves all cutting teeth of a cutter, which increases its resistance, but reduces the number of machining accuracy due to errors in the kinematic connection.

Wheels with corrugated teeth are milled by conventional mills. At that the spacing between the cutter and the cut wheel change, providing a predetermined thickness of the teeth.

When milling the teeth with undercut the cutters with a thickened tooth head are used.

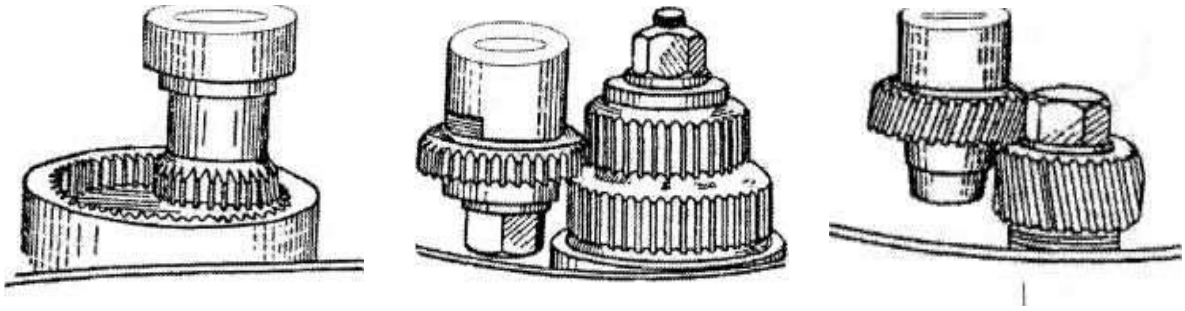
12.1.4.2.2. Cutting Teeth of Cylindrical Wheels by Gear-Tooth Shaping

Gear-tooth shaping is used for cutting straight and helical tooth cylindrical gears (Fig. 12.16). This is the only method of direct cutting teeth inwardly of the engagement (Fig. 12.16, a), and the outer wheel engaging block if there is no escape for the modular hob with hobbing cutter (Fig. 12.16, b).

Machines: gear shaping, mostly vertical, plant "Komsomolets" mod. 512, 5A12, 514, 516, as well as foreign "Fellow" firms, "Lorenz" and others. Tools of mod. 512 and

5A12 – high-speed, have four steps of cutter spindle velocities in the range of 250 ... 600 double strokes per minute.

For cutting teeth and involute spline on long shafts horizontal gear shaping machines are used.



a b c

Fig. 12.16. Slotting of Teeth on the Gear Shaping Machines:
a - wheels with inner teeth; b - block; c - outer helical gearing

Tools. When cutting teeth of aircraft engines' cylindrical wheels with a module m<9, round shapers with direct and oblique tooth are used: disc, pan, sleeve and tail (Fig. 12.17.)

Gear shaper cutters' material - speed steel.

Cutting part is backed off gear teeth of the same module and the angle of engagement as and the cut wheel, the 6 – 7th degrees of accuracy. Each tooth is a pre-cutter with a rake angle γ = 5° and back – α= 6°.

Disk shaper cutters with straight teeth (see. Fig. 12.17, a) are intended for processing gears with external and internal gearing.

Disk shapers with spur teeth (see. Fig. 12.17, b) are intended for cutting helical teeth. They are made as a set of two cutters with left and right tilt of teeth.

Deep Counterbore-Type Cutters (see Fig. 12.17, d) are used for cylindrical wheels with straight teeth of the outer 20 ° gearing, construction of which prevents the passage of a nut, attaching the cutter.

Hub-Type Gear Cutters (Fig. 12.17, c) are used for cylindrical wheels with straight teeth of internal 20 ° engagement and for wheels of external engagement with small modules – from 1 to 3.5 mm and a diameter of the pitch circle to 50 mm.

Shank-type cutter (see. Fig. 12.17, e) are used to spur and skew-toothed spur gears of internal 20 ° engagement with gear teeth slope of 15 °.

As you can see in Fig. 12.16, a cutter and cut wheel are cinematically-bound pair, i.e., are conditionally engaged and rotate about their axes at a speed equal to the circular feed of a wheel – S_c mm / motion of cutter.

When cutting spur gears a cutter performs a reciprocating movement, when helical cutting ram receives additional-body rotation on the value of the tooth helix deviation using the copier. In the reverse (no-load) motion of a cutter wheel moves away from its axis by about 0.1 mm to avoid rubbing on the back side.

A cutter is fed radially at the start of treatment at absence of a circular-feed or without it. At that feed "for cutting-in" is usually taken in percentage of the rotation feed.

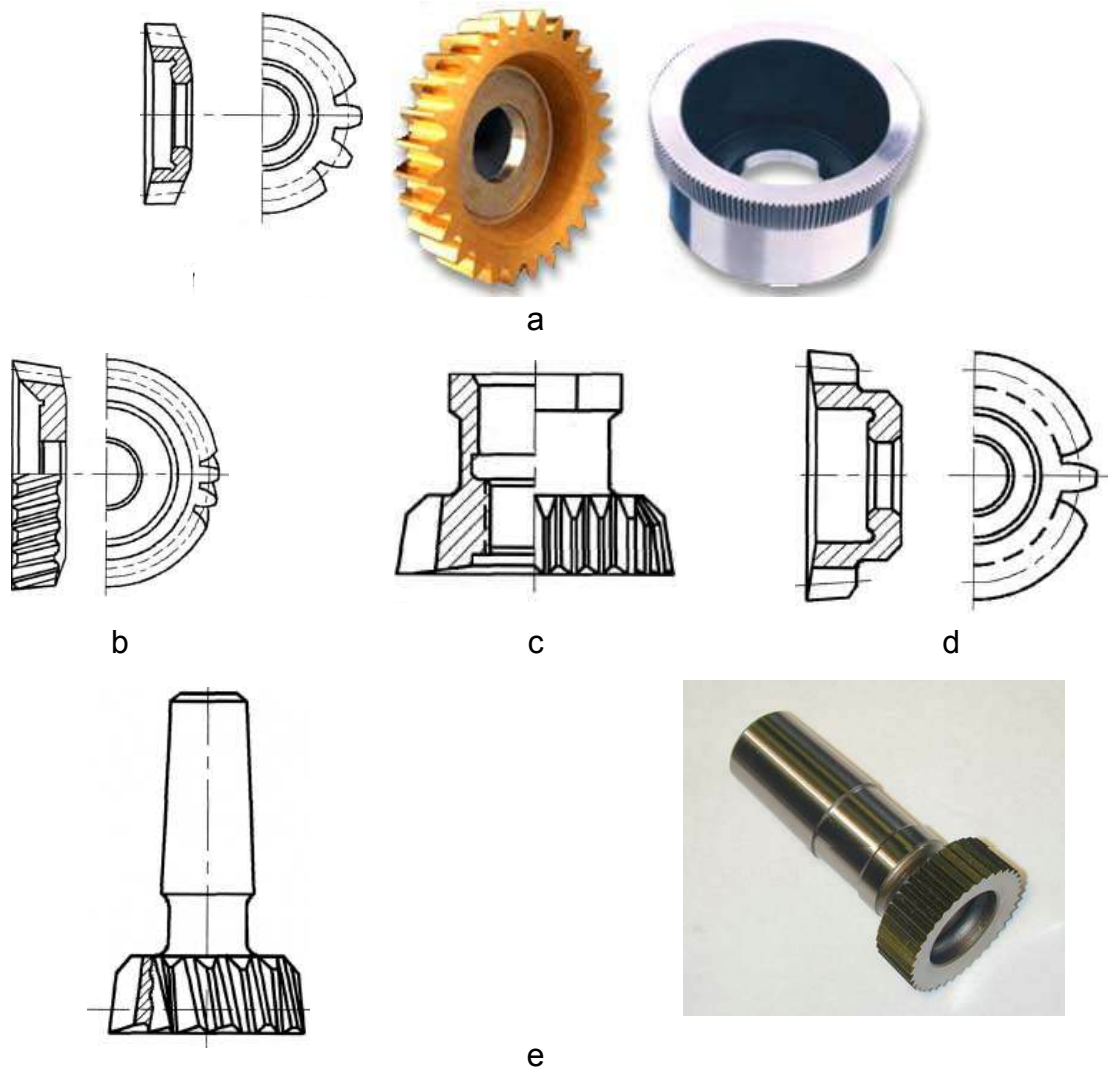


Fig. 12.17 . Designs of Cutters: a – disk shaper cutters with straight teeth; b – disk shapers with spur teeth; c – hub-type cutter; d – deep counterbore-type cutter; e – shank-type cutter

Cutting Conditions. If cutting wheels of steel teeth of middle firmness (HRC 32 ... 38):

a) For roughing take:

– Cutting speed - V = 12 ... 18 m / min;

– Circular feed – S_c = 0.25...0.5 mm / cutter blow;

b) For finishing:

V = 12 ... 18 m / min and S_c = 0.15 ... 0.30 mm / cutter with transverse feed of cutting-in - S_{vr} = (0.1 ... 0.3) S_c in both cases.

Precision machining. Higher is the 7th and even 6th degree as compared to the hobbing in the manufacture of wheels with a module of 5 mm due to the greater simplicity and, consequently, higher precision manufacturing of tool-cutter.

The surface roughness after chiselling corresponds to roughness of R_a = 3.2 ... 1.6 m GOST2789-73 due to the continuous swarf removal.

Treatment performance. If cutting wheels with the module m < 2.5 performance is the same as in hobbing, but the accuracy is one degree higher. If m> 2.5, it is advisable to apply hobbing, as idle tool paths are absent.

Corrected gears with negative and positive bias are cut by conventional standard profile cutter with change of center distance between the wheel and the tool.

When gear-tooth shaping, as when hobbing, one may obtain tip relieved tooth - with cut-off profile at the apex on 0.02 ... 0.04 mm for module m = 2.5 m ... 5.0. That is ensured by cutters with doubled angle profiles or cutting in two operations with the usual cutter and then a cutter designed for this purpose.

12.1.4.2.3. Gear-Tooth Planing

Typically used for cutting teeth of chevron, as well as spur and helical gears with larger diameters up to 60 mm modules. Cutting scheme is shown in Fig. 12.18. Because of the tool no-load passes and reindexing (moving to the starting position in the processing of the nearest tooth) performance is significantly reduced, which limits the use of the method in manufacturing of gears of medium (2.5 ... 90 mm) modules.

Used are machines "Komsomolets" mod. 515, as well as the company's "MAAG" and "Parkinson." The tool is a comb, that is a rail with 6 - 8 straight-sided cutting teeth, which enables to produce it with high accuracy. As you can see from Fig. 12.19, running occurs when cutting a tooth. When cutting the comb and the wheel are in a conditional engagement. For the next tooth treatment the tool should return to its original position.

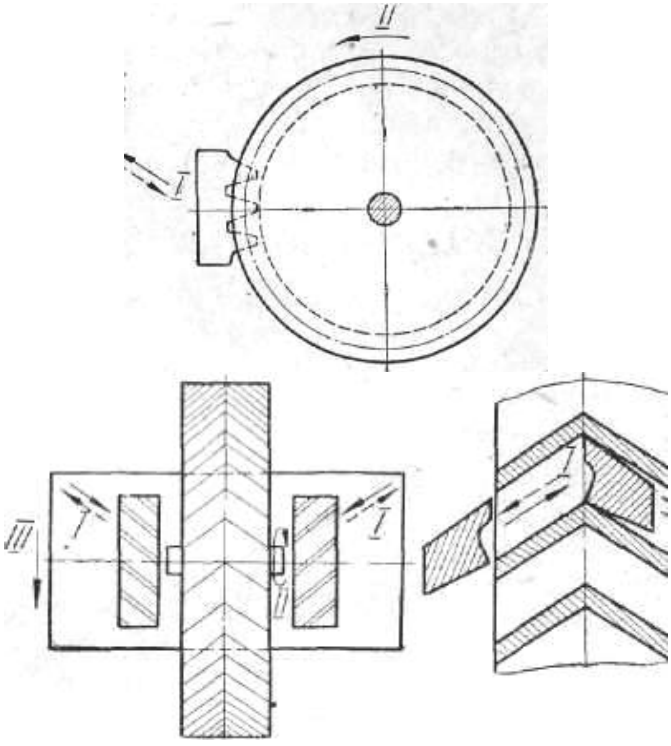


Fig. 12.18. Schemes of Cutting Herringbone Tooth. Wheels Movement



Fig. 12.19 Cutting Tools

For cutting herringbone tooth wheel machines have two racks (mod. 515). Since the teeth of chevron wheels can not be grinded and the only finishing method after complex heat processing (carburizing, quenching and tempering or nitriding) is an ointment, the high-precision cutting must be done. As for the pitch of the teeth, it is ensured by the machine divider. The accuracy of the profile depends on the accuracy of tool. Cutting is provided in two or three passes. Finishing pass is sometimes isolated in a separate operation on more accurate machines.

12.1.4.3. Finishing Methods of Machining Cylindrical Gear Teeth

Finishing is used to improve reliability during long-term usage of engines - strength and wear resistance of the teeth, as well as to reduce vibration and noise.

Finishing treatment methods. Depending on the size, shape, kind of engagement (external or internal), material hardness, required accuracy and quality of the tooth surface are used: shaving, grinding, gear honing, ointment, polishing.

12.1.4.3.1. Shaving

Shaving - precision machining process of straight and helical tooth of cylindrical gears by metal or abrasive tool – a shaver on gear shaving machines.

In engineering, the models of machines 5712, 571 A, 571 B, 5714, 5715, 5716, 5717, etc. are used for shaving of exterior and interior (5717) teeth. Shaving process is shown in Fig. 12.20 *b.*

Metal shaver treats wheels with a hardness not higher than HRC40 units, i.e., before quenching and tempering – from hardened steel and cemented and before nitriding – from azotized steels.

Shavers are flat or round, made of high-speed shear steel with high hardness after heat treatment – HRC 62 ... 64 units.

In the aircraft engine building shaving of teeth is carried out by round shavers made in the form of gears with straight or helical teeth, on the surface of which the grooves with cutting edges are located (Fig. 12.20, a). Usually they are made of the most common modules – m = 2...8 mm, with a pitch line diameter of 180...240 mm and the angles of teeth inclination of 10...15 °.

Number of the shaver and treated wheel should not be divisible, which eliminates periodical copying of tool teeth errors on the wheel teeth.

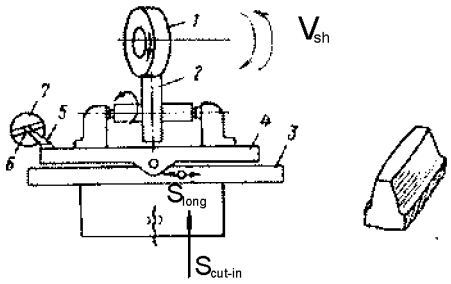
In processing shaver and the wheels are in gapless engagement and with rotating them by relative sliding along the forming teeth the thin chip cutting with a thickness of 0.005 ... 0.01 mm occurs by a very sharp-edged grooves of a shaver (Fig. 12.21).

For finishing machining of spur gears a shaver must be with helical tooth, and for "helical" wheels - with straight teeth.

Rotation of shaver is reversible at 100 ... 300 rev / min, which make it possible to carry out the finishing treatment of the tooth on both sides.



a



b c

Fig. 12.20. Process of Shaving of Barrel Tooth by Round Shaver:
a – shaver; b - processing scheme and machine construction elements: 1 – shaver; 2 – gear; 3 – machine table with longitudinal displacement; 4 – rocking device; 5 – arm of device; 6 – finger; 7 – groove of copier ensuring swing to get the barrel surface of the tooth; c – barrel tooth

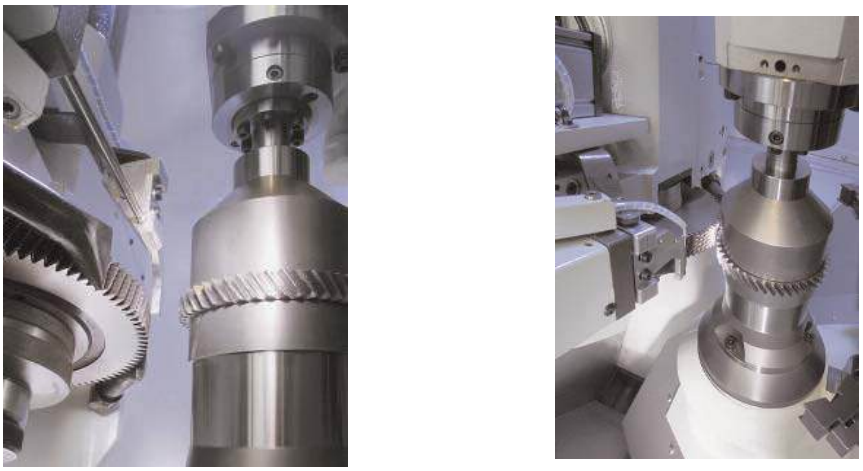


Fig. 12.21. Working Machine Zone

To handle it along the table entire length the direct and inverse displacement occurs at a speed of 0.1...0.3 mm per a revolution of part. At the end of each pass, there is a "cutting-in" with a vertical feed of 0.02 ... 0.03 mm. Allowance for shaving – 0.1 ... 0.2 mm for the tooth thickness. It is removed during 10 ... 12 table double blows. To improve the surface roughness of tooth four to eight double strokes without incision are transmitted to a table.

Shaving accuracy depends on the accuracy of previous processing and generally increases by one degree. The error on the involute is of 8...15 mm; by pitch – 8...10 mm; concentricity starting cylinder and base surface – 5 ... 10 μm.

If one needs a barrel surface of the tooth fluctuations are transmitted to machine table with a special device. The difference in the tooth thickness on the pitch circle diameter along the length is of 0.02...0.04 mm.

It is inappropriate to eliminate large errors from previous processing by shaving. The economic feasibility of given method of finishing treatment is obvious. They allow 10...12 resharpenings and allow to treat 40...60 parts per hour.

Shaving compared to the grinding eliminates the forming of the tooth-adverse tensile residual stresses in the surface layer.

For shaving wheels with high hardness is HRC 58 ... 64 – after the carburizing, quenching and tempering, and after nitriding round abrasive shavers with a ring gear without grooves are applied. Shaving machines, the principle of which is the same as with a metal shaver are used.

Within one – two minutes roughness is reduced by two to three classes. Resistance of mean modulus shaver is *m* = 3 mm, up to full deterioration is 1500 ... 2000 wheels.

Allowance for finishing processing by abrasive shaver is 0.01 ... 0.02 mm per a side. Therefore, to increase productivity to complex thermal treatment a metal shaver is used for shaving, after complex thermal treatment – an abrasive shaver. Approximate duration of shaving gears with module m = 3 ... 4 mm, number of teeth 25, 35 ... pcs. and tooth length B = 20 mm, by abrasive shaver is 30 ... 40 seconds.

12.1.4.3.2. Grinding

This method of finishing treatment of internal and external teeth engagement is applied to get high accuracy and to decrease surface roughness after thermic treatment – hardening and wheels tempering made from hardened steel (HRC 50 ... 59); cemented wheels – after cementation, hardening and tempering – (HRC 58 ... 62); azotized wheels – after azotizing (HRC62 ... 64).

The grinding provides with the 6-th and even 5-th degree of accuracy.

Deflections of main characteristics defining the accuracy of engagement do not exceed: tooth thickness – 0.03 mm; main step – 0.005 ... 0.006mm; tooth profile – 0.002 ... 0.005 mm; flapping of pitch cylinder – 0.01 ...0.02mm.

The allowance for grinding depends on previous treatment and installation errors – location and fixation. Usually at exist methods of teeth treatment the allowance for grinding for wheels with a module 2.5 ... 6 mm is 0.3 ... 0.4 mm at tooth thickness.

Unreasonable allowance growth for grinding increases work-content and manufacturing cost of tooth-wheels.

Methods of Grinding

By means of tool effect on treated tooth surface and depending on kinematics there are two methods: form-copying method and form-generating method. The point of each method remains the same as in cutting teeth.

Grinding of Cylinder Wheels Teeth by Copying Method

Using copying method only straight teeth of wheels with internal and external clutches are grinded with abrasive disks, the working surface of which is performed with accuracy of hollow profile between teeth (Figs 12.22, 12.23). Machine mod. 586, 5860, 5861 types are used for grinding of external teeth and machines 5860B и 586B types are used for grinding of teeth of wheels with internal engagement as well as machines «Orkut» and «Gir-griding».

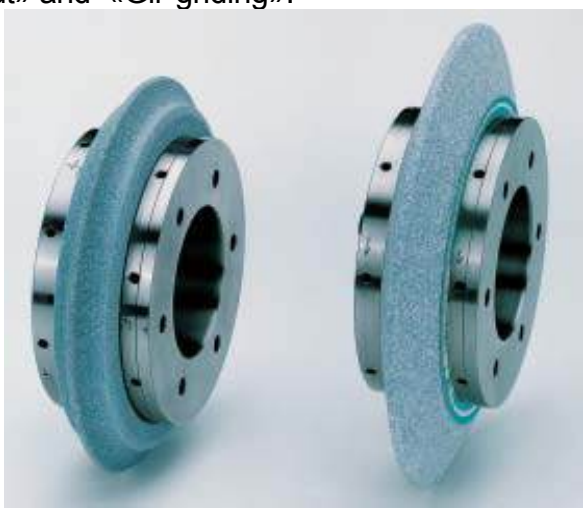


Fig. 12.22. Abrasive Discs for Teeth Wheels Treatment by Copying Method

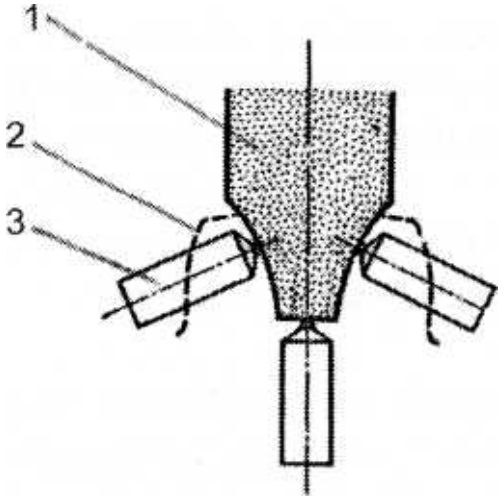


Fig. 12.23. Wheel Dressing by Three Cutting Diamonds Strictly to Hollow Profile Between Teeth: 1– abrasive disc; 2 – diamond points; 3 – teeth profiles forming hollow are shown with dotted line

The working area of machine for grinding internal teeth by copying method is shown in Fig.12.24. The process of grinding is carried out by movement of abrasive disc along generating line of the tooth. After each double pass the wheel reindexing to the next hollow with the help of automatic division device is carried out.

Treatment conditions. The disc rotation velocity is recommended to be V_g = 20...25 m/sec; velocity of axial moving – V_{ax} = 12 ... 20 m/min. The depth of cut along normal to tooth profile - t = 0.005 ... 0.01 mm.



Fig. 12.24. Working Area of Machine

One pass embraces grinding of all teeth of the wheel. The amount of passes depends on values of stocks and the depth of cut. The last few (from 1 to 3 passes that are sparking-out) is usually carried out without cut-in.

This method provides high performance, but accuracy of treatment one degree lower than at teeth grinding by form-generating method. The accuracy of treatment depends on accuracy of division and correction – disc profiling that can be implemented by special device – pantograph.

Aluminum oxide wheel with ceramic bond at external grinding with diameter 450 mm, rotation velocity 30 ... 35 m/sec are usually used.

Grinding of Cylinder Wheels Teeth by Rolling Method

In aircraft engine building two methods got wide spread occurrence: a) grinding with two dish wheel on machines with bend rolling mechanism; b) grinding with worm abrasive discs on machines with steady rolling mechanism.

The straight and spiral teeth of internal and external clutch wheels are grinded by the first method. Machines mod. 584, 5831 and other types as well as «MAAG» и «Niles» are applied. There are two schemes of abrasive discs. The first scheme dominating earlier: when discs located under angle 15° each and in the section of their working surfaces are spur rack along which teeth rolling are implemented (Fig. 12.25, a). The second scheme (null method) is machines with parallel discs (Fig. 12.25, b) and is widely spread since exclude undercutting at tooth leg.

The pitch disk block-tape mechanism of machines rolling is shown in Fig. 12.25.

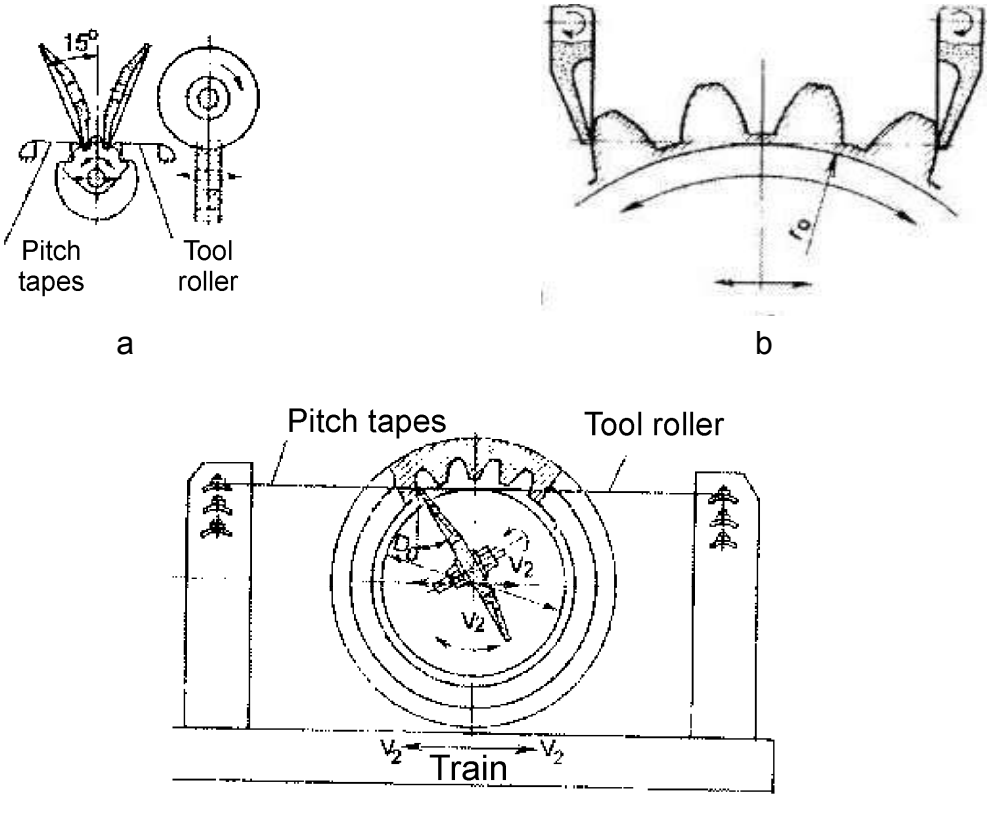


Fig. 12.25. Pitch Disk Block-Tape Schemes of Machines Rolling For Grinding Teeth of Cylindrical Wheels: *a и b* – external engagement; *c* – internal engagement

The treated wheel fixed on mandrel in the center of movable table gives slow reversing move in the direction of length tooth. During double pass the surfaces of two teeth are grinded simultaneously, whereupon automatic wheel reindexing on two next teeth.

The process of form-generation consists of two coordinated wheel motion – they are reciprocative motion of the wheel in the direction that is perpendicular to tooth length and reversing rotation relatively to its axis provided with the block-tape mechanism.

The machines for teeth grinding by this method have device for correcting of discs with diamond point before treatment and for corrective maintenance automatically compensating wear.

The scheme of working area of the machine for teeth grinding of cylindrical wheels by two discs with the block-tape epicyclic train is shown in Fig. 12.26.

The grinding stock on the machines of such kind is 0.2 ... 0.4 mm per tooth thickness. The pass number is from 2 to 4 that provides the 6th even 5th degree of accuracy and roughness of the surface R_a – 0.4 micron.

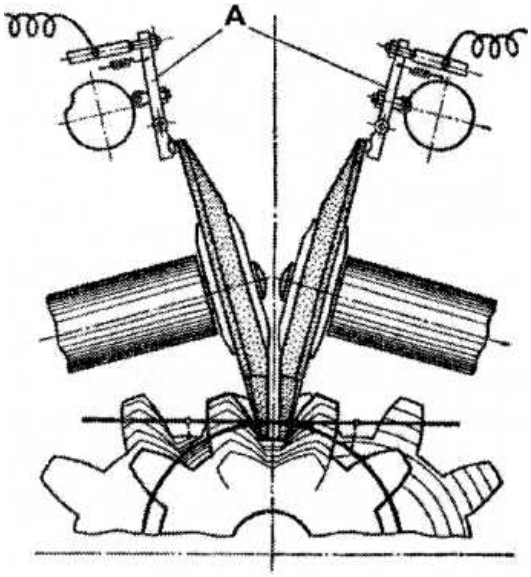


Fig. 12.26. . Teeth Grinding of Cylindrical Wheels by Two Discs with Block-Tape Epicyclic Train: A – the device for automatic correction of discs

When using this method for grinding of external teeth of tempered, cemented and azotized wheels with module 1.0 ... 3.0 mm by this method it is possible to receive deviation of tooth thickness more than 0.03 mm; main step and profile – 0.006 mm; not-concentricity of pitch cylinder and engineering base that corresponding 5th degree of accuracy.

Grinding by Helically Formed Wheel

In serial manufacturing of aviation engines it is reasonable to use heavy-duty gear tooth profile grinder working on method of steady rolling by helically formed wheel, they are models 5830, 5832, 5833 as well as Racehour company" – Switzerland and "Matrix" company– Great Britain (Fig. 12.27).

The worm disc in the section has rectilinear profile formation of which is carried out with special diamond device.

The aluminum oxide wheels with ceramic bond have diameter 450 mm. The velocity of disc rotation is 30...35 m/sec. The teeth wheels are grinded with module 1.5...3.0 mm diameter to 200 mm. The axial movement - feeding is imparted to the part.

The grinding takes place at sledge movement up. At reverse accelerated pass down the disc is taken off the wheel and at the end of the wheel takes position for the next pass. This method of grinding provides the 6th degree of accuracy and maximum performance in comparison with other methods considered earlier. At that roughness of

surface after grinding corresponds to surface with R_a = 0.8 ... 0.4 micron – GOST 2789-73 and GOST 2.309-73.

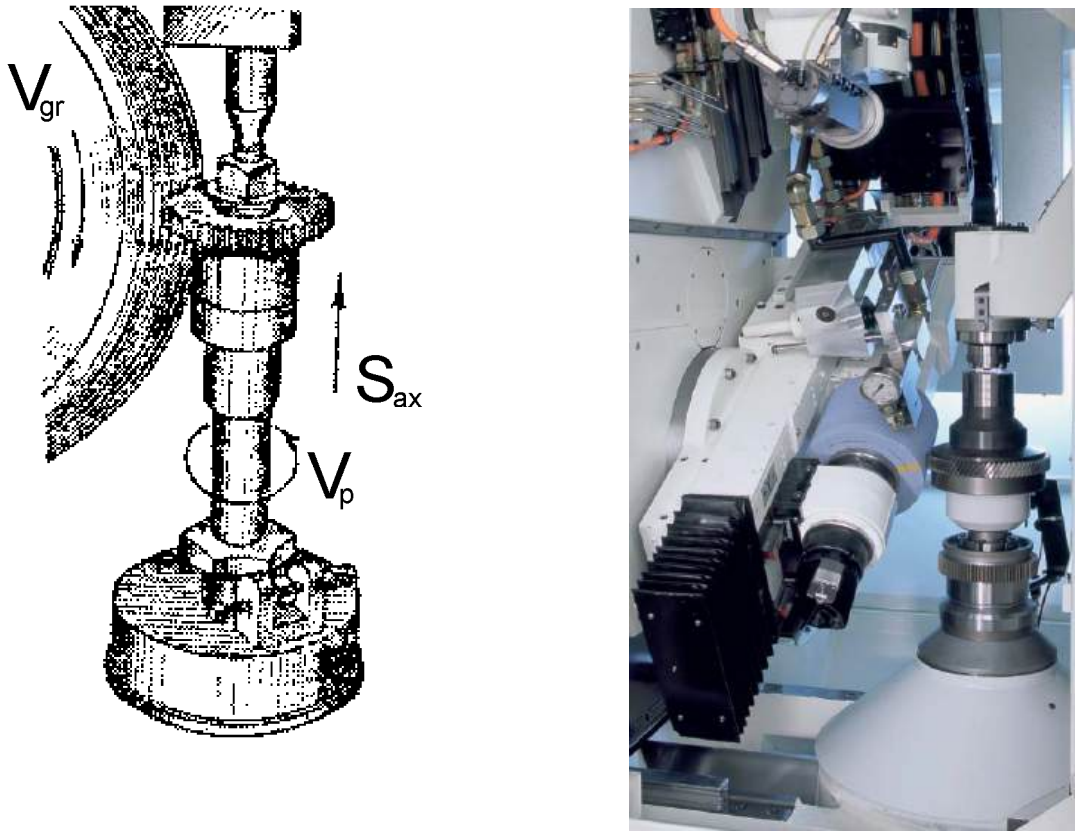


Fig. 12.27 Grinding of Gear Teeth By Helically Formed Wheel

The grinding stock area is 0.3 ... 0.4 mm on thickness tooth.

Gear honing. This method of last finishing treatment practically supplants con-frication of internal and external teeth of cylindrical wheels. The gear honing is carried out after complex thermal treatment such as cementation, hardening, tempering, nitriding and finishing grinding, and after cementation, gear shaving, hardening and tempering – for non-grinded teeth.

The **machines** are similar to shaving machines at crossing axis of the hone and treated wheel, but without radial feeding. The gear hone is helicoidal wheel made from plastic and charged with abrasive disc. The stock on treatment is 0.025 ... 0.05 mm.

The toothed wheel and hone remains in tight mesh. Reversing rotation is imparted to hone from electric motor thus making the wheel perform reciprocating motion along the its axis.

Both straight and spiral teeth with module 1.25 ... 6.0 mm: ordinary, tubby, flanked are honed.

The process is carried out at constant pressure between the hone and the wheel with heavy cooling of liquid coolant. The honing removes visible defects and reduces

errors of pitch, profile and non-concentricity of initial circle and basic surface of the wheel.

The machinig time is 25 second per tooth.

12.1.4.3.3. Teeth Confrication

Teeth confrication. This is one of the methods of finishing treatment of toothed wheels of internal and external engagement when thermally treating them in complex with hardness more than HRC 58 after grinding.

The goal is to increase the accuracy and improve roughness of working surface of the tooth of loaded wheels.

The stock on the confrication is 0.005 ... 0.02 mm on a side and practically metal-removing action is made within tolerance limit.

The best choice of technological process is the teeth confrication after cementation, shaving, hardening and tempering excluded teeth grinding that sharply raises performance of the wheels manufacturing made from cemented steels at final state.

Confrication Methods:

1. The confrication by cast-iron lap carried out in the form of cylindrical wheels.
2. The confrication by cast-iron «worm».

The confrication by cylindrical laps has wide spread occurrence not only in aircraft engine building, but in the other branches of industry such as automobile industry tractor industry and is implemented by two schemes:

– the axes of the part and lap is parallel the confrication with one lap is used (Fig. 12.28);
– the axes of the parts and laps cross – the confrication with three laps is used (Fig. 12.29).

In the first case internal and external teeth of cylindrical wheels are lapped (Fig. 12.28, a, b*).* As can be seen from figure the toothed wheel receives reverse rotation from electric motor and oscillating movement along axis for cutting off equal stock along length of the tooth. The lap works «on cut-in» under load made by mechanical brake system or hydro pump. The lap can be straight-toothed and with spiral tooth.

On the machines working by second scheme the teeth of the wheel are lapped by three laps: one lap is straight-toothed and two laps are helical with different spiral direction. The cross of axes of the lap and wheel usually under the angle 10° ... 15° causes at rotation relative longitudinal sliding of the teeth like at gear shaving. The rotation is reverse, the first half machining time is in one direction, the second – in the other direction for equal removal of the stock on each side of the tooth.

The speed of toothed wheel is n = 150 ... 300 rpm. The resistance of one set of the laps is 60 hours. The treatment performance is 3 ... 6 sec per one tooth of the wheel.

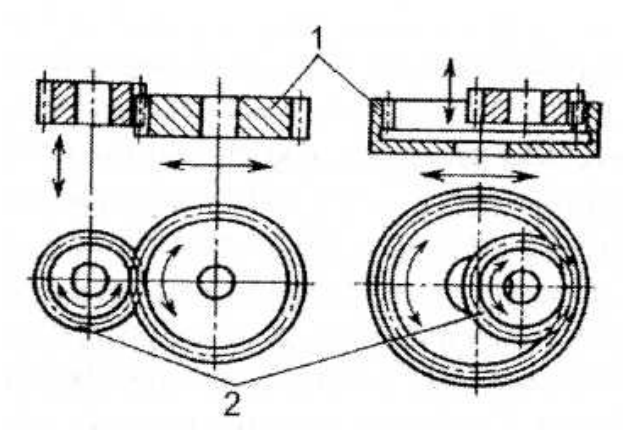


Fig. 12.28 Teeth Confrication of Cylindrical Wheels by One Lap: a – external engagement; b – internal engagement; 1 – lapped wheel; 2 – laps [2]

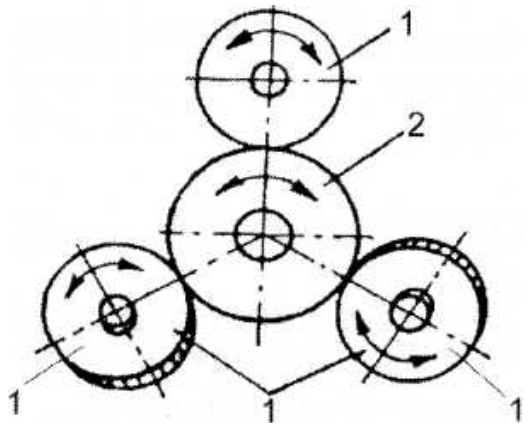


Fig. 12.29. Teeth Confrication of Cylindrical Wheels by Three Laps: 1 – laps; 2 – lapped wheel [2]

The confrication by worm lap is usually carried out on gear-hobbing machines. The lap is precisely manufactured cast-iron worm with diameter of 200 ... 400 mm in section is spur rack. The rotational velocity of the lap is *V*= 10 ... 20 m/min. The axial advance along generating line of the tooth is fixed depending on the size of abrasive grain and the value of the stock. Active pastes with next composition are applied as abrasive material in all methods:

- abrasive dust – 35 %;
- turbine oil – 50 %;
- solid oil – 10 %;
- chemical reagent – oleinic or stearic acid – 5 %.

The wheel accuracy after lapping must be with specified degree that is provided with previous finishing treatment – grinding or shaving.

The wheel confrication of 6th and 5th degree of accuracy became ineffective at errors existence in the pitch and profile of the teeth.

The surface roughness of the teeth after confrication improves. One can receive R_a = 0.4...0.2 micron on GOST2789-73 and GOST2.309-73.

The increase in the stock on confrication causes performance slowdown.

12.1.4.3.4. Polishing Teeth

Working surface is not polished as unmanageable mode when polishing does not preclude the reduction of tooth profile accuracy.

Polished are normally hollow and edges at the ends of the teeth after rounding in order to remove traces of the previous processing by blade and abrasive tools - sharp adverse stress raisers, reducing cyclic strength of the tooth.

Polishing is carried out manually with soft wheels with fine abrasive paste (GOI), pneumatic tools or polishing pasterns with wheel frequency of rotation n = 1500 rev / min. A "damming" is allowed at the end face profile, but not more than 0.02 mm, i.e. a value that provides barreling.

12.1.4.3.4. Tooth Flank

Flank of the teeth of cylindrical wheels – cutting of working surfaces at the head of the tooth (Fig. 12.30.), which is carried out by : hob; gear cutter, double angle profiles provided on milling devices by GOST 9324-81; on rams – by GOST 9323-81.

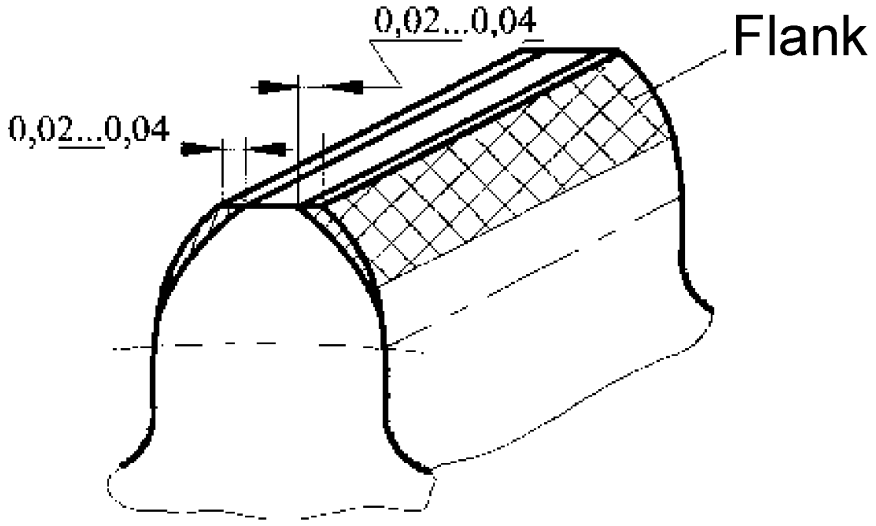


Fig. 12.30. Driving Tooth Flank

These same tools with double angles and profiles are required for grinding, shaving, and gear honing and confrication.

In the absence of cutting tools with double flank angles flanking is performed in two operations. First tooth is treated by a conventional tool, then by the profile one – such as modular cutter, copy method. At that flank is removed from two teeth adjacent to one cavity simultaneously. For flanking by form-generated method special hobs or shapers with a complex profile are required.

Flank of the teeth as a finishing step is the "damming" at the top of the profile in order to distribute the load evenly over the surface of the tooth. Normally flanked are gear - wheels with few teeth – 16 to 24, operating in conjunction with the internal gear wheel.

For gear module 2 ... 4 mm damming at the tip of the tooth on each side is of 0.01 ... 0.04 mm and gradually decreased by 2 ... 4 mm in height.

The pilot production of tooth flank is performed manually by abrasive bars; in series – when grinding them on the machines of the company MAAG, Orkut, and others.

12.1.4.4. Roll Forming of Teeth

Roll forming of teeth can be hot and cold.

Hot roll forming is performed on a special mill with blank heating to 1000 ... 1100 ° C. The blanks are deformed by rolls, having a form of corrected gears with chosen sizes of involute profile teeth. The rollers are made of hardened alloy steel. The teeth of the rolls are carefully polished. Diameter is greater than the diameter of the rolled wheel to increase the degree of engagement overlap. After roll forming the structure of the metal surface layer (cold-worked layer is generated) improves, which significantly increases the strength and durability of gears.

The rotation of the blank in rolling of the teeth can be of forced and free. In forced

rotation rolls and blank get coordinated rotational movement of the gear with respect to to gear ratio of their teeth number. During free rotation mandrel with blanks, receiving traverse advance along its axis, is provided with the driving gear, providing the required number of turns of rolled blank in engagement with the rolls.

After the release of the guide gear out of engagement with the rolls the latest are engaged with the knurled part of the workpiece and rotate it.

Hot roll forming can be carried out by processing the workpiece by a radial feed of rolls and just a few pieces by the axis feed.

Piece roll forming (Fig. 12.31, *a)* is performed by gradual pressing rotating rolls *1* in a rotating blank *2*. Rolls are moving to each other via spindles. Billets are heated by HF currents.

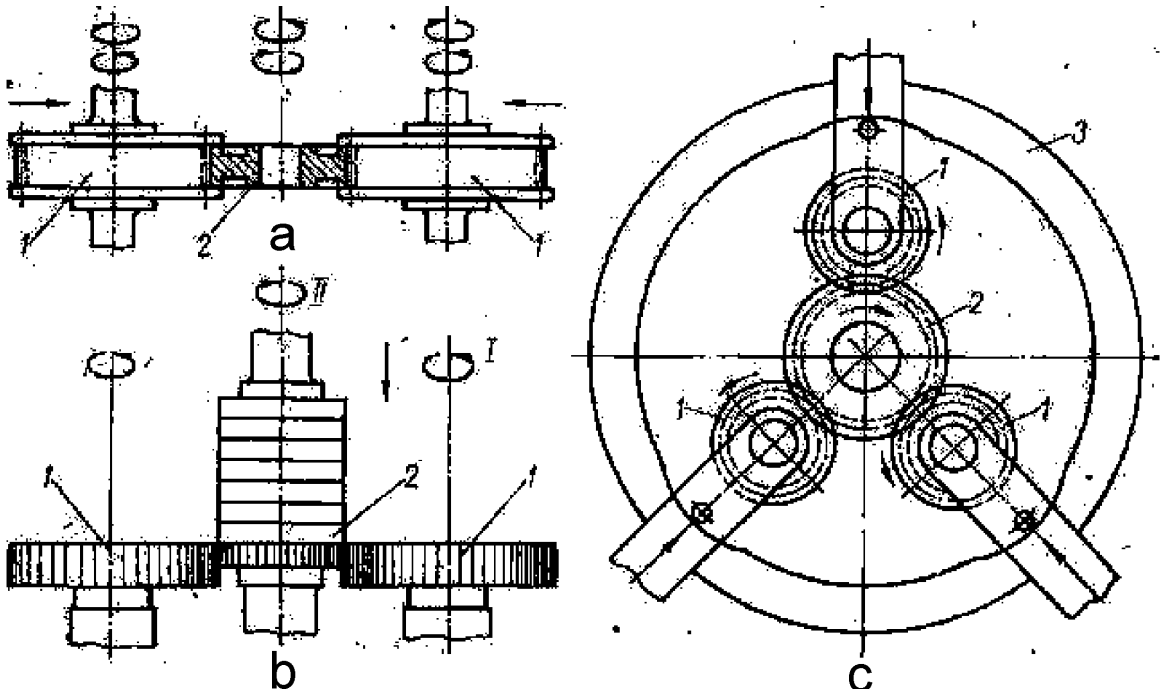


Fig. 12.31. Toothed Profiles Roll Forming:
a - piece; *b* - batch; *c*- calibration of teeth by cold rolling [2]

After installation and heating of a blank the form rollers start rotating with the reversal of automatic cycle, receiving a radial feed. When rollers reach the established center distance, feed is turned off and rolls make a few turns for teeth calibration.

When rolling a few pieces at once (Fig. 12.31) working motions are synchronized rotations I and II of the rolls 1 and blanks 2 and axial movement of the workpiece package relative to rolls (top to bottom). Interaxis distance between rollers and blanks package is constant.

Rolled rolls with axial feed of blanks, which are either bars or the package of several blanks, have special cut bevel within which the tooth height is increased gradually.

Gears rolled on mill are subjected further only to two machining operations - boring of the central hole, and deburring (if available).

Mill capacity is of 240 ... 300 gears per hour (m = 1,5 ... 3 mm). Productivity of gear hobbing machine when cutting the same gears is – 7.5 pieces per hour.

Hot-cold roll forming of the 7-8-th degree of accuracy gears with the module

2 ... 5 mm and an outer diameter of 40 to 250 mm (method of two-time plastic deformation) developed by the institute "Orgstankinprom". In the stamped workpiece the tooth is formed by the method of hot rolling on the special machine, followed by calibration of the machine for cold-rolling. Further processing of gears is carried out on multi-spindle aggregate-milling machines and multi-spindle diamond-boring machines.

Hot roll forming teeth on the first machine is performed by two geared rolls (a batch and single-piece roll forming is possible). To avoid formation of scale roll forming is carried out in a neutral (protective) environment. Machine performance during batch processing of module gears of 2 ... 5 mm is 250 ... 350 parts per hour.

Calibration of teeth (finishing) on the second machine is carried out by the method of centerless cold plastic deformation (Fig. 12.31, c). Tools are 3 recuperators *1*, located at an angle of 120 ° and set freely on their axes. Recuperator are rotated by blank *2*.

The working movement at this are rotation of treated engaged gear with the recuperator and radial feed of recuperators to the item (variable spacing). Recuperator convergence is made by rotating the centering ring 3 having eccentric portions. In this recuperator spindles are moved radially to the abutments.

Several plants use roll forming of bevel gears with straight and curved teeth with a module up to 10 mm and a diameter of 300 *mm*. According to the method of hot rolling the 8-9-th degree of accuracy is achieved; method of hot-cold rolling – the 7-8-th degree of accuracy.

12.1.4.5. Methods of Bevel Gears Teeth Treatment

The aircraft engine industry uses bevel gears with straight and rotation tooth. The latter provide a smoother engagement, thereby, reducing the noise and vibration. By their design features they both can be in the form of bushings and rollers, gears, that is the tooth form virtually does not effect on the content of the production process before and after the teeth cutting. However, the processes of cutting and finishing processing lines and circular teeth are significantly different.

12.1.4.5.1. Cutting and Finishing of Bevel Gears with Straight Teeth

Straight bevel gears with module m = 2 ... 8 mm are usually cut in two steps: preliminary and final.

Rough pre-cutting –each cavity is cut individually to the full depth by copying, leaving allowances for final treatment of 0.8 ... 1.2 mm on the tooth thickness.

Finally-finish cutting is performed by form generating method on gear planing machines.

In turn, to improve the accuracy of treatment and conservation of expensive equipment, for a long time in good condition, finish cutting is performed in two passes, leaving an allowance for the latter 0.4 ... 0.5 mm on the tooth thickness.

Bevel wheels with small modules (m <2,5 mm) are pre-cut and cut on a single machine.

Isolation of pretreatment in a separate operation allows to avoid premature wear – loss of accuracy of gear planing machines, mod 526, 534, and others working by form generating method.

Preliminary Cutting of Bevel Gears' Straight Teeth

For small-scale production, operation is carried out on the horizontal-milling machines by copying with means of universal de-inflammatory devices.

Cutter – disk module, is constructed on the smaller module of the wheel.

Material Cutters – high-speed steels: P9, P6M5 and others.

The processing mode is set, as at the conventional milling, on the horizontal mills.

Also gear planing machines are used after long-term operation - have lost their accuracy, but is cut by dividing the cavity.

For large-scale production of rough cutting of spur bevel gears operate on multi-spindle machines, "Komsomolets" and mode EZ-1 modular disc cutters (Fig. 12.32) while three wheels by method of division with overlapping auxiliary installation time and removing them after processing.

It is advisable to use in mass production semi-automatic 3-spindle machines of Saratov factory and firm "Gleason" with a modular tool head (Fig. 12.33), simultaneously processing three wheels with a small number of teeth, and the average unit 2 ... 3.5 mm method of division.

Machines have high productivity. Processing Time one cavity does not exceed 30 seconds. Finish cutting teeth is performed on machine tools of plant "Komsomolets" mod. 526.523, 5A26,532, and machines of firms "Glisson" and "Garbek".

Each tooth is treated by running two planing cutters, engaged with the wheel during its reverse rotation (Fig. 12.34). After running in the opposite direction of the blades are disengaged and performed re-indexing wheel for the next tooth. The contour of the cutting edges of cutters is a spur tooth rack; their movement – directly opposite.

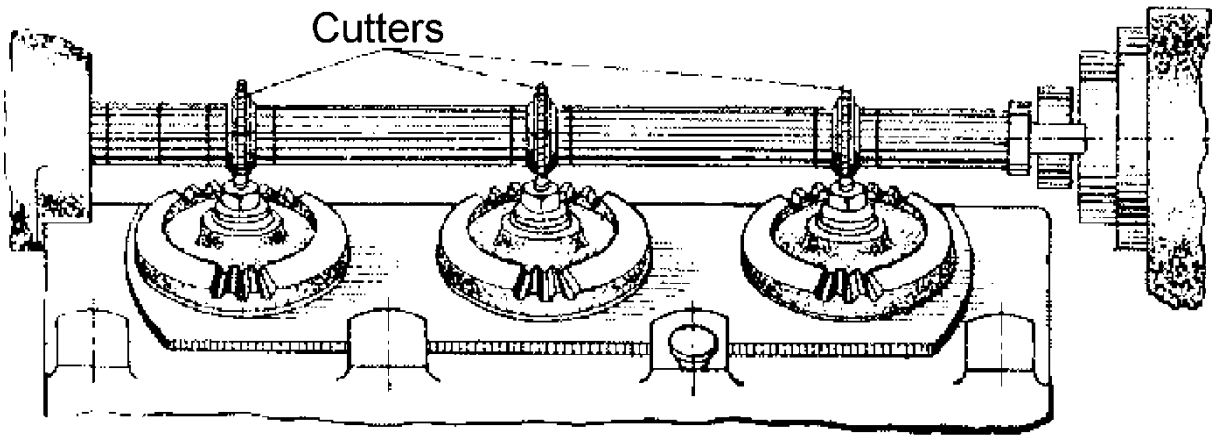


Fig. 12.32. Rough Cutting Tooth Bevel Gears with Multi-Local Devices

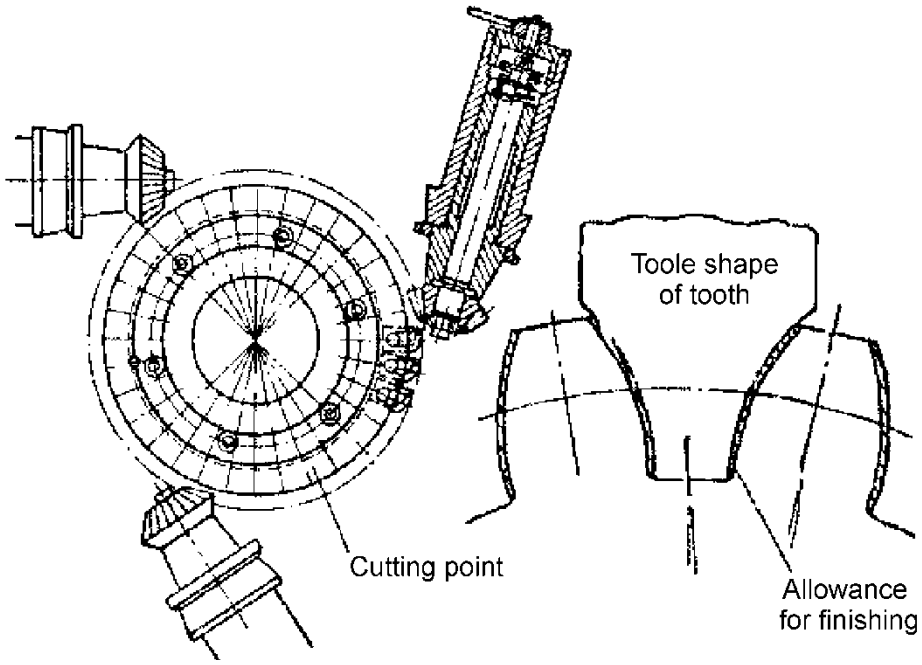


Fig. 12.33. Rough Cutting Cavities Bevel Gears on the Semi-Automatic Device with Cutter Head

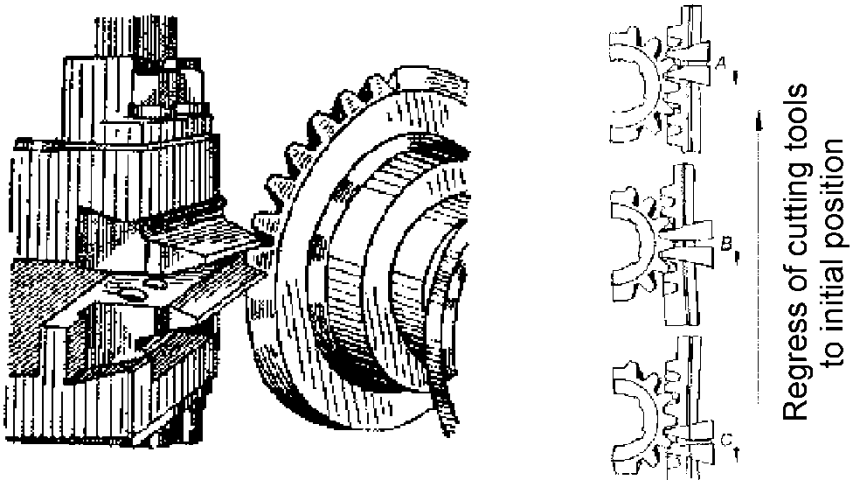


Fig. 12.34. 12.34. Bevel Gear Planing by Two Cutters by Form-Generating Method: A – B – C – the beginning and the end of a full treatment of each tooth

When finishing form-generating method a fairly high accuracy of the tooth thickness is achieved. For example, for the modulus of 3 mm and the number of teeth 18…25 the error of the tooth thickness is 0.04 mm, the gash jumping is 0.03 mm, roughness – 0.015 mm. Planing process is fully automated, but is intermittent, which explains the relatively low productivity of these machines. Fig. 12.35 shows examples of installing and fixing the gears on the machine.

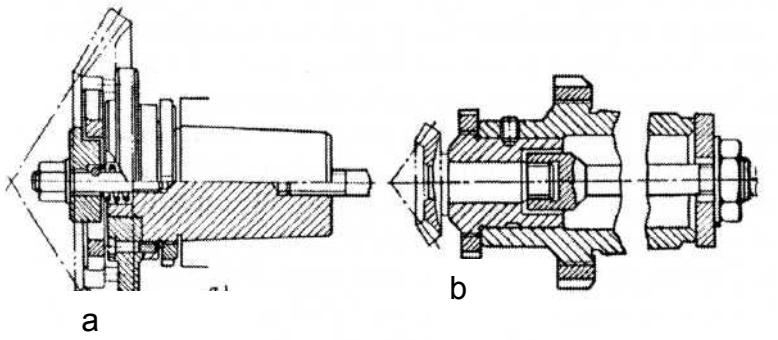


Fig. 12.35. Installing Toothed Bevel Gears in Gear Planing Machine: *a* – by opening with front fastening; *b* – on the shank with rod tightening

Grinding of Bevel Gears Straight Teeth

Grinding of spur bevel gears is performed by form-generating method on the machines ENIMS by one wheel – mode. 5870, and two wheels, mod. 5A84 with traverse wheels advance (Fig. 12.36).

On one wheel machines the tooth working surface is grinded. When optional grinding of teeth on both sides of the processing is carried out in two steps: a machine changeover to the second side, which is made in the pilot production; with two machines configured separately for each side of the tooth – in series production. The machines with two wheels, the teeth are grinded simultaneously from two sides as in grinding the cylindrical wheels. The serial production of GTE does not exclude to use gear grinding machines, working with two wheels by for-generating method, of well-known firms: "Gleason" – mod. 105 in the United States, "and Gedenreyh Garbek", mod. 60TS – in Germany; "Hitachi" mod. 600V-1, in Japan; "MAAG", mod. KS-42, Switzerland.

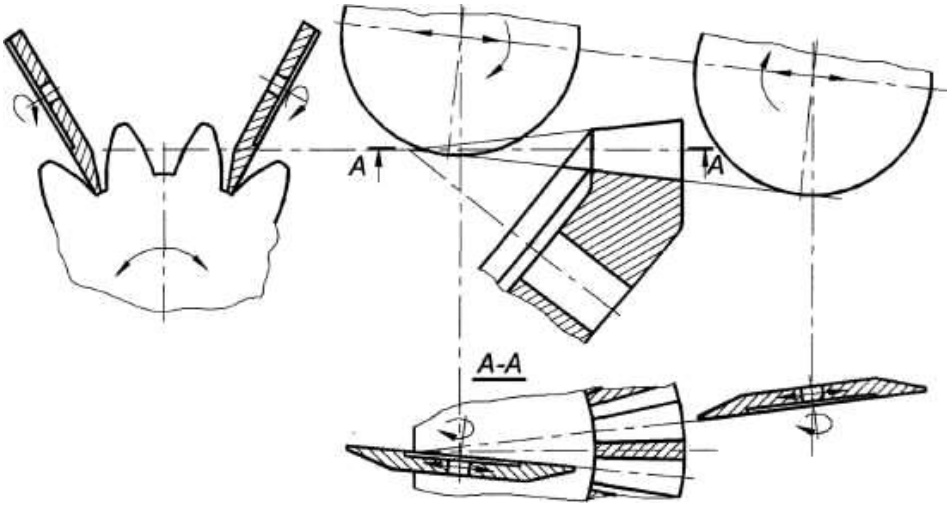


Fig. 12.36. Grinding of Bevel Gear Straight Tooth

12.*1.4.5.2. Cutting and Finishing of Spiral Teeth of Bevel Wheels*

Spiral teeth of bevel wheels are cut by form-generating method on plant`s "Komsomolets" machines, mod. 527, by cutter head, the relation position of which with wheel is shown in Fig. 12.37. The head radius depends on teeth curving.

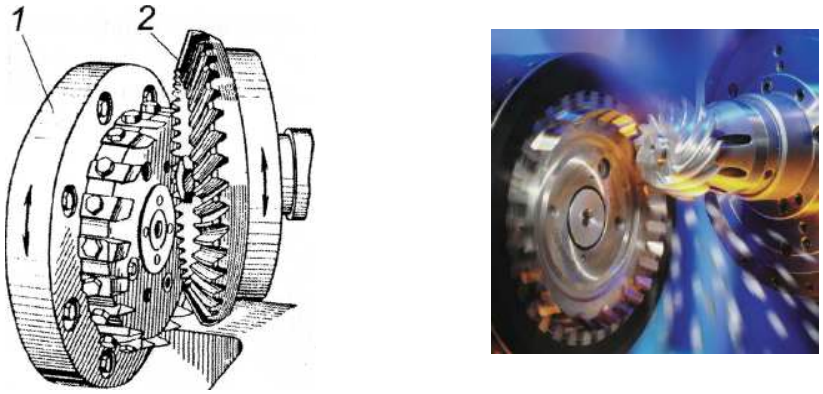


Fig. 12.37. Spiral Teeth of Bevel Wheels Are Cut by Form-Generating Method
1 – cutter head; 2 – cut wheel

Cutter heads are unilateral and bilateral.

Unilateral cutter heads: cutters cut edges on only one side – on the outside or from the inside. They can treat one side of the tooth – convex or concave.

Bilateral cutter heads: cutters alternately cut edges on both sides and can handle at once both sides of the tooth.

A method of cutting is selected for both of mating wheels.

Usually the roughing process – cutting of teeth, is performed by bilateral heads what provides high performance.

Finish cutting is performed by three ways.

The unilateral method. Both sides of each tooth of the mating wheel are cut with unilateral head. Accuracy of engagement is high; productivity is low.

Simple bilateral method. Unilateral heads treat only smaller mating wheel of mated ones; larger is cut by bilateral head.

Doublebilateral method. Both mating wheels are cut by bilateral heads. It increases productivity but low accuracy is obtained, which limits its use.

Grinding of spiral teeth is performed by form-generating method on machines ENIMS, mod. 5872, by the company "Gleason", mod. 15 and 17 (Figs 12.38, 12.39).

Tool – cup wheel , which in axial section is a rack tooth. The mutual position of wheel and disc in the grinding process is shown in Fig. 12.40.

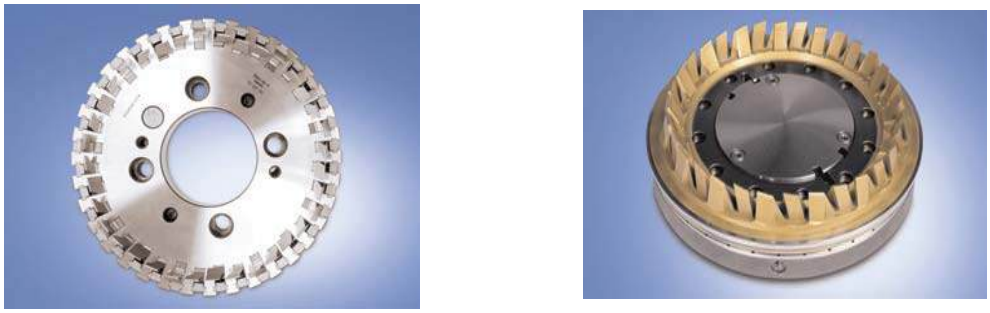


Fig. 12.38. Cutter Heads



Fig. 12.39. Machine Working Site

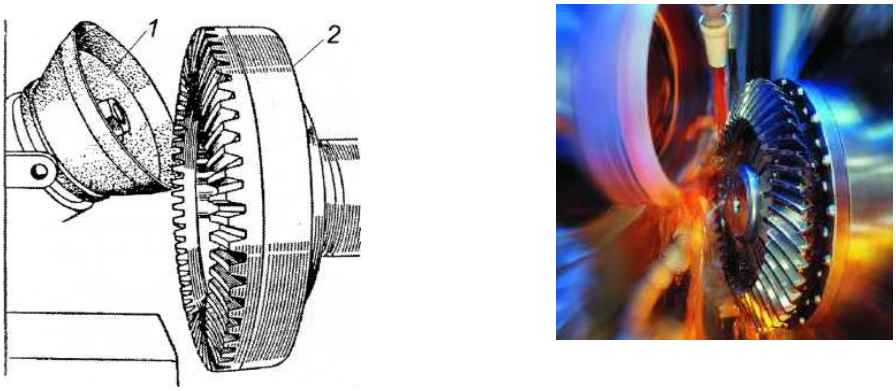


Fig. 12.40. Grinding of Spiral Teeth of Bevel Wheels by Form-Generating Method : 1 – cup grinder; 2 – gear wheel

Generating motion is created by slow reverse rotation of the wheel about its axis and rotation of the circle center being in engagement with the wheel about the same axis. To reduce the complexity of machining sometimes it is advisable to grind only the working side of the tooth.

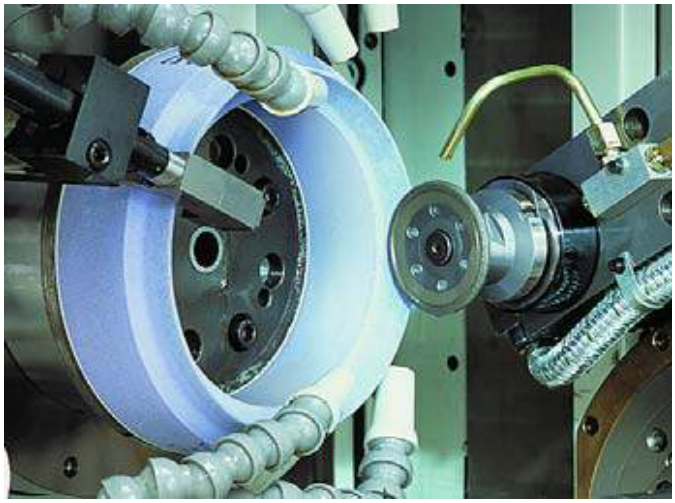


Fig. 12.41. Machine Working Site

12.*1.4.5.3. Toothing of Worm-Wheels*

Worm wheels are cut on gear-hobbing machines with worm gear tooth cutters with radial wheel infeed (Fig. 12.42, a) or with tangential infeed of miller (Fig. 12.42, b).

The diameter of the hob must be greater than the outer diameter to double the amount of radial clearance, i.e. 0.25 *m* while maintaining the center distance (see Fig. 12.43).

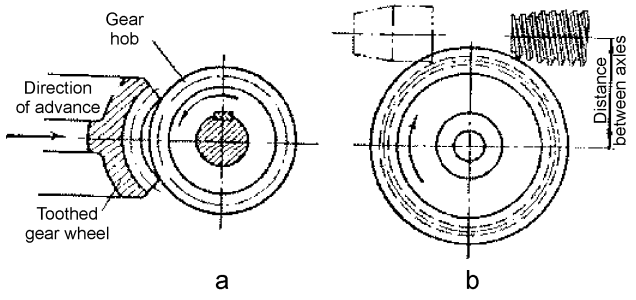


Fig. 12.42. Toothing of Worm-Wheels



Fig. 12.43. Tool for Machining of Worm Wheels

12.1.5. Treatment of Splineway

Splineway elements in conjunction of parts are used for safe center adjustment and their high pulling torque.

The shape and nature of the ganging splineways are rectangular, involute and triangular (Fig. 12.44).

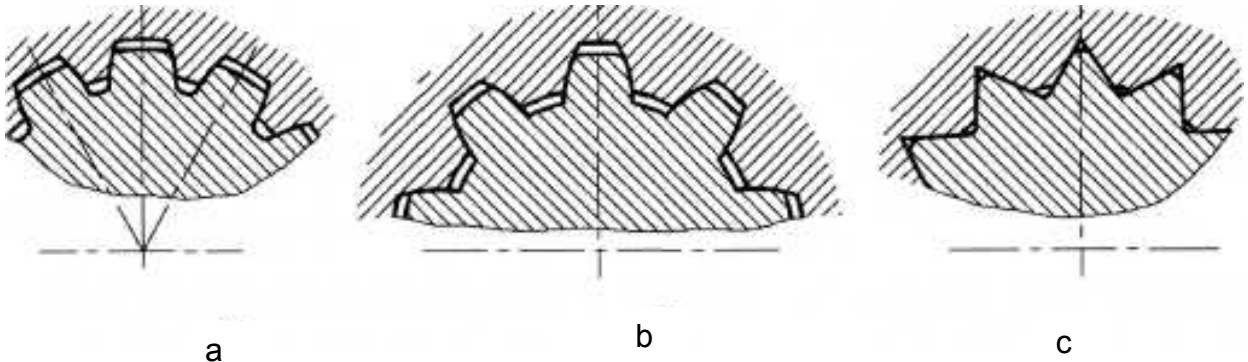


Fig. 12. 44. Scheme of Spline Joints:
a – rectangular; b – involute; c – triangular

Rectangular spline joints may be arranged centering on flank surface at the inner or outer shaft diameter .The most accurate joint can be obtained by centering the outside diameter of the splines, it`s easy provide the required accuracy both in pulling the port hole and at the outer diameter of the grinding shaft splines.

Centering on the inside diameter is useful when processing mating parts hardness exceeding HRC 40 units , sanding the bottom of cavities at the outer slots easier than the bottom of the cavity between the splines in the port holes, high accuracy is ensured by pulling (see Fig. 12.45).

Involute splines are used for the transmission of high pulling torque. They provide high accuracy of centering of joined parts. They can be cut into a well-known gear production machinery.

Triangular spline joints are used in fixed joints of thin-walled parts of small dimensions. Their centering control is performed only on the flank surface of splines.

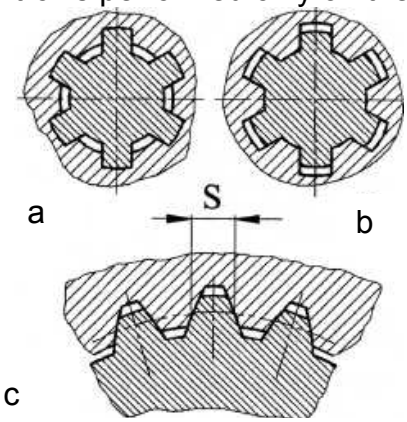


Fig. 12.45. Types of Mutual Centering of Parts in Spline Joint:
a and *b* – on the outer and inner shaft diameters, and *c* – flank surface of involute splines

Accuracy of splined crown is determined by:

– accuracy of diameter of centering sleeve;

– accuracy of the profile and uniformity of splines pitches;

– accuracy of spline pitch position relative to design bases of joined parts, i.e., – spline runout.

The requirements for precision splined joints elements depend on the method of mutual centering of parts.

When used in products construction rectangular splines centering is carried out more often on the outer diameter.

Involute and triangular splineways are centered along the profile.

12.1.5.1. Machining of External Splines

The outer cylindrical surface of the splined crown – the surface for splines, is typically treated separately from the actual machining spline runout. At the same time the required accuracy of their relative position is ensured by compliance with the principle of the constancy of the technological bases and accuracy of the installation of equipment parts and accessories.

The outer rectangular splineways can be cut in two ways:

– by division method;

– by form-generating method .

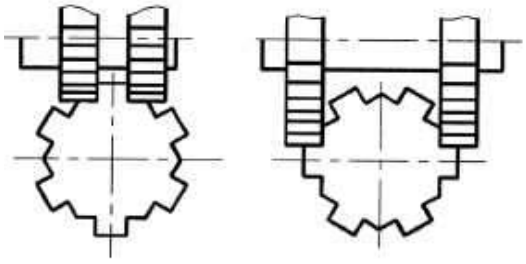
When processing by division method they are cut on plane horizontal milling machines with dividing devices in two ways (Fig. 12.46 and 12.47).

The first variant. By using doubled millers (see. Fig. 12.46) side surfaces of splineways are got, then (Fig. 12.46) bottom of the cavity is treated with radius miller (12.47, a). *The second variant* – (Figure 12.47 b) slot is made by form milling cutter.

The first variant provides higher accuracy; the second – greater performance. It is used when having a large number of splineways, which are grinded in further work.

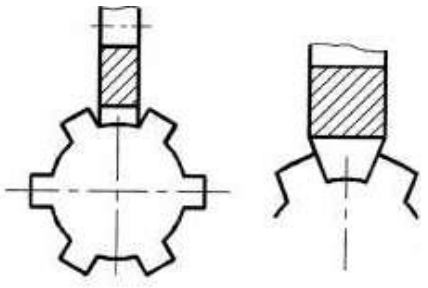
Better ways of tapping of the outer rectangular splineways is to cut them by form-generating method by a splined worm metal slitting saw, which profile and design are determined by the graphic-analytical method.

For that horizontal spline-milling machines of mod. 5434 are used. In short part splineways can be cut in the gear hobbing machines.



a b

Fig. 12.46. Treatment of Splineways on Plane Horizontal Milling Machine by Doubled Millers



a b

Fig. 12.47. Methods of Finishing Processing of Splineways

External involute splines are cut by on gear hobbing or machines, or by form-generating method worm modular metal slitting saw.

Examples of slot milling machine installation pieces are shown in Fig. 12.48. With no outlet for milling rectangular and involute splines are cut by cutter splined to conventional vertical gear-shaping machine of company "Gleason".

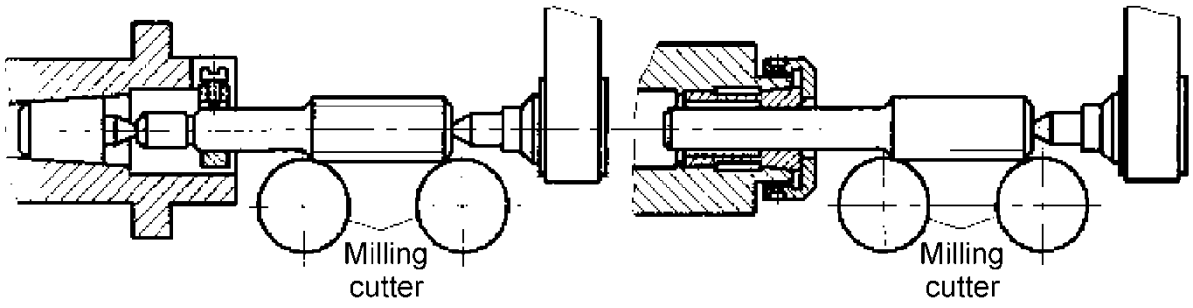
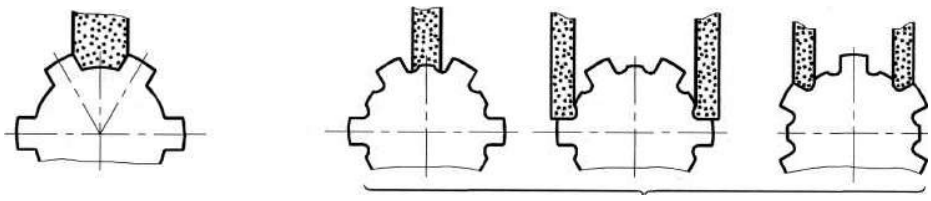


Fig. 12.48. Examples of Mounting of Workpieces on Spline Shaft Milling Machines

Rectangular splineways are grinded: on the outer surface of cylindrical grinding , and the depressions and the bottom sides (in the case of joint centering by inner diameter) are made by the scheme shown in Fig. 12.49, a, as well, or by the scheme shown in Fig. 12.49, b. Grinding by the scheme *a* provides better performance than with a split-grinding of sides carried out by the scheme *b* (see. Fig. 12.49).

Involute external splines in most cases are grinded by form wheel on copy machines by copying method. Slots with module m> 2.5 mm are grinded by dish grinding wheel on gear grinding machines by form-generating method.



a *b*

Fig. 12.49. Methods for Grinding of Outer Square Splines

The triangular splines in the aircraft engine building are used rarely. They are treated on gear machines by copying machines or by form-generating method, depending on the size of a specially designed tool.

12.1.5.2. Treatment of Internal Splines

The through-holes most often are got by pulling them on the horizontal or vertical broaching machines, if the items are designed to use this high-performance processing method. The formation of the splined hole is carried out in a single operation or two.

In the first case it is pre-bored or even grinded on the size of the internal diameter of the splines, and then by pulling the profile of both rectangular splines and involute and triangular splines are formed.

When processing in one operation the outer and inner diameters are got by pulling. This significantly enhances broach design and there is an additional requirement in the development of the technological process of manufacturing parts – the use of the spline hole as a technological basis in the processing of concentric towards it surfaces.

In the through holes all kinds of slots are cut only by slotting. At the same time involute splines are preferred, which without difficulty can be got on gear shaping machines by running a standard tool.

Processing of slotted holes in the parts with a hardness of more than 40 HRC units, i.e. after thermal treatment – quenching and tempering, is complicated. To reduce warpage it is necessary to quench them on special arbors or under pressure, and to carry ointment by special lapping. Elements of large diameter slotted holes are grinded by profiled wheel by method of copying.

For large produce programs fine-grained splines on the shafts with m <3 mm is advisable to get by plastic deformation – rolling before heat treatment at a hardness of material not exceeding HB 220. After heat treatment they need to be grinded. Along with the increase flow rate this spline treatment method enhances their strength characteristics.

12.1.6. Ways to Improve Quality of Cogwheels' Treatment

Factors affecting the quality of gears:

Tools: use of hobs with a reduced profile angle (precision machining of wheels is increased by one or two degrees); finishing and roughing cutting of hardened wheels by worm milling cutters with larger inclination angles (λ=-15…60°) of front surfaces of the teeth; use of hobs, allowing to combine pre and finish cutting, elastic hons and others.

Machines: introduction of the cycle adaptive machining control systems by multipass cycles with automatic change of modes in the process of treatment with their optimization; operational correction systems for machines and electronic systems of profiling of grinding wheels; increase of stiffness of kinematic chains and dynamic stiffness of bearing systems; the use of the automatic allowance distribution in gear grinding; adherence of the workpiece and flywheel tool to spindle, reducing the natural frequency of the tool and the wheel at the teeth-finishing by method of break-free; reduction of gaps in joints of machine components; installation of machines on antivibration foundations and others.

Treatment methods: wider use of diagonal hobbing – transfer of continuous movement to hob along its axis for load balancing all the teeth; combination of gear cutting wheels with treatment of wheel bases; reverse roll forming by rack tool; axial permutations of hobs; roughing and finishing of wheels in one setup step by stepped ram et al.

Processing methods: the use of intermittent grinding, allowing using of grinding intervals to reduce the cutting temperature; obtaining high-quality blanks with the proper arrangement of the fibers and the same metal density in backer or closed dies; utilization of the chemical-thermal preliminary and final processing; higher-hardness of the material to be treated (up HRC_3 45 ... 50); reduction in errors of workpieces basing by means of a holding force directed perpendicular to design datum; application of settings to maintain the ambient temperature within 20 ° ± 1 ° C and others.

12.2. FINISHING AND STRENGTHENING TREATMENT

12.2.1. Machine Parts' Wear Resisting Property

Types of wear. The mechanism of the surface layer destruction is different due to the variety of changes that occur in the contact layer. There are such types of wear: **mechanical** (fatigue, abrasion), **mechanomolecular** and **mechano-corrosive** (oxidative, fretting, ect.). According to the intervening medium nature one may distinguish **non-lubricant chafing, boundary friction wear**, **wear with an abrasive material**. By the surface deformation nature wear can occur while elastic and plastic contact and at microcutting.

1. ***The abrasion wear*** is a common type of the parts' surface damage of the transport, road, agricultural, mining and other machines that work in process media containing abrasive parts. The abrasion wear is the result of the cut-off and plastic deformation of microasperities (rough edges) by hard foreign particles at mating surfaces relative movement. These particles are usually inorganic and have a non-metallic atomic bonds, that cause relative simplicity of the physical processes of such wear type. A particle separation occurs when a single or multi effect of the abrasive body. The wear occurs consequently in the form of microcutting or as a fatigue damage (low-cycle under elastoplastic straining, fatigue damage at multicyclic influence). To reduce the abrasion wear the level of abrasive effect is reduced and the surface hardness of the parts is increased through the thermohardening, surface plastic deformation and powdered carbide sputtering. To prevent the excessive abrasion (mechanical) wear limits a specific power raiting, required to overcome the resistance in the contact area:

$$w = \sigma_c f \upsilon_{ss} \le [w],$$

where f is a friction coefficient (resistance factor) between the contacting parts; V_{ss} – sliding speed; $[w]$ is an accepted friction power, $[w]$ = 150…250 N•mm/(mm^2•s) for spline joints, etc.

1. ***Mechanomolecular*** **wear** occurs at high contact voltages in the place of homogenous materials parts mating (gear and hyperboloidal transmissions, screw joints, ect.)**.** It begins with a local plastic deformation and oxide layers fracture on certain areas of the contact surface, and ends with a molecular adhesion (grasp) of the material parts of these areas and the subsequent destruction of the seizure areas with relative motion. The development of the parts rubbing surfaces damage by seizure is called jamming. The seizure intensity increases with the growth of contact stress (pressure), slippage speed, contact zone temperature and other factors. In order to prevent seizure of contacting parts on a surface the protective coating and the oxide layers are applied, the lubricant is supplied to the contact zone, the surface strength (hardness) of parts is increased, the contact stresses and the relative movement speed are limited.

The wear calculations for prevention the mechanomolecular wear are due to the following rules:

$$\sigma_c \le [\sigma_c]; \theta_c \le [\theta_c],$$

where $[\sigma_c]$ и $[\Theta_c]$ – admissible contact voltage and contact zone voltage.

When applying the $[\sigma_c]$ and $[\Theta_c]$ values, the previous experience of such designing plays decisive role.

2. ***The mechano-corrosive wear*** is observed in machines and devices where the friction parts chemically interact with the environment. The surface of the friction parts is destroyed under the influence of two simultaneous processes – corrosion and mechanical wear. When vibrating in a metal and air contact conditions the mechano-corrosive wear occurs in the form of fretting. The thin metal oxidized surface layers that are not removable from the friction zone and in turn abrasive particles (black powder) are broken as a result of a small cyclic relative parts displacement. The oxidation process in the open air is continuous, so the destruction is progressive in nature. Fretting enables the destruction of the riveting, press, screw, slotted and keyed joints. Various methods of surface hardening of the contact zones are used to prevent fretting, applying soft electrodeposited coating, spraying teflon and rubber cover, etc. In chemically active environments, in liquids and various gases, where corrosion processes are active, the mechano-corrosion wear of parts causes substantial damage. Corrosion-resistant materials are used to prevent corrosion and mechanical wear.

12.2.2. Material Electromechanical Processing Technology

Technology main point: the electromechanical processing (EMP) is a highly effective technology of surface hardening by concentrated flows of energy, based on the complex thermal deformation effects with an electric current of high density (108...109 A/m^2) and low voltage (2...6) through the zone of parts contact and deforming the electrode, tool (roller or plate), moving in mutually perpendicular directions at a speed V and supply S (Fig. 12.50).

As a result of a large amount of Joule heat liberation a high (106 °C/s) local heating of micro-volume surface with its simultaneous plastic deformation and subsequent

intensive cooling occurs (105 ° C / s) due to the removal of heat in the interior of the metal.

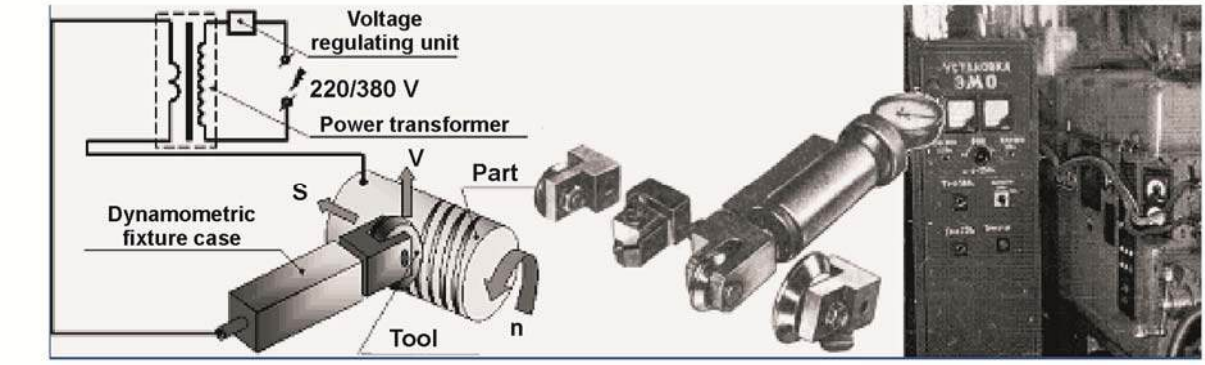


Fig. 12.50. Electromechanical Processing Technological Complex and Its Scheme [2]

As a result of a powerful heat shock the hardened "white coat" is formed on the surface of the material. It is a unique martensite structure (hardenite, nanostructural martensite) with a high strength and wear resistance (Fig 12.51.).

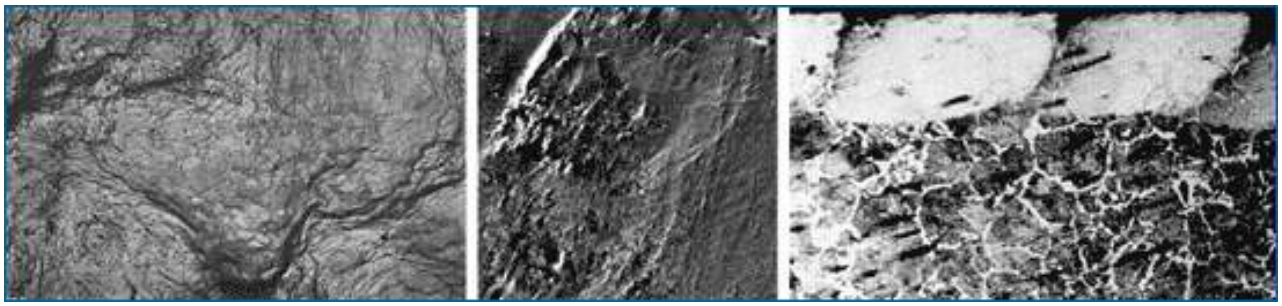


Fig. 12.51. Metallographic Structure of White Layer (electron, scanning and optical microscopy) [8]

Purpose. The main purpose of EMP is a metal products surfacing for improving their performance properties such as wear resistance, fatigue resistance, corrosion resistance, etc., i.e. the wearing capacity of movable matings increase in 2-6 times depending on friction and wear conditions; an endurance strength improving to 30 ... 70 % and longevity (more than an order of magnitude) of components, operating under cyclic load; back-to-back endurance increasing (i.e. steel SH15 in 1.8-2 times compared to carbonitriding) [8].

Area of Application:

- external and internal cylindrical and flat surfaces of steel parts hardening to a depth of 0.2 mm with the increase of surface microhardness up to four times and simultaneous reducing of the roughness to one or two classes;
- hardening of parts surface layer to a depth of 0.2 ... 5 mm, followed by finishing the surface by grinding or running-in;

- hardening of gears, spline shafts, cylinder line bores of internal combustion engines, piston rings, thread spindles, large bulks journals;
- work surface electromechanical recovery according to a hot plastic landing of some metal volume from a treatment zone area;
- momentum electromechanical hardening, that ensures the formation of regular structures of surface layer to the strength and plastic properties distribution on the local volumes of the surface under specific operating conditions;
- combined treatment of stress concentrators, which includes electromechanic drawback and subsequent surface deformation;
- plasma coatings processing to form of amorphous and nanostructure of reinforcing particles and increasing adhesive and cohesive strength of the coating to the level of solid materials.

Main Advantages of the Method:

- environmental cleanliness and electrical processing, lack of radiation (including X-rays and secondary) and emission of harmful substances;
- flux absorbing coatings, protective gases, vacuum, electrolytes and other special consumables apply omission;
- oxidation and decarburization of the processed surface omission, due to the fact that the hardening process takes place in a closed area of contact of the tool with the work surface;
- high quality (low roughness) of a hardened surface;
- treatment of various shapes and sizes products to obtain the required quality, the possibility of processing the hollow, long non-rigid parts without warpage (because of the high temperature heating zone localized at the point of contact with the tool surface), as well as large parts;
- "individual approach" to any particular surface treatment, taking into account the operating conditions and loading scheme, possibility of surface treatment to form a predetermined distribution of physical and mechanical properties for its local volume, as well as the establishment of regular discrete structures and regular surface microrelief;
- the limited areas processing without affecting on the neighboring use of EMP for local heating to apply in further processing;
- flexible and quick change of modes and processing schemes, as well as tool change, a minor cost of manufacturing, maintenance and operation of equipment;
- possibility to use the machining transition as the final (finishing and reinforcing) casting, at the same time the problem of special operations (heat treatment and finishing).

12.2.3. Diamond Smoothing

Diamond smoothing (DS) is a process intended to reduce the surface roughness and surface hardening of metals to improve the fatigue resistance of gas turbine engine compressor shaft.

The diamond smoothing is a variation of the gas-turbine engine parts surface layer hardening technology applying mechanical methods. The gas-turbine engine parts diamond smoothing method bases on the ability of metals to deform in a cold state.

Under the pressure of a diamond tip, with a high hardness and a small friction factor as well as higher thermal conductivity the protruding uneven surface of the shaft is plastically deformed. At the same time, the surface roughness is reduced, and a new micro-relief is obtained.

The main features of the surface obtained because of a diamond burnishing, are:

- Improvement of roughness to R_a = 0.15...0.2 mm; formation of a surface layer with specified parameters values and depth of hardening;

- Formation of surface texture microroughness with corner radius of lugs and protrusions that is considerably greater than in the machining process and the adjustment amount of the protrusions and lugs per unit area;

- The formation of a surface that keeps well the lubricating oil and resists wear, seizing and corrosion.

Diamond smoothing of shafts is carried out due to the following scheme (see Fig. 12.52.): forced circulation ω of the blank 2 while advancing of S instrument 1 along the contacted surface.

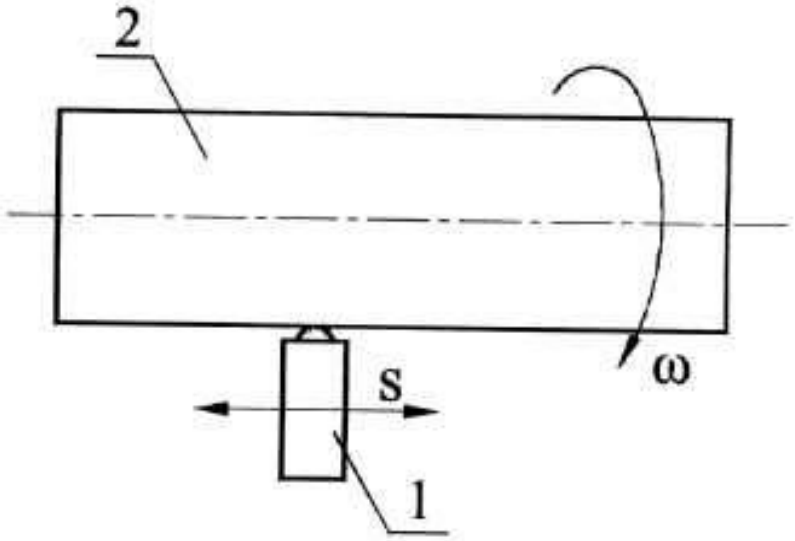


Fig. 12.52. Shaft Diamond Smoothing Process Scheme:
1 – instrument, 2 – blank

The efficiency of a diamond smoothing is defined by technological process modes, as well as views and modes prior surface treatment, which are regulated by sectoral recommendations.

A hardening process differs with its tool simplicity and allows to use universal equipment, on which machining of turbine engine shafts is carried out.

It is recommended to do a DS on turners, boring and diamond-borings of a high accuracy. The usage of a normal machine accuracy 1M63, 163 is possible. The infeeder machine must ensure the application of small feed within 0.03 ... 0.075 mm / rev.

Selection of Smoothing Parameters

The smoothing parameters that influence the roughness of the blank surface are: smoothing force, supply and working tool radius. At the smoothing force increase to a certain value the roughness decreases.

Under optimal values of power source smoothing, the irregularities are smoothed and formed roughness caused by the process of smoothing related to the supply. With further increase of irregularities, the height of burnishing force increases as compared with the lower one, cracks may appear in the surface layers.

The value of smoothing power R_y is determined by the following formulas:

for hardened steels:

$$P_y = 0.013 \cdot HB\left(\frac{DR}{D+R}\right) \ ;$$

for materials of a high and medium hardness:

$$P = 0.008 \cdot HV\left(\frac{DR}{D+R}\right) \ ,$$

where HV– hardness of the processed surface by Wikkers; D – diameter of the processed surface, mm; R – the diamond working part radius, mm.

The smoothing strength is within 200 ... 250 N for parts of high-strength materials and in the range of 100 ... 150 parts of N for medium hard materials. The main criterion for the tool radius selecting of the sphere is the hardness of the blank material: for parts of mild steels and non-ferrous alloys, it should be 2.5 ... 3.5 mm, for parts of materials of medium hardness – 1.5 ... 2.5 mm, for parts of high-strength steel (HRC> 60) – 1.0 ... 1.5 mm.

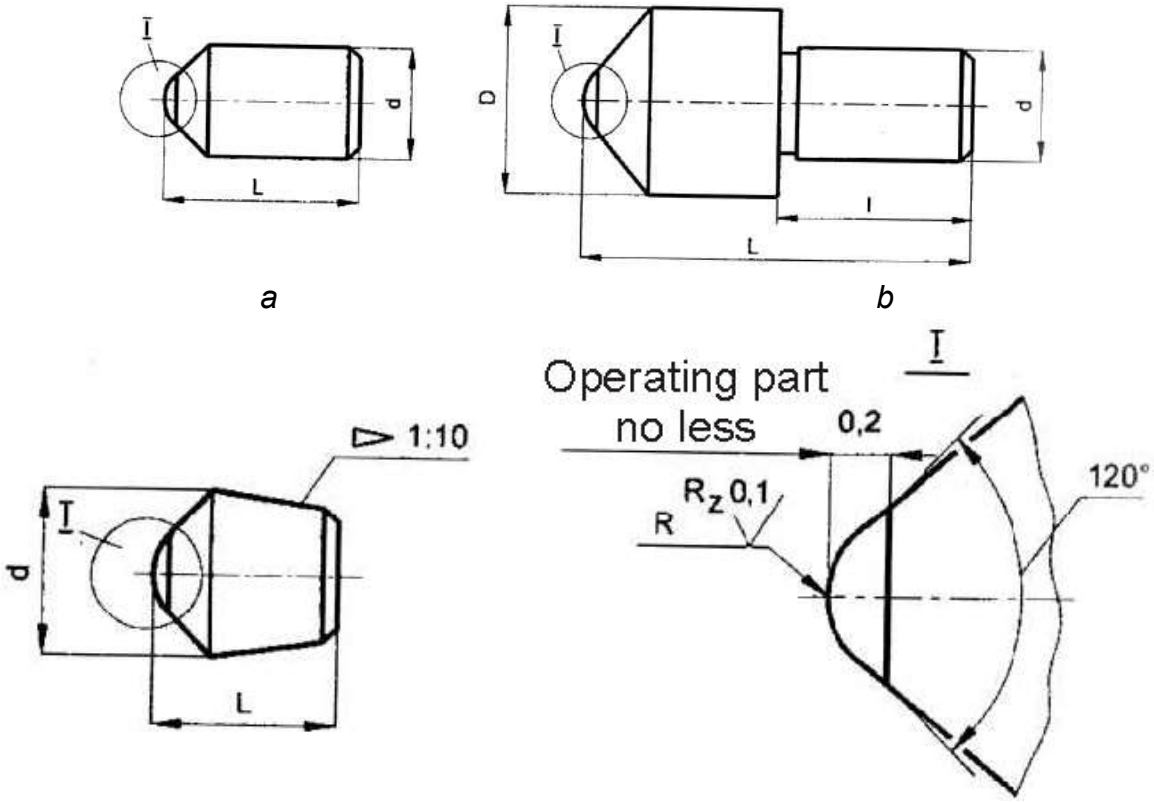
Smoothing Tools

Tools for material smoothing are characterized by a work item form of a working surface, the method of an operating element fastening (chip) in the tool body.

Spherical and cylindrical smoother are serially produced. Smoothers of a spherical shape with radius of 0.5 ... 4 mm can be of two types: without head (Fig. 12.53, a, b) and with head (Fig.12.53, b).

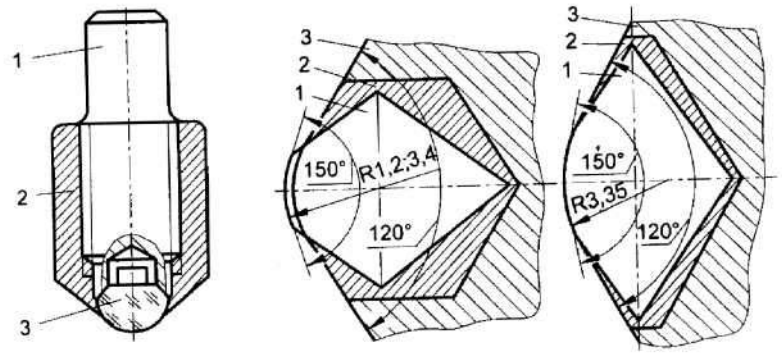
The diamond crystals are attached mechanically to a tool body (Fig. 12.54, a) by soldering (Fig. 12.54, b, c) and filling material with electrical insulating properties, such as plastic. When using adhesives with high mechanical properties a diamond can be fixed by gluing. Polycrystals of synthetic superhard materials generally have a defined size and shape, so they can be fixed mechanically, by means of pressing and folding in the body or compression of the screw plug (Fig. 12.54, a).

The less the roughness parameter of the working surface of the tool is, the higher the quality of the parts is achieved, so it should not be less than R_a = 0.063 μm, which is not always provided by the supplying factory. It is advisable for an tool consumer to save on production polished smoother using a special machine or attachment to the Universal tool grinding machines.



a *b* *c*

Fig. 12.53. Smoothing Instrument With Spherical Working Part [2]



a b c

Fig. 12.54 Examples of Smoother Construction and Ways of Diamonds Installation: *a* – with a screw plug; 1 – plug; 2 – body; 3 – diamond; *b, c – the diamond soldering station of a cylindrical and spherical shapes*; 1 – diamond; 2 – solder; 3 – body [2]

Formation of Stress State of Shafts' Surface Layer

Residual compressive stresses formed in the surface layer in the mechanical processing, have a decisive influence on their carrying capacity. Therefore, the design of processes should be prescribed for a hardening mode that allows obtaining a favorable distribution of the residual stresses over the section details.

When diamond burnishing residual stresses arise due to non-uniform plastic deformation of the surface layers and the presence of heat. Accompanied by shear deformation of interrelated processes reorientation cleavage structure components, which reduces the density of the plastically deformed layer. The increase in the specific volume during plastic deformation can reach 0.3 ... 0.8 %. The aspiration of plastically deformed layer to increase its volume hinder below the elastically deformed layers of metal. Thus, the interaction between plastically and elastically deformed layers creates a compressive residual stress, and balancing their tensile residual stresses in depth.

When diamond burnishing with the average speed, a significant heating of the surface layer occurs, which has almost no effect on the formation of residual stresses. However, increasing the speed of burnishing results in significant by heating the surface layer, may change the character of the diagrams of residual stresses at the surface.

As it can be seen from Fig. 12.55, *a*, the increase of the smoothing strength causes an increase in the depth of compressive residual stresses and offset deep into the surface layer of the highest level. For steel 13H11N2V2MFSH maximum level of compressive stress is formed by burnishing with the power of P_y = 300 N.

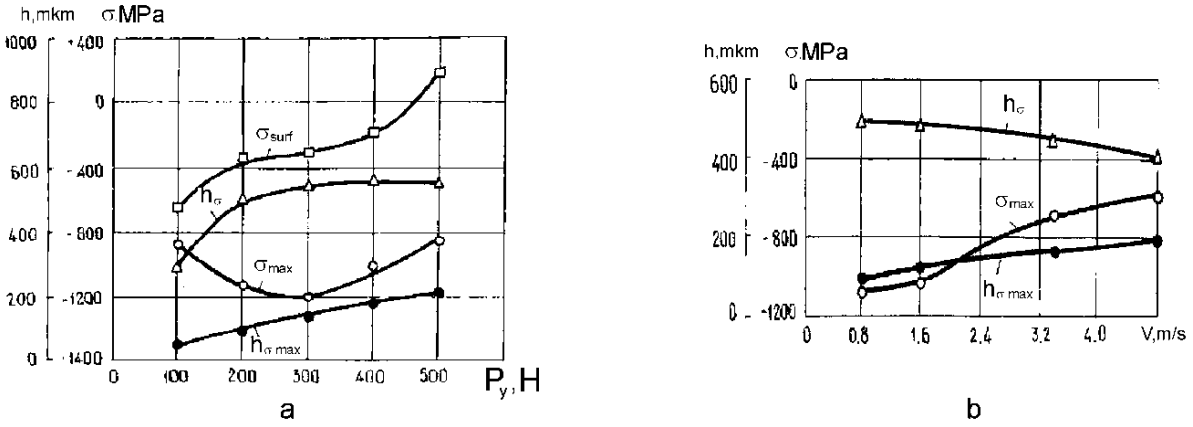


Fig. 12.55. Dependence of depth of tangential residual stresses and their level (V = 1.3 m / s) on the force (*a*) and speed of smoothing (*Py* = 400H) 13H11N2V2MFSH steel shaft (*b*) [2]

With small resources and temperatures not exceeding the recrystallization temperature of the metal, a significant effect is the realization of reinforcement mode, providing greater depth of strain hardening, which contributes to the transfer chamber of origin of the fatigue cracks in the depth of part and a significant manifestation of the residual compressive stress even a low level. In this case the diagram of compressive residual stresses at the surface has a maximum, which has an inhibitory effect on fatigue crack growth, which increases as it moves to the outer surface.

The Effect of Smoothing on Contact Fatigue Resistance of Parts

The **contact fatigue** refers to the process of destruction of the surface layer (chipping) as a result of shear stresses. Moreover, these stresses are concentrated in relatively small areas and may occur under static and dynamic loading, as well as roll-

ing without sliding or rolling with slippage. Contact fatigue in general has the same characteristic properties as any other type of fatigue. For a contact fatigue there is a certain limit of the load – **limit contact fatigue** that loads less than the destruction of the surface which does not occur at an arbitrarily large number of cycles. Contact fatigue phenomena may develop on the surface of a large number of parts that can be hardened by a diamond burnishing. Those should include parts of the type of gear, cage needle bearings, gear couplings, roller cam shafts and others.

Among the technological methods to improve the resistance of the fatigue of the contact parts, a significant place is occupied by a diamond burnishing. The processing of this method improves the quality parameters of the surface layer, its physical and mechanical properties, increases the uniformity of the structure and thereby increases the contact resistance and fatigue parts. Moreover, this increase may be a substantial amount. Strengthening burnishing treadmills angular contact bearings high-torque turbo-drill provided an increase in the durability of two - three times.

When diamond burnishing of steel samples ShKh-15, the surface quality significantly improves: micro-hardness is increased with 9500 to 11500 MPa; R_a roughness parameter is decreased from 0.32 to 0.16...0.040 μm (P_y modes at 150 H, S = 0.038 mm / rev, V = 0.3 m / s); increasing the value (up to 700 MPa) and depth (up to 260 mm) of residual compressive stress, residual stress gradient is reduced; on the surface at a certain depth and the presence of finely dispersed carbides is observed.

All this allowed to increase the samples' durability at 10 % (about 3.5 times) and at 50 % (70 %) probability of failure as compared to samples whose surface is superfinished.

Similar results were obtained with test specimens of 6 mm diameter (σ_{Zmax}= 6000 MPa load) of the steel SH-15 of electroslag remelting (HRC61 ... 65 MPa), treated with polishing and diamond burnishing by two working passages (R_s = 1.2 mm; P_y = 150H; S = 0.04 mm / rev).

For samples treated with polishing, the durability N_{50} = 17.28 • 10^6 cycles (N_{10}= 1.73 • 106 cycles), after diamond burnishing N_{50} = 29.78 • 10^6 cycles and, accordingly, N_{10}= 6.44 • 10^6 cycles.

The diamond burnishing also increases the resistance to the contact fatigue of chrome-plated surfaces. This contact fatigue resistance depends on the film thickness and the processing method. With increase of the coating thickness, the durability of the surface layer increases. Burnishing with chrome plating gives a greater increase in durability compared to chrome plating and subsequent polishing (endurance limit is the same in both cases). Durability samples ironed before and after coating, the durability of the above samples, after chroming ironed.

Smoothing (by modes P_y = 120...180 H, S = 0.05 mm / rev, R_s = 1.0 ... 1.8 mm) the chromium coat 25 and 80 microns thick after polishing prior to R_a = 0.63 microns increasing component life of 1.25 ... 1.5.

Smoothing also removes the uneven surface of the deposits (formed more favorable micro-relief and somewhat reduced porosity of the coating). In some cases, when smoothing the coatings with unevenly spray-coated surface, chipping of small areas had a place. Therefore, before smoothing it is advisable to grind or polish the detonated coats.

Choosing the Smoothing Modes of the Nitrided Parts

The use of the thermochemical processing and diamond burnishing become more widespread in the industry.

The studies have been conducted which were related to the determination of the optimal mode of diamond burnishing and its impact on the quality of the surface of the nitrided parts, which was carried out on the inner surfaces of the steel 38H2MJuA cages, that have nitriding hardness of 28 HRC…35 MPa, after nitriding HRC 60 ... 63 MPa.

The surface of the holders after polishing had a roughness parameter R_a = 0.32 microns, after grinding R_a = 0.63 microns. Diamond burnishing of holders was performed after tool grinding R_s with sphere radius of 2 mm. The inner surface of the holder during operation is in contact with a graphite seal, durability and reliability that is determined by the shape and roughness, and the degree of saturation of the abrasive surface.

As the output parameter of optimization for holder parts was adopted machined surface roughness. From a relatively large number of influencing factors that have been selected as principal: the strength of burnishing P_y, burnishing speed V, feed S. The second experiment was performed in order to obtain the power dependence of the form

$$R_a = \left(0.095 \cdot S^{0.06} \cdot V^{0.12}\right) \Big/ P^{0.01}.$$

Based on the research results we can recommend the following modes of diamond burnishing of nitrided GTE parts, allowing to obtain a parameter of surface roughness in the range R_a = 0.32 ... 0.16 mm: diamond with R_s = 2 mm P_y = 150 ... 200 N; S = 0.05 ... 0.1 mm / rev; V <1.7 m / s.

After grinding and polishing the surface of a nitride, abrasive grain length of 10 ... 20 m, width of 2 ... 3 mm are embedded, which are, apparently, the primary causes of wear of the graphite seal. In addition, the risks existing on the surface are the centers of origin of the fatigue cracks on parts operating under cyclic loading.

After diamond burnishing of grinding nitrided surface as a result of plastic deformation of the roughness embedded abrasive is almost completely removed. Separate seamed abrasive particles are small – 1.5×0.2 microns. The absence of abrasive particles is larger and more favorable microprofile surfaces significantly reduce the wear of graphite seals.

To evaluate the effectiveness of smoothing nitrided parts four groups of fatigue samples of steel 40HN2MA-W were tested: polished (the original); nitrided after polishing; ironed after grinding and subjected to diamond burnishing after nitriding. As it can be seen from Fig. 12.56, smoothing rough after grinding increased the endurance limit to 590 MPa, which is 20 % higher than after polishing. When burnishing nitrided samples endurance the limit has not changed, but life at voltages above σ_{-1} increased in two to four times. Partly hardening efficiency is due to the relatively large depth of the nitrided layer (h = 0.5 mm).

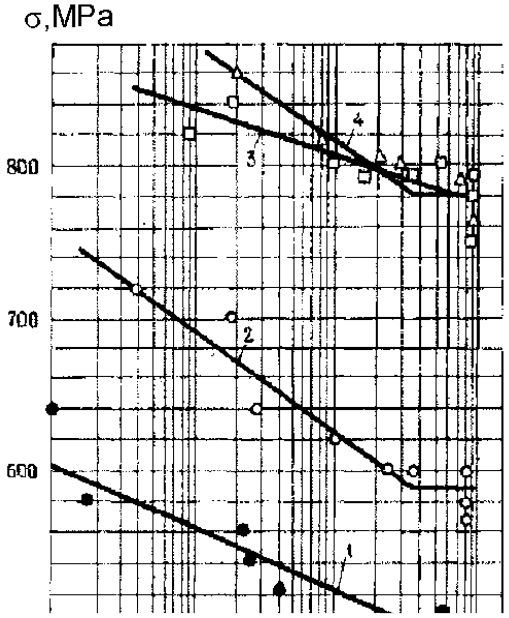


Fig. 12.56 Fatigue curves of steel 40ХН2МА-Ш samples *(P* = 150 Н; L_s = 2 mm; *S* = 0, 05 mm/v): 1 – after grinding ; 2 – after smoothing, following the grinding , 3 – after nitriding, following the polishing ; 4 – after smoothing, following the nitriding [2]

12.2.4. Hydro Shot Blasting of Parts and its Technological Features

The essence of the hydraulic shot blasting hardening is treatment of parts with balls ejected with the cooling lubricants. Fig. 12.57 shows a schematic diagram of the hydraulic shot blasting ejector unit.

Workpiece 1 is mounted on the spindle in a chamber 2 divided by a grid 3 into two cavities A and B. The grid places pellets (balls), in the cavity B – lubricating and cooling fluid is. A nozzle-ejector 4 (from one to several dozen) is placed in the cavity A. A pump 5 supplies a jet of cooling lubricant, which picks up the pellets and sends it to the workpiece, treating a surface of the workpiece. The grid 3 separates the liquid from pellets and it is drained through pipe 6 into the tank 7.

Filter 8, located in the tank, separates small particles of the workpiece material formed by the "peeling" of the surface layer, and worn pellets that have passed through the grid 3. Then, the pump 5 sucks liquid again and the process continues.

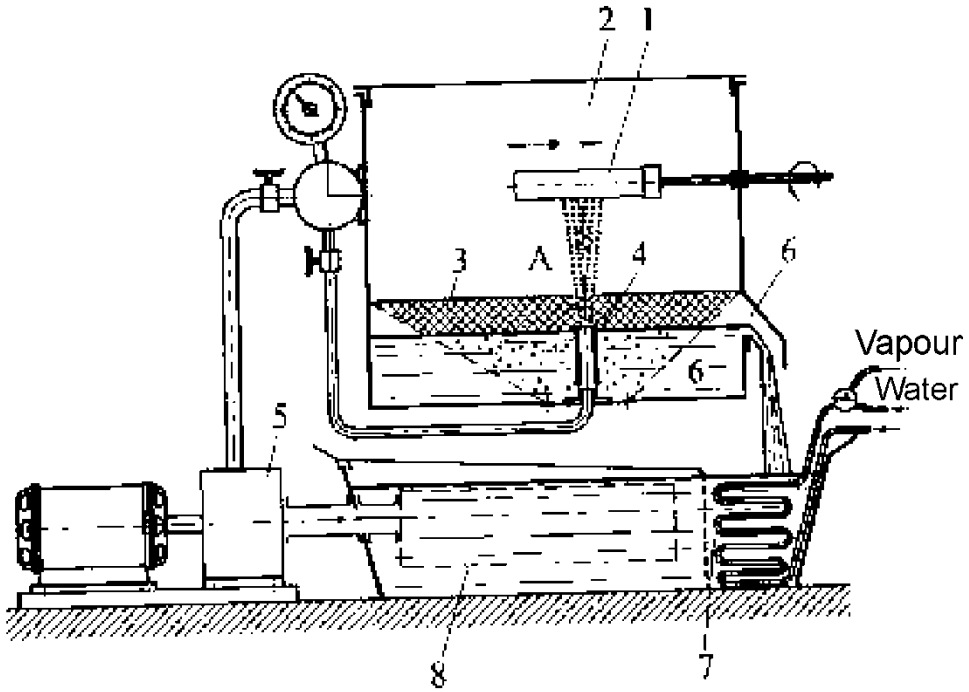


Fig. 12.57. Schematic Diagram of Hydraulic Shot Blasting

Hydraulic shot blasting method of hardening is a new process which has main features.

In the surface layer and on the workpiece surface are formed:

– only compressive residual stresses, and, as a rule, they have little subsurface maximum;

– relatively low surface roughness (R_a = 1.25 ... 0.16 microns) of the surface is retained, high (R_a = 2.5 ... 10 microns) is reduced to R_a = 2.5 ... 1.25 microns;

– the micro-geometry is improved as the radii of curvature of cavities (holes from balls) and protrusions defining a support surface increases;

– the gas saturation process is excluded due to lower temperature in the contact zone and the presence of isolation from the atmosphere of the surface layer of continuous liquid film.

12.2.5. Magnetic and Abrasive Polishing of GTE Parts

One of perspective methods of the finishing and strengthening treatment is the method of magnetic abrasive polishing.

Distinctive features of magnetic and abrasive polishing **(**MAP) are the following: the universality of the abrasive tool allowing to process the parts from various materials and configurations with formation of the surface of parts having favorable characteristics; a combination in one process of both finishing and strengthening operations; ability to mechanize treatment of practically any part.

MAP represents treatment of parts in the environment of the abrasive powder withheld by forces of magnetic field in a working zone. Magnetic and abrasive material settles down between poles of electromagnets, creating the cutting tool ("brushes") which density can be varied over a wide range, changing intensity of a magnetic field in gaps. At the movement of a part through a working zone abrasive powder covers the processed part, putting pressure upon her in each point of a surface. Presence of the

pressure component upon a part from powder on a normal (magnetic origin) and the tangential component caused by movement of a product relative to powder brings to removal of metal and to smoothing of microasperity on a part surface.

The major factor influencing MAP indicators along with technological parameters and material is the initial condition of a surface of the processed product.

So, for example, titanic alloys at high temperatures possess the increased chemical activity to the oxygen, nitrogen, hydrogen which are airborne. When grinding to a hydrogen absorption it leads to emergence of hybrid films and layers which are the reason of emergence of the stretching tension, fragility of a surface and formation of microcracks.

So, MAP mode: B = 0.75 tesla, $V_{r.1\ disc}$ = 2.1 m/s, and sample $V_{r.2\ sample}$ = 0.43 m/s, duration of polishing of t = 4 min, MAN-23M40Fe80 for processing of titanic alloy allows to lower relative physical broadening at VT5 alloy with $\beta = 1.63 \cdot 10^{-3}$, taking place after grinding, to $\beta = 0.79 \cdot 10^{-4}$. Reduction of physical broadening shows that there is an increase in degree of heterogeneity of distribution of microtension (the 2nd sort) and reduction of area size of coherent dispersion. Reduction of the size of blocks increases resistance of plastic deformation of metals owing to reduction of areas through which sliding happens.

Results of measurement of hardness of titanic alloy VT5 after grinding and after MAP on the PMT-3 device are shown in Fig. 12.58. Measurements were taken at the polished end of samples.

Apparently from Fig. 12.58, the microasperity of samples after MAP is more, than polished samples with an identical depth (for example, at micron h =2 μm after MAP Hμ = 3080 MPa, after grinding of Hμ = 2800 MPa).

Cumulative influence of microasperity increase of a surface, hardening at the expense of thin structure, removal of a defective layer, and also change of the direction of microasperity after MAP on wear resistance investigated on cylindrical samples from a titanic alloy VT5 after grinding, turning and MAP.

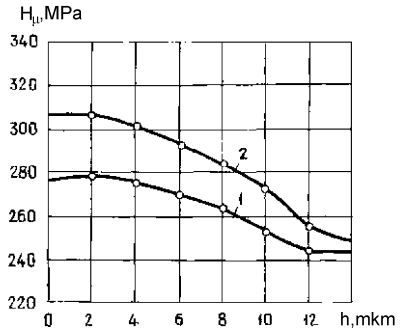


Fig. 12.58. Change of Microasperity of Titanic Alloy VT5:
1 – after grinding; 2 – after MAP [2]

Wear resistance is functionally connected with the hardness, structure, a tension of a blanket of material and roughness of a surface.

The roughness of samples after grinding and turning made R_a=0.7... 0.8 microns, a roughness of samples after micron MAP R_a = 0.6. Wear of polished samples for 32600 cycles is 31.6 mg, fine-molded – 26 mg, samples processed by MAP: 1) 19.2 mg; 2) 16.0 mg (Fig. 12.59).

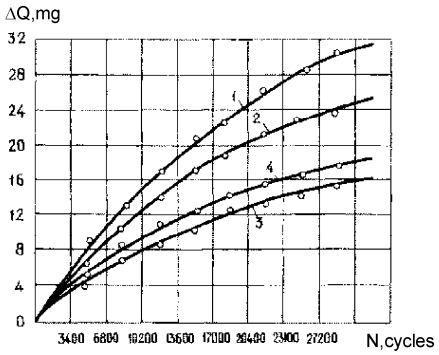


Fig. 12.59. Curves of Wear of Titanic Alloy VT5 Depending on Number of Cycles of Attrition;

1 – grinding; 2 – turning; 3 – MAP at *Ni* – 1.24; 4 – MAP at *Ni* = 17.5

12.2.6. Processing of GTE Parts in the Abrasive Semifluidized Layer (Turboabrasive Treatment)

Physical essence of a method of turboabrasive treatment of parts. The essence of a way of turboabrasive treatment (TAT) consists in creation of the so-called "boiling" layer of abrasive particles in a working chamber of installation 1 by means of a stream of the air given through a support-but-distributive porous lattice 4 under a layer of an abrasive 5. It has very small viscosity and high fluidity. The processed blank 3 is placed in the abrasive semifluidized layer and transmit rotation to it with a high circumference speed (V_p =15...30 m/s) (Fig. 12.60).

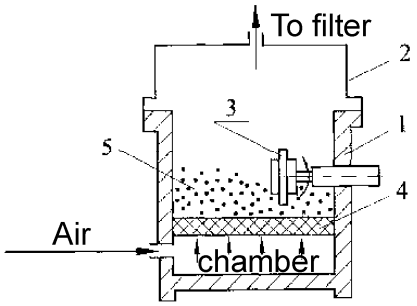


Fig. 12.60. Scheme of Turboabrasive Treatment:
1 – installation case; 2 – cover; 3 – treated part; 4 – air distributor cap;
5 – boiling layer of abrasive grains

The layer of particles possessing such properties can be effectively used as a peculiar abrasive tool at impact of particles to a surface of the rotating part there is intensive removal of metal and formation of a new microrelief of surface.

The high fluidity of the boiling layer provides processing of figurine-shaped surfaces. At the same time all open surfaces of blank are processed that causes achievement of high efficiency.

Sources of intensive removal of metal are the layer pressure upon a surface, the high frequency and density of impact of grains with a surface reaching 300...500 1/μm•mm^2.

Intensity of removal at TAT can reach 2...4 μm / min and more that is much more, than, for example, at the vibroabrasive treatment which is usually 0.05...0.1 μm /min.

Small efforts and short duration of interaction of grains with the processed surface, and absence of friction against a linking of the cutting tool determine low-temperature nature of process of turboabrasive treatment.

The properties of process stated above completely exclude a possibility of mechanical and thermal deformations of the processed parts that allow to mechanize processing of not only figurine-shaped, but also perforated, thin-walled and easily deformable parts (like hollow thin-walled shaft, rings of the guiding devices with cutting-out for blades, thin-walled GTE disks, etc.).

The factors defining efficiency of turboabrasive treatment. Efficiency of process (treatment productivity, roughness of surfaces, uniformity of treatment, etc.) is defined by the following input parameters:

– scheme of arrangement of the treated part in relation to the plane of air distribution grill (kinematic scheme of treatment);

– speed of movement of the treated V_p part;

– time of treatment of τ;

– pressure P_a (flow rate Q_a) of the air supplied to installation for liquefaction of abrasive layer;

– granularity and type (grade) of abrasive grain;

– physicomechanical properties of the treated material (hardness, durability);

– character of the initial surface which is subject to treatment (height of microroughnesses and its direction, a way of preliminary processing – turning, milling, molding, stamping, etc.).

Kinematic Scheme of Treatment

Considerable influence on intensity of removal of metal and uniformity of treatment renders the choice of the optimum kinematic scheme of treatment in dependence on a form and other geometrical parameters of parts (Fig. 12.61).

The scheme of treatment with horizontal axis of rotation (Fig. 12.61, *a*) in comparison with the scheme of treatment with vertical axis has advantages (Fig. 12.61, *b*). Treatment productivity according to the first scheme, other things being equal conditions, in three-four times higher than at the second. The first scheme is applied when the parts are treated as rotation bodies (cylinders, rings, disks, etc.). The scheme with oscillating motion of a part (Fig. 12.61, *c*) in three-five times less effective on productivity than the scheme in Fig. 12.61, *b,* but it can be used when treating the parts which don't have the forms of bodies of rotation (for example, when treating blades of GTE, etc.). Substantial increase of productivity and, the main thing, uniformity of treating blades of GTE, cogwheels and other parts of mainly small overall dimensions is provided by the scheme of their movement with the planetary movement.

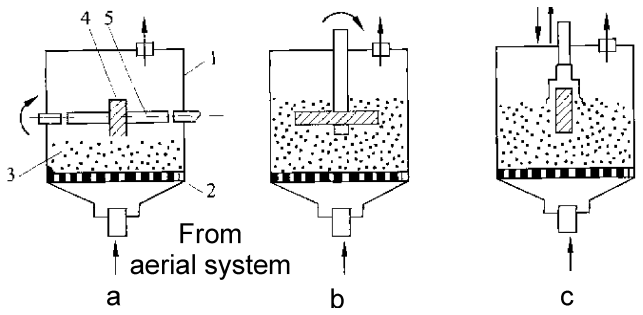


Fig. 12.61. Schemes of Experimental Installations For TurboAbrasive Treatment:
a – with horizontal axis of rotation: 1 – the device of the boiling layer;
2 – air distributor cap; 3 – abrasive layer; 4 – treated part; 5 – mandrel; *b* – with vertical axis of rotation; *c* – with oscillatory rotation of a part

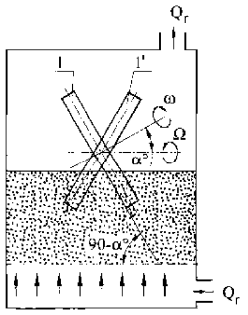


Fig. 12.62. Scheme of Processing in the Boiling Abrasive Layer with Variable Angle of Installation Of Axis of Rotation to the Plane of Air Distributor Cap

The movement of parts relative to own axes can be swinging or be carried out with variable frequency of rotation.

Small roughness of the treated surfaces with its high reference value, reduction of time of obtaining the roughness and more uniform processing of elements of surfaces is provided by the scheme at which parts transmit rotation at the same time around of two axes which are crossed at an angle of the axes, one of which is parallel to the plane of an air distributor cap (Fig. 12.62).

Quality of Surface while Turboabrasive Treatment

Quality of a surface characterizes operational properties of parts of machines, devices and instruments. In spite of the fact that quality of a surface is formed at all stages of processing, the defining impact is exerted by finishing operations.

In the process of turboabrasive treatment way in the "boiling" abrasive layer is used when carrying out finishing operations of parts including critical parts of aircraft engines, studying and generalization of results of research of regularities of microrelief formation of the processed surface and condition of metal surface coating is very important.

Surface Roughness

Speed of blanking and granularity of the applied abrasive have the most significant effect on formation of microrelief.

The initial surface roughness of preparations exerts impact only for the period of getting a stationary final roughness, but not on its size.

In the process of turboabrasive treatment (polishing) of samples from not tempered steel 45 achievement of the following as-turned finish is provided:

– by grain 63C63 – R_a = 0.6 ... 1.0 μm;

– by grain 63C40 – R_a = 0.4 ... 0.6 μm;

– by grain 63C16 – R_a = 0.2 ... 0.4 μm.

When processing the tempered steel the values of R_a decrease on 30...50 %.

Time of receiving a stationary final roughness of previously polished surface (R_a = 0.8...1.0 microns) is 3...5 minutes, and previously fine-molded (R_a = 2...5 microns) is 8...12 min. Increase of speed of part rotation from 10 to 20 m/s reduces the reached roughness from R_a = 0.3 ... 0.4 μm to R_a = 0.18 ... 0.20 μm (when processing crude steel 45 by grain 63C16 within 15 min).

Researches and know-how of the enterprises show that stable receiving a roughness of R_a = 0.2...0.4 μm (when treated by grain No. 16) can be reached also on such materials as stainless steels, titanic alloys, brass, heat resisting alloys, etc.

In the process of turboabrasive treatment of disks of compressors and turbines (materials – ВТ3-1 and *ЭИ*698-ВД) by abrasive grain of electrocorundum group 14A No. 63...80 during 4...10 min roughness of surfaces of disk cloth is R_a = 0.6...0.4 mkm, while after the metalwork operations R_a =1.25...0.8 microns.

Condition of Metal Surface Coating

The condition of surface coating of machined parts depends on the following main characteristics: residual tension of the sort I, microasperity, parameters of crystal structure of a metal-environment interface, limit of endurance, extent of saturation by an abrasive (charging)

After turbo-abrasive treatment of the annealed samples (material – steel 45) in all cases the residual tensions of compression (Fig. 12.63) which depth of distribution is 15 to 40 microns, and the maximum size – 300 to 700 MPa.

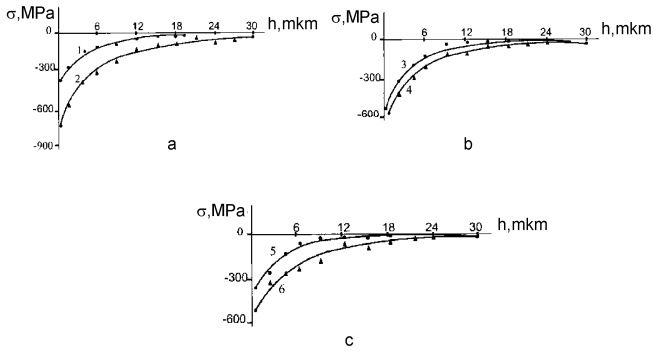


Fig. 12.63. Residual Tension of Compression σ after Turboabrasive Treatment *(a – c):*
1 – V_c = 10 m/s; 2 – V_c = 20 m/s; 3 – τ_m = 2 min; 4 – τ_m = 10 min;
5 – d_3 = 80 μm; 6 – d_3, = 630 μm [2]

The most significant effect on formation of intense layer perform part speed (Fig. 12.63, a) and abrasive grain sizes (Fig. 12.63, c): with increase in speed from 10 to 20 m/s the maximum tension of G_{max} increases twice and the depth of intense layer to three times, and with increase in abrasive grain sizes from 80 to 630 μm of σ_{max} max increases by 1.7 – 1.8 times and the depth of intense layer – by 2.0–2.5 times. Processing duration (over 2... 4 min.) does not exert noticeable impact on formation of residual tension (Fig. 12.63, *c*).

Microasperity of the annealed samples after turboabrasive treatment increases on 20...50 % (1000 to 1200 till 1200 to 1800 MPa). Extent of influence of technology factors on microasperity same, as well as on residual tension.

Similar researches of the samples previously processed by grinding, and then turboabrasive method show that if grinding creates the residual stretching of tension in a layer about 10 to15 μm in depth to σ_{max} = 500... 600 MPa, after turboabrasive treatment a zone of stretching tension considerably decreases and is about 3 to 5 μm, and in un-

derlying layers (h < 20...10 μm) the compressive residual tension σ_{res} = -250 to -300 MPa are observed.

Technological Capabilities of the Turboabrasive Treatment Method

The analysis of results of works and researches stated in the previous sections, and the experience of industrial application of turboabrasive treatment accumulated by the enterprises show that this method allows to solve the following technological problems effectively:

– finishing processing (polishing) of surfaces of formed parts, easily deformable, thin-walled parts with reduction of initial roughness with Ra = 2.5... 5.0 μm to Ra = 0.2... 0.4 μm;
– removal of agnails after various operations of processing by cutting;
– rounding off of sharp edges;
– finishing strengthening processing of elements of surfaces, especially formed parts of GTE;
– preparation of surfaces under coating of various types (galvanic, plasma, etc.);
– removal of scale after heat treatment, traces of corrosion, deposit, oxide films, etc.

The specified technological problems are solved in relation to processing of various materials: constructional and stainless steel, color, titanic and heat resisting alloys.

Comparative characteristics of various methods of surface plastic treatment of parts are presented in Table 12.5.

Table 12.5

Characteristics of Methods of Superficial and Plastic Processing of Parts

Method of SPP	Accuracy of processed surface	Ra (Rz), μm	Increasing of microhadness, %	Compressive residual stresses, MPa	Depth of hardened layer, mm
Static methods					
Whetting hardening	6...8	2.5...0.5	20...30	500...800	0.5
Surface rolling	5...7	0.5...0.1	20...50	600... 800	5.0
Diamond burnishing	5... 7	0.2...0.4	30...60	300...700	0.2
Dynamic methods					
Vibration treatment	5...7	0.2...0.02	10...15	100...200	0.2
Centrifugal treatment	5...7	0.3...0.05	15...60	400...800	0.7
Stamping	11 ... 13	(100)	15...60	400...800	3.5
Hydraulic polishing	5...7	0.6...0.05	20...30	300...700	0.2
Shot blasting	10... 13	(100)	20...40	400... 800	1.0

12.2.7. Application of Protective Coating

Role of coating in modern aircraft equipment is more and more important, because requirement for increasing of product resource, ecological safety, takes place increasing of operational temperatures, forces etc. Solving of numerous problems become possible while using special systems of protective coating (Fig. 12.64).

Type of covering, its thickness and method of covering depend on functional using operational conditions etc. (see Tables 12.5, 12.6, 12.7).

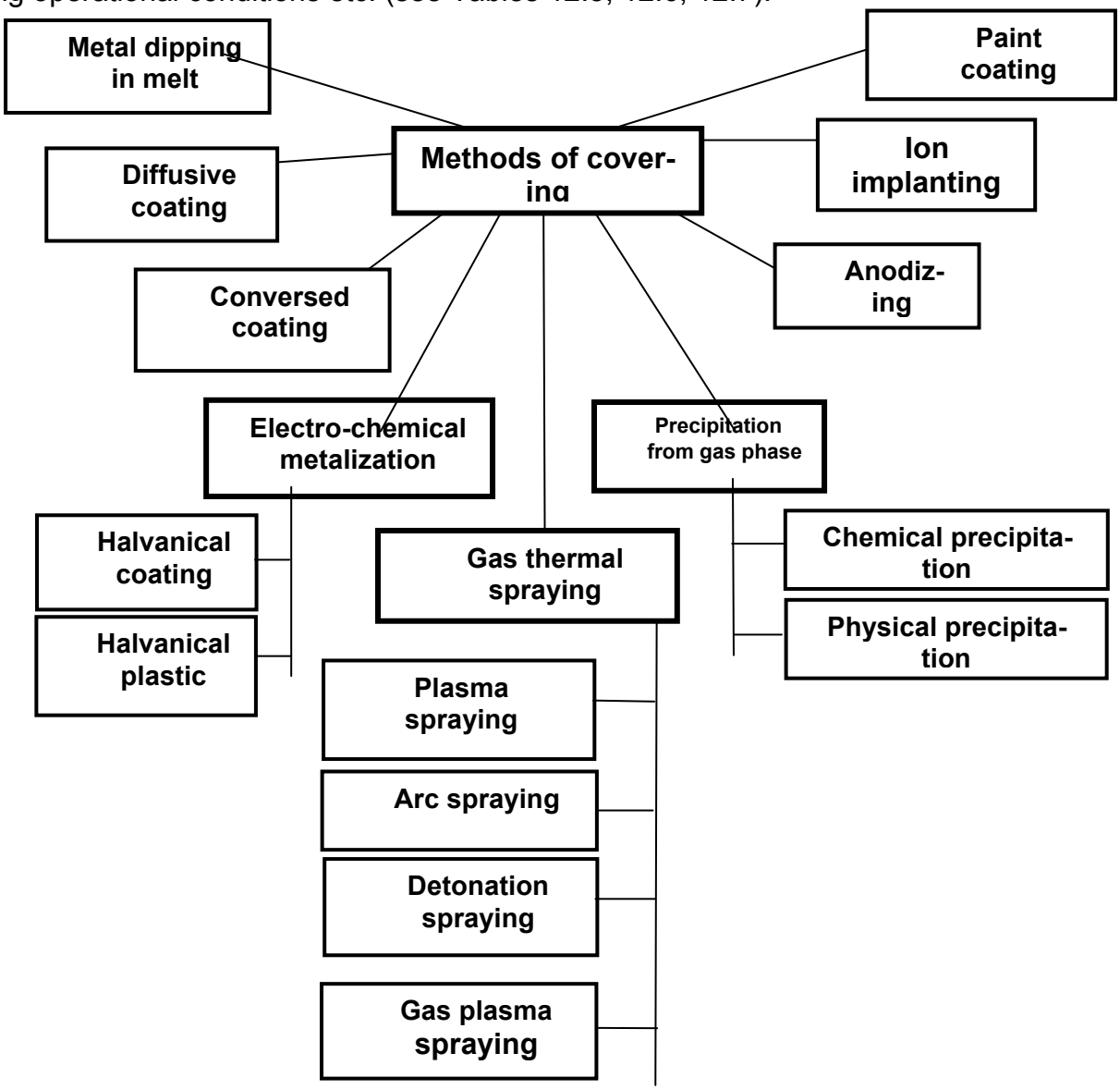


Fig. 12.64. Schematic Diagram of Covering Methods

Factors that are considered while choosing covering:

- Cost and resource;
- Corrosion activity of operational environment, influence of spraying process on substrate material;
- Surface profile;

- Operational temperature, lubrication conditions and availability of abrasive on operational environment, loading and speed, availability of force impact etc.;
- Porosity, heat-proofing, electro-proofing etc.

Table 12.6

Main Functions of Coating [10]

Function	Coating
Decreasing of abrasive disorientation	Titanium carbide, titanium nitrite
Friction decreasing	Molybdenum disulfide
Increasing of friction	Titanium, connected abrasives
Lubrication improvement	Copper, lead
Increasing of operational temperatures or forces	Chemically plated nickel
Adhesion decreasing	Silver, gold plating
Corrosion disorientation decreasing	Plating with chrome
Drip erosion protection	Polyurethane, neoprene
Piece erosion protection	Cobalt alloy, molybdenum

Table 12.7

Surface Processing of Different Materials [10]

Metal	Treatment type
Aluminum	Chrome plating, phosphating, chromate conversion covering
Beryllium	Anoding, chromate conversion covering
Cadmium	Phosphating, chromate conversion covering
Stamping steels	Borating, ion nitrating, nitrating
High temperature steels	Diffusive coating
Magnium	Anoding, chromate conversion covering
Less carbon steel	Borating, phosphating, carbonization, liquid nitrating, nitro cementing, cyanidation
Molybdenum	Chrome plating
Alloys based on nickel and cobalt	Borating, diffusion coating
Refractory metals	Borating
Stainless steel	Precipitation from gas phase, ion nitrating, diffusive covering, liquid nitrating, nitrating.

Table 12.7. (to be cont'nd)

Metal	Treatment type
Steel	Precipitation from gas phase, chrome plaking, phosphating, ion nitrating, high frequency hardening, fire hardening
Titanium	Chrome plaking, anoding, ion nitrating
Instrumental steels	Borating, ion nitrating, diffusive covering, nitrating
Zink	Precipitation from gas phase, anoding, phosphating, conversion coating

12.2.7.1. Plasma Spraying

While plasma spraying, high temperature of ionized inert gas is used. Most often used plasma forming gas is argon (Fig. 12.65).

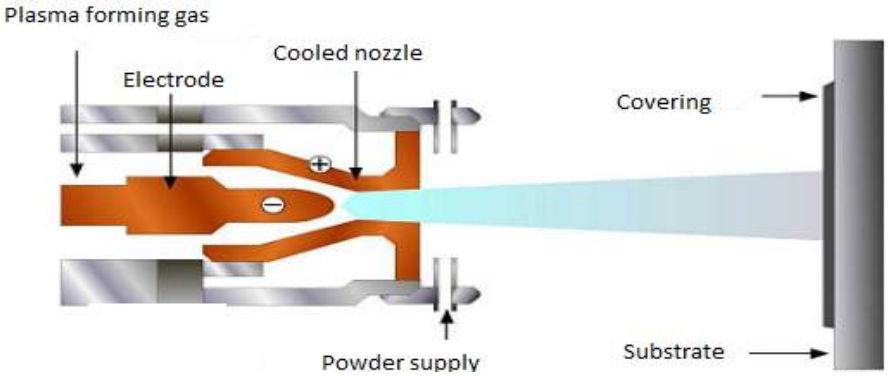


Fig. 12.65 Plasma Spraying Scheme
(Source: Plasma and Thermal Coatings Ltd.)

Atmospheric plasma spraying – APS

In this technology plasma with atmospheric pressure for melting and powder transportation is used. Temperature while APS is higher than 10000°C, that allows spraying a wide range of materials (Fig. 12.66, 12.67).

Using:

- Hard wearing coating;
- Corrosion resistant coating;
- Thermal barrier coating (TBC);
- Product recovery etc.

Materials:

- Ceramics (for example Al_2O_3, TiO_2, Cr_2O_3, ZrO_2);
- Metals (for example, cobalt, nickel, molybdenum), alloys (for example based on Fe).

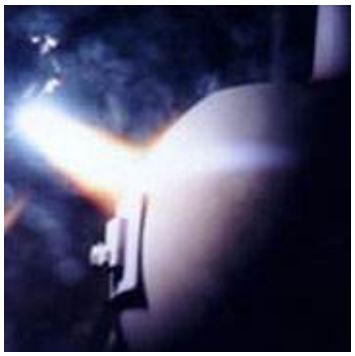


Fig. 12.66. Atmospheric Process Plasma Spraying *(Source: Advanced Powders and Coatings Inc)*

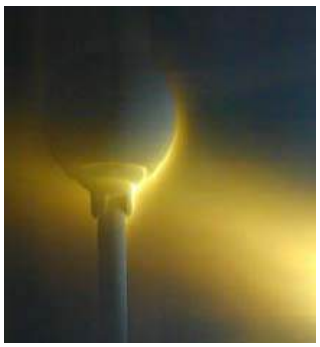


Fig. 12.67. Vacuum Process of Plasma Spraying *(Source: APC Advanced Powders and Coatings Inc)*

Vacuum Plasma Spraying – VPS

In this technology plasma with low pressure is used (nearly 0.1 atm.) (see Table 12.9).

Table 12.9

Plasma Coating (*Source: Plasma Group Science*)

Covering	Characteristics	Using
Wolfram carbide/ cobalt (83/17)	Macro hardness – Rc 50-55 Micro hardness – DPH300 750-950 Adhesion strength – 62MPs (9000psi)	Used in low temperature environments to 500°C on parts, exposed to fretting or abrasive wearing
Chrome oxide	Macro hardness – Rc 70-72 Micro hardness – DPH300 900-1300 Adhesion strength – 41MPs (6000psi)	Used on areas with intensive abrasive wearing; in pump bolts; impellers, zones of friction of teeth operating wheels, shafts etc.
Yttrium, stabilized by zirconium	Macro hardness – Rc 30-35	Used as thermal barrier covering, where is important resistance to high temperatures, corrosion and heating impact. Components, that are covered by this material include gas pieces, turbines, exhaust port and other units that operate in temperature 850°C
Nickel/ Chrome (80/20) + 6 % Aluminum	Macro hardness – RB 90 Adhesion strength – 28MPs (4000psi)	High temperature coating for minimizing oxidation and corrosion of carbonic steels to 980°C.
Cobalt / Chrome/ Nickel/ Wolfram	Macro hardness – Rc 59	Can be used in temperatures to 540…840°C, this material has high resistance to erosion and fretting-wearing
Aluminium / Polyester	Macro hardness – Rc 60…80	Used as wearing resistant covering in temperature 345°C

Temperature that is reached during VPS is higher than 10000°C, that allows operating spraying of any material. This process is done in controlled inert environment that allows getting high quality coating with low porosity, lower oxidizing and excellent adhesion (see fig. 12.67).

Using:

- Wear resistant coating;
- Corrosion resistant coating;
- Thermal barrier coating (VPS);
- Product recovery etc.

Materials:

- Ceramics (for example Al_2O_3, TiO_2, Cr_2O_3, ZrO_2);
- Metals (for example Ta, Ti, nickel, copper, W), alloys (for example, Ti based, Fe-based).

12.2.7.2. Detonation Spraying

Detonation spraying of coating belongs to number of methods that use diacritical sources if energy and is characterized by spraying cycles frequency. Traditional sequence of operating cycle elements is in next. Ignition chamber, usually looks like pipe, closed in one side, is filled by operating gas mixture that can detonate (Figs 12.68, 12.69). At the same time or in some delay sprayed powder is entered. Then a bung and gas mixing place are created of phlegmatizing gas for preventing back hitting between point of initiating of ignition and gas mixing place. Detonating mixture in which in weighed condition sprayed powder is, is fired by electric spark using fire fount. Firing process is finished by burning that is spread with increasing speed. After passing certain distance, burning turns to detonation. During interaction with detonation wave and detonation products, pieces of sprayed material are heated and head to product surface with high speed (see Fig. 12.70). Interaction of sprayed pieces current with substrate causes formation of unit spot of spraying. Operating cycle structure of detonation installing is shown in Fig. 12.71 [12].

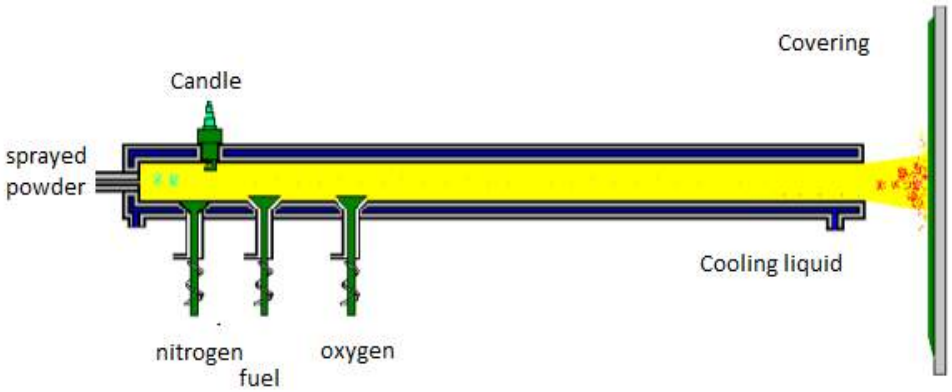


Fig. 12.68. Scheme of Installing FOR Detonation Spraying Coating



Fig. 12.69 Appearance of Facility for Spraying

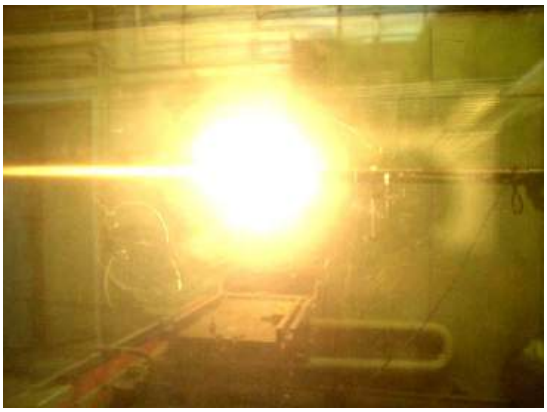


Fig. 12.70. Process of Detonation Coating

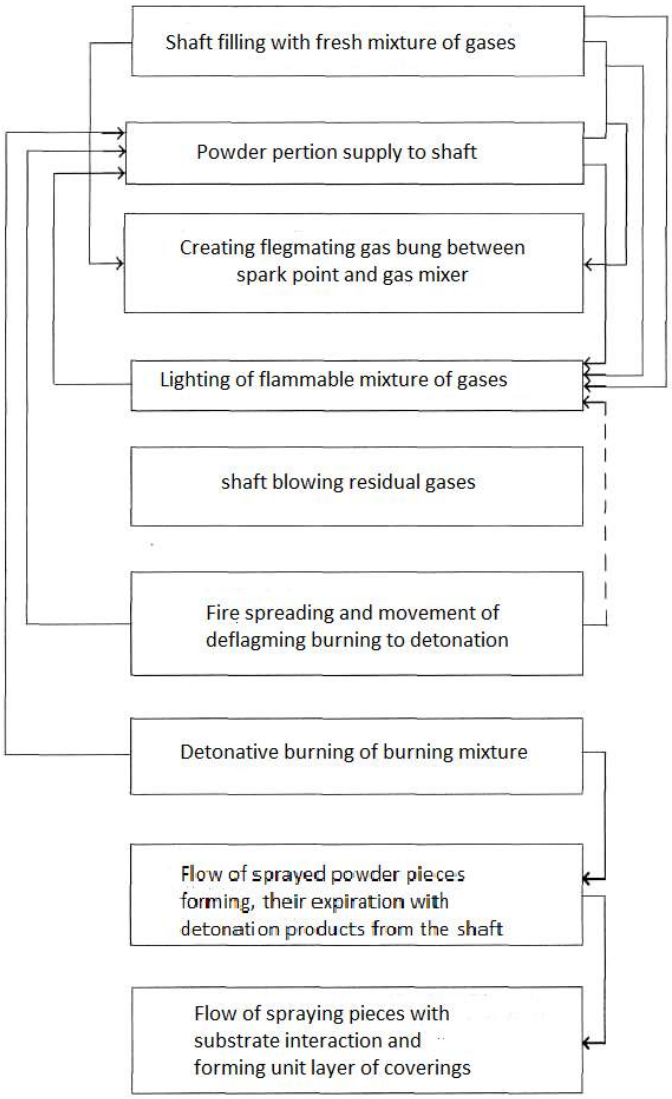


Fig. 12.71. Operating Cycle Scheme of Detonation Spraying Covering

Detonation coating is successfully used in engineering, aircraft industry. Comparing with shown higher this method has advantages [11, 12]:

- Strength of adhesion of the coating with product is from 180 to 360 MPs that is higher than of coating that are applied by other alloys while using them for hardening and recovering of parts, operating in extreme conditions under impact of high contact and striking loadings.
- Thickness of applied layer isn't limited and is within 5 … 5000 μm;
- Part temperature while spraying depends on its sizes and doesn't exceed 530 K that complies low tempering. In view of short detonation spraying process temperature effect is insignificant.
- Insignificant thermal effect on material to sprayed because of short cycle duration gives an opportunity to form physical and mechanical properties of coating at its surface and thickness.
- Ability of smooth and stable regulation of parameters of the process in wide ranges allows for every material to set most optimally spraying mode.
- Using of fine composites for spraying gives an opportunity to form surfaces with roughness that in some cases doesn't need additional machining operation.
- High energy of detonation products that speeds up pieces, increases their temperature, allows to form coatings of refractory materials not only on metal parts with hardness of surface 60HRC and higher but of non-metal materials (glass, ceramic, wool, carton etc.).

Selection of Spraying Materials

For selection of materials for coating two factors are followed – service appointment of coating and its properties.

By service appointment detonation coating can be divided into next main groups:

1) For saving from wearing;
2) For saving from destroying on high temperatures;
3) For protecting from corrosion;
4) With special electro-physic and optic properties;
5) For creating self-contained gaps;
6) For recovering sizes of worn parts;

By grain composition the powders can be divided into next groups:

– Up to 5 μm – very small;
– From 5 to 15 μm – small;
– From 15 to 30 μm – medium;
– From 30 to 60 μm – large;
– From 15 to 60 μm – medium large;
– From 5 to 60 μm – little big.

For detonation spraying most suitable are fractions by grain metrical composition up to 60 μm.

While endurance failure of surface layers, fretting-corrosion, gas abrasive wearing of aircraft engines operating during temperature up to 840°C, unless wide used is hard wearing resistant coating based on carbide wolfram with cobalt and nickel connections – WC +15% Co, WC + 18% Co, WC + 20% Ni and other. This coating has high strength of adhesion, resistance to shock loadings.

In range of high temperatures up to 1300 K the parts of aircraft engines are protected by coating, based on chrome carbide – Cr_3C_2 + 15% Ni and O_2C_3 + 15% N/O.

Most heat resistant is oxide coating – ZrO_2 + Al_2O_3, O_2O_3, T/O. Oxide coating is also used for protecting from corrosion.

For decreasing of wasting and overflowing of gas in rotor seals through radial gaps run in extra coating Ni + 25% of graphite; 69% Ni + 14% O + 8% Fe + 3.5% Al + 5.5% of powder are used. For repairing and recovering parts coating based on Ni is used.

Detonation and plasma coatings are wide used both in foreign and in domestic constructions. Thus, in engines of airplanes “Boeing” DSV-50 600 parts with plasma coatings sprayed by development of company “Metco” are used. Snekma uses plasma coating for protecting 90 units in engines of airplanes “Concord”, “Airbus”, “Mirage” and others [11].

For such units as rail flaps, temperature joints of aircraft engine, seal of air compressor, segments of ventilating duct and stiffeners, middle supports of ventilating blades, such systems as WC + 12% Co, WC + 17 % Co, mechanical mixture WC + 12 % Co with self-fluxing nickel alloy and composite powder, Ni-Al (80% Ni + 20% Al) [11] are used.

Distinctive properties of these systems are high resistance to wearing by abrasive pieces and in conditions of fretting-corrosion, high strength of adhesion while moderate temperatures (up to 540°C).

Powder of wolfram carbide with Ni-Al composite powder 95 % Ni – 55 % Al and nickel chrome boric silicon alloy and also alloy of wolfram carbide with 12% of cobalt is used for spraying on compressor blades, compressor sleeves, teeth mechanism jack, shaft pin, etc.

12.2.7.3. Technology of Integrated Vacuum Ion-Plasma Surface Modification

Purpose

The technology is designed to harden the surface of structural steel and can be used for treatment of friction pairs of interacting surfaces of machines and mechanisms operating in conditions of high dynamic loads, for example to increase the contact strength of screw wheels and for wear resistance of shafts, plugs or inner cylinders.

Brief Description and Technical-and-Economic Indexes

The essence of the technology is in the fact that the three-layer wear-resistant composition is formed in a single work cycle with the original plasma generator and the evaporator on a part''s surface made of structural steel. The three-layer wear-resistant composition consists of two-phase zone of inner nitriding and the finely divided nitrides covering zone or carbonitrides of transition metals like TiN.

The presence of two-phase nitriding sublayer provides high adhesion to the substrate TiN with bedding, dimensional stability of the bedding in contact with the counterbody during friction and it also provides heat resistance. In consequence of the complex ion-plasma treatment of the parts' surface made of structural steel, they acquire an increased durability and corrosion resistance.

The effectiveness of the proposed technology can be illustrated by the example of structural steel 40X (C – 0,40 %; Cr – 1,0 %; Mn – 0,5-0,8 %; Si – 0,17-0,37 %) (see Fig. 12.72).

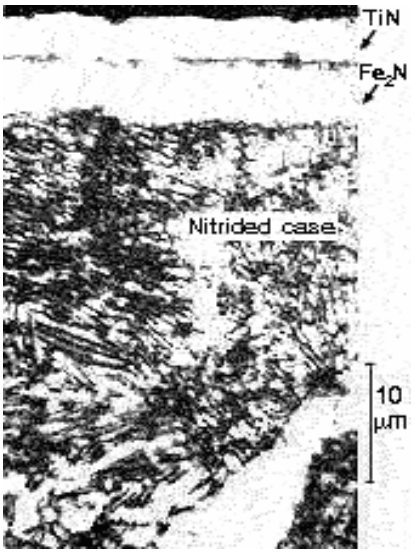


Fig. 12.72. Structure of Modified Steel Layers 40x after Vacuum Ion-Plasma Treatment [9]

The graphs (Fig. 12.73, 12.74) show the measurement results of the micro-hardness of steel sample 40X subjected to ion-plasma treatment by the developed technology and the measurement results of the steel samples 40X wear resistance, subjected to various types of treatment. Wearing tests were performed on a laboratory friction machine IP-3 without lubrication according to scheme "rotor-flat specimen" at a contact load of 200 N and a sliding speed of 0.36 m/s. The criterion of deterioration was a weight loss of the sample. Tests show that the modification of the steel samples sur-

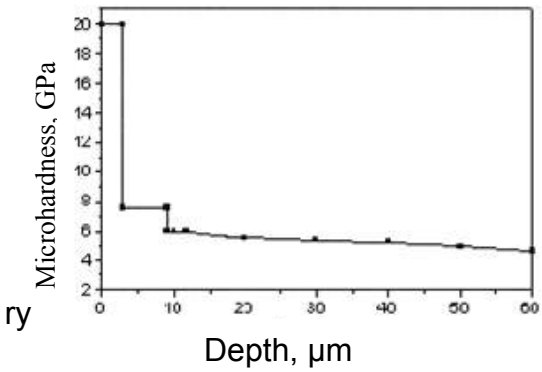


Fig. 12.73. Profile of Sample Micro-Hardness for Steel 40X Subjected to Ion Nitriding in Semi-Self Arc Discharge and Plasma-Assisted Deposition of TiN Coating

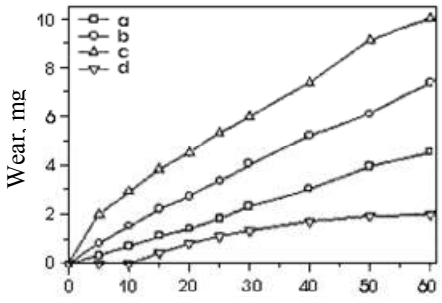


Fig. 12.74. Wear of the Surface Layers of Steel 40x Dependence on the Time of Test in Conditions Of Dry Friction in Various Ways After Treatment: a – quenching, b – on nitriding, c – deposition of TiN coating, d – complex treatment by developed technology.

face by the developed technology allows to increase their microasperity of three - four times, and durability – of four - five times [9].

12.3. MANUFACTURING TECHNOLOGY OF TOOTH WHEELS

The gears machining operation is divided into three phases – draft, finishing and trimming.

Draft stage consists of rough operations, during which 60 – 70% of the total allowance for processing is removed.

At the finishing stage 30% of allowance is removed. The final shape of the detail is made. At this stage the machines of high accuracy are used. If teeth are not obtained in the procuring phase, they are cut at the finishing stage.

The trimming stage includes operations that provide the required accuracy and cleanliness of the gear surfaces. Mounting external and internal cylindrical surfaces and edges are used as settling surfaces. Splines, threads and keyways are treated after heat treatment in order to prevent damage during transportation.

The place of the heat treatment in the technical process is determined by its type and dimensions of the tooth wheel.

Cementation and hardening is carried out at the end of the finishing stage. The hardening is carried out in a stamp on hardening presses in order to reduce twisted effect of disk tooth wheel.

Sample Treatment Plan of Casehardened Gear

1. Work piece heat treatment (normalization).
2. Draft stage: facing, boring holes, turning of the outer surfaces.
3. Finishing stage: facing cut, holes boring, precise boring or grinding of holes from one setting block, turning or grinding of the other end, turning of the external surfaces, turning of the grooves and notches, teeth cutting and their cleaning.
4. Copper plating of surfaces that can not be casehardened.
5. Cementation, hardening, tempering.
6. Removal of copper.
7. The trimming stage: grinding the surface of the shank and the holes, grinding or turning in surfaces for the splines and the threads, threading and splines, grinding teeth, locksmith cleaning teeth.
8. Final inspection.

Sample Treatment Plan of Nitrided Gear

1. Work piece heat treatment (normalization).
2. Draft stage: facing, boring holes, turning of the outer surfaces.
3. Heat treatment (quenching + tempering).
4. Finishing phase: facing cut, holes boring, precise boring or grinding of holes from one setting block, turning or grinding of the other end, turning of the external surfaces, turning of the grooves and notches, teeth cutting and their cleaning, cutting of splines.
5. Heat treatment (stabilization abatement).
6. The pre-nitriding (basic grinding of cylindrical surfaces and the reference face), pre-grinding of teeth with remaining allowance for final processing.
7. Tinning surfaces not subject to nitriding.

8. Nitriding.
9. Galvanic tin removal.
10. The final stage: the grinding of holes and surface of the shank, the final grinding of the tooth surface, turning surfaces for thread and splines, cutting splines and thread, metalwork sweep.
11. Final inspection.

12.4. SHAFT PRODUCTION PROCESS

The machining operation of shafts is usually divided into three stages: draft (roughing), finishing and final processing.

Due to rough machining the equal distribution of allowances for post-processing is achieved and surface defects are removed. It is carried out under strenuous conditions, which are limited by permissible shaft deformation under the influence of the cutting and fixing forces. If the scale production is large, it is often used powerful and tough machine performance.

When finishing significantly smaller allowances than for roughing are relieved. Modes are administered less stressful to prevent deformation of the whole shaft, as well as its surface layers. During the mass production at this stage the productive multiple-tool lathes are used and they are often equipped with copiers. The main purpose of this stage is to obtain a shaft with small allowances for final processing. Here, the error must be corrected after the first stage of machining and warpage after heat treatment.

In the final processing it is required to obtain the specified accuracy and cleanliness of surface, so there are unacceptable conditions that can lead to significant errors due to deformation. The largest share in this step is the processing in grinding machines; in addition, the use of various methods of trimming (honing, superfinishing, smoothing and polishing). At the same stage, they usually process thread, small slots and other surfaces that might be damaged if they were treated before.

In constructing the shaft machining process there is usually the sequence where the last stages of the internal surfaces were treated before the exterior ones. This is due to the concentricity of the inner and outer surfaces which is easier to achieve if to treat initially the internal surfaces, and then using them as bases for mounting the external processing.

Much attention in the design process is given to the choice of the heat treatment space. Thermally treated shafts, for example, can be subjected to quenching and tempering both before machining and after its first stage. In the first case, obviously no need to carry out mechanical shafts in the thermal plant and back and thereby reducing the duration of the processing cycle. However, this is not always possible: with large machining allowances there is a danger of unpuncturable shafts. First, such shaft are skinned and then subjected to heat treatment.

For shafts which have separate portions cemented, the place of the heat treatment in technological process depends mainly on the way of uncemented surfaces protection.

The following engine security methods are used:

– copper plating of uncemented surfaces before the cementation;

– increase in the allowance of uncemented surfaces, which is removed after cementation, but before quenching.

The combination of the first and second ways is in copper plating of surfaces with an increased allowance, which is removed after hardening of the details (double protec-

tion).

12.4.1. Exemplary Shafts Treatment Plans

Thermally treated shafts:

1. Heat treatment (normalization).
2. Trim of the ends and center-drilling.
3. Turning the external surfaces of a backrest (if long shaft).
4. Draft stage:
   a) drilling axial holes;
   b) bore of a hole, grooving pivot chamfers;
   c) cutting of the test specimen (for the first control group of shafts);
   d) turning of the outer surfaces.
5. Heat treatment (quenching and tempering).
6. Restoration of bases.
7. Grinding necks under the backrest (if long shaft).
8. Finishing stage:
   a) boring holes;
   b) turning of the external surfaces;
   c) milling the grooves, flats and other similar outer surfaces;
   d) drilling radial holes.
9. The final stage:
   a) polishing the opening;
   b) grinding or boring hole sections with high accuracy;
   c) grinding of external surfaces;
   d) cutting the slots;
   e) threading;
   f) polishing the outer surface;
   g) the final finishing of work surfaces.
10. Final inspection.

Hardening and tempering the shafts with a diameter of less than 50 mm with small margins on all levels are carried out before the first stage. If the shafts have a small length, for example up to 500 mm, they are not center-drilled, a roughing step starts with the hole treatment.

Cemented shafts:

1. Heat treatment (normalization).
2. Trim of the ends and center-drilling.
3. Turning the external surfaces for a backrest (if long shaft).
4. Draft stage:
   a) drilling axial holes;
   b) bore a hole, turning pivot chamfers;
   c) cutting of the test specimen (for the first control group of trees);
   d) turning of the external surfaces;
   e) grinding hole cemented sections;
   f) grinding the external cemented surfaces.
5. The copper plating of not cemented surfaces.
6. Cementation, hardening, tempering.
7. Dressing and peeling of basic chamfers of the hole.

8. Grinding necks for the backrest (if long shaft).
9. Finishing stage:
   a) boring holes;
   b) turning of the non-cemented outer surfaces;
   c) cutting slots, cutting fillets etc.;
   d) drilling radial holes.
10. The final stage:
   a) hole boring;
   b) grinding cemented areas of a hole, restoring basic surfaces;
   c) grinding the outer center shafts;
   d) grinding slots;
   e) threading;
   f) polishing the outer surfaces;
   g) final finishing of external work surfaces.
11. Final inspection.

Possible ways of obtaining the preform:

1. Hammer forging in pilot production.

2. Toggle clamping die forging. Die line is along the axis. Without a central opening.

3. Toggle clamping by direct or indirect extrusion of a preform with axial hole. Initial preform – a wheel, a profiled material.

4. On horizontal forging machines (GFM). Upsetting of a flange without axial bore. The initial preform – round profiled material. For stepped shafts rotary extrusion is used.

5. GFM – from the initial pipe blank. Shafts with an axial bore.

All Templates without axial and with axial bore are subjected to heat treatment – hardening and tempering to a hardness of 280...320 HB.

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Chapter 13. ENGINEERING PROCESS FOR HELICOPTER TRANSMISSION GEARBOX MOUNTING

13.1. MONITORED QUALITY PARAMETERS

According to the standard definition, the engineering process is the process of preparing the plant for manufacturing a new product line. Special attention is given to the issues of product high quality, reliability, and life assurance. Upon that, the largest amount of work accounts for monitoring the geometrical parameters, related to the assurance of accessories interchangeability, accurate replication of exterior shapes and kinematics of assembly operations (Table 13.1).

Table 13.1

Quality Parameters Monitored at Engineering Process Stages

Stages of engineering process	Monitored quality parameters, %		
	physical	geometrical	functional
Structure workability optimization	12	68	20
Prescriptive and testing techniques development	5	70	25
Jigs, fixtures, and tools development	2	93	5
Jigs, fixtures, and tools production	1	90	9
Manufacturing techniques optimization and full-scale production technology introduction	4	40	56

Different techniques and tools are used to mount assembly fixtures. These techniques and tools are chosen according to the following factors:

– standard technique of assuring the interchangeability during the engineering process;

– structural features of tools;

– accuracy requirements to the production and arrangement of fixing and basic equipment members.

The classification of mounting tools used to produce assembly fixtures is based on the design features of instruments and devices, as well as their precision considerations.

All the mounting tools fall into three classes:

– hard carriers of shape and size;

– coordinate racks;

– noncontacting tools for mounting and size coordination.

The field of monitoring and brassboard equipment utilization for mounting and correlation with the setting base is limited by:

– the 7 to 12 accuracy degrees for production tolerance;

– the dimensions of correlated joints: the accuracy degrees 7 and 8 – under 1250 mm; the accuracy degrees 9 and 10 – under 2500 mm, the accuracy degrees 11and 12 – under 4000 mm.

If the correlated parts dimensions exceed the specified ones, the hard size carriers production is not recommended. In such cases, optical and laser systems are to be used to monitor the equipment mounting.

The development of helicopter and airplane engineering causes the increase in requirements to the production quality of aircraft assemblies and accessories, as well as

the techniques and tools for monitoring their mounting.

The most advanced monitoring techniques are optical and laser ones. They have a number of advantages over the contact techniques. These techniques make it possible to monitor the mounting process at a considerable distance with high accuracy and slight inertia. For example, it is possible to find the distance from a point to a straight line with an accuracy of ± 0.025 mm when the point is 5000 mm away from the tube. When the holding frame equipment for heavy helicopters and airplanes is mounted, the full-scale range of laser instruments is unrestrained.

Optical and laser techniques can be used to do linear and angular measurements, to control the mounting of aircraft accessories and assemblies, and get the most adequate results.

By using laser and optical monitoring techniques in manufacturing it is possible to reach a higher degree of accuracy in meeting the production requirements during the assembling work, as well as provide reliable operation of accessories when in service. The present-day optical devices make it possible to measure dimensions, monitor shapes, as well as mutual axes and surfaces arrangement at a high accuracy degree.

A helicopter differs from an airplane in structure considerably, and some of its assemblies and accessories, such as the transmission gearbox, operate at specific conditions and require special attention during the mounting process.

13.2. ENGINEERING FEATURES OF TRANSMISSION GEARBOX ACCESSORIES INSTALLATION MONITORING

The process of mounting the helicopter transmission gearbox accessories is labour-intensive. The mounting difficulties are determined by the fact that the transmission gearbox accessories (intermediate and tail gearboxes, shaft bearings) are located at a considerable distance from each other, and especially from the main gearbox.

When accessories are located in such a way, it is necessary to provide their coaxiality with accuracy better than a tenth of a millimetre and the shaft alignment error within several arc minutes.

For example, finding an appropriate place for an intermediate gearbox requires shifting it both horizontally and vertically, finding the necessary angle of mounting against the main gearbox. Pre-setting the intermediate gearbox with bolts of minor diameter, going through the transfer sleeves, does not always provide its shifting at the necessary distance because of large dimensions and mass.

It is inconvenient and labour-intensive to shift the intermediate gearbox manually along the mounting surface, as the workers have to slacken the bolts attaching it to the rib, i.e. to disturb its initial coaxial and angular alignment.

The fittings of the tail gearbox bearings are mounted on the fuselage central section holding frame of some helicopters not finally. They are attached with test bolts with a view to correct their position in the final assembly shop during the process of transmission gearbox accessories mounting according to the intermediate gearbox final position. Such a technological solution does not reduce the production expenses in the process of transmission gearbox accessories mounting.

The bearing fittings are mounted on bulkheads with acceptable errors in coaxiality according to the drawing, but their angular fixation and mounting against the plane vertical to the tail shaft axis are flexible and have considerable errors. Moreover, the flexible fixation of fittings during riveting the fuselage outer skin to bulkheads causes additional errors, and in some cases even facilitates the fittings deformation.

The tail pylon is jointed with the fuselage center section according to the drawing. However, being affected by gravity after removing the retention pins from the holding frame, it does not remain in its initial position, but droops. Moreover, the pylon overhangs after mounting standard accessories on it. The pylon overhang changes the position of ribs, on which the intermediate and tail gearboxes are mounted, which results in occurrence of additional coaxial and angular errors.

This can cause undesirable consequences. During the transmission gearbox accessories mounting the unexpected labour-intensive works occur, such as filing the mounting surfaces of the gearbox ribs etc.

As experience has shown, some facts must be considered during such works.

The ambient temperature effect must be taken into account. For example, if sun rays heat a side of the pylon, it becomes deformed due to the heating and the position of the tail rotor drive shaft axis changes respectively. The pylon deformation in the intermediate gearbox plane against the fuselage longitudinal plane can reach more than 10 mm. The transmission gearbox can be adjusted more accurately if the helicopter is standing on the hoisters, not on the landing gear.

The correlation degree of basis planes influences the mounting quality considerably. The complexity of such correlation is determined by the fact that aggregate assembly of a fuselage is usually done on two, three, or even four holding frames. That is why negligible errors of each holding frame summarize into considerable errors during the final mounting of the transmission gearbox accessories mounting.

At this rate a careful analysis of holding frames monitoring schemes and assemblies characteristics must be done, and the requirements to all the base surfaces of transmission gearbox accessories must be brought into accord, etc. A special mounting scheme for holding frames must be developed, specifying the arrangement of the monitoring instruments, technological coordinate planes, and the table containing the basic mounting points coordinates. The monitoring scheme of the holding frame basic mounting points must accompany the workshop drawing of the holding frame.

To avoid unexpected errors, it is advisable to monitor the base planes and coordinate points of the holding frames using the same techniques and tools. The introduction of improved techniques and tools for monitoring the laser centering measuring systems or optomechanical instruments – sighting telescopes, provides higher quality mounting of the helicopter transmission gearbox accessories.

By using models, which imitate the transmission gearbox accessories, it is possible to expand the spread of mounting work. In particular cases, models can be effectively used as retainers of separate basic fuselage assemblies and surfaces for optical instruments struts.

For example, the main gearbox model must have the leads of the basic tail gearbox shaft mounting drive and the fan shaft drive. It can be used to fixture the assemblies of the sub-gearbox frame directly on the holding frame or on the brassboards. In this case there is no need to develop and produce a special appliance for fixing the assemblies of the sub-gearbox frame on the holding frame.

The transmission gearbox tail shaft drive of such a model can be a starting basis for arranging the optical instruments appliances when monitoring the process of mounting of the intermediate and tail gearboxes, as well as the tail shaft bearings.

The fan shaft drive can be a starting basis for mounting the fan. Such a model will make it possible to fixture the sub-gearbox frame assemblies securely on the holding frame in the main gearbox mounting area, to fixture the fittings of tail shaft bearings on the holding frame (directly on the base bulkheads planes).

13.3. ENGINEERING PROCESS ADMINISTRATIVE AND TECHNICAL MEASURES ON OPTICAL AND LASER TOOLS USAGE

Using improved optical and laser monitoring tools in the manufacturing process results in complication of the organizational preparation stage. The introduction of these tools must be planned at the stage of aircraft designing and large-sized technological equipment developing.

When designing a product, a monitoring technique is carefully developed, an optical instrument is chosen and the direction of the following factors are specified: the main sighting lines direction, the hole for these sighting lines, the areas for instruments struts, the areas for target marks on parts and assemblies.

The holding frame structure must be developed especially carefully. This task must be fulfilled by the technologists both from the aggregate assembly workshop and final assembly workshop. The structure development must include the making of fixing assemblies list, choosing the appropriate fixing techniques, as well as monitoring techniques and tools, choosing the mounting scheme of the holding frame and the basic places for monitoring instruments.

Choosing the measuring base is the key condition for doing measurements and monitoring the assembly work. The measuring base is the product's surfaces, which are the mounting surfaces for the products being matched and according to which the dimensions are found.

Choosing the measuring bases depends on the engineering process stage, at which the monitoring process is fulfilled. During the operational monitoring process the measuring bases are those technological bases which were used in the manufacturing process. When monitoring the finished product the measuring base is the main mounting base, i.e. the product surface, which is the mounting surface.

In almost every case of the product's engineering process a centerline or a baseline (or surface) is taken as a parts dimensions datum. It is also necessary to have the models of the intermediate and tail gearboxes. The models make it possible to make the complete markup for mounting the standard accessories, i.e. to find the mounting places for the intermediate and tail gearboxes.

Basing on the practical use of models and other appliances in the transmission gearbox accessories mounting with the use of optical instruments, the recommendations on technological equipment usage have been developed to promote the mounting quality and operational comfort during the full-scale production process. For example, the intermediate gearbox model must have two moveable flange heads (see Fig. 13.1). Each head can be moved horizontally and vertically with the use of rotors.

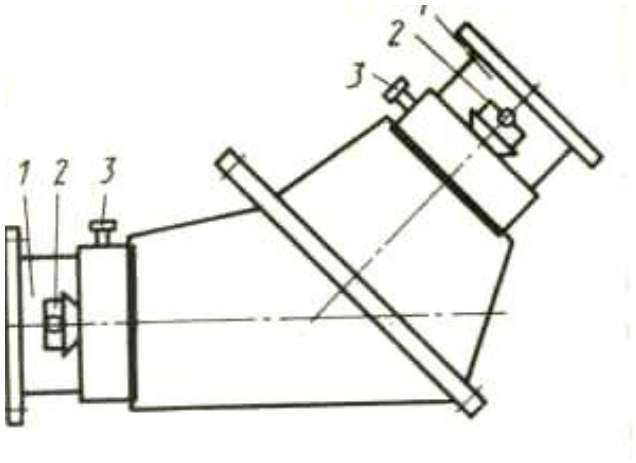


Fig. 13.1. Intermediate Gearbox Model:
1 – moveable head; 2 – horizontal movement rotor; 3 – vertical movement rotor

Such model structure with a scale on the model case makes it possible to find the distance, at which the intermediate gearbox must be shifted vertically or horizontally, to be arranged coaxially with the optical instrument. Upon that, the model itself remains attached to the rib with bolts and there is no need to change its position on the mounting surface.

The production accuracy of such a model, i.e. its base and bearing surfaces, as well as their positions, must meet the tolerances and fittings, specified in the workshop drawing for the standard intermediate gearbox.

The structural engineer of the technological tools and the technologist must accomplish the following tasks during the engineering process:

– to develop a general layout of optical and laser instruments for base sighting lines and base laser beams in the accessory;

– to develop a general layout of coordinate location holes in the parts and assemblies slated for base sighting lines and base laser beams location;

– to work out the equipment design;

– to design the holding frame equipment.

The technologist must accomplish the following tasks during the development of the engineering process for the accessory mounting:

– to find the most effective techniques for technological monitoring of both the aggregate and the final assembly of a product;

– to make a list of works both for the aggregate and the final assembly of a product, which require using optical and laser measuring tools;

– to develop a correlation scheme of coordinate location holes in the technological equipment;

– to make a list of necessary measuring tools;

– to develop a checkout scheme of the standard measuring system;

– to make a task for developing the base mounting assemblies for the installation of optical and laser monitoring instruments;

– to develop technical specifications for designing the monitoring technological equipment;

– to develop a manufacturing technique, specifying the ways of parts mounting according to the base sighting lines and base laser beams;

– to organize the training of technologists, foremen, assembly fitters, who take part in the process of assembling, to teach them how to use new measuring systems.

One of the advantages of the laser measuring systems introduction lies in the fact that the parts arrangement is monitored continuously during the process of assembling. This process can be changed, if necessary, which reduces the quantity of faulty parts. During the usual assembling processes the inaccuracy in the basic parts arrangement is generally revealed when jointing accessories and it is impossible to reconstruct these faulty parts. The assembling technique using base laser beams can be effectively implemented both into the pilot and the full-scale production when assembling the most important structure members.

Some technical requirements must be met when implementing optical and laser instruments into the production. These requirements are determined by the fact that the instruments are generally complex in structure and operate in specific conditions in contrast to the contact measuring tools. These instruments are expensive and require highly-skilled maintenance staff. It is recommended to store such instruments and their appliances in separate storage cabinets and rack cabinets in a dry heated space.

Besides, it is recommended to have a team of workers (not less than 3 people) in the final assembly shop for continuous measuring and monitoring. In this case optical and laser instruments will be used in the most appropriate way.

In the full-scale production, where the amount of assembly work is always recurrent, it is recommended to have a fixed team of workers responsible for the assembly work monitoring in the final assembly shop. The foreman of this team must organize and manage all this work.

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Chapter 14. SELECTION OF OPTICAL AND LASER MEANS OF TRANSMISSION INSTALLATION AND CONTROL

It is expedient to carry out a selection of means of optical and laser control in such sequence: measurement method selection, selection of measurement diagram and selection of the optical or laser device.

When selecting a ***measurement method*** it is necessary to provide the accuracy of measurements demanded for technical process, little measurement time, convenient control and check of optical and laser devices.

The selection of the ***measurement diagram*** should be begun with working out choices of such diagrams taking into account production requirements. Thus it is necessary to consider a base choice for installation of the optical and laser devices, installation possibility of target marks, plumbs, etc., to provide the smallest errors at measurements, convenience in work with optical and laser devices when measuring, the minimum time expenses for control operations.

Selection of optical or laser control devices is carried out taking into account such requirements:

– it is necessary to provide compliance to the chosen measurements diagram;

– range of a measuring scale or the optical compensator have to provide necessary counting at measurements;

– it is necessary that technical characteristics of the device correspond to vising distance;

– the absolute error of the device has to correspond to the chosen measurement method.

14.1. SELECTION OF OPTICAL MEANS OF INSTALLATION AND CONTROL

The universal device PPS-11 is intended for straightforwardness and planeness check of large-size products, control of coaxiality of openings, parallelism or perpendicularity of the planes and axes of openings. The absolute measurement error of shift in the plane, a perpendicular sight axis is equal ± (0.01 ± to L/200 mm), where L is a line-of-sight distance, m.

The device represents a telescope with a range of focusing from 0.03 to 30 m. The telescope (Fig. 14.1) has handles of focusing 1 and reference mechanism 2, a view lens 3.

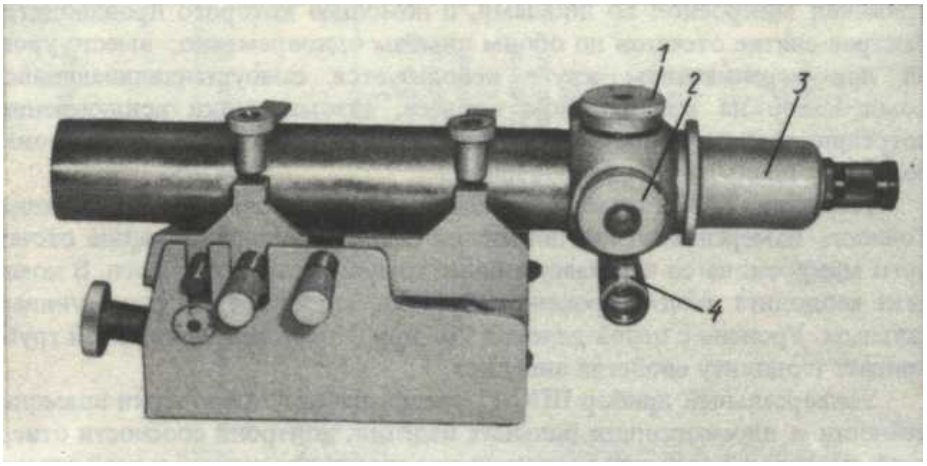


Fig. 14.1. PPS-11 Telescope on Support

The carrier 4 serves for installation of illumination of telescope target mark. On scales count shifts of the image of vising object of rather optical axis of a telescope. The existence of two handles of the reference mechanism allows carrying out measurement in two mutually perpendicular planes. An view lens 3 is replaceable, instead of it, it is possible to establish the view lens which refracts an optical axis on 90 °. In two mutually perpendicular directions an armored penta prism is applied.

The VLC device is intended for regulation of coaxiality of engine shaft and a gear box by helicopters sight and collimating methods.

Practice showed that VLC can be used also for measurement of non-axiality of shafts and openings, measurement of angular displacement at installation of separate units of transmission, and the collimator VLC – for measurement of corners in a combination to other optical devices, for example with the PPS-11 device.

The telescope of the VLC device (Fig. 14.2) has a landing core 4 by means of which the telescope is installed in the special adaptation.

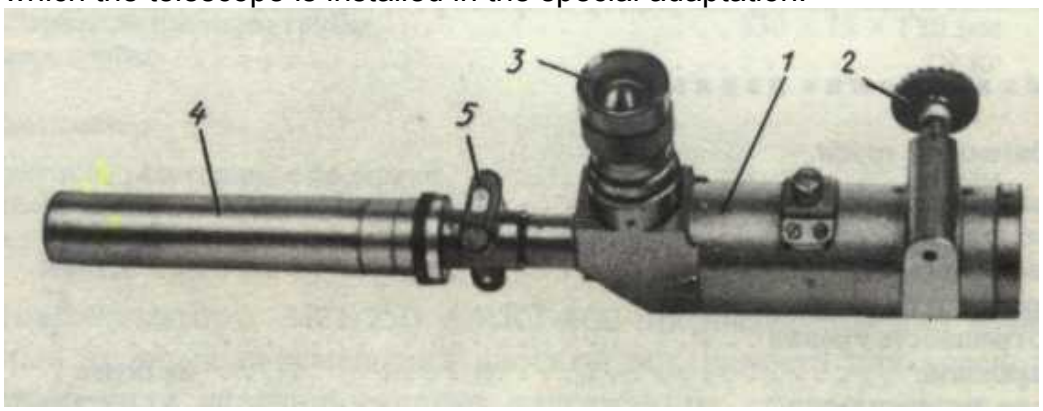


Fig. 14.2. VLC Device Telescope:
1 – hull; 2 – handle;
3 – view lens; 4 – landing core; 5 – level

On a setting core there is a level 5 intended for coordination in the cross direction of a pipe grid with collimator grids. In the hull 1 of telescope the armored mobile lens is mounted which can be moved along an optical axis by means of the handle 2 that gives opportunity to focus the image on a grid for measurement of inclinations or on a grid for measurement of the collimator shifts. On the hull is mounted view lens 3 which has a diopter scale.

The collimator has two grids. One grid with a stroke – a bisector is intended for non-axiality measurement, and another with a crosswise stroke – for measurement of angular shifts. Coaxially to a telescope carries out installation of the collimator serial focusing of a pipe on a grid for measurement of shifts and on a grid for measurement of tilt angles.

If the telescope is focused on a collimator grid for measurement of tilt angles, the grid won't be visible to measurement of shifts. At such diagram it is possible to install the collimator coaxially with the line of telescope vising.

This possibility of optical system *the collimator – the telescope* allows using such optical systems for adjustment of shaft coaxiality of the engine and a gear of helicopters, and also for control of installation accuracy of intermediate gears on helicopters.

The collimator of the VLC device (Fig. 14.3) has a landing core 1 with a diameter of 52 mm which is installed in an opening of the special adaptation. In a setting core there is a assembly 4 of illuminations with the plug socket.

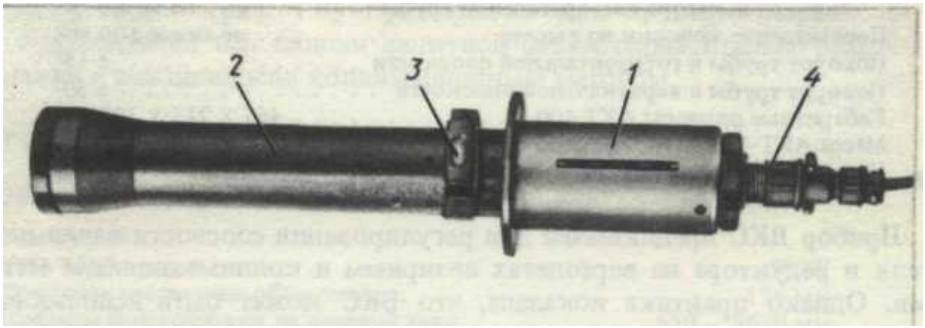


Fig. 14.3. BKC Device Collimator:
1 – core landing;
2 – pipe; 3 – level; 4 – illumination assembly

On a pipe 2 of collimator the level 3 is intended for coordination of collimator grids in the cross direction with a grid of a sight pipe. The lens, a grid for measurement of collimator inclinations, a lens and a grid for measurement of shifts of the collimator are built in a pipe.

To mount a telescope and the collimator on roller couplings, the set from two devices which are based on a shaft is developed. Devices are fixed in two provisions by splined couplings of a shaft with the roller coupling and a stop of the flange face plane of the device in a face surface of an external holder of the roller coupling.

Each device is fixed with three bolts on a special collar, which is mounted on the roller coupling.

At installation optical axes of a telescope and the collimator of the VLC device have to coincide with shaft axes of gear and engine shaft. It is provided with the accuracy of processing of telescope landing cores and the collimator of the VLC device and accuracy of production and assembly of the device for telescope installation (Fig. 14.4). The landing core of a telescope 1 is established in a shaft of the device 2 and is fixed with the tightening screw.

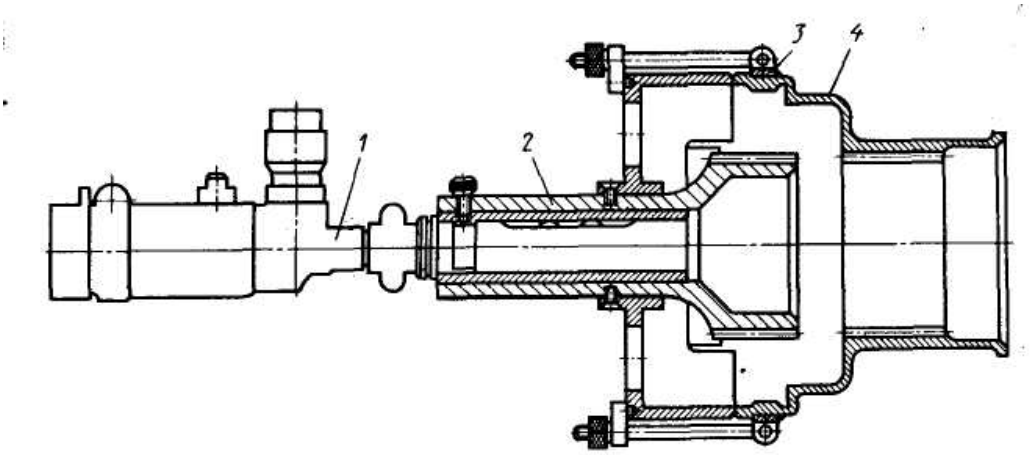


Fig. 14.4. Installation Diagram of Telescope on Gear:
1 – telescope; 2 – device for installation of telescope on the cardan coupling of gear; 3 – collar with clips for fastening of device for telescope installation;
4 – coupling of cardan gear

Base for telescope installation of VLC at optical control of coaxiality of the engine shaft and a gear is the roller coupling of a gear. At this diagram of the VLC installation as the basic line is accepted an optical axis of a telescope which is as if continuation of an axis of gear shaft.

The collimator 4 (Fig. 14.5) is fixed on the device 3 with two bolts passing through a special flange on its hull.

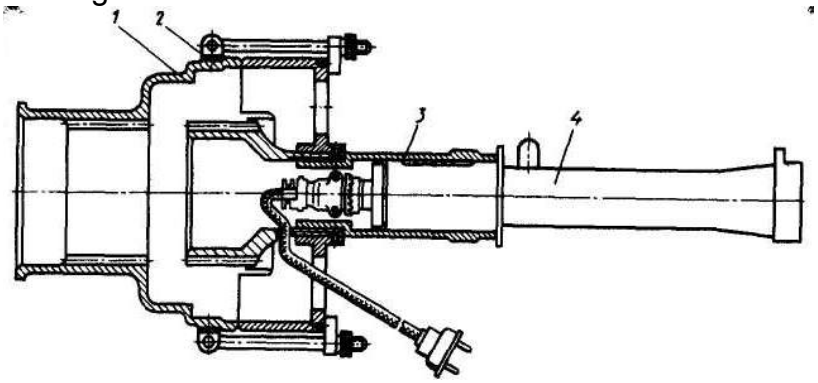


Fig. 14.5. Installation Diagram of Engine Collimator:

1 – coupling cardan engine; 2 – a collar with clips; 3 – the device for installation of the collimator; 4 – collimator

In Fig. 14.6 the installation diagram of a telescope and the collimator of the VLC device on an intermediate gear of transmission is shown.

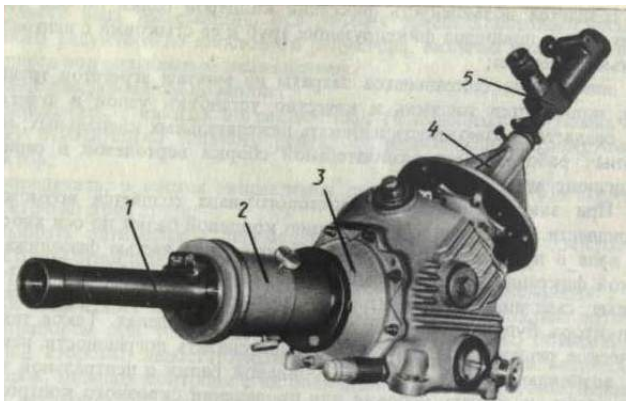


Fig. 14.6. Installation Diagram of VLC Device on Intermediate Gear:

1 – collimator; 2 – the device with displaced head on a corner; 3 – intermediate gear; 4 – the device for telescope installation; 5 – telescope

At optical measurements angular displacements lead to axial displacements. Therefore at control of installation accuracy of an intermediate gear with the help of optical devices it is necessary to achieve parallelism of axes of shaft in the beginning, and then to eliminate axial shift.

14.2. SELECTION OF LASER MEANS OF INSTALLATION AND CONTROL

In measuring systems, as a rule, low-power gas lasers are used (power of radiation 1... 2 mW).

These lasers have the minimum width of the generation line, the smallest angle of divergence of measurement beam (about 1'), the single-mode and single-frequency

mode of radiation, the stable power and geometrical parameters of emission of a beam. The radiation trace of helium neon gas laser is steady at vising distance to 100 m.

The single-mode emission mode is characterized by uniform distribution of density of laser radiation on the cross section of a beam that creates evenly lit spot on the screen. Such distribution of density allows to measure shifts of object with the greatest accuracy with the help as photodetector, and a usual target mark when the beam of laser radiation is projected on the measuring screen.

The laser sizes shouldn't exceed the ordinary optical devices sizes. The laser hull has to be of a cylindrical form, and the radiation beam should be settled down concentric to the hull with a margin error no more than ± 0.01 mm or ± 0.5". The laser hull has to be tight to exclude possibility of hit of moisture, dust and oil on resonator mirrors. Service life of a gas-discharge pipe has to be not less than 5000 h.

The stability of provision of an axis of the directional pattern of radiation is very important requirement. This drift shouldn't exceed a share of angular seconds during 8 h operation of the laser.

In Fig. 14.7 is shown serial helium neon laser of model ILGN-207 which technical characteristics conform to the requirements given above.

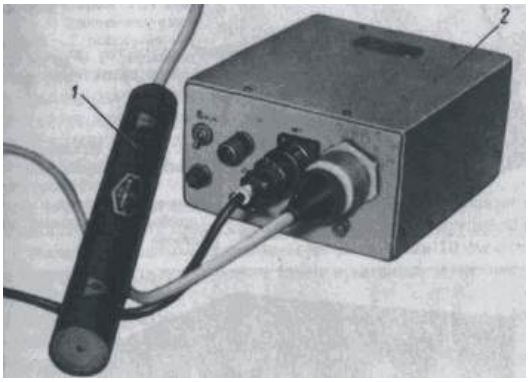


Fig. 14.7. Serial Laser of Model ILGN-207: 1 – radiator; 2 – power unit

Photodetectors of laser emission are applied to more exact laser measurements. The principle of their action is based on interaction of an electromagnetic field with substance of a sensitive element. The photodetector is the main assembly of the position sensitive target mark (PSTM).

Depending on arrangement according to assembly or reference diagrams they can play a role of both basic, and working position sensitive target mark. Adjusting devices for position sensitive target mark have to comply with the technical requirements shown accepted scheme of measurements and provide the most exact control.

To display information on the provision of a beam of laser radiation the position sensitive target marks are supplied with blocks of pointer or numeric display in linear or angular units of measure.

The increase in density of emission power in the direction of an axis of the leaving laser beam by means of the collimator allows to increase long-range action of the laser aligning measuring system (LAMS) to hundreds of meters. In measuring systems the collimator is installed in one device with a laser radiator.

LAMS which are widely applied in aircraft construction differ from other measuring systems of similar appointment with the increased accuracy thanks to creation of stable in space and time ring interferential structure in a beam which axis is taken for a straightforwardness standard.

In LAMS are used also the visual target marks (VTM), simpler in a design, reliable and rather exact. In dependence on an objective of PSTM and VTM can be structurally executed so that to control two, three, four or five coordinates – linear, angular and a distance to object at the same time.

Design of devices is defined by the technical requirements, purpose and working conditions. For control of assembly installation works and accuracy of installation of separate units laser devices are built in building berths, coordinate and assembly stands and other industrial equipment.

The way of laser fixing in the adjusting device is defined by the measurement diagram. For example, in case of installation or control of an arrangement of assemblies of building berth equipment the laser is usually aligned on two extreme points of the technological coordinate planes (TCP).

The beam of laser emission passing between TCP is a basic axis for installation or control of building berth equipment. In this case the laser device of the corresponding complete set, for example, as the index of the direction (Fig. 14.8), can be installed on an adjustable support. Special requirements aren't imposed to a design of its case.

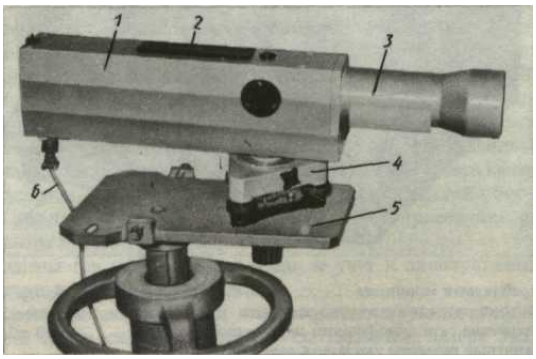


Fig. 14.8. Laser Direction Sign: 1 – laser hull; 2 – level; 3 – collimator hull; 4 –support for leveling; 5 – platform; 6 – laser power supply cable

In a case when the laser beam of LAMS has to display or create the basic axis of a shaft passing, for example, through the center of a flange of the unit drive, measurements are carried out according to other diagram. Such measuring diagram is necessary at transmission units installation on helicopters (Fig. 14.9).

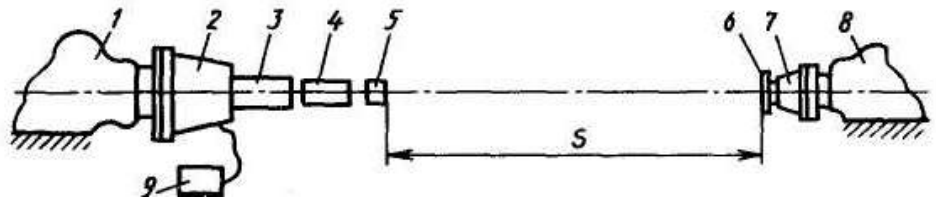


Fig. 14.9. Arrangement Diagram of LAMS at Control of Installation Accuracy of Intermediate Gear:
1 – main gear box; 2 – the device for laser installation; 3 – laser radiator; 4 – collimator; 5 – diaphragm; 6 – target mark; 7 – the device for target mark installation; 8 – intermediate gearbox; 9 – laser power unit

In this case the laser hull has to have basic seating faces on which the adjusting device is mounted.

In Fig. 14.10 is shown one of device designs for installation of laser radiator and the collimator on the main gear box of the helicopter. The laser and the collimator are placed in the device hull. Centering of the device concerning an axis of a gear box is carried out by adjusting screws.

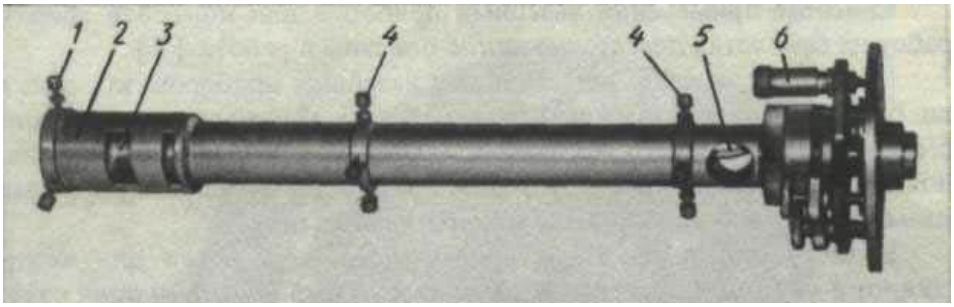


Fig. 14.10. Device for Laser Radiator and Collimator Installation on Main Gear Box of Helicopter: 1 – fastening screw of the collimator; 2 – device hull; 3 – collimator; 4 – fastening screw of laser radiator in device; 5 – laser power wire; 6 – adjusting screw

The production experience of optical and laser means of installation and control shows their high efficiency and universality.

In the course of assembly works on helicopter and the plane it is possible to control quality of installation of building berth equipment, accuracy of fuselage installation, tail and keels, a wing, the stabilizer, landing gear, the main and intermediate gear boxes, separate units relatively to each other, not parallelism of the gear and fan shaft, axes of openings of tail shaft supports.

The provision of basic beams of laser radiation is set by TCP, which are mounted on end faces of units in the locations of building berth plates or basic sections of switch actuators.

When assembling the unit on the basis of basic beams of laser radiation necessity for production of the model frames, ribs and other elements of a building berth, which temporarily support a framework of the assembled unit, disappears.

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Chapter 15. STANDARD TRANSMISSION INSTALLATION AND CONTROL OPERATIONS BASED ON OPTICAL AND LASER SYSTEMS

The optical and laser means which were earlier applied only for control began to be used for performance of units installation. Installation process of units becomes operated, at the same time the accuracy of units installation increases not less than on two orders. By means of devices it is possible to observe visually unit axis deviations from the optical line of vising or a laser axis of a radiator, and also not parallelism of axes, i.e. angular shifts.

The main condition for performance of measurements and control of assembly works is the choice of measuring base. At optical and laser measurements they usually use control lines from which they carry out measurements. It is desirable that the vising line of the telescope or laser beam axis was the control line.

15.1. INSTALLATION AND CONTROL OF ROTOR GEARBOXES INSTALLATION ACCURACY

By means of optical devices there is an opportunity to observe a real picture of intermediate gearbox location as regard to rather main and tail gearboxes, and location of transmission tail shaft. All working route of a tail shaft from a main gearbox to a tail gearbox, including a shaft axis break, is under visual observation.

The method of vising allows to define:

– with what errors (in linear and angular units) as regard to the main gearbox the intermediate gearbox is located;

– in what direction transmission units have to move relatively to each other;

– what size of intermediate and tail gearboxes optimum shifting as regard to each other.

When installing the transmission units on helicopter, the base for installation of optical and laser devices is the main gearbox (Fig. 15.1).

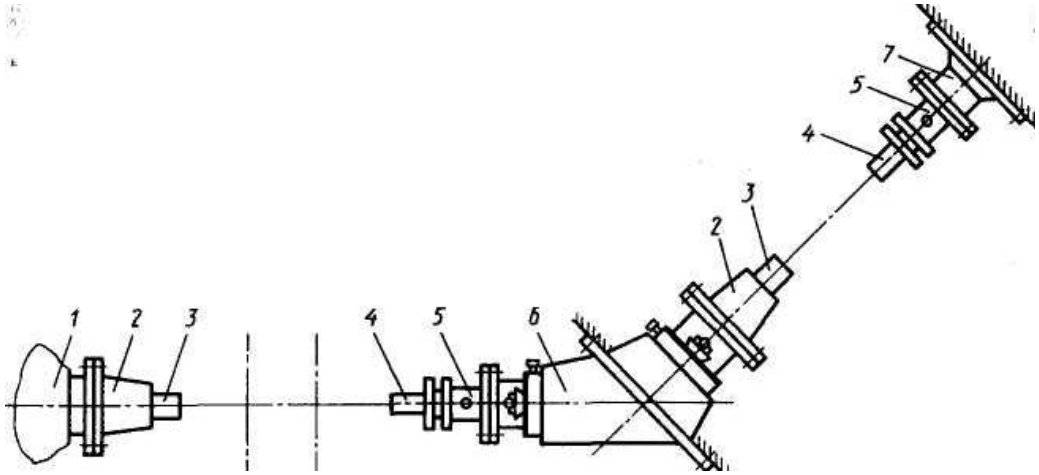


Fig. 15.1. Scheme of Optical Devices Location:
1 – main gearbox;
2 – device for telescope installation; 3 – telescope;
4 – collimator; 5 – device for collimator;
6 – model of intermediate gearbox; 7 – model of tail gearbox

The telescope 3 of the optical device (PPS-11 or VLC) in device 2 is installed on a output flange of main gearbox tail shaft 1. The model 6 of intermediate gearbox is put

on the place of a regular intermediate gearbox, and model 7 of tail gearbox – on the place of a regular tail gearbox. The model of the tail gearbox can be the usual plate imitating the basis of the tail gearbox with a drive output to a tail shaft.

In the course of primary assessment of model 6 location as regard to a boresight, it is possible to establish a target mark, which grid is graduated in millimeters, instead of the collimator 4. It will allow to measure model axis shift size from the boresight in the horizontal and vertical planes.

In Fig. 15.2 the measurement scheme of angular shifts which is used for calculation of angular errors of intermediate transmission gearbox installation is shown.

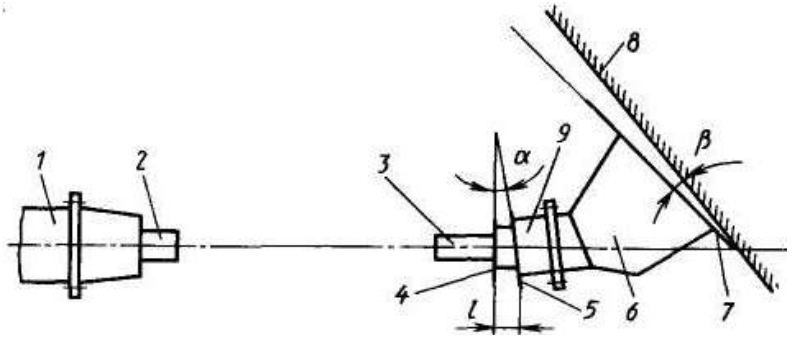


Fig. 15.2. Measurement Scheme of Intermediate Gearbox Angular Shifts:
1 – main gearbox; 2 – telescope; 3 – collimator; 4 – mobile flange; 5 – motionless flange;
6 – intermediate gearbox; 7 – gearbox case basis;
8 – rib plane; 9 – device for collimator installation

A scope 2 is installed on the main gearbox 1 and a collimator 3 is installed on the accessory gearbox 6 in the device 9 with two flanges – a fixed 5 one and movable 4 one. Knowing diameter of a flange 4 and displacement distance 1, it is possible to define a collimator axis tilt angle α.

Symmetry of angles α and β (see Fig. 15.2) allows to find the degree of tilt and angle on which the body of the accessory gearbox should be displaced, in other words how much a rib should be filed down to install an accessory gearbox against the boresight with allowable operational margins.

To find the angle of installation of the accessory gearbox on a basic surface, special device (Fig. 15.3) with angle head, displacing relative to the needed plane for installation of the collimator, is used.

The device consists of a body 1 with flange 5. The head 3 is displaced by handles 2, and the device is fixed to the accessory gearbox with bolts 4. Internal surface of the body is spherical. There is a head 3 inside and the collimator is installed in this head.

Due to the head 3 it is possible to find a required angle and the plane in which the accessory gearbox has to be installed. Knowing of asymmetry of a flange 6 and head 3 axes, it is possible to find degree of scraping the mounting surface of bearing fitting.

Having such a device for the collimator, it is possible to define the plane in which it is necessary to move a gearbox without tilting the model set over its mounting seat. The model set of the accessory gearbox at the same time remains fixed by bolts on a rib.

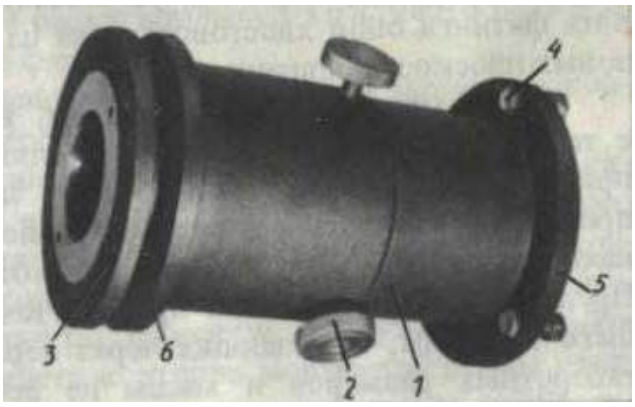


Fig. 15.3. Device for Angle Displacement of Collimator: 1 – body; 2 – handle to move the head; 3 – head for collimator installation; 4 – screw for device installation; 5, 6 – flanges

According to axis misalignment and angle of the accessory gearbox displacement against the main reduction drive and the tail gearbox against the accessory gearbox, the comparative analysis of measurements is carried out to find an optimal option of possible unit movement against each other taking into account the minimum finalization of bearing surface area of ribs.

Also such a variant when the tail gearbox can be taken for the second base during installation and units check isn't excluded. In this case the device with a scope is installed on the model of the tail gearbox, and the device with the collimator is installed on the model of the accessory gearbox.

If such an option is true only one mounting surface for the accessory gearbox is reworked. After a test check of installation accuracy of the units the bearing surfaces are filed and the accessory gearbox and tail gearboxes of transmission are installed.

Knowing average value of sagging of a tail rotor pylon after loading with units, it is possible to calculate and change the angle of installation of basic ribs on which the accessory gearbox and tail gearboxes are mounted, i.e. to perform corresponding adjustment of sagging angle. Such change allows to avoid wasteful expenditure of working hours and considerably increases technological effectiveness of transmission units.

15.2. INSTALLATION AND CONTROL OF MOUNTING ACCURACY OF TRANSMISSION SHAFT BEARINGS

Mounting and control of alignment accuracy of transmission shaft bearings for longitudinal and cross layout drawing of the helicopter can be carried out with the help of the optical and laser devices (Fig.15.4).

The target marks with two operation meshes are recommended to use when controlling the accuracy of setting of shaft bearings by sight-size method. In this case linear displacement of bearing is controlled by the mesh of target mark that is in the plane of bearing setting and angle displacement of bearing is controlled by the second mesh.

The angular displacement of bearing can be calculated, knowing the distance between the meshes of target mark and linear displacement of the second mesh of the target mark relative to the line of device sight-size. The allowable offset of the second mesh of the target mark relative to the boresight can be set knowing the tolerances of the angular deviations of the tail shaft bearings.

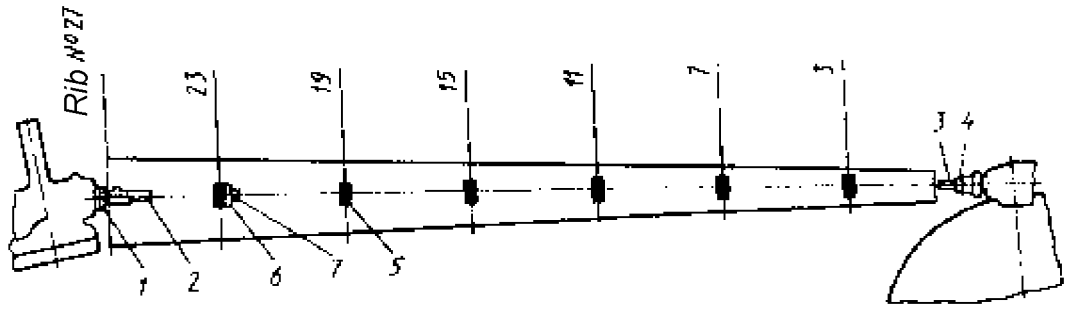


Fig. 15.4. Diagram of Mounting and Control of Alignment Accuracy of Transmission Shaft Bearings:
1 – device for setting of optical tube; 2 – optical tube;
3 – collimator; 4 – device for setting of collimator; 5 – seat of shaft bearings;
6 – device for setting of target mark; 7 – target mark

The marking of the setting of transmission shaft bearings is recommended to carry out after final adjustment and setting of the intermediate gearbox.

The mounting or control of alignment accuracy of transmission shaft bearings is carried out in the following order:

– device for mounting of the optical tube of the devices PPS-11 or VLC is set on the output shaft of the main gearbox; the optical tube is set on the device;

– verification of setting of intermediate gearbox is carried out, device for setting of collimator is fixed to the shaft output of intermediate gearbox; the collimator is fixed;

– optical tube is focused on the mesh mark for tilt measurement of the collimator. With proper installation of the intermediate gearbox the center of the mesh mark for tilt measurement of the collimator coincides with the center of the mesh mark of the optical tube;

– optical tube is focused on the mesh mark to measure displacement of the collimator. At coaxially output of shafts of the main and intermediate gearboxes the center of the mesh mark of the collimator to measure displacement is matched the center of the mesh mark of the optical tube;

– collimator and a device for its setting are taken off the intermediate gearbox;

– device for setting of target mark is fixed on one of the ribs of the wing or one of the frames tail boom, the target mark is set in a device;

– optical tube is focused on the target mark. The center of mesh mark of the optical tube is combined with the center of the plate mark of the target mark by displacement of the devices with a target mark;

– marking for setting the shaft bearing of the transmission.

The device of target mark is set at the places of fastening of the bearings by turns and the next shaft bearings are set similar.

If the fittings of the shaft bearings in the process of installation are fixed tightly, then coaxial marking can be performed by the target mark, as in this case, the angular errors are minor. With the help of a collimator it is possible to perform a control check of angular displacements and, if it is necessary to make an appropriate adjustment.

If the fittings of the bearings of the shaft are not fixed tightly in the process of installation then coaxial and angle marking can be carried out only with the help of a collimator.

The marking of the fittings for installation of bearings of tail shaft on frames of fuselage and tail boom is appropriate after finishing of the bearing surfaces and the final setting of the gearboxes.

To drill and ream the holes, it is necessary to have a special device for moving the collimator (or target mark) in the horizontal and vertical planes and for its rotation (Fig. 15.5). The device is equipped with a set of quick-change jig bushings and simultaneously functions as the template jig.

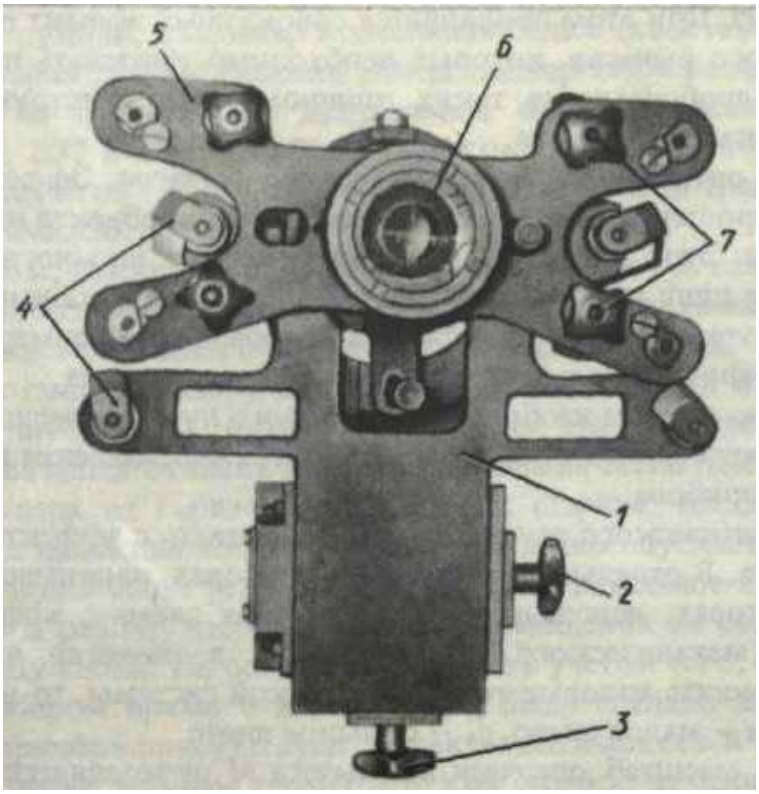


Fig. 15.5. Device for Setting of Bearings of Tail Shaft:
1 – case; 2 – handle for movement of the case in horizontal plane;
3 – handle for movement of the case in vertical plane;
4 – lock of device fastening on the frame; 5 – movable head;
6 – bushing for setting of collimator; 7 – handle for offset of movable head

The consuming and multi-step installation process of the fittings of the bearings can be significantly simplified to ensure rigid fixation of fittings of the bearings on the jigs by using special pipes with flanges that replace the tail shaft section. This fixation will not cause difficulties and hindrances when carrying out riveting operations in the central part of the fuselage and tail boom.

When rigid fixation of the fittings of the tail shaft bearings the following technological advantages explore:

– conditions of final installation of the fittings of the bearings are created directly on the assembly jig;

– installation of fittings with minimum angle errors relative to the plane that is perpendicular to the axis of the tail shaft is ensured;

– the possibility to avoid unexpected fitting works in the workshop of final assembly of helicopters in serial production is created.

When attaching the bearings of the tail shaft the ability to do more rigid fixation of the end beams along the axis of the tail shaft in the process of docking with the central part of the fuselage is created.

Such a technological solution will significantly reduce errors that inevitably arise when docking of end beams with the central part of the fuselage, and create conditions for the cross-cutting monitoring of the installation of all transmission units with optical and laser devices.

Systematic observation, recording and analysis of errors committed during mounting works and measurements, create the preconditions for adjustment of technological processes in order to improve them.

15.2.1. Control of Misalignment of Intermediate and Main Gearboxes

When installing the bearings the shafts error for misalignment of the intermediate and main gears boxes must be taken into account. Fig. 15.6 shows a diagram of the arrangement of the bearings of the tail shaft transmission, which is made with consideration of misalignment of the axis of shafts of intermediate and main gearboxes.

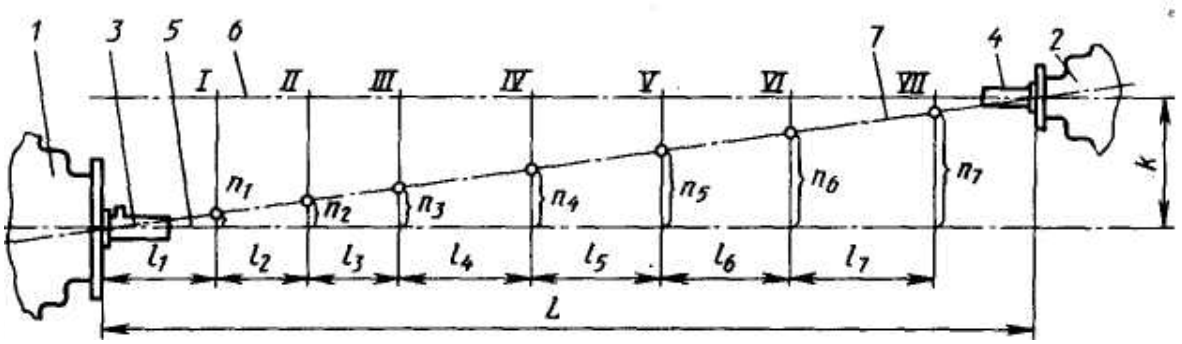


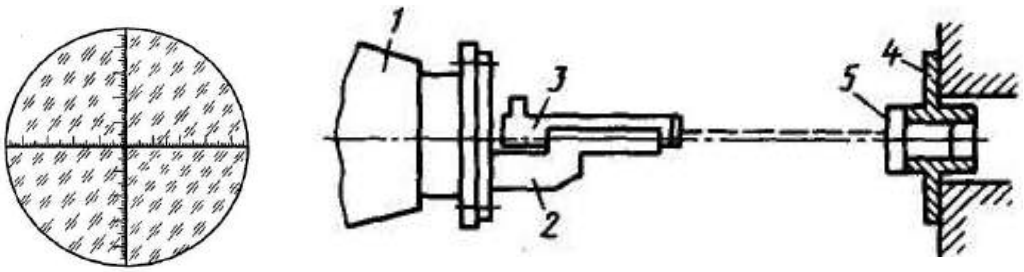
Fig. 15.6. Diagram of Arrangement of Tail Shaft Transmission Bearings:
1 – main gearbox; 2 – intermediate gearbox; 3 – optical tube;
4 – collimator; 5 – boresight; 6 – axis of the shaft of intermediate gearbox; 7 – axis of tail shaft;
I–VII – bearings

The axis of the drive shaft of the intermediate gearbox 6 is at a distance k from the boresight 5, adopted for base. The diagram also shows that the working (actual) axis of the tail shaft 7 does not coincide with the boresight, neither with the axis of the drive shaft of the intermediate gearbox.

Control of the location of the tail shaft bearings in this case is possible by the combined method of optical inspection using the optical tube with auto collimator, and mirror target mark.

The case of the target mark, regardless of design should provide the exact coincidence of the stroke center with the axis of the setting nozzle.

Mirror target mark with the markings on the plate (Fig. 15.7 a) allows to set the tail shaft bearing offset corresponding calculated and measure its angular deflection. Fig. 15.7, *b* shows the diagram of installation of the optical tube and the target mark.



a b

Fig. 15.7. Diagram of Control of Tail Shaft Bearing Mounting

The device 2 for mounting the optical tube3 is based on the output of the shaft of the main gearbox 1. The device 4 for set of the mirror target mark 5 is attached to the place of the tail shaft bearing. The optical tube at first is focused on the mesh mark of the target mark.

With offset of the device 4 with the target mark relative to the axis of the optical tube they try to obtain that the mark of the target mark plate coincides with the mesh mark of the optical tube sight. Then the illumination of auto-collimator of sight tube is switched in and focused it on the mirror surface of the target mark plate. Moving the device for setting target mark in the direction that is perpendicular to the optical axis optical tube it is necessary to achieve alignment reflected from the mark mirror with mesh mark of sight of the optical tube.

15.3. CONTROL OF TAIL PYLON JIG PLATE PARALLELISM

Modeling of aerodynamic shapes and sizes of helicopters and airplanes is determined mainly by the accuracy of manufacturing of jigs in which are mounted both separate compartments and units as a whole.

Large dimensions of controlled objects and high precision surface requirements have led to the fact that non-contact optical and laser tools have become basic in the manufacture and control of jig equipment.

Control of plates parallelism and coaxiality of the holes of the jigs is performed by a collimation method using the optical system from the optical tube and collimator. When you set and control the parallelism of the plates the devices are adapted, on which optical devices are mounted (Fig. 15.8).

The intermediate gearbox 8 is set on the plate 7, the tail gearbox 1 is set on the plate 2. The optical tube 5 is set on the intermediate gearbox, and the collimator 4 is set the tail gearbox.

The optical tube is focused on the mesh mark for measuring angles of inclination of the collimator. Observing through the eyepiece, you can see that if the mesh of the optical tube and the collimator mesh are not aligned, the plates have an angular deflection. The offset angle can be measured on a micrometer of the mesh of optical tube of the optical device.

Misalignment of the holes in the plates can be determined, by focusing the optical tube at a mesh mark of the collimator displacement measure.

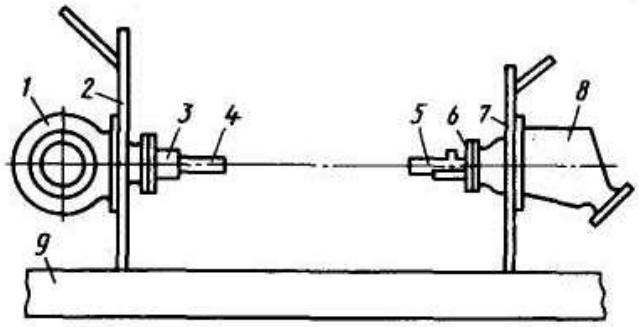


Fig. 15.8. Diagram of Control of Jig Plate Parallelism for Assembly of Tail Pylon:
1 – tail gearbox; 2, 7 – plates; 3 – device for collimator setting;
4 – collimator; 5 – optical tube; 6 – device for optical tube setting;
8 – intermediate gearbox; 9 – jig case

If the mesh for measurement of displacement of the collimator does not match the mesh of the optical tube, the holes are out of line. The value of misalignment is defined similarly by a micrometer of the optical tube mesh.

The errors that appear when using optical and laser devices, can be reduced for the following activities:

– careful study of the designs of jig tackle during its design;

– thorough control and alignment of base planes and dimensions of the jig relating to the transmission units;

– during installation use of the models of the main, intermediate and tail gearboxes of transmission.

15.4. CONTROL OF HELICOPTER TAIL BOOM SAGGING

When installing the transmission units on the helicopter tail boom is loaded with regular units, in result it turns down.

When controlling the deflection of the helicopter tail boom the next conditions is to be followed:

– the boresight of the optical tube coincides with the geometric axis of the tail shaft;

– a device for optical tube is mounted on the flange of the output shaft of the main gearbox;

– the target mark is installed on the second-to-main gearbox bearing shaft;

– a device for measuring the sagging of a boom is mounted on the last bearing of transmission shaft.

Control of deflection of the helicopter tail boom (Fig. 15.9) is executed in such technological order:

1. The device is installed on the flange of the main gearbox 1.

2. The optical tube 2 is fixed on the device.

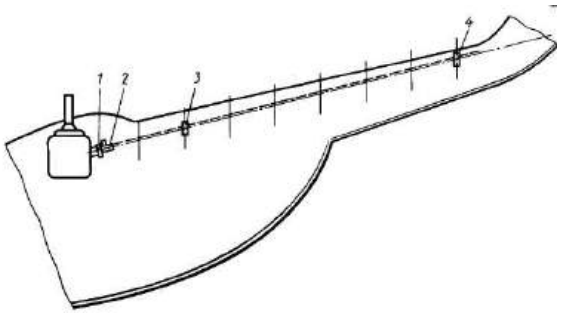


Fig. 15.9. Diagram of Helicopter Tail Boom Sagging Control:
1 – device for setting of optical tube; 2 – optical tube; 3 – device for setting of target mark;
4 – device for sagging measurement

3. The device 3 with a target mark is fixed on the second bearing from the main gearbox transmission shaft.

4. The optical tube is focused on the target mark. The target mark is set so that the center of mark of its plate aligns with the center of the mesh mark of the optical tube.

5. A final check of accuracy of mounting the optical tube is conducted on the flange of the main gearbox, for which the optical tube is rotated around its axis with the roller clutch of the gearbox with step 90° to return to its original position. It is necessary to observe the lack of displacement of the mesh marks relative to each other in the eye-piece of the optical tube.

6. The target mark is removed from the device.

7. The optical tube is focused on the device to measure the sagging of the tail boom.

15.5. STRUCTURAL ASSEMBLY BASED ON REFERENCE LASER BEAMS

The assembly, as is known, is a set of operations according to setting of parts in the assembly position and connecting them in assemblies, panels, units and airframe or helicopter in a whole.

The main positions determining the method and order of units assembly – location chart of units, assembly bases, the terms of delivery of subassemblies and scheme of alignment, the location of the holes for reference laser beams – must be represented at the stage of engineering process.

To control the assembly of individual structures and the accuracy of setting individual units the laser devices are built into the jigs on the technological master plate. The alignment of set of tackle is recommended to hold by one pair of the technological master planes. In developing schemes of non-model alignment of jig tackle it is necessary to follow the principle of unity of construction, design and mounting bases.

Initial technical documentation for the design of the technological master planes are: drawings of the unit (assembly); layout and schematic wiring diagram of the location of the assemblies to be aligned, and means providing alignment; the general scheme of non-model alignment of the joint, connector or the unit.

Obtaining the spot of small sizes of the laser beam on different distances creates an opportunity to control the installation of the transmission units according to their alignment with the main gear (see Fig. 15.9, 15.10).

Reference laser beams, LAMS in conjunction with working PSTS or VTS are simultaneously bases for the installation of jigs and assembly of units, sections and panels, which ensures the unity of instrumental and assembly bases (Fig. 15.10).

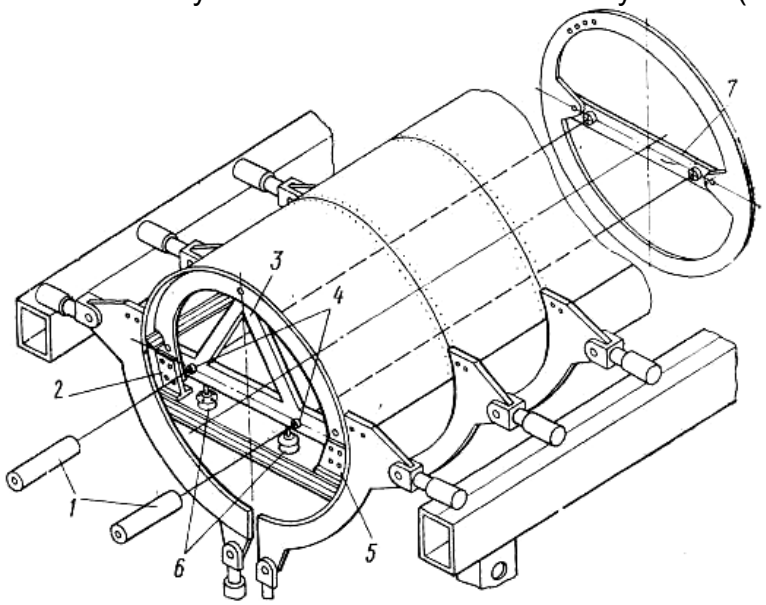


Fig. 15.10. Fuselage Top-to-Bottom Section. Assembly Diagram:
1– radiators of LAMS; 2, 5 – fastener assembly of the section; 3 – technological holders;
4 – working PSTS (VTS); 6 – control jack; 7 – technological master plane

In the method of creating of a reference direction using an annular apertures the laser radiation is collimated, and an annular diaphragm is mounted at the exit of the collimator. The case of the radiator laser with a collimator is based on the opening in the TCP by using a special device (Fig. 15.11). The adjustment screws and bearing retainers are served for the angular setup of the reference laser beam.

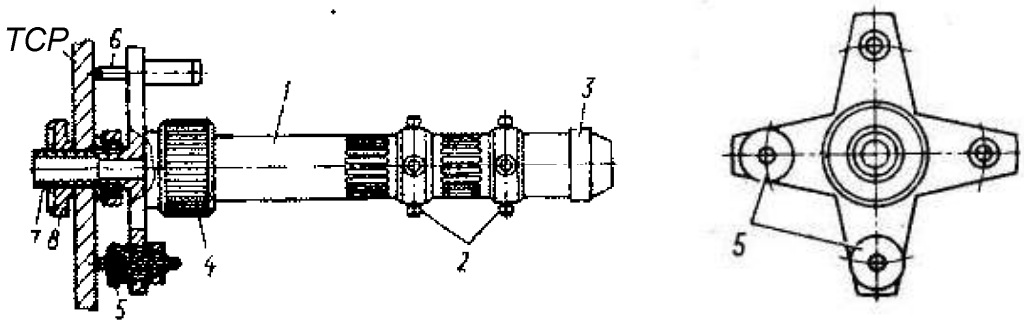


Fig. 15.11. Device for Installation of Laser Radiator on TCP:
1 – radiator case;
2 – screws for alignment; 3 – annular aperture;
4 – split terminal; 5 – screw for angular installation of beam axis;
6 – supporting holder; 7 – bushing of setting base of radiator; 8 – union nut

The annual apertures allow obtain the light spot of smallest diameter on target mark. In the process of alignment of the collimator it is necessary to achieve steady con-

trast of annular patterns of radiation. Basing laser radiators on the holes of the TCP, a system of reference laser beams to align the position in space of the assemblies of the jigs is created. TCP simultaneously performs two important functions: align the axes of the structural elements of the devices during installation and provide identical playback sizes when you transfer the coordinates to fixtures aligned.

Assembly method on the basis of the reference laser beams can be effectively applied both in prototype production and in mass production in the assembly of the most critical structural elements.

15.6. TAIL ROTOR PYLON SHIFT CONTROL

After the riveting works on the fixture and removing off the supports check the amount of displacement of the tail rotor pylon in the zone of installation of the tail gearbox. Measurement of linear and angular displacement of tail rotor pylon at full-scale static tests is similar to a standard task.

The diagram of control of the tail rotor pylon shift of the helicopter in the tail gearbox installation zone is shown in Fig.15.12.

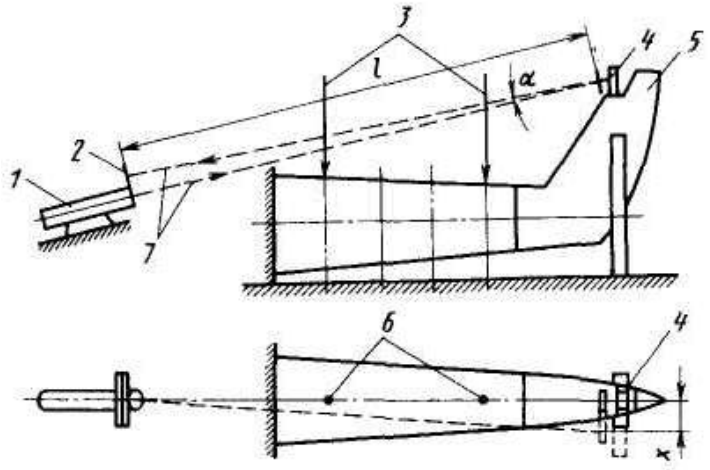


Fig. 15.12. Diagram of Helicopter Tail Rotor Pylon Shift Control:
1 – emitter of laser; 2 – screen; 3 – plumblines;
4 – mirror target mark; 5 – tail rotor pylon;
6 – point of reference; 7 – direct and reflected laser beam

This arrangement of measuring instruments allows to use geometric construction to find a mathematical relation to determine the linear and angular displacements of the tail rotor pylon of the helicopter. The basis for the reference measuring offset tail rotor pylon would be the laser beam axis.

Control of displacement of the tail rotor pylon at the tail gearbox mounting area after removing of bearings is performed in the following sequence:

1. Two plumblines 3 are lowered on the pylon point of reference 6.

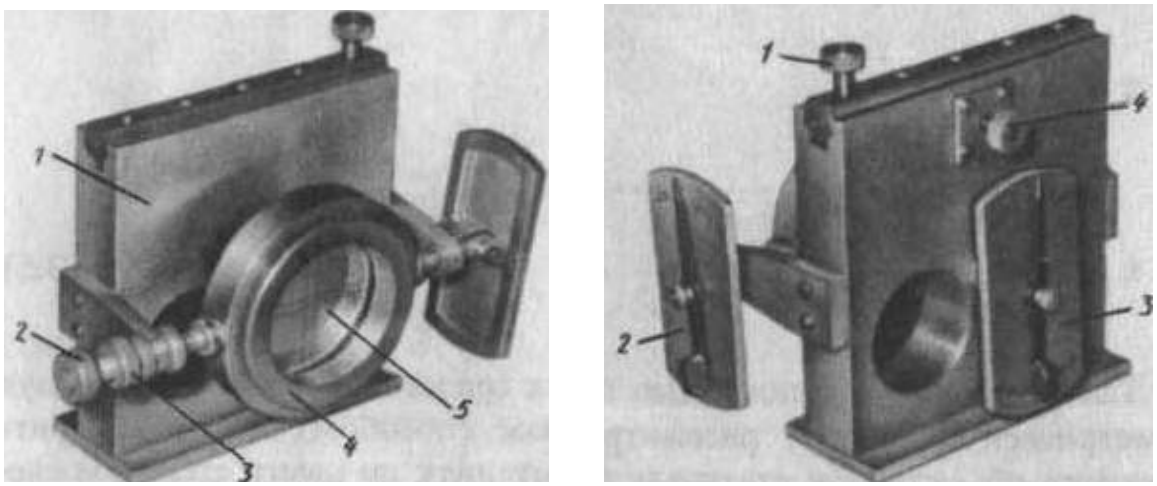
2. Mirror target mark 4 is set at the end of the tail rotor pylon 5 and fixed so that the mark center at its mirror is located on the center line of the tail boom pylon and the tail rotor pylon.

3. In the housing of the laser emitter 1, the screen 2 is set so that vertical line of the screen mark passes through the laser beam axis.

4. The housing 1 of a laser emitter is installed so that the laser beams are in a plane of plumblines 3 and alternately projected in mark centers of mirror target mark 4 and the screen 2.

5. After removing the bearings, fixing the tail rotor pylon on the jigs, the displacement of the target mark in the horizontal and vertical scales is measured at the mirror target mark 4 and the screen 2.

With a special device with a mirrored target mark (Fig. 15.13), which is mounted on an object to control the displacement, it is possible to measure linear displacement on the coordinate axes directly on the mirror mesh, and by gravity protractors – angular rotations about these axes.



a b

Fig. 15.13. Device with Mirror Target Mark and Gravity Protractor:
a – view from mirror side: 1 – housing; 2 – adjusting screw;
3 – locking nut; 4 – case of target mark; 5 – mirror with linear scale;
b – view from goniometers side: 1 – retaining screw; 2, 3 – gravity protractors;
4 –adjusting screw

This laser method of control is easier that optical, it allows you to measure not only linear, but also the angular displacement of the test object, and visual observation of the laser ray spot greatly simplifies the measurement process and increases productivity.

15.7. LASER METHODS OF ASSEMBLY AND CONTROL BASED ON COMPUTER INTEGRATED SYSTEMS

Computerization of engineering problems – one of the main ways to improve the competitiveness of products of any engineering enterprise, regardless of the type of production. The key to modern engineering production stage is the transition from a discrete to the electronic description of the product (EDP). EDP is a paperless basis of representation of the information model products, including all product data.

The enterprises of the aerospace industry use high-level systems Unigraphics, CATIA, CADDS-5, EUCLID, etc. as a basic means of computer integrated technologies.

The composition of these systems include the programs of constructing a mathematical model of the surface of the projected plane or helicopter, the development of

the internal layout of the compartments, the aggregates and airframe as a whole, maintaining project databases, provide the engineering process, the planning and scheduling of work and many others.

Construction of parts, workpieces, tools, jigs and fixtures based on solid modeling, development of drawing documentation, preparation of control programs for processing and control for CNC equipment – all these problems are solved with the use of a number of computer integrated *CAD / CAM*-systems.

As a result of creating the reference coordinate system of the visible laser rays in space a new way to solve the problem of basing aircraft and helicopter parts in the assembly is represented. The emergence of small-sized laser light diodes further enhances the advantages of the laser.

There are new standalone laser devices – trackers, that set the position of reference laser beams in a spherical coordinate system and at the same time perform the function of a laser range finder (Fig. 15.14, a).

One advantage of this method is that the assembly process is carried out in a continuous active control of the location of items collected (Fig. 15.14, b).

The non-contact nature of the control, a high level of discreteness laser points allows for both a measurement of geometric parameters of the object, and a comparison of the measured object with CAD-models, calculating the extent of deviation of each measured point (Fig. 15.14, c).



a b c

Fig. 15.14. Examples of Use of Laser Trackers
in Assembly and Control Aircraft Structures

To control the manufacturing of products and equipment commonly used are stationary coordinate measuring machines (CMMs). When measuring the large components, as well as when there is a necessity to carry out control of geometrical parameters of aggregates, compartments directly with the use of the jigs such machine utilization is impossible. For the purpose of effectively using of mobile CMMs, which are constructed as hinge-arm lever arrangement (Fig. 15.15, a).

The manipulator is fixed in a convenient place to work with a magnetic base, after which the operator manually brings the probe tip into contact with the surface. With sensors located in the joints of the manipulator, the computer system receives information about the spatial coordinates of the probe tip. Thus, readout coordinates of points of the surface being measured with the help of a computer or laptop deviations from the nominal size (when it is compared with computer model) or the actual geometric parameters of the product are calculated.

One of the world's leading manufacturers of portable CMM is a US company CIMCORE, producing three series of manipulators.

Models Stinger and INFINITE are designed to work in the contact measurement mode, and the SC INFINITE model is designed for use with laser scanning Perceptron ScanWorks system. The system is intended for fast control of product sizes (measurement speed – up to 30 thousand points / s), as well as for three-dimensional scanning of objects (Fig 15.15 b.).



a b

Fig. 15.15. Mobile CMM of Hinged-Lever Arrangement

Mobile CMM of INFINITE series provide the absolute measurement error throughout the volume of the working zone from 15 micrometer. The measurement distance from a fixed position is for different models from 1.2 to 3.6 m. In order to increase the measurement area special LeapFrog positioning system, SpaceLok and Gridlok are intended.

All models of manipulators work with Delcam software company – PowerINSPECT system designed specifically for use with various types of CMMs. When working with laser scanning head Perceptron the PowerINSPECT Point Cloud module, allowing to work with a large array of pixels, is used.

The system supports control of surfaces of products on the cross sections, provides a measure of standard geometric elements, deviations of the form and mutual arrangement of surfaces, and also allows exporting of points, wireframe models and surfaces, to prepare measurement reports.

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Chapter 16. DEVELOPMENT OF MAINTENANCE MANUAL ON TRANSMISSION (ON THE EXAMPLE OF HELICOPTER MI-26)

An important stage of the aviation vehicles development is the development of the technical documentation allowing it to implement safe service under defined conditions. In accordance with the requirements of the AP-29 the *Instruction on Airworthiness Maintenance* should be designed, which should be introduced as a branch of maintenance guide.

The maintenance guide of rotorcraft must include [1]:

- background information containing explanations of the rotorcraft design features;
- description of the rotorcraft structures, its systems and installations;
- the main guidance and background information describing control and operation of rotorcraft elements and systems;
- maintenance information on rotorcraft;
- information on how to find places of failures and damages with description of possible failure modes, methods of detection and corrective actions;
- information on the procedures and methods of removing and replacing the items or its components with all the necessary precautions;
- hatch and access panels layout for maintenance and repairs, and the information needed to provide access for inspection and examination in case of absence of viewing panels;
- the information needed to carry out the final works and protective treatment of the structure after checks and inspections;
- all data relating to fixing elements and units design, such as their marking, rejection guidelines and allowable torque values;
- list of the necessary special tools.

Let us consider the maintenance guide for helicopter Mi-26 (Fig.16.1).

This is a single-rotor helicopter with an anti-torque rotor, two gas turbine engines and tricycle landing gear. The body is made of aluminum-lithium alloys with wide application of steel, titanium alloys and composite materials.

Main rotor has 8 blades with a hinge mount and hydraulic dampers. The blades are rectangular in plan, they have a chord of 0.835 m and improved airfoils TSAGI D2 and P65 with relative thickness of 12 % at the root and 9 % at the end of the blade and moderate twist. The blades of mixed construction with tubular steel spar, faced with the CM, to which 26 sections of ribs are attached, as well as a shell of fiberglass and a honeycomb core. Blade nose is protected with anti-erosion cover of titanium alloy with the electrical anti-icing system.



Fig. 16.1. Helicopter MI-26

Anti-torque rotor has five blades of diameter 7.67 m. The fiberglass blades have rectangular shape in the plan.

The engine unit consists of two turboshafts GTE D-136, installed close on the top of the fuselage in isolated compartments with a titanium fire seal. Engine air intakes are equipped with dust protection devices and electrical anti-icing systems. Engine length is 3.75 m, width is 1.4 m, height is 1.13 m, dry weight – 1050 kg, exceeding power – 8.390 kW / 11.400 hp, takeoff output – 7.360 kW / 10.000 hp, maximum duration – 6.256 kW / 8500 hp, specific fuel consumption – 0.206 kg / hp / h.

Transmission consists of main gearbox VR-26 with two freewheel clutches and a main rotor brake, a tail drive shaft, an accessory gearbox and anti-torque gearbox. The main gearbox VR-26 has a cooling fan drive gear and oil systems installed on engines and helicopter units. The gearbox has a length of 2.5 m, a width of 1.95 m and a height of 3.02 m, the dry weight – 3640 kg.

Helicopters are serial produced since 1980 and has more than ten versions for both civilian and military applications.

Mi-26 Application Requirements

Size, (m)	
Main rotor diameter	32
helicopter length with driven propellers	40.03
helicopter width on outer landing gear wheels	8.15
rotor disc area (m)	810
Engines	2-CCD-E-136
takeoff rated power, (kW/ h. p)	2x7355 / 2x10000
Weights and loads (kg)	
gross take-off weight	56 000
standard weight	49 600
helicopter base weight	28 200
gross inboard	
payload	20 000
external load	14 900
internal fuel, (l)	12 000
Performance:	
maximum speed km/h	295
cruising speed, km/h	255
hover ceiling without ground effect, (m)	1800
absolute ceiling, (m)	4600
practical range, (km)	595
extra-long-range fuel tank flying range, (km)	2000

The maintenance guide consists of 7 books of the following context
Book 1 GENERAL INFORMATION ON HELICOPTER

6.00.00	Size and Space
7.00.00	Jack System
8.00.00	Leveling and Weighing
9.00.00	Towing and Taxiing
10.00.00	Parking and Tiedown
11.00.00	Captions and Placards
12.00.00	Servicing
20.00.00	Master Production Schedule

Book 2 FRAMEWORK (2 CHAPTERS)

Part 1

52.00.00	Entry Doors, Hatch Doors, Covers
53.00.00	Fuselage
55.00.00	Stabilizer
56.00.00	Canopy, Windows

Part 2

65.00.00	Main Rotor and Anti-Torque Propeller

Book 3. HELICOPTER SYSTEMS (4 CHAPTERS)

Part 1

21.00.00	Air Conditioning System
25.00.00	Utility Services.
26.00.00	Fire protection

Часть 2

28.00.00	Fuel System
29.00.00	Hydraulic Power System

Часть 3

30.00.00	Anti-Icing System
32.00.00	Landing Gear

Часть 4

36.00.00	Pneumatic System
148.00.00	Equipment Cooler System
155.00.00	Motor Fuel Consumer Distribution

Book 4. POWER UNIT (3 CHAPTERS)

Part 1

49.00.00	Airborne auxiliary power
71.00.00	Power unit

Part 2

72.00.00	Engine
73.00.00	Engine fuel system

75.00.00 Air bleed offtake system
76.00.00 Engine-control system
77.00.00 Engine instrument
79.00.00 Oil system
80.00.00 Starting system
Part 3
84.00.00 Transmission

Book 5. AVIONICS
23.00.00 Communications
110.00.00 Helicopter Pilotage Radio Equipment
113.00.00 Identification Warning and Response Radio Equipment
132.51.00 Airborne Television Equipment BTU-GB
133.30.00 Process Flow Diagram of Preparing, Examination and Terminative Works of Radio Equipment During Preparation of the Helicopter to Flight and Scheduled Operation. Test Equipment List.

Book 6. PROTECTION MEANS AND TRANSPORT EQUIPMENT
121.40.00 Pintle Mount
127.00.00 Means of Creating Clutter
129.00.00 Pyrotechnical Devices
130.00.00 Radiological Monitoring Aviation Assets
131.00.00 Waste Equipment
132.00.00 Aerial-Delivery System

Book 7. AIRCRAFT EQUIPMENT (2 CHAPTERS)
Part 1
24.00.00 Electrical System
33.00.00 Lighting
Part 2
31.00.00 Instrumentation System
34.00.00 Flight and Navigation Equipment
35.00.00 Oxygen Equipment
110.24.07 Radio Magnetic Indicator RMI-2
141.00.00 Piloting and facility units
142.00.00 Flight Data Recorder and Airborne Control Device

The third part of Book 4 – 84.00.00 – Transmission contains the essential data for the works on technical service and ground operations of the helicopter transmission and accessories, including the data on the description and the works, troubleshooting guidance and the transmission operation.

The transmission is intended for deceleration and torque transfer from two gas-turbine engines to the main rotor and anti-torque rotor, fan and accessories of the main gearbox. Transmission (Fig.16.2) consists of:

- main gearbox
- main rotor brake
- fan power shaft
- tail rotor drive shaft horizontal portion

- accessory gearbox
- tail rotor drive shaft inclined portion
- tail rotor drive shaft

The main gearbox is upside the fuselage central section and is mounted to the fuselage strong frames Nos 13 и 17 through the geared frame. The main rotor brake is mounted to the main gearbox wet sump tail part by pins.

Tail drive shaft horizontal portion is upside the fuselage central section and the tail beam. The drive shaft is mounted to the frames Nos 21a, 25, 30, 35, 40 to the fuselage central section and to frame No. 5 of the tail beam.

The accessory gearbox is mounted to the fin beam rib No. 2

The tail rotor drive shaft inclined part is in the fin upper part and is mounted to the accessory and tail gearboxes and to rib support No.8.

The tail gearbox is mounted to the fin beam upper part on the strong rib No. 14.

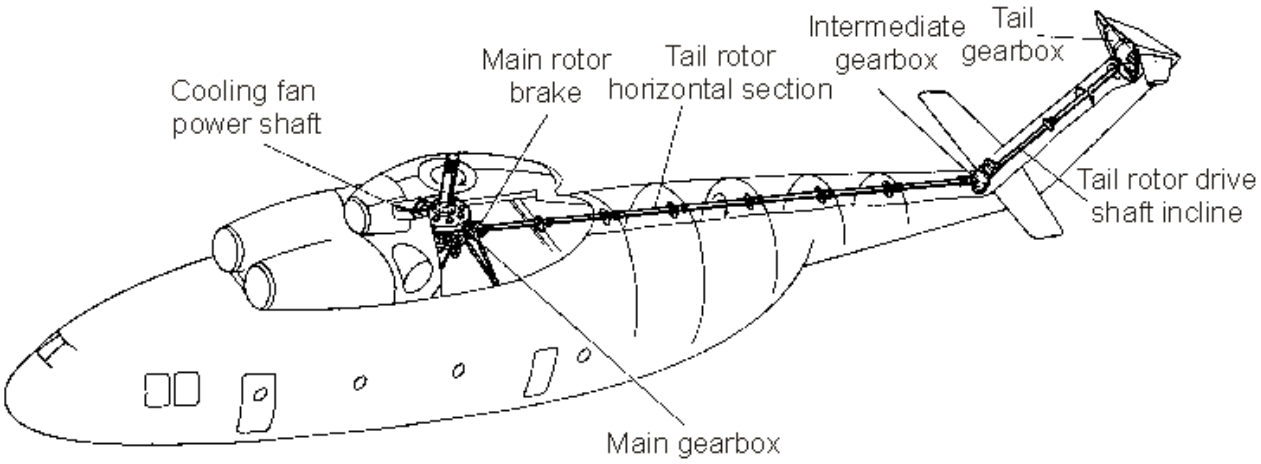


Fig. 16.2. Helicopter Appearance and Transmission

16.1. MAIN GEARBOX

The main gearbox (Fig. 16.3) is used for two gas turbine engines power adding, its transfer to the rotor mast and anti-torque rotor at appropriate spin speeds and provides a accessory assembly drive, located on the gearbox.

To make helicopter flight with one inoperative engine, as well as in the autorotation mode a gearbox has two freewheel clutches that automatically shut off one or both engines from a helicopter transmission. The gearbox is independent of the engine oil system in which the oil tank is the oil sump of the gearbox.

The main gearbox is located on the central part of the fuselage and is fixed to power fittings on the frames Nos 13 and 17 of the fuselage with undergear frame.

The main gearbox is a separate accessory that consists of bodies with mechanisms and an oil system.

The main gearbox consists of freewheel clutches, front and rear bevel gear speed gearboxes, tail shaft drive boxes, upper gearbox, fan drive, oil sump, oil system blocks and a safety cover.

The main gearbox mechanisms are designed as separate units that are interconnected with flanges and springs.

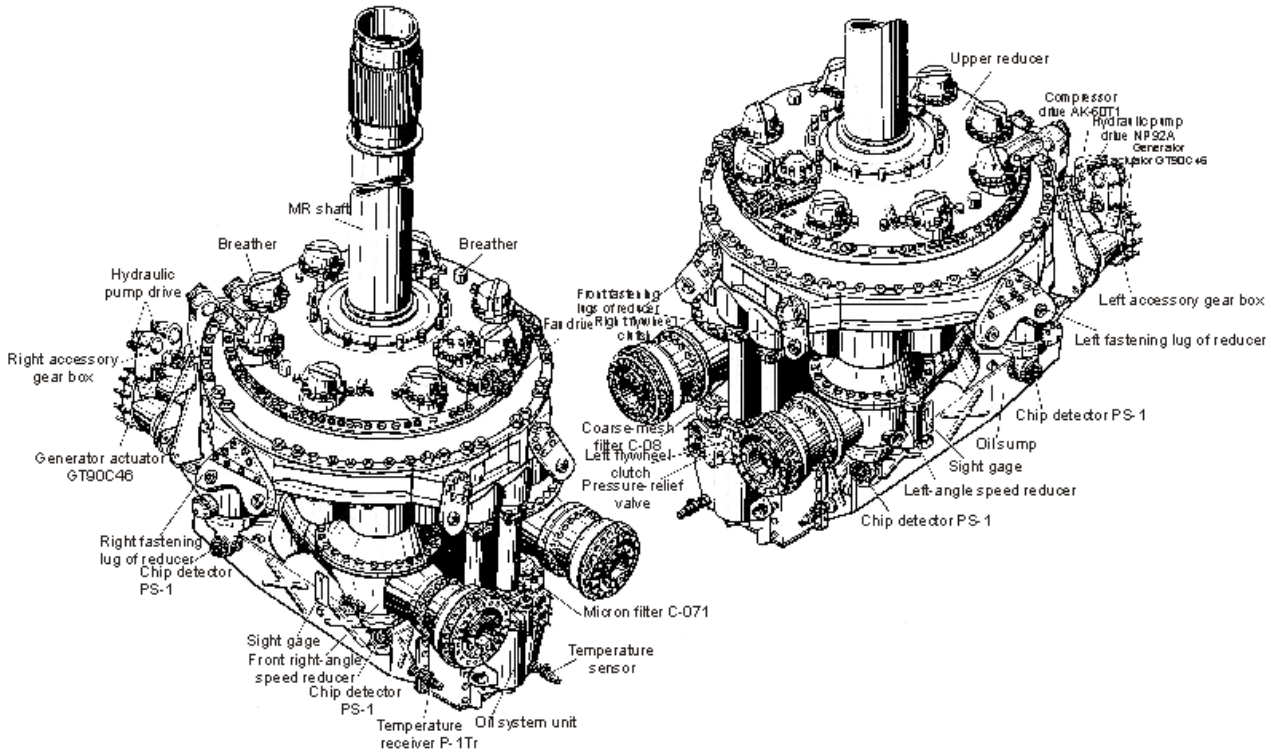


Fig.16.3. Main Gearbox. General View

Basic Technical Data

Gearbox designation VR-26

Rotor type ..... gearing
three-stage

first stage ..... bevel
spiral

second stage ..... cylindric
spur

third stage ..... cylindrical
helical

Nominal speed, r/min:

Input shafts 8300

Main rotor shaft 132.7

Gear ratio:

to main rotor shaft ..... 1/62.53=0.016

to tail rotor shaft ..... 1/3.1=0.323

to fan drive ..... 1/2.916=0.343

Main rotor oil system:

oil pump ..... geared with two main stages and two scavenge stages

The oil pressure behind the main stages:

in operating modes ..... 4.0 ± 0.5 kg/cm^2

shortly in working states ..... 2.5 kg/cm^2

in idle mode, not less than ..... 2 kg/cm^2

in a rotor warm up mode with oil from -15 °C temperature to recommended temperatures the accessible oil pressure..... up to 6 kg/cm^2

The oil temperature in rotor input

maximum, no more than ..... 90 °C

minimum in flight ..... 10 °C

recommended ..... 30 … 70 °C

minimum, at engine starting without heating..... - 40 °C

minimum for entering the higher modes during engines testing…. -15 °C

Rotor dimensional specifications:

height ..... 2995 mm

length..... 2505 mm

width ..... 2000 мм

Rotor mass (dry) ..... 3575 kg $\pm$ 2 %

The torque of two engines is delivered to the gear input through two plate clutches to compensate the inaccuracy of installation and relative movement during

operation of the engine and rotor. To make helicopter fly with one operating engine, as well as on the mode of self-rotation of the rotor freewheel clutches 1 (Fig. 16.4) are used that automatically disconnect one or two non-operating engines from the transmission. The torque from each engine is transmitted through the one-way clutch and then to the two bevel gear speed gearboxes (front and rear).

In bevel gear speed gearboxes the rotations are decreased (torque increases) and the rotation axis from a horizontal direction becomes vertical.

From the driven gears 3 of bevel gear speed gearboxes the torque is transmitted to four inputs of the upper gearbox, where the final reduction in speed occurs (torque increase). The torque from the four input gears 6 is transmitted to eight gears 7 mounted in the same shaft with the gears 5 and 9, which transmit torque to gears 12, 23 of the rotor shaft. From the gears 11 the torque is transmitted to the gear 10 of the fan drive shaft. From the lower pinion 23 torque is transmitted to gears 4, which transmit the moment to oil pumps. The torque from the engine through a freewheel clutch is transmitted to the gear 19 by splined shafts, which transfers it to the gear 20 of the tail rotor shaft and a gearbox. From the gear 20 on the one side gear 21 is transmitted, on the other – to the gear 18. The crown wheel 21 transmits the torque to the gear 22 of tail shaft drive. The gear 18 through 17 and 16 transmits the torque to gear 15 on the gearbox drive. The gear 15 transmits the torque to gears 14, which transmit the torque to the gears 13.

The Freewheel Clutch

The freewheel clutch (Fig. 16.5) is a mechanism that provides a torque transmission from the engine to the rotor and prevents the transmission of torque from the gearbox to the engine. A freewheel clutch is provided with rollers, the main operating elements of it, that during rotation of sprocket in operative direction are wedged between the shroud and wedge sprocket teeth.

When the shroud rotates at higher speed than the sprocket, the rollers come out of a jammed condition, a free rotation occurs with respect to the shaft. The shroud is connected with the flange by bolts on one side, on the other one – with the bearings.

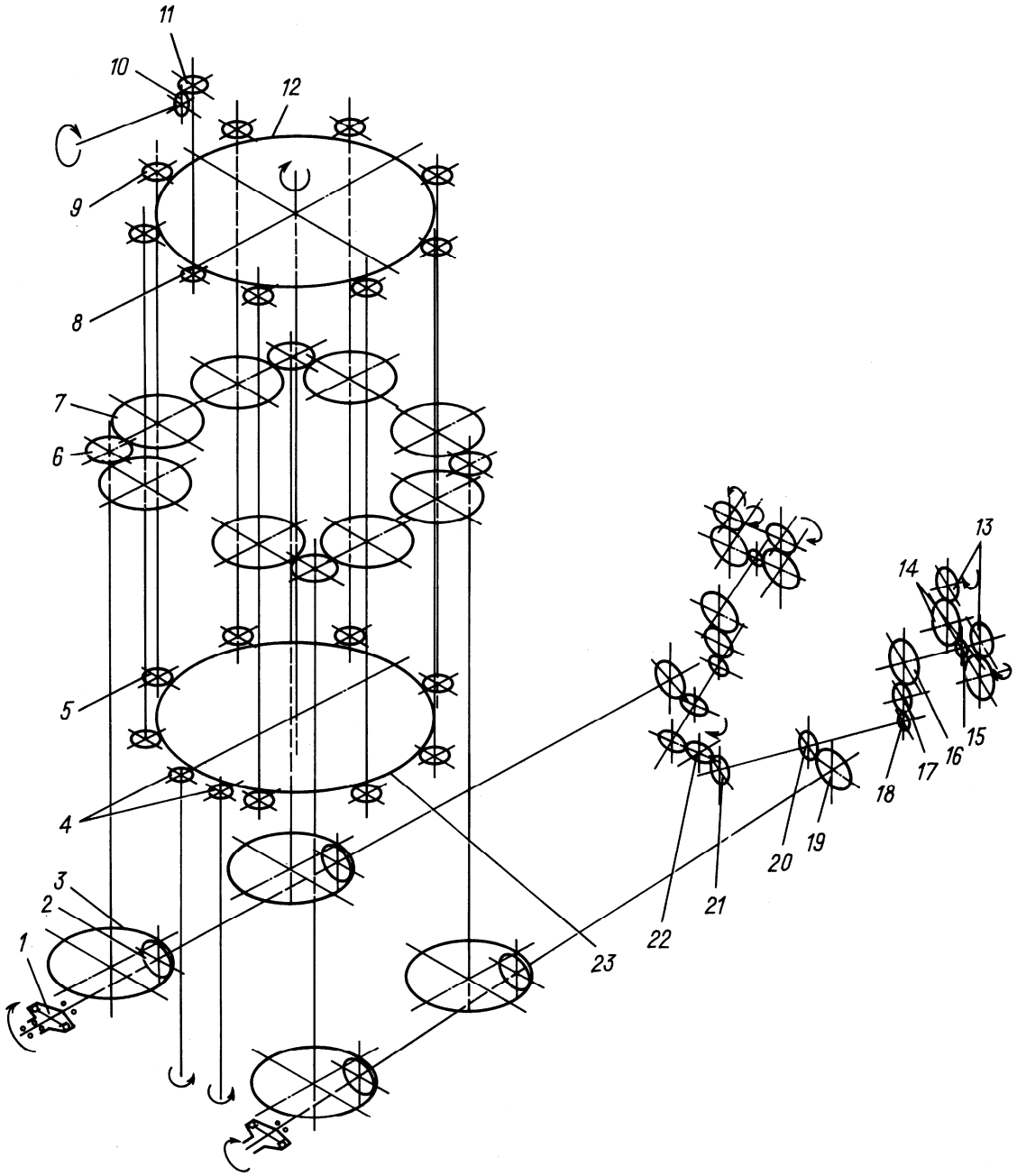


Fig. 16.4. Kinematic Scheme:

1 –	freewheel clutch;	4 – 21	gear;
2 –	driving bevel gear;	22 –	bevel gear;
3 –	driven bevel pinion;	23 –	gear;

Bevel gear speed gearboxes

Bevel gear speed gearboxes (Fig. 16.6) serve to transmit torque, coming from an engine via a freewheels to a tail rotor drive and the upper gearbox. In addition, in bevel gear speed gearboxes the number of rotations decrease (torque increases) and the rotation axis changes its direction – from a horizontal to a vertical one.

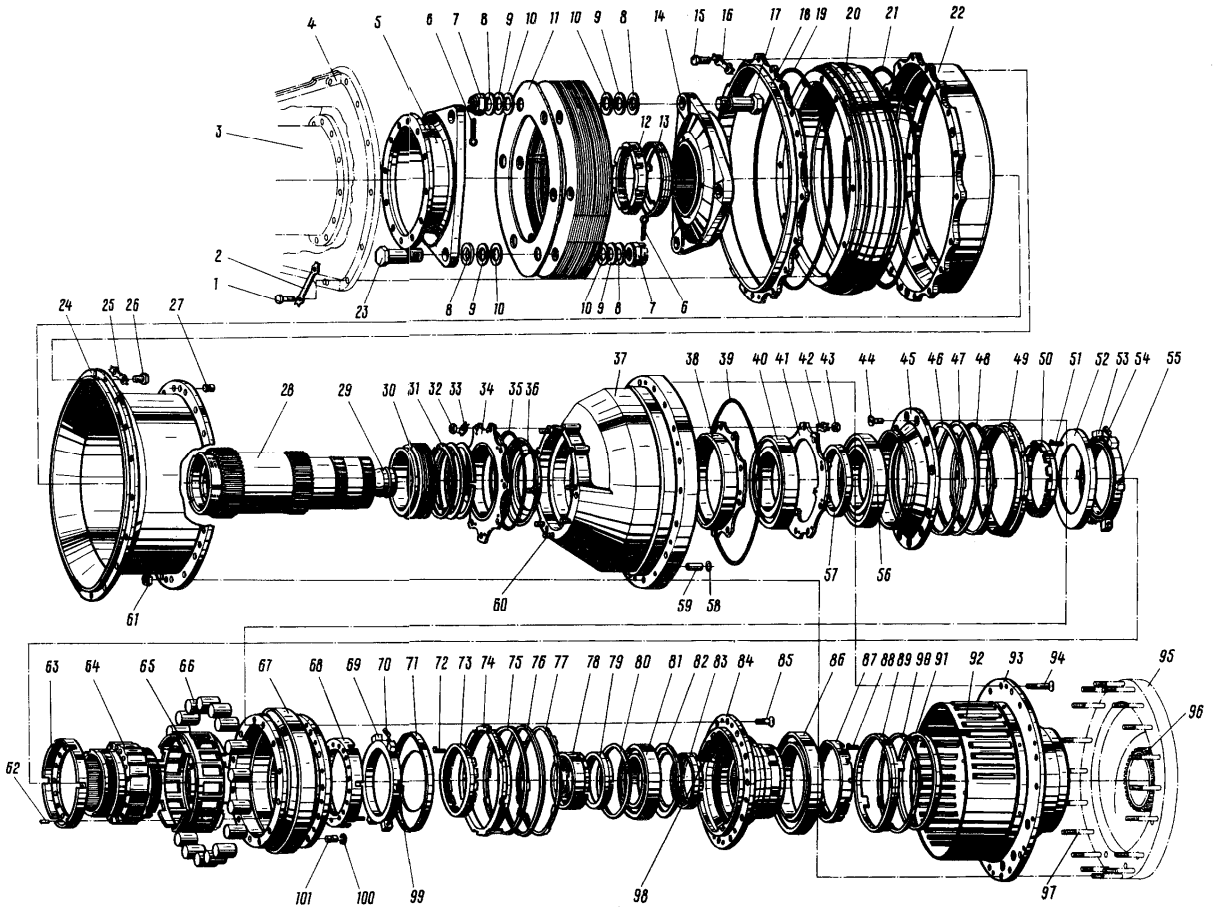


Fig. 16.5. Freewheel Clutch

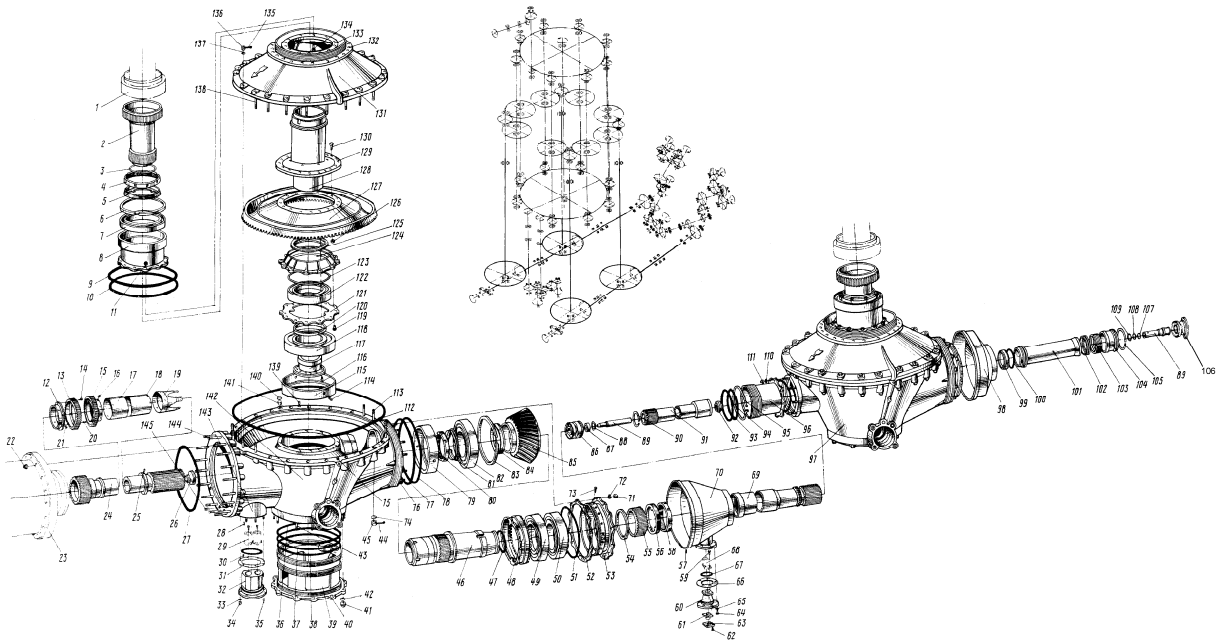


Fig. 16.6. Bevel Gear Speed Reducer

Filler Neck

Filler neck (Fig. 16.7.) is a casted pipe made of a magnesium alloy ML5. In the upper part a neck has a lug 17 with an opening to fasten a chain 18, retaining the rubber cover 1 in removed position. On the bottom of a filler neck is a flange 12 for attachment to the body of the left rear bevel gear. Filler cap is secured using washers 14, nuts 15 to pins 16 of left rear bevel gear speed reducer.

A pad 13, made of rubber IRP 1287 is installed between filler neck and a bevel gear speed reducer. On the top of the inner part a filler neck has fits for the cover 19 and oil filter. Oil filter consists of a insert 6, frame 9, grids 7, 10, bottom 8. The insert 6 is connected to the frame 9 and bottom 8. The bottomed frame is covered with nets 7, 10, made of brass net number 0.8. To prevent the ingress of dirt, moisture and other foreign particles a filler neck at the top of a fit shoulder for the cover 1, made of rubber H0-68-1. The cover 1 has a belt for attaching the chain 18.

Tail rotor shaft

Tail rotor shaft (Fig. 16.8) is used to transmit torque from two independent engines by free flows with the help of two springs 12, 199 to tail rotor drive 13. The drives from two motors are arranged symmetrically to the longitudinal axis of the gear unit and are identical in construction.

For a description of the depth and fullness of the unit structure consider the full description of the structure of the tail drive in the Manual.

Spring 199 connects with the driving gear (205) by splines. A driving gear (205) is made of steel 12X2N4A-Sh and installed in the housing (112) of a drive box. The housing 112 is made of magnesium alloy ML5. The sleeves, which are supports for the bearings of gears and shafts of 30KhGSA steel, are pressed in a body. Spring (199) made of steel 12Х2Н4А-Ш, outside, on both sides, is splined, and inside – seamed plug (196). The axial spring 199 is fixed by a spring 195 placed in housing (194) and retaining ring (193).

The pinion gear (205) is connected with spring via the internal splines (199). At seats of gears roller (202) is pressed from one hand which is fixed in a steel sleeve 203 by retaining ring (205). Roller bearing (202) through the retaining ring (201) is clamped by the nut (200). The nut (200) is wrapped with a torque wrench calibration of 60...80 kg • m and locked by retaining ring by putting the ring into the nut slots. On the other end of a gear 205 the thrust ring (206), a sleeve (209), ball bearing (211), adjustment ring (210), wafer ring (212), flow nozzle (213), roller bearing (214) and sleeve (215) are installed. The above-mentioned pack is clamped with a nut (218) through the retaining ring 217. The nut 218 is wrapped by torque wrench with a torque of 50...60 kg·m and locked by retaining ring by putting the ring into the nut slots. Between the sleeve (215) and a cover (219) adjustment ring (216) is installed.

The sleeves (209, 215), the adjusting rings 210, 216 and a flow nozzle 213 is attached to the cover 219 with eight pin screw nuts 207 wrapped by torque wrench with a torque of 0.9...1.1 kg·m and locked with laminar washers 208. The cover 219 is made of magnesium alloy ML5, has three grooves for rubber O-rings outside (220-222), and twelve mounting holes inside. The cover 219 is attached to the housing 112 of the box (224) drives with screw nuts wrapped by torque wrench with a torque of 0.9...1.1 kg·m and locked with laminar washers (223).

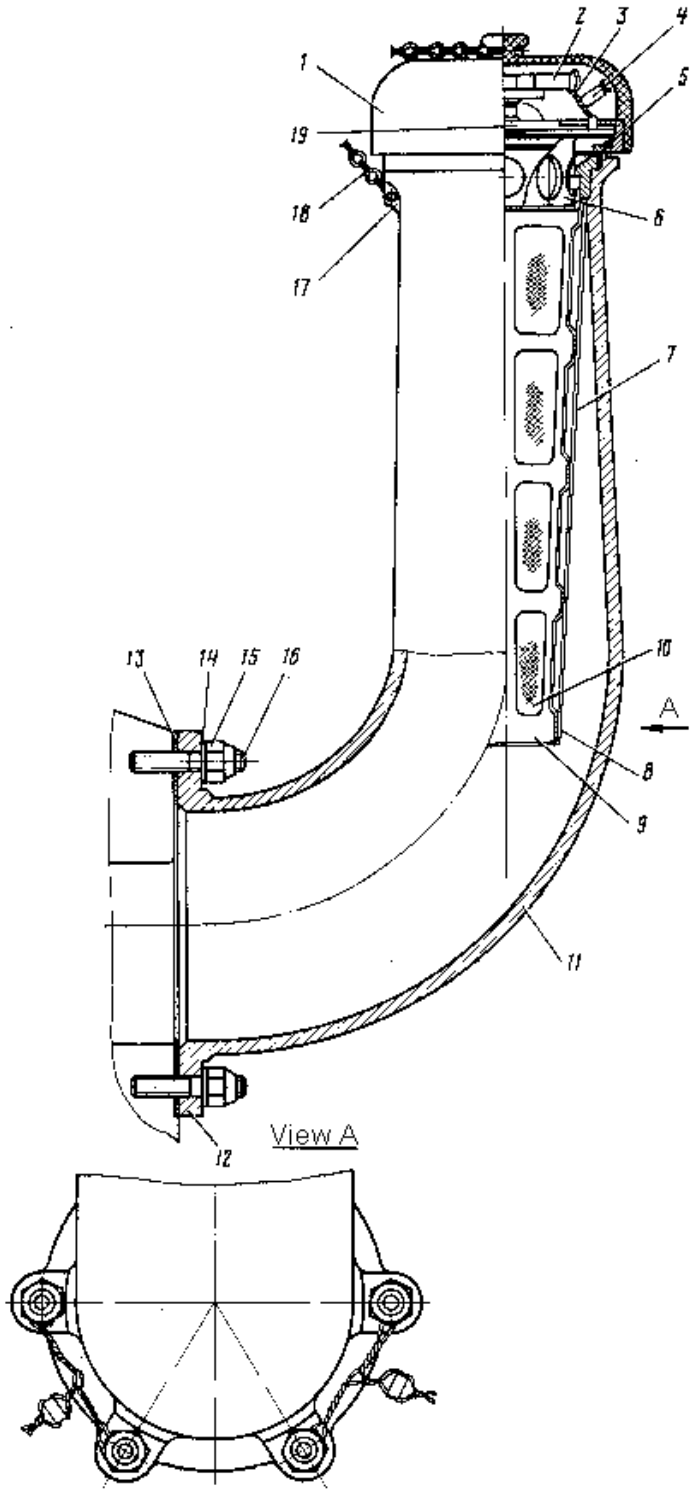


Fig. 16.7. Filler Neck:
1 – rubber cover; 2 – cover handle; 3 – copying wire;
4 – stopping; 5 – gasket; 6 – oil filter insert; 7 – oil filter net:
8 – oil filter base; 9 – oil filter structure; 10 – oil filter net;
11 – filler neck; 12 – flange; 13 – gasket; 14 – washer; 15 – screw nut;
16 – pin; 17 – lug; 18 – chain; 19 filler neck cover.

A driven gear 98 is mounted in the body 112, made of special steel 12Kh2N4A-Sh and having fit seats for bearings and gears at both ends. There are slots inside the gears 98. The gear 98 engages with the gear 205. The contact of gear 98 and 205 spot is regulated by the selection of shims 216, 89. At the fit seat of the gear 98 the gear 187 is installed with cotter pin 99 of unit drive, roller bearing 183, clamped by the retaining ring 181 with nut 180. Nut 180 is screwed by torque wrench to a torque of 60...80 kg·m and locked by retaining ring by putting the ring into the nut slots. Inside the gear 98 installed is hub 178, which is fixed by a retaining ring 101 and is the generator drive 170.

Steel sleeve 182 is pressed into the drive housing and constitutes a support roller 183, which is fixed with retaining ring 184 in the sleeve.

On the other end of the gear 98 thrust washer 97, a glass 96, ball bearing 95, wafer ring 92, adjusting ring 94, 89, flow nozzle 93, roller bearing 91, sleeve 90 are installed. This pack is tightened with nut 87 screwed with a torque wrench to 50...60 kg·m through the retaining ring 88, and locked by retaining ring 88 by putting the ring into the nut 87 slots. The pack is attached to the adapter 83 by eight pins tightened with screw nuts 197.

The nuts 197 are screwed by a torque wrench with a torque of 0.9...1.1 kg·m and locked by plate washers 198. The adapter 83 is made of magnesium alloy ML5. On the outer surface there are three grooves for rubber sealing rings 84, 82, 81, within the fixing holes in the housing 7 are installed.

The driving gear 67 is made of steel 12Kh2N4A-Sh and has two ends with fit seats, undercuts and threads, and splines inside. On the gear 67 thrust washer 66, a sleeve 63, ball bearing 62, adjustment ring 61, wafer ring 60, flow nozzle 59, roller bearing 58, a sleeve 57, adjustment ring 56 are installed. The pack is clamped through a retainer ring 55 by a nut 54 screwed with torque wrench to a torque of 50...60 kg•m locked by retaining ring 55 by putting the ring into the nut slots.

On the other end of the gear 67 a roller bearing 70 is mounted, which through a lock ring 71 is clamped with nut 72 screwed with torque wrench to a torque of 60...80 kg·m and locked by a retaining ring by putting the ring into the nut slots. Roller bearing outer race 70 rests on a steel glass 73, pressed into the housing 7. The bearing is fixed in the sleeve with retaining ring 69.

Parts, located between the sleeves 57 and 63, are set with sleeves and a gear on the adapter 83 with eight pins and screwed with screw nuts 65 by a torque wrench with a torque of 0.9...1.1 kg·m. Nuts are locked with plate washers 64.

A driven gear 98 and driving gear 67 are connected with the spring 80 through spline coupling at both ends. Spring 80 in gear 98 is fixed by a locking ring 100 and in the gear 67 – with a retaining ring 68. Spring is made of steel 40KhN2MA-Sh.

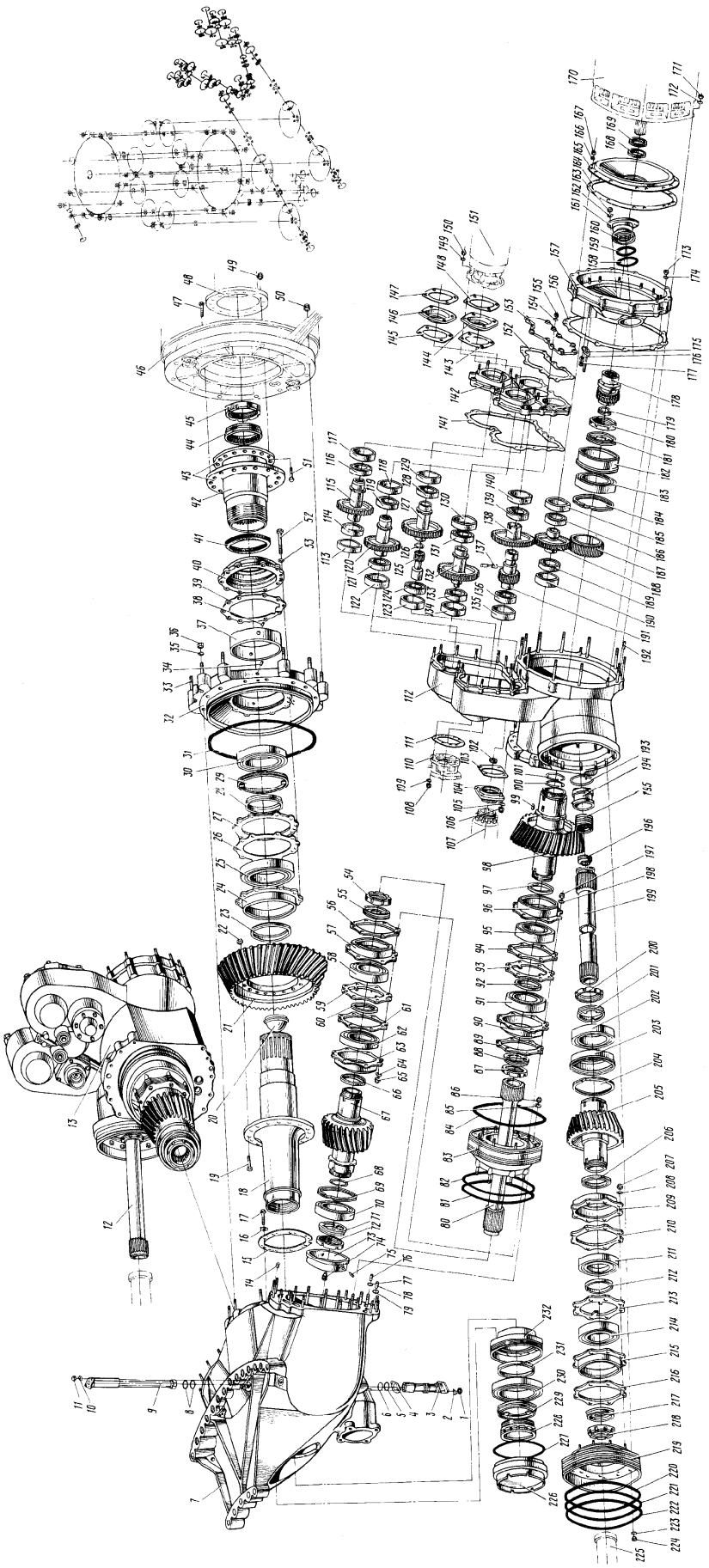


Fig. 16.8. Tail Rotor Drive

In Fig. 16.8 the following elements are marked by figures:

1 – screw nut;
2 – washer;
3 – hub;
4 – pad;
5 – sealing ring;
6 – sealing ring;
7 – tail rotor drive spring drive body;
8 – sealing rings;
9 – flow nozzle;
10 – washer;
11 – screw nut;
12 – spring;
13 – tail rotor drive spring;
14 – hub;
15 – adjustment ring;
16 – washer;
17 – bolt
18 – drive spring;
19 – bolt;
20 – bank;
21 – follower;
22 – screw nut;
23 – wafer ring;
24 – sleeve;
25 – ball bearing;
26 – adjustment ring;
27 – wafer ring;
28 – thrust hub;
29 – flow nozzle;
30 – roller bearing;
31 – sealing ring;
32 – cover;
33 – pin;
34 – bank;
35 – washer;
36 – screw nut;
37 – sleeve;
38 – lock pin;
39 – pad;
40 – cover;
41 – сальник;
42 – flange adaptor;
43 – fixing hole;
44 – retainer ring;
45 – screw nut;
46 – main rotor brake;
47 – bolt;
48 – tail drive spring flange adaptor;
49 – screw nut;
50 – screw nut;
51 – bolt;
52 – bolt;
53 – washer;
54 – screw nut;
55 – retainer ring;
56 – adjustment ring;
57 – sleeve;
58 – roller bearing;
59 – flow nozzle;
60 – wafer ring;
61 – adjustment ring;
62 – ball bearing;
63 – sleeve;
64 – laminar washer;
65 – screw nut;
66 – thrust washer;
67 – leading gear;
68 – retainer ring;
69 – retainer ring;
70 – roller bearing;
71 – retainer ring;
72 – screw nut;
73 – sleeve;
74 – hub;
75 – flow nozzle;
76 – hub;
77 – sealing ring;
78 – hub;
79 – sealing ring;
80 – spring;
81 – sealing ring;
82 – sealing ring;
83 – adapter;
84 – sealing ring;
85 – washer;
86 – screw nut;
87 – screw nut;
88 – retainer ring;
89 – adjustment ring;
90 – sleeve;
91 – roller bearing;
92 – wafer ring;
93 – flow nozzle;
94 – adjustment ring;
95 – ball bearing;
96 – sleeve;
97 – thrust washer;
98 – follower;
99 – cotter;
100 – retainer ring;
101 – retainer ring;
102 – sleeve-type seal;
103 – pad;
104 – flange adaptor;
105 – washer;
106 – screw nut;
107 – tachometer transmitter
108 – screw nut;
109 – washer;
100 – compressor;
111 – pad;
112 – gearbox body
113 – sleeve;
114 – ball bearing;
115 – gear;
116 – ball bearing;
117 – sleeve;
118 – sleeve;
119 – ball bearing;
120 – gear;
121 – ball bearing;
122 – sleeve;
123 – sleeve;
124 – ball bearing;
125 – spring;
126 – retainer ring;
127 – gear;
128 – ball bearing;
129 – sleeve;
130 – sleeve;
131 – ball bearing;
132 – gear;
133 – ball bearing;
134 – sleeve;
135 – sleeve;
136 – ball bearing;
137 – lock pin;
138 – gear;

139 – ball bearing;
140 – sleeve;
141 – pad;
142 – box body cover;
143 – pad;
144 – cover;
145 – pad;
146 – cover;
147 – pad;
148 – pad;
149 – washer;
150 – screw nut;
151 – hydraulic pump;
152 – pad;
153 – cover;
154 – washer;
155 – screw nut;
156 – pad;
157 – left cover;
158 – sealing ring;
159 – sealing ring;
160 – flow nozzle;
161 – flow nozzle;
162 – washer;
163 – screw nut;
164 – pad;
165 – cover;
166 – washer;
167 – screw nut;
168 – sleeve-type seal;
169 – sleeve-type seal;
170 – generator GT-90C46;
171 – screw nut;
172 – washer;
173 – screw nut;
174 – washer;
175 – flow nozzle;
176 – washer;
177 – screw nut;
178 – hub;
179 – retainer ring;
180 – screw nut;
181 – retainer ring;
182 – sleeve;
183 – roller bearing;
184 – retainer ring;
185 – sleeve;
186 – ball bearing;
187 – gear;
188 – gear;
189 – ball bearing;
190 – sleeve;
191 – gear;
192 – hub;
193 – retainer ring;
194 – spring body;
195 – spring group;
196 – bank;
197 – screw nut;
198 – laminar washer;
199 – spring;
200 – screw nut;
201 – retainer ring;
202 – roller bearing;
203 – sleeve;
204 – retainer ring;
205 – driving gear;
206 – thrust washer;
207 – screw nut;
208 – laminar washer;
209 – sleeve;
210 – adjustment ring;
211 – ball bearing;
212 – wafer ring;
213 – flow nozzle;
214 – roller bearing;
215 – sleeve;
216 – adjustment ring;
217 – retainer ring;
218 – screw nut;
219 – cover;
220 – sealing ring;
221 – sealing ring;
222 – sealing ring;
223 – laminar washer;
224 – screw nut;
225 – spring;
226 – screw nut;
227 – pad;
228 – screw nut;
229 – retainer ring;
230 – roller bearing;
231 – wafer ring;
232 – sleeve

The driven gear 21 is engaged with the driving gear 67. Gear 21 is made of special steel 12Kh2N4A-Sh and is attached to the shafts with 18 screw nuts 19 and 22, screwed with torque wrench with torque 8...10 kg • m and locked by putting bolt into nut. Shaft 18 is made of l2Kh2N4A-Sh steel and has a thread, splines, fit seats for bearings and inside expanded plug 20.

At one end of a shaft 18 a wafer ring 231, roller bearing 230, retainer ring 229 are set. These parts 228 are clamped with nut screwed by torque wrench to a torque of 80 ... 100 kg • m and locked by retaining ring by putting the ring into the nut slots. Pack is screwed with nut 226, screwed in a sleeve 232. Sleeve 232 is secured to the housing 7 by means of the adjusting ring 15 with bolts 17. Bolts are tightened by washers 16. To protect from the oil leakage between the nut 226 and sleeve 232 pad 227 is installed. On the other end ball bearing 25 which rests on the sleeve 24 with

outer race is mounted on the shaft 18, a spacer 28, nozzle 29, roller bearing 30, adjustment ring 26, the wafer rings 23, 27. Roller bearing 30 outer race rests on the sleeve 37 pressed into the cover 32 and fixed lock pins 38.

Cover is made of magnesium alloy ML5, it has fourteen holes for fastening to the body 7 and ten pins for mounting the main rotor brake support 46.

Upper gearbox

The upper gearbox – a gear unit having two reduction stages is the main power unit. The upper gearbox body consists of an upper cover 70, upper body 114, lower body 39, bottom cover 7. The upper gear body is made of aluminum alloy AK4-1.

Fan drive

Fan drive serves to transmit torque from main gearbox to the fan drive shaft. A fan drive is located in the front portion of the upper gear cover.

The description and operation of an oil sump, oil system blocks, oil pump, coarse and fine filters are described further in the Manual text.

The following mandatory sections of the Manual are sections devoted to troubleshooting, maintenance services, rules of storage and transportation. Consider its content.

16.1.1. Main Gearbox – Troubleshooting

1. Oil leaking on the gearbox body flange connections or seals of drive units.
   1.1. Possible causes – wear of seals during operation.
   1.2. Search of defective item.
   Oil leakage is detected visually with inspections of the main rotor.
   1.3. Troubleshooting – main gearbox should be sent in for repair.

2. Cracks on the brackets of the gearbox frame mounting and the main gearbox housing
   2.1 Possible causes:
      1) manufacturing defects;
      2) exceeding the permissible operating loads.
   2.2 Detection of a faulty element.

   Cracks are detected by visual inspection. Places with suspected crack are inspected with a magnifying glass.

   1.4. Troubleshooting.

   The main gearbox should be repaired.

16.1.2. Main gearbox VR-26 – Maintenance Practices

The maintenance services include the following task cards:

1. Filling the main gearbox with oil.
2. Draining the oil from the main gearbox.
3. Removal and installation of the main gearbox.
4. Inspection of the main gearbox.
5. Cleaning of the main gearbox.

16.1.3. Main Gearbox VR-26 – Storage Rules

1. Main requirements

Permission is granted to store the main gearbox in its original packaging in fundamental unheated warehouses or under a canopy with a mandatory protection from rain and corrosive environments.

The main gearbox storage platform level under a canopy must be above ground level not less than by 200 ... 300 mm.

2. Conservation and depreservation of the main gearbox

2.1. Conservation of the main gearbox:

1) conserve the oil system of the main gearbox. Remove the gearbox from the helicopter;
2) clean the main gearbox;
3) apply lubricant K-17 to all surfaces not having paint coating;
4) wrap gearbox in a waxed paper, pack in a plastic bag;
5) pack the gearbox unit in a container, fulfill documentation for work done.

2.2. Depreservation of the main gearbox:

1) unpack the gearbox, clear it off conservation grease. Set the gearbox on the helicopter.
2) to depreserve the oil system of the main gearbox.

For preservation and depreservation, it is necessary to have: a hair brush, grease K-17, waxed paper, plastic film.

For example, Table. 16.1 shows a task card of the works on the oil drain from the main gearbox.

Table 16.1

Task Card

Name of Operation: Draining Oil From Main Gear		
Operations and Technical Requirements (TR)	Corrective Actions at Deviations from TR	Checked by
I. Remove the lower chip detector, for that unlock, press and turn it counter-clockwise and remove from the main gearbox housing. Avoid contact with oil on the parts of construction and wiring. Use a pan. 2. Set the hose end into the chip detector seat to drain the oil from the main gearbox. 3. Lower the free end of the hose in the tank for the oil collection and drain the oil from the main gearbox. 4. Pull out the hose tip from the chip detector seat to drain the oil. 5. Refit the chip detector, for that insert the cover into the seat, press and turn it clockwise until it is locked. 6. Unlock the chip detector. NOTE. To reduce the drainage time: a) pour hot oil immediately after switching off the engine; b) use another lower chip detector by doing work of para. 1-6.		

Test Equipment (TE)	Tools and Fixtures	Consumables
	Combination pliers Hose for oil drain from the main gearbox 90-9960-00 Drain tank Pan for oil drainage	Safety wire KO 0.8

16.2. GEARBOX FRAME

16.2.1. Description and Operation

The gearbox frame (Fig. 16.9) is used for fixing the main gearbox on the helicopter. It takes static and variable dynamic loads encountered in flight. The frame consists of eight struts 7, 9, 10, 11, 17, 18, 32, 33 that are secured to four foots 35, two foots 34 of the gearbox and to strong frames 13 and 17 of the central part of the fuselage. The front 9, 10 and rear 18, 32 struts are secured to strong frames with bolts 8 which are fixed with nuts 1. Two spherical washers 6 and 5 are located between the nut and the frame unit. Nuts 1 are screwed with a torque wrench with a moment of 280+20 kg • m. After installing the frame on the helicopter in 5 hours check the nuts 1 tightening moment. Further check of nuts 1 tightening moment is carried out in 100 flight hours. Struts 18, 32 are secured to the bolts 8 with the eccentric bolt 24, fixed with a special washer 23 against rotation. Bolt is tightened with nut 19, screwed with a torque wrench with a moment of 8 ± 2 kg • m and locked with a pin 20, through washer 21. Struts 9, 10 are secured with a bolt 12 to the bolts and clamped via a nut 13, screwed with a torque wrench with the moment 8 + 2 kg • m and locked with cotter pin 14 through washer 15. The side front 7, 11 and rear side 17, 33 struts are secured to the fuselage joints similar for bolt 8. The side front and rear side struts are secured to feet 34 of the gearbox by bolts 31 tightened through the washers 27 with nuts 26 by torque wrench with a moment of 8 ± 2 kg • m locked with cotter pins 25. Teflon pads 29, 30 are put between the foot and eyelet of struts.

16.2.2. Troubleshooting

1. Cracks at welded joints, junction holes or permanent deformation of struts

1.1. Possible causes.

Production and operational defects (loosening the nuts 1, nicks, teasers).

1.2. Detection of a defective item – defects are detected visually during inspections.

1.3. Troubleshooting - if there is a defect reduction gearing frame replace.

ATTENTION! DO NOT CHANGE INDIVIDUAL STRUTS.

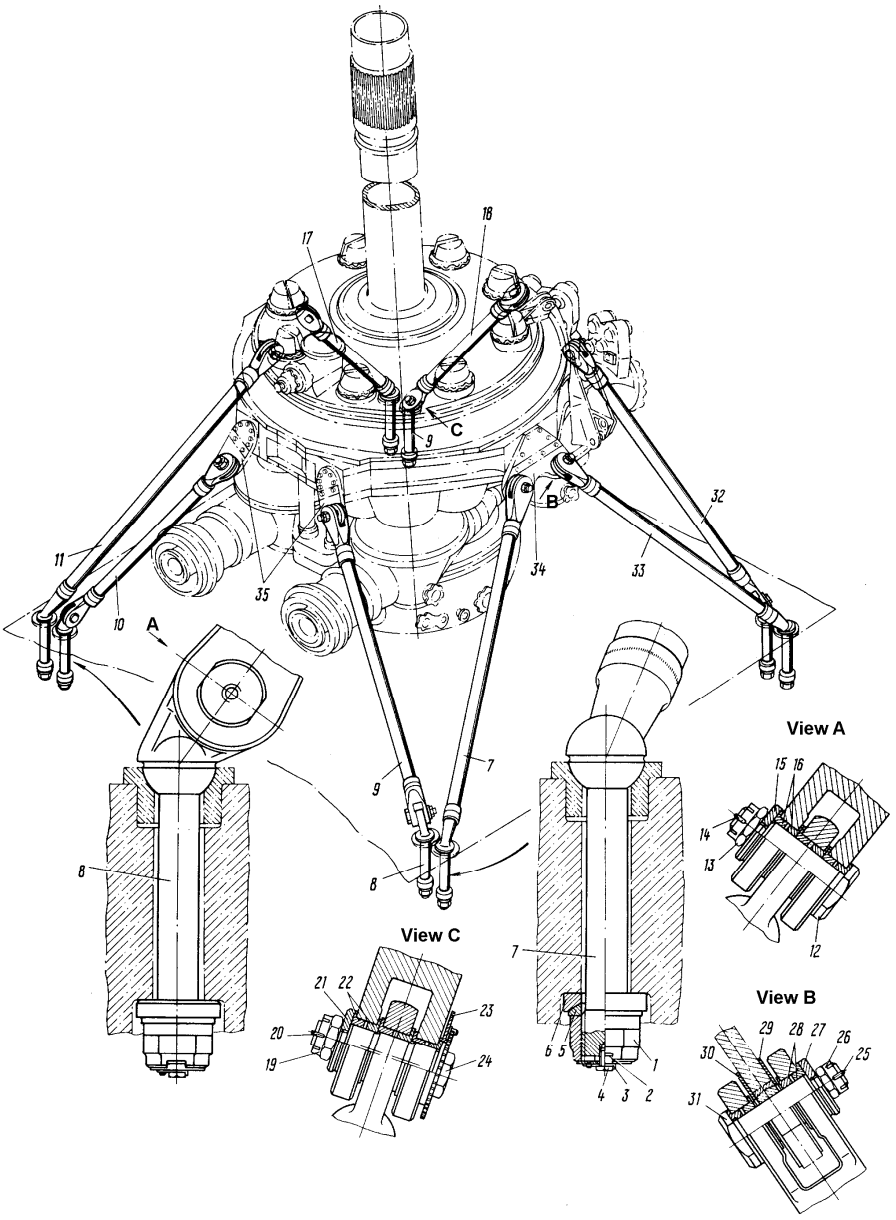


Fig. 16.9. Gear Rack:

1 – nut;
2 – plate nut;
3 – plate nut;
4 – bolt;
5 – spherical washer;
6 – spherical washer;
7 – lateral brace;
8 – bolt;
9 – front brace;
10 – front brace;
11 – lateral brace;
12 – bolt;
13 – nut;
14 – pin;
15 – shim;
16 – bush;
17 – side rear brace;
18 – rear brace;
19 – nut;
20 – pin;
21 – shim;
22 – bush;
23 – special shim;
24 – eccentric bolt;
25 – cotter pin
25 – nut;
26 – shim;
27 – bush;
28 – pad;
29 – pad;
30 – bolt;
31 – rear brace;
32 – lateral brace;
33 – gearbox foot;
34 – gearbox foot

16.2.3. Oil System of Main Gearbox VR-26 – Description and Operation

Main part

The main gearbox has a forced lubrication system. Gearbox oil system is designed for the lubrication of bearings, gears and cooling the working units and components. Oil gear system (Fig. 16.10) consists of two parts – external and internal. The outer part of the oil system is designed to cool the hot oil in the air-water heat exchangers. The inner part of the oil system is designed to lubricate and cool the gearbox components and parts. Air-oil heat exchanger is attached to the frame with bolts. The frame is attached to the bonnet frame. Gear unit with air oil heat exchangers are connected with aluminum pipes.

Description

The internal oil system of the main gearbox consists of a gearbox sump 37, functioning as the oil tank and a "hot" compartment, oil system block 45, functioning as a "cold" compartment, which houses the oil pump, having two pressure 48 and two scavenge 47 stages, fine filter 50 with by-pass valve 53, relief valve 52, pressure sensor 25 with oil separator, pressure switch 26 with oil separator pressure, oil jets 3-7, 10, 12-15, 17, 18, 21-24, 27-36, 38-40, 42, 54, 56, jet orifice 16, 19, 41, connected by channels with the pressure stages of oil pump, breathers 8, 11, magnetic plugs, chip detectors 44 and temperature sensors 43.

External oil system of the main gearbox (see. Fig. 16.10) consists of four air-oil heat exchangers 1 with thermoreducing valve 5991T, temperature receiver 46, coarse mesh filter 49 and oil pipe 2 and 51.

16.2.4. Oil System of Main Gearbox – Troubleshooting

1. The oil temperature in the oil system of the main gearbox is above the allowable

1.1. Possible causes:

1) insufficient amount of oil in the oil system;
2) blockage of cells of air-oil heat exchanger;
3) failure of the thermo-valve of air-oil heat exchanger;
4) failure of the alarm system.

1.2. Detection of a defective item:

1) on the oil sight glass make sure in sufficient oil quantity in the main gearbox oil system;
2) examine the flow sections of the cooling air in the oil cooler cells, make sure in absence of contaminants;
3) replace the thermostatic valve or air-water heat exchanger;
4) check the serviceability of the sensor and feed circuits of oil high temperature indication in the main gearbox.

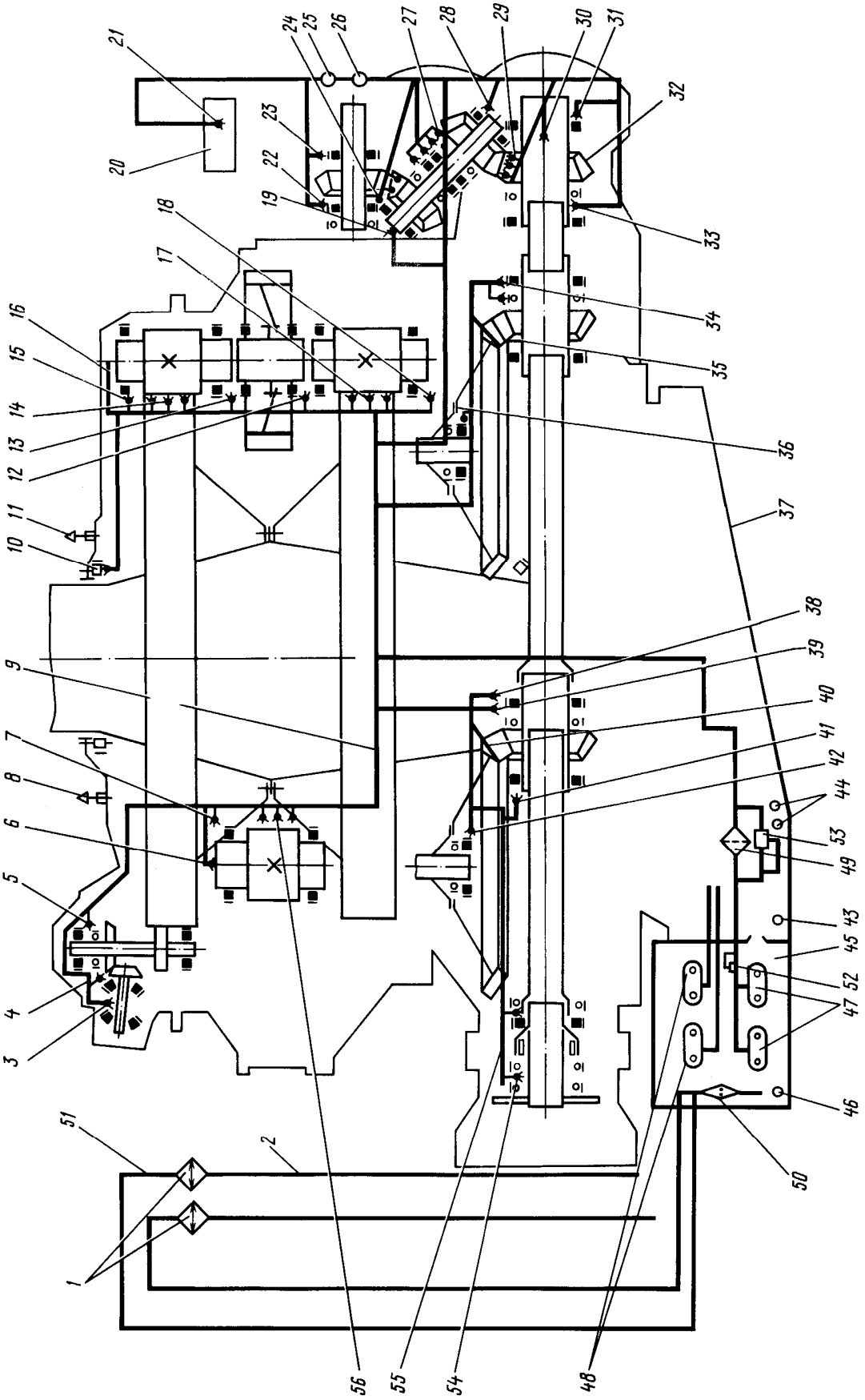


Fig. 16.10. Oil System of Main Gearbox

Fig. 16.10 shows next elements which are indicated under numbers:

1 – air-oil heat exchanger;
2 – oil pipeline;
3–7 – jet;
8 – breather;
9 – ring collector;
10 – jet;
11 – breather;
12–15 – jet;
16 – jet orifice;
17, 18 – jet;
19 – jet orifice;
20 – drive box;
21–24 – jet;
25 – pressure sensor with oil separator
26 – pressure switch with oil separator;
27–36 – jet;
37 – gearbox sump;
38–40 – jet;
41 – jet orifice;
42 – jet;
43 – temperature sensor;
44 – chip detector;
45 – oil system block (cold compartment)
46 – the temperature receiver;
47 – oil pump scavenge stage;
48 – oil pump pressure stage;
49 – coarse filter;
50 – fine filter;
51 – oil pipeline;
52 – reducing valve;
53 – bypass valve;
54 – jet;
55 – oil channel;
56 – jet.

2. Leaking oil in the nipple or hose connections

2.1. Possible causes:
1) loosening of the joint;
2) damage to the attaching parts.

2.2. Detection of the defective item.
Faulty connection is detected by the presence of traces of oil leaks.

2.3. Troubleshooting: Tighten loose joint and lock.

If the leak is not eliminated by tightening, disassemble the joint, make sure of serviceability of the attaching parts. Replace the defected attaching parts.

3. Sampling valve after sampling does not close (leaks).
3.1. Possible causes:
1) solid particles ingress in valve seat;
2) the weakening of the valve spring.
3.2. Detection of the defected item:
The defect is detected by the presence of traces of oil leaks.
3.3. Debug:
1) to drain a small amount of oil in the sampler to flush foreign particles out of the valve seat;
2) when the spring is loosened replace the sampling valve.

4. The presence of metal chips in the oil

4.1. Possible causes:
1) increased wear of gearbox components;

2) defective oil chip indication system.

4.2. Detection of the defected item:

1) when triggered indicator about the appearance of the chips in the oil make sure in serviceability of indication system;
2) the chip is detected during the inspection of filters, magnetic plugs and chip sensors installed in the oil system of the main gearbox.

4.3. Debug

When chips are detected the issue of further operation of the main gearbox is solved with Supplier representative.

16.2.5. Oil System of Main Gearbox – Maintenance Practices

This Manual includes the following task cards:

1. Removal and installation of thermoreductive valve.
2. Removal and installation of pipelines of the main gearbox oil system.
3. Removal and installation of fillers.
4. Removal and installation of air-oil heat exchangers.
5. Inspection of the main gearbox oil system.
6. Inspect the oil fine and coarse filters of oil system.
7. Check the amount of oil in the main gearbox oil system.

16.3 INTERMEDIATE GEARBOX

The intermediate gearbox is intended for change of the direction of tail shaft axis on a corner of 44° and torque transmission from main gearbox to tail gearbox.

Gearing is carried out without change of speed. The intermediate gearbox fastens to rib No. 2 of an inclined beam by means of feet with four bolts. Greasing of the rubbing parts and their cooling is provided with special oil system of a gearbox.

The torque coming from the main gearbox through horizontal part of a tail shaft comes to the spline flange 5 of pinion gear mounted on vents of a shaft of the leading conic gear wheel 6. From flange 5 torque is transferred to the leading shaft 4 of gear wheel 6, engaged with driven conic gear wheel 15 which transfers a torque to the driven shaft 13 of gear wheel 15. The driven shaft transfers a torque on one hand to a spline flange 11 of the driven gear, connected to inclined part of a tail shaft, on the other hand – to the leading gear wheel 14 of oil pump drive which transfers a torque to the driven gear wheel 16 of the oil pump drive. The gear wheel 16 is executed as a part of the

driven shaft 17 of the oil pump drive. The shaft 17 transfers a torque to the leading shaft 3 of oil pump which is connected to the leading gear wheels 2 of oil pump and through them transfers a torque to the driven gear wheels 1 of oil pump. From a shaft 13 of gear wheel 15 the torque is transferred to the leading gear wheel 12 of the fan drive which through an intermediate gear wheel 7 transfers the moment to the leading gear wheel 8 of driving shaft 9 of the fan drive and with it on a wheel 10 of the fan. The fan wheel forces air, pumps it via oil heaters, cooling oil of intermediate and tail gearboxes.

The gearbox consists of a forward cover of a case (assembled with shaft of the leading gear wheel), a case, a back cover of a case (assembled with the driven gear wheel) and the fan, the pallet and the air duct. The kinematic scheme of a gearbox is shown in Fig. 16.11.

For an example the design of a forward cover of a case assembled with shaft of the leading gear wheel is represented in Fig. 16.12.

Gearbox basic technical data	
Gear ratio	1.0
Crown wheel nominal speed	2684 rpm
Air-blast wheel nominal speed	6334 rpm
Oil system	forced
Oil pump	
Number	1 pcs.
Type	Double-stage geared
Guide roller rotary speed	2243 rpm
Sprocket number of teeth	8 pcs
Gear ratio of the leading gear wheels to driven	1:1
Mass of gearbox	120.5 kg

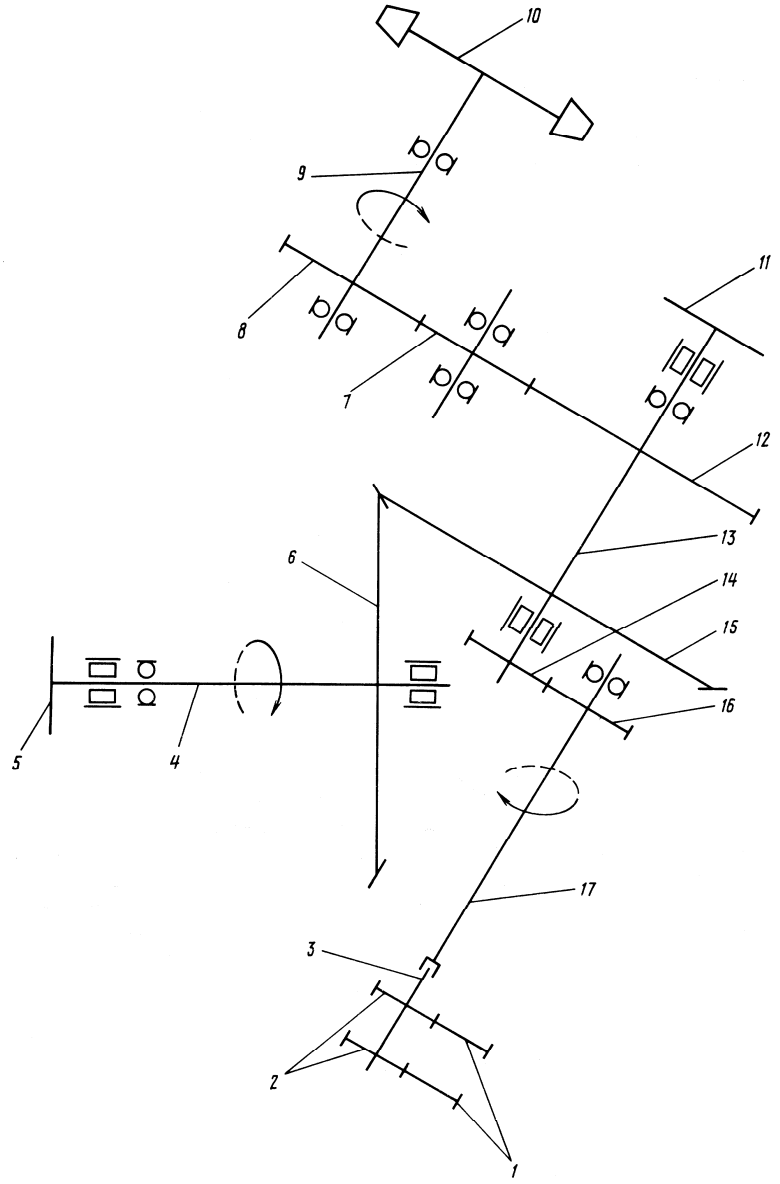


Fig. 16.11. Kinematic Scheme of Intermediate Gearbox:

1 - oil pump driven gears;
2 - oil pump driving gear
3 - oil pump main shaft;
4 - main shaft;
5 - driving gear splined flange;
6 - driving bevel gear
7 - intermediate gear
8 - fan driving gear;
9 - fan drive main shaft;
10 - fan impeller
11 - driven gear splined flange;
12 - fan drive driving gear
13 - driven shaft
14 - oil pump driving gear
15 - driven bevel pinion
16 - oil pump driven gear
17 - oil pump driven shaft

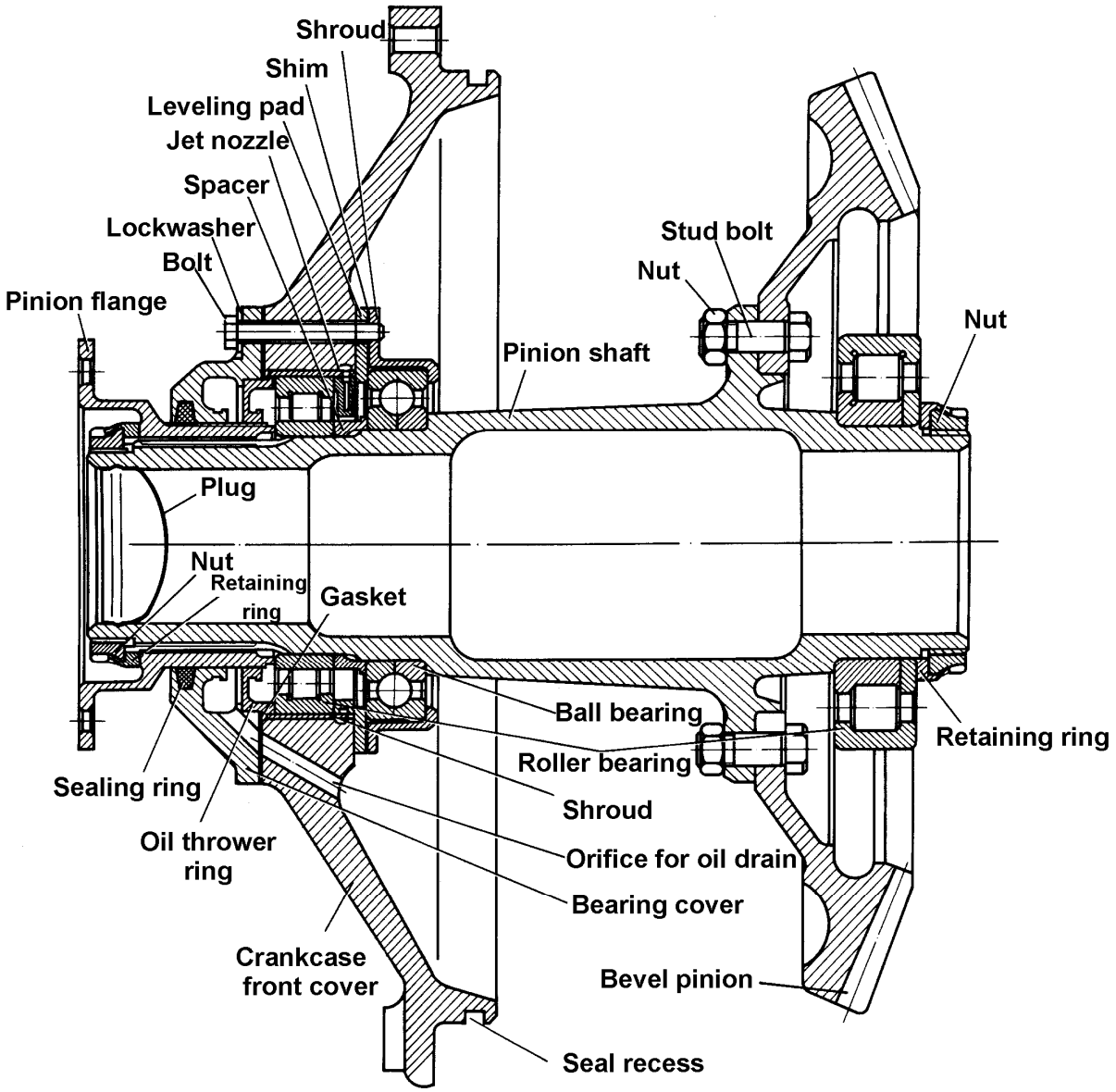


Fig.16.12. Driving Gear with Gearbox Front Cover

16.3.1. Intermediate Gearbox – Troubleshooting

Dribble of oil on sockets of gearbox and from under consolidations of the leading shaft

1.1. Possible causes: production defect.

1.2. Detection of a faulty element: defect is found visually at survey.

1.3. Malfunction elimination.

The issue of further operation of a reducer is resolved together with the representative of the industry.

16.3.2. Intermediate Gearbox – Maintenance Practices

The real maintenance practices include the following task cards:

1. Filling with oil of intermediate gearbox.
2. Discharge of oil from intermediate gearbox.
3. Dismantle and installation of intermediate gearbox.
4. Survey of intermediate gearbox.
5. Check of amount of oil in oil system of intermediate gearbox.

16.3.3. Intermediate Gearbox – Storage Rules

It is allowed to store an intermediate gearbox in original packing in the fundamental not heated warehouse or under a canopy with obligatory protection against atmospheric precipitation and corrosion environments.

Platform level at storage of intermediate gearbox under a canopy has to be higher than the level of the earth not less than on 200 … 300 mm.

16.3.4. Intermediate Gearbox – Shipment

It is allowed to transport intermediate gearbox in original packing by all means of transport without restriction of speed and distance.

During shipment it is necessary to protect intermediate gearbox from influence of shock loadings.

ATTENTION! TRANSPORTATION BY DRAGGING OF THE CONTAINER WITH THE INTERMEDIATE GEARBOX IS FORBIDDEN.

16.3.5. Oil System of Intermediate Gearbox –Description and Operation

General part

The intermediate gearbox has the forced lubrication system. The oil system of a reducer is intended for greasing of bearings, gear wheels and cooling of the working assemblies and parts of a gearbox. The oil system includes the air-oil heat exchanger, a two-level oil pump and hoses connecting the air-oil heat exchanger to gearbox pallet unions. The oil system of an intermediate gearbox consists of two parts. The external oil system is intended for cooling of hot oil in the air-oil heat exchanger. Internal part of an oil system (see Fig. 16.13) it is intended for greasing and cooling of assemblies and parts of a gearbox. The air-oil heat exchanger fastens on a rib No. 4 of fin in the block

with the air-oil heat exchanger of tail gearbox. Flexible hoses are fastened with collars to ribs No. 2 and 3 of tail fin.

Description

The external part of oil system includes a flexible hose of removal of hot oil from the pallet to air oil heat exchanger, the air-oil heat exchanger, a flexible hose of the cooled oil supply from the air-oil heat exchanger in the pallet. Flexible hoses on the design identical, and on length different.

The flexible hose consists of the sleeve with two cotton braids and inside rubber layer closed up in tips. The sleeve on both ends is closed up in nipples and pressed out by the coupling. The nut on the angular nipple is stopped by a wire stopper.

The hose has the marking put with yellow special enamel. Marking specifies number of the drawing and date of production. Hoses are fastened on ribs Nos 2, 3 of tail fin by means of collars, nuts, washers and screws.

Operation

Hot oil through a scoop comes to the oil pump scavenge stage, then on the internal channel in the pallet moves in a hose and comes to the air-oil heat exchanger and is cooled in it. The cooled oil on a flexible hose comes to the gearbox pallet again from the air-oil heat exchanger.

The internal part of oil system includes scoop of the cooled and hot oil, the pressure and scavenge stages of oil pump, the reducing valve, the filter, pressure indicator and jets. The internal part of the oil system contains also oil channels system in forward and back covers of a case, a case and a pallet.

Functioning of internal part of an oil system is illustrated in Fig. 16.13.

16.3.6. Oil System of Intermediate Gearbox– Troubleshooting

1. Chips on a magnetic plug or the oil system filter

1.1. The possible causes – internal destructions of a gearbox.

1.2. Search of a faulty element: defect is found at survey of the filter or a magnetic plug.

1.3. Malfunction elimination – to resolve an issue of further operation of intermediate gearbox with the representative.

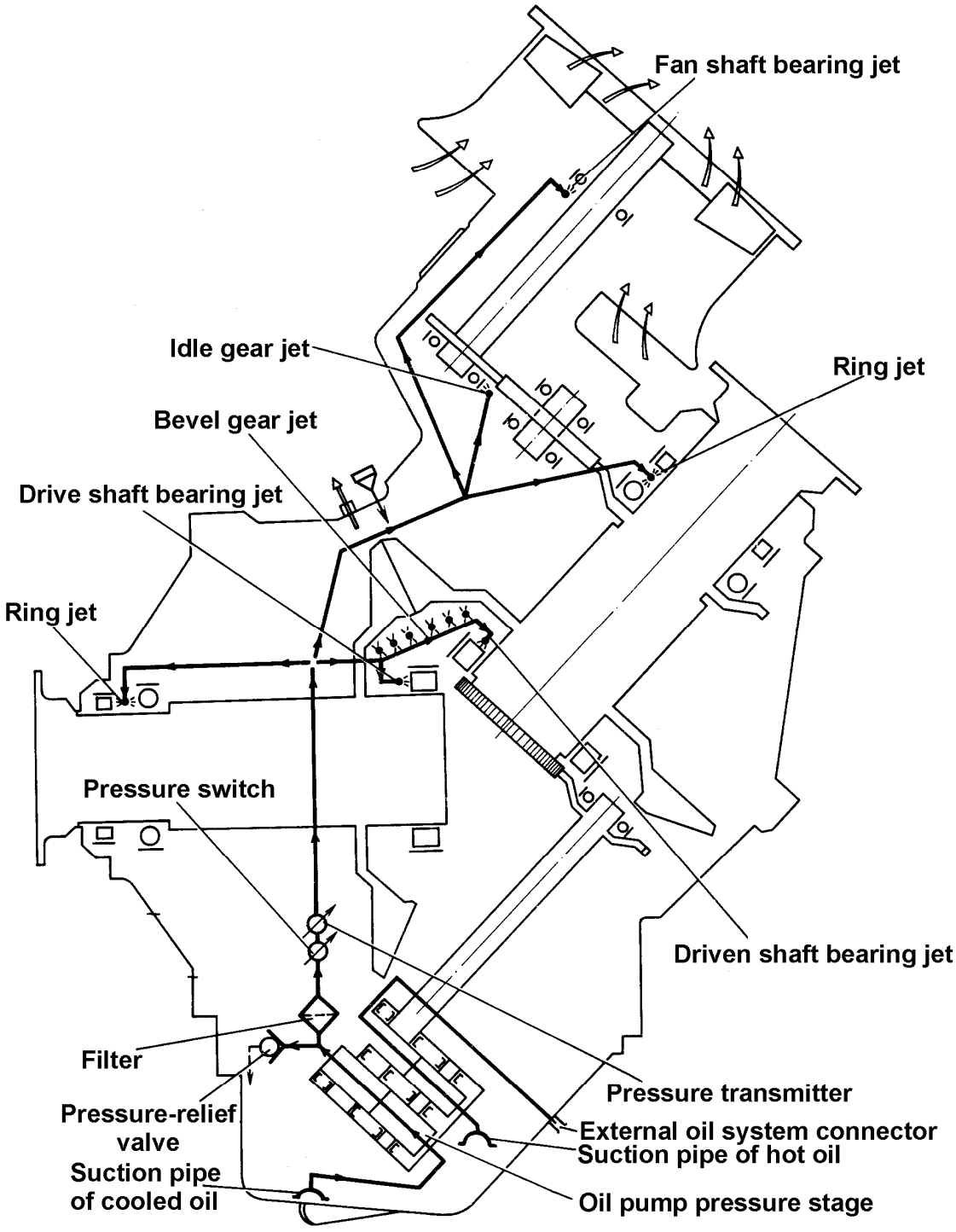


Fig. 16.13. Internal Part of Intermediate Gearbox Oil System

2. Leak of cells of air-oil heat exchanger
2.1. The possible causes – damages in use.

2.2. Detection of a faulty element: defect is detected visually at surveys.
2.3. Malfunction elimination – to replace the air-oil heat exchanger.

3. Dribble of oil from under cap nuts the nipple joints and pipelines.

3.1. Possible causes:
1) damages in use;
2) low-quality production.

3.2 Search of a faulty element: defect is detected visually.

3.3. Elimination of malfunctions – to tighten a cap nut in case of its loosening.

To replace the pipeline in case of malfunction of a part of nipple joint or the pipeline.

16.3.7. Oil System of Intermediate Gearbox– Maintenance Practices

The real maintenance practices include the following task cards:

1. Removal and installation of intermediate gearbox oil system.
2. Removal and installation of intermediate gearbox oil filter.
3. Inspection of intermediate gearbox oil system.
4. Inspection of intermediate gearbox magnetic plug.
5. Inspection of intermediate gearbox oil filter.

16.4. Tail Gearbox– Description and Operation

General part

The tail gearbox (Fig. 16.14) is intended for rotation of the tail rotors with the set speed. A gearbox is fixed with eighteen bolts to rib No. 14 of fin. Greasing of units, parts and their cooling is provided with special gearbox oil system independent of other units. The gearbox consists of the following assembly units: a case, the leading gear wheel with a back cover of a case, the driven gear wheel with shaft and the bearing, a shaft of the tail rotor with a forward cover (neck) of a case, satellites, the ram, oil pump, oil sump and rod.

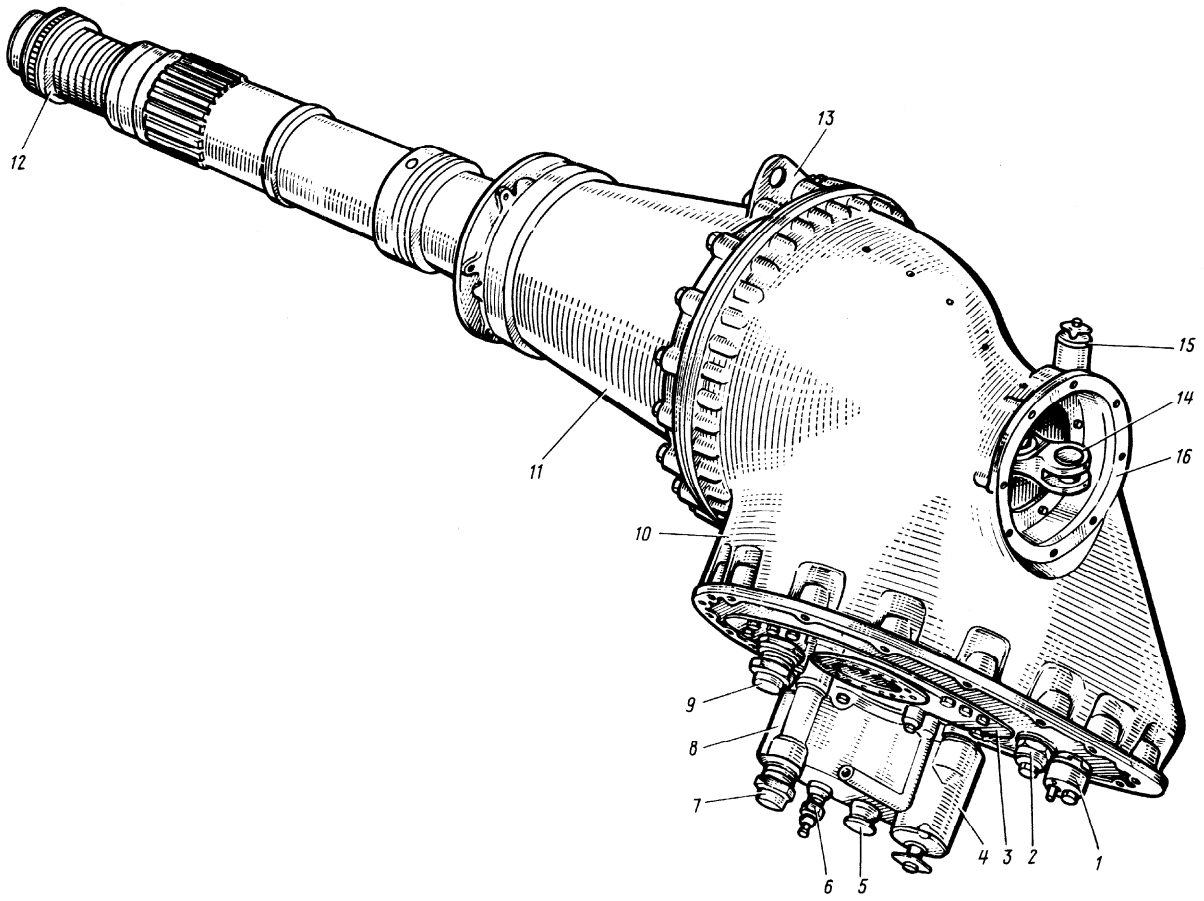


Fig. 16.14. Tail Gearbox:

1 – oil pressure sensor;
2 – pressure indicator;
3 – back cover of a case;
4 – oil filter;
5 – magnetic plug;
6 – temperature sensor;
7 – the connection of cold oil supply from a radiator in oil sump;
8 – oil sump;
9 – the connection of hot oil removal in radiator;
10 – case;
11 – forward cover of a case;
12 – ram;
13 –eye for lifting of gearbox;
14 – bail;
15 – drainage pipe line;
16 – flange of fastening of hydraulic actuator

The torque from tail shaft comes to flange 1 (Fig. 16.15) of the leading gear wheel 3. The gear wheel 3 mounted on a shaft 9 transfers a torque to the driven gear wheel 4. The gear wheel 4 transfers a torque to satellites 8 and drove 6. Drove 6 transfers a torque to a shaft 7 of tail rotor. Leading gear wheel 2 which is at the same time is mounted on a shaft 9 of oil pump drive transfers a torque to the driven gear wheel 10 of oil pump drive connected with the main shaft 11 of oil pump. The shaft 11 transfers a torque to the leading gear wheel 12 of oil pump and the driven gear wheel 15 of oil pump of the scavenge stage, and the leading gear wheel 13 of oil pump and

the driven gear wheel 14 of oil pump of the pressure stage. Decrease of speed happens in gear wheels 3 and 4 and the central gear wheel 5 with satellites 8.

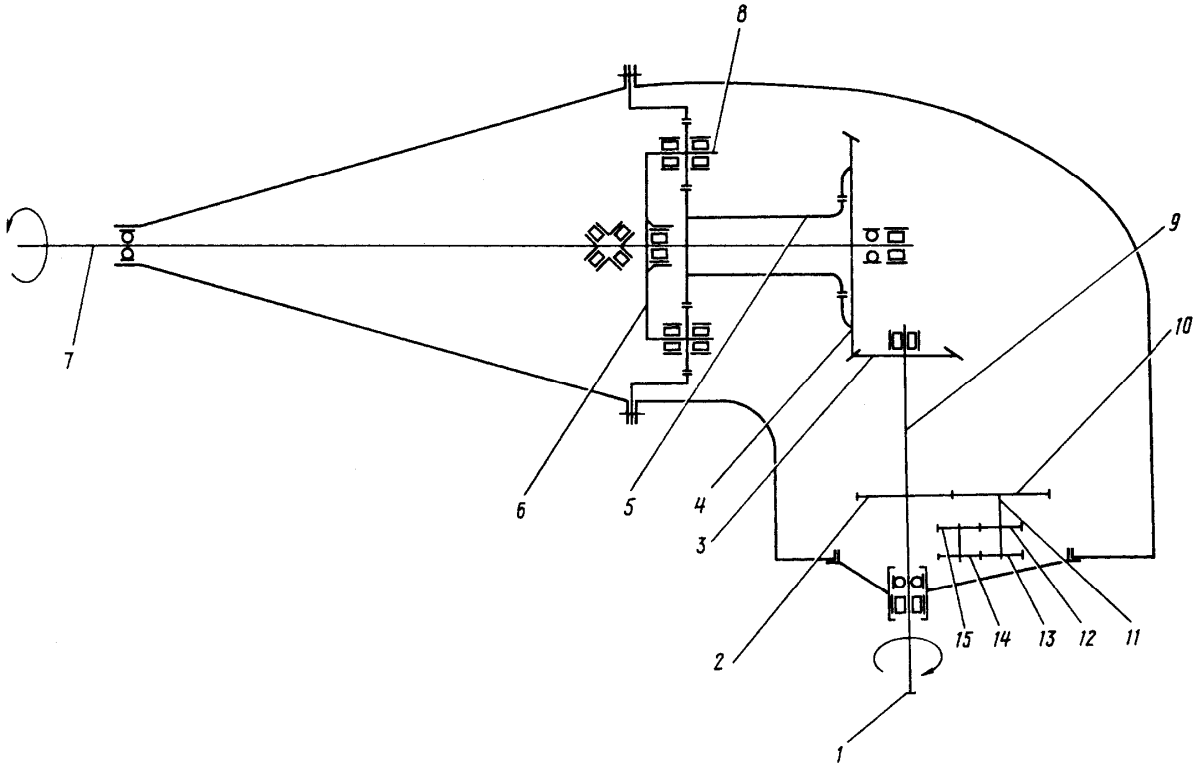


Fig.16.15. Tail Gearbox Kinematic Scheme

1 – flange of leading gear wheel;
2 – leading gear wheel of oil pump drive;
3 – leading gear wheel;
4 – driven gear wheel;
5 – central gear wheel;
6 – drove;
7 – shaft of the tail rotor;
8 – satellites;
9 – shaft of leading gear wheel;
10 – oil pump drive driven gear wheel;
11 – leading shaft of oil pump;
12 – leading gear wheel of oil pump;
13 – leading gear wheel of oil pump;
14 – oil pump driven gear wheel;
15 – oil pump driven gear wheel

Basic Specifications of Tail Gearbox

Gear ratio 4.848
Nominal speed of the driving gear ..... 2684 rpm
Nominal speed of the tail rotor shaft ..... 554 rpm
Lubricating system……forced
Oil pump:
q-ty ..... 1 pcs.
type ……..….. double-stage,

geared

The oil pressure behind the main stages

in working modes ..... 5 ± 1 kg/cm^2

idle power rating ……no more 2.0 kg/cm^2

Oil temperature ……no higher than 90 °C

Oil B-3C

Oil quantity ..... 22.5 lit.

Gearbox weight..... 342 ± 6.5 kg

Section of tail gearbox is demonstrated in Fig. 16.16

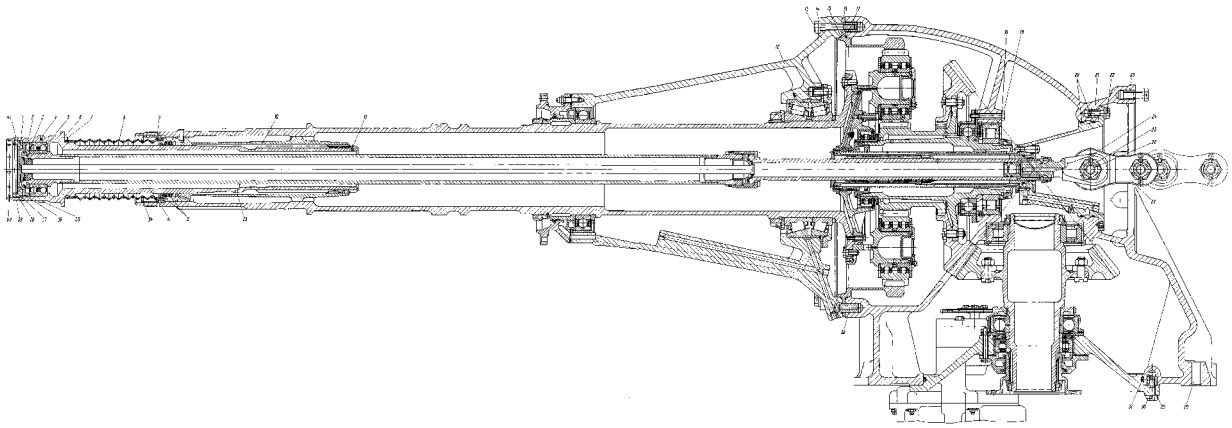


Fig. 16.16. Longitudinal Section of Tail Gearbox

16.4.1. Oil System of Tail Gearbox

The tail gearbox has a forced lubricating system. Gearbox oil system is used to lubricate the bearings, gears, spline joints and to cool working units and parts.

Oil system includes an air-oil heat exchanger, a two-stage oil pump, pipes and flexible hoses, which connect the air-water heat exchanger with reducing nipples of sump well. Tail gearbox oil system consists of two parts. The outer part of the oil system is designed to cool the hot oil in the air-oil heat exchanger. Air-oil heat exchangers are attached to fin rib No. 4 in the unit with air-oil heat exchanger of accessory gearbox. The inner part of the oil system (Fig. 16.17) is designed for the lubrication and cooling of units and gearbox parts.

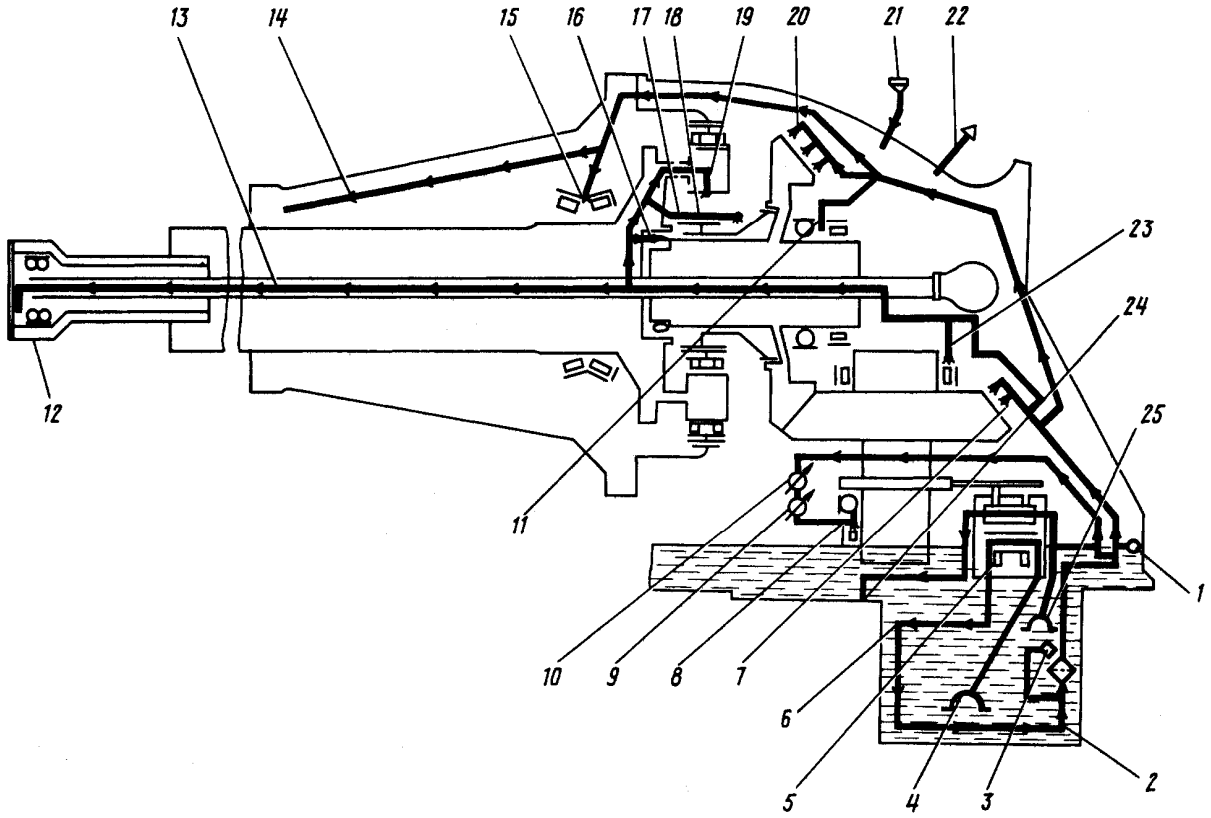


Fig. 16.17. Inner Part of Tail Gearbox Oil System:

1 – pressure gearbox valve;
2 – oil filter;
3 – safety relief valve;
4 – cold oil breeze plug;
5 – oil pump;
6 – dolly;
7 – jet d = 1.2 mm;
8 – jet d = 1 mm;
9 – pressure sensor IMD–8c;
10 – pressure switch MSTV-1.7;
11 – jet d = 1 mm;
12 – jet = 1 mm;
13 – oil passage of anti-torque propeller shaft;
14 – jet d = 1.2 mm;
15 – jet d = 1.2 mm;
16 – jet d = 0.8 mm;
17 – oil passage of sattelites;
18 – jet d = 0.8 mm;
19 – jet d = 0.8 mm;
20 – jet d = 1.2 mm;
21 – filler neck;
22 – breather;
23 – jet;
24 – connecting fitting of outer oil system;
25 – hot oil suction pipe

16.5. TRANSMISSION SHAFTS – DESCRIPTION AND OPERATION

Transmission shafts are designed to transmit torque moment from the main gearbox via an accessory gearbox to the tail gearbox and from the main gearbox to the fan system. The tail rotor drive shaft is attached on the seven supports to the fuselage

bulkheads and ribs of the tail boom and fin and to the outlet flange of the main, accessory and tail gearboxes.

Main Data of Tail Rotor Drive Shaft

Rated revolutions	2664 r.p.m.
The mass of tail rotor drive shaft (dry)	227 ± 4.5 kg
Lubricating oil	oil TC for hypoid gear

The tail rotor drive shaft consists of a fore part which connects the main gearbox with the accessory gearbox, and of the sloping part which connects the accessory gearbox with the tail one. Tail drive shaft consists of nine tubes connected together by flanges through groove sleeves. Each flange joint is tightened by sixteen bolts due to which lining-up of pipes occurs. Between the pipe flange and bolt heads slot washers are set. The bolts' nuts are tightened with a torque wrench with the moment of 3.0...3.5 kg • m. The pipes of tail rotor drive shaft are made from the AK6 material. They are specially balanced and are marked at the ends. The tail rotor drive shaft has seven supports on which it is mounted on the helicopter. Supports are attached through the cross over flanges to the formers Nos 21a, 25, 30, 40 of the fuselage, to former No. 5 of the tail boom and to rib No. 8 of the fin.

16.5.1. Transmission Shafts – Troubleshooting

1. Oil dribble from under the sealing of spline joints
   1.1. Potential causes are damages or worn sealing.
   1.2. Detection of a failed element.
   The dribble is detected visually during inspections.

2. Overheating of spline coupling
   2.1. Potential causes:
      1) insufficient lubrication;
      2) increased break of a shaft axle.
   2.2. Detection of a failed element.
   Overheating is detected by touch. Overheating is a heating during which it is impossible to keep a hand on the clutch for a long time.
   2.3. Malfunction repair
   Check the shaft axle break and the shaft's pipe whipping.

3. Shift's twisting
   3.1. Potential causes – the excess of admissible workloads on a shaft.
   3.2. The detection of a failed element
   The twisting is detected visually during inspections due to the location of check line on shaft's pipes.
   3.3. Malfunction repair
   If a shaft is twisted, the transmission must be replaced.

4. Excessive axle's breakage or excessive shaft's pipe whipping
   4.1. Potential causes – the shaft deformation during excessive workloads.

4.2. Detection of a failed element.
The defect is detected via a display device.
4.3. Malfunction repair
If a shaft is twisted, the transmission must be replaced.

5. The displacement of gauge marks on bolts and nuts of shaft's parts fixation.
5.1. Potential causes – the loosening of bolted joint.
5.2. The detection of a failed element
It is detected visually during inspections.
5.3. Malfunction repair
To tighten the loose bolted joint to stated moment.

16.5.2. Transmission Shafts – Maintenance Practices

These current maintenance practices include the following task cards:
1. Lubrication of spline couplings of transmission tail drive shaft.
2. Repair of calibrated bolt tightening.
3. Disassembly and assembly of the tail drive shaft's pipe.
4. Disassembly and assembly of the tail drive shaft's support.
5. Disassembly and assembly of the transmission tail drive shaft.
6. Inspection of the transmission tail shaft.
7. Installation of device for measuring the torsion angles in the immobilized clutches.
8. Installation of device for measuring the torsion angles in the bearing sleeves.
9. Installation of device for measuring the torsion angles in the elastic clutch.
10. Installation of device for check of the transmission shaft's whipping.
11. Cleaning of the transmission shafts.

16.6. MAIN ROTOR BRAKING SYSTEM – DESCRIPTION AND OPERATION

The braking system is designed for speed stop of main rotor after engines stop and for the transmission brakage during parking of the helicopter. It includes:
- brake;
- control handle;
- cabling.

Brake transmission is controlled by a handle, which is connected with brake lever via cable. Brake control system is interlocked with the engine start system, resulting in the start-up which is possible only when the transmission brakes are completely released. Hand controller of main rotor's brake is installed in the cockpit at the right side of flight engineer's seat. The main rotor's brake is located at the main gearbox rear output, where the tail rotor drive shaft is attached.

Two turnbuckles are installed in the area of the former No. 1F of fuselage center section, in order to control the cable adjustment and tension.

16.6.1. Brake – Description and Operation

Brake (Fig. 16.18) consists of a caliper on which two pads with levers are mounted.

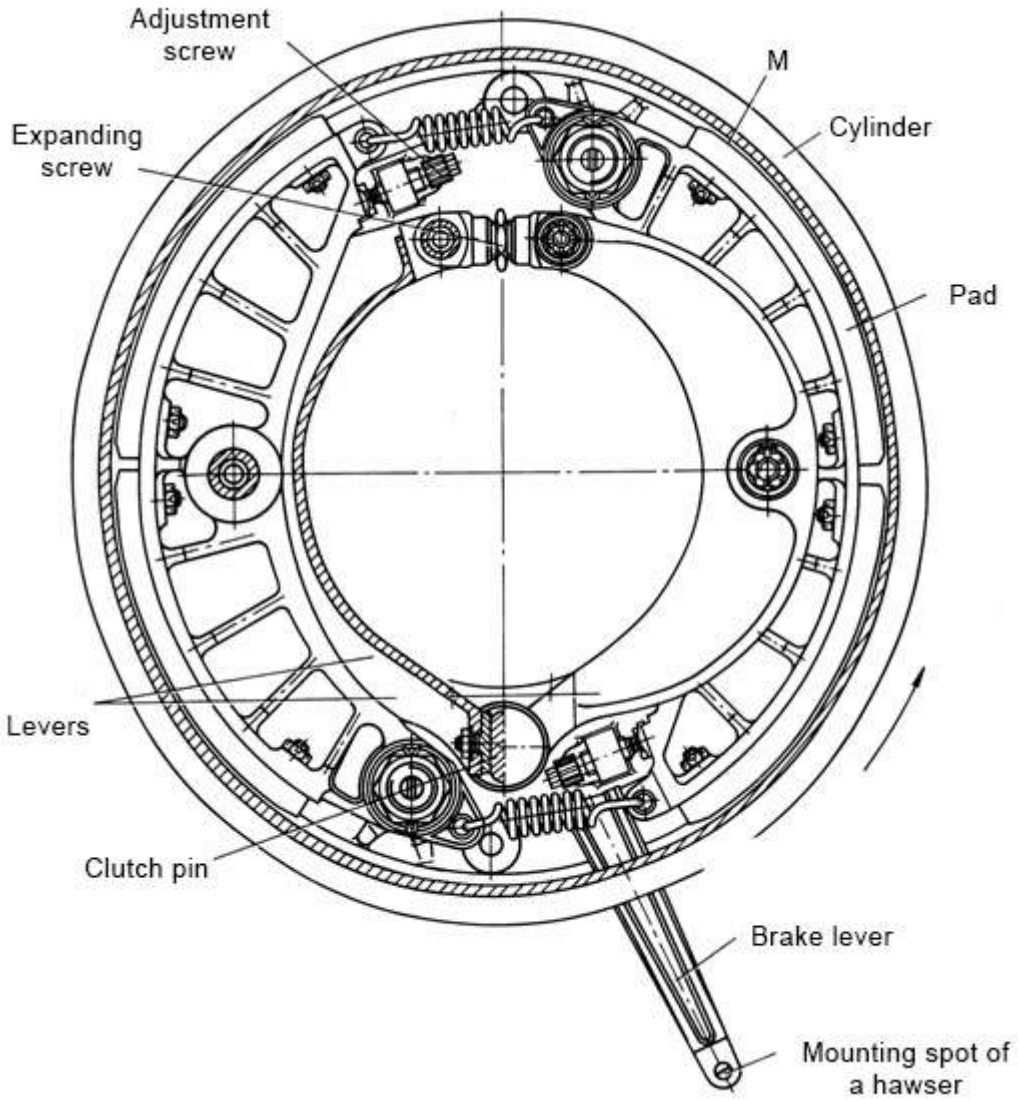


Fig. 16.18. Main Rotor Brake

The caliper is made of magnesium alloy ML5 and attached to the case of the main gearbox at the attachment point of the tail rotor drive shaft. The plate brake has mechanical cable control. The cylinder of brake 43 is made of steel 40HN2MA and it is mounted via bolts on flange adaptor of tail output of main gearbox together with the tail rotor drive shaft. The cylinder has ribs for cooling.

As a result of the force application to the lever, the clutch lifter, connected to the lever via cam slots, starts to turn. This clutch lifter puts pressure on levers, hinge joined to the pads. Herewith the pads are rotated about their axes, fixed to a caliper and, after

their gap selection are pressed with their friction linings against the cylinder so that the rotating cylinder starts to brake. With the release of the brake lever the brake pads and levers, connected to them, cast back by the action of springs, that is how the brake release occurs.

16.6.2. Brake – Troubleshooting

Table 16.2 shows main rotor brake maintenance practices.

Table 16.2

Main Rotor Brake – Maintenance Practices

No.	Defect	Potential causes and methods of detection	Troubleshooting method
1	Increased wear-out of the brake friction linings	The wear-out during the operation. It is detected when measuring the gap between friction linings and the cylinder of brake.	To change the linings
2	Oil ingress on brake-shoe linings	Poor-quality maintenance. Inefficient braking	To clean the linings
3	Corrosion and locking damage of the nuts and bolts of brake parts fasteners	Poor-quality maintenance. Must be detected visually	The locking must be replaced. If necessary the nuts must be retightened
4	Cracks of main rotor brake parts. Traces of paint coat burning or discoloration on the parts	Overheating and increased loads on the brake components as a result of heavy braking	To replace defective parts
5	Violation of paint coat of main rotor brake	Poor-quality maintenance and storage. Must be detected visually	To repaint

16.7. TRANSMISSION INSPECTION TOOLS – DESCRIPTION AND OPERATION

Information about the work of transmission units is showed by pointers, indicators and warning devices. Pointers and red annunciators are located on the dashboard of flight technician (see Fig 16.19), sensors and transducers are in the appropriate gearboxes.

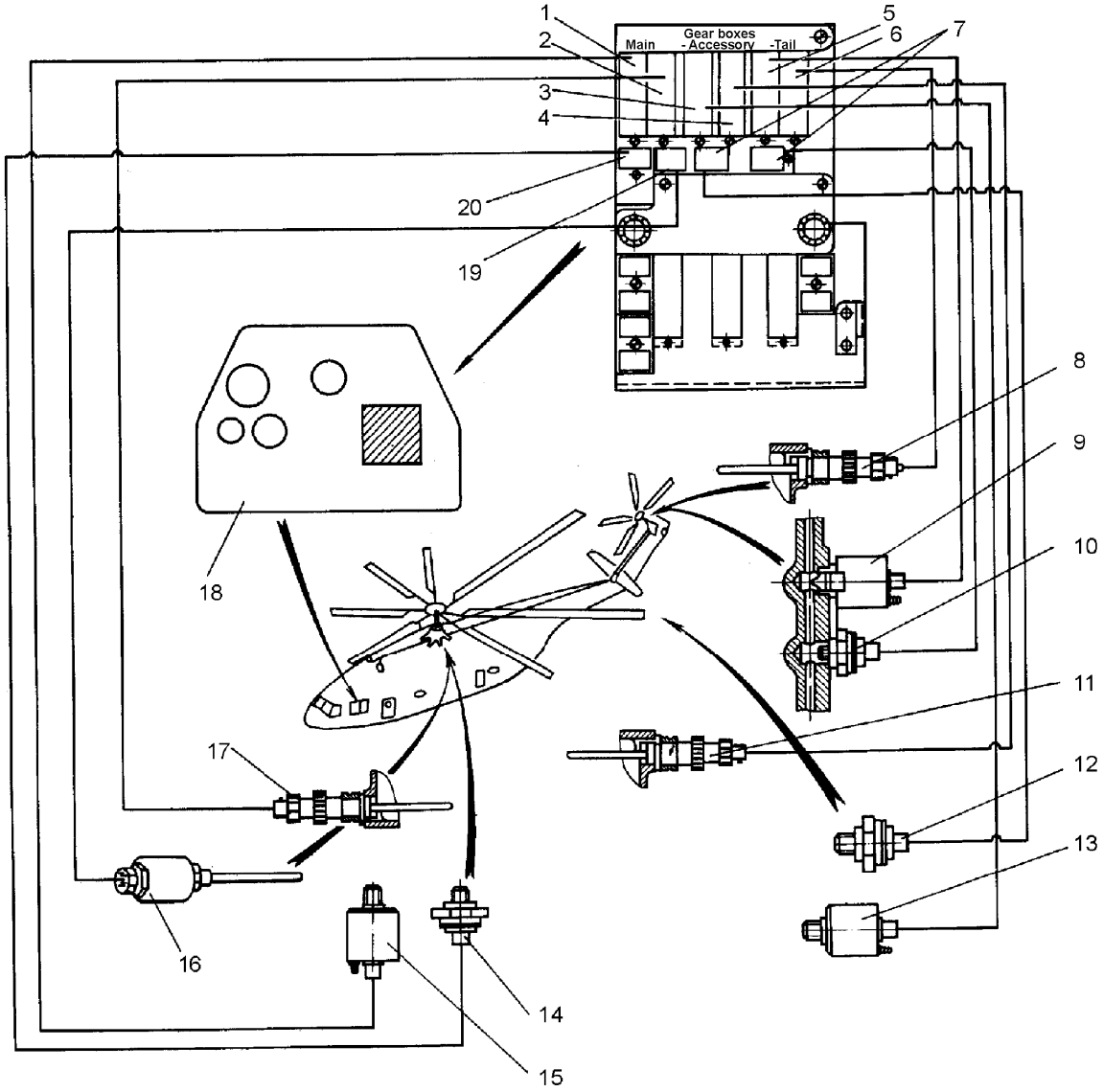


Fig. 16.19. Transmission's Inspection Tools:

1 – oil pressure indicator;
2 – oil temperature indicator;
3 – oil pressure indicator;
4 – oil temperature indicator;
5 – oil pressure indicator;
6 – oil temperature indicator;
7 – oil pressure indicator board;
8 – oil temperature sensor;
9 – oil pressure sensor;
10 – oil pressure warning device;
11 – oil temperature sensor;
12 – oil pressure warning device;
13 – oil pressure sensor;
14 – oil pressure warning device with oil distributing box;
15 – oil pressure warning device with oil distributing box;
16 – oil temperature warning device;
17 – oil temperature sensor;
18 – dash board of flight technician;
19 – oil temperature panel;
20 – oil pressure panel

Temperature receivers are located on the main, accessory and tail gearboxes. Oil temperature warning device can be only in the main gearbox.

Pressure indicators and oil pressure warning devices are located both at the accessory and tail gearbox and at geared compartment in the area of former No. 17.

Red annunciator of oil temperature warning device is located on a dash board of flight technician.

Thus, we have considered the structure of the third part of the fourth maintenance Manual of helicopter Mi-26 – Helicopter Transmission. The authors do not claim to give full statement for maintenance of helicopter transmission in operation and they aimed to familiarize with the list of questions submitted to the relevant sections of the Manual.

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Chapter 17. PROVISION OF RELIABILITY OF HELICOPTER TRANSMISSION DURING DESIGN PROCESS

17.1. AVIATION VEHICLES RELIABILITY ELEMENTS

While in operation, aircraft parts and accessories are subjected to natural wearing, affected by the ambient atmosphere, undergo physical and mechanical changes. As a result, the reliability of aircraft output parameters decreases, and their output characteristics get worse too. The rate of these changes can reach such values, at which the flight performance is at its very worst and the flight operating safety cannot be guaranteed. This is the reason for setting specific terms of operating life for aircraft, engines, accessories, and instruments. Within these terms their reliable operation and the necessary output parameters level are guaranteed.

Helicopter life of operation. The aircraft operating life is the time to the limit behavior, specified in the technical specification and stated in airborne hours, years of storage, cycles of operation, number of firing, etc. The operating life may also be envisioned as the product reliability margin, which is designed in, proved true during the manufacturing process, and used during the operation. It is an engineering factor, set on the basis of the field experience or results of special test conducted under natural or laboratory conditions. Thereunder, a distinction is made between assigned life, overhaul life, and warranty service.

Assigned life is the performance period upon reaching which the product use must be stopped irrespective of the product state. It is set in the technical specification for reasons of safety and economy. The Ministry of Civil Aviation operators are allowed to extend the operating life for some aircraft types in the following terms: up to 5 % for helicopters with turboshaft engines; up to 10 % for helicopters with reciprocating engines.

Overhaul life is the performance period of a product between two sequent repair actions. The aviation vehicles with exhausted operating life are directed for repair or discarded from operation.

Operating life before the first major repair is the performance period from the beginning of product use to the first major repair.

The interrepair life is calculated in the same way, but the period of the reliable operation of the product is measured in the calendar time (years, months) in this case.

Elapsed time is the period or volume of product operation measured in airborne hours, number of landings, amount of fire, number of operation cycles etc. For airplane airframes only airborne operation is counted as the elapsed time, for helicopters the whole airborne operation (100 %) and one fifth (20 %) of the rotor system ground operation are counted.

For aircraft engines and rotors elapsed time is equal to the sum of the whole amount of flight operation (100 %) and one fifth (20 %) of their ground operation. For airborne systems, which are tested on the ground under flight oriented conditions, elapsed time is equal to the sum of the whole ground operation (100 %) and the whole flight operation.

Warranty service is the performance period of a product, before the end of which the manufacturer guarantees and provides meeting specified requirements to the

product, upon condition that the customer meets the operating procedures, including storage and transport regulations. Warranty service is specified in the technical specification or contracts between the manufacturer and the customer. It is a technical legal parameter, about which the manufacturer and the customer make an agreement.

Warranty period is the term within which the manufacturer guarantees and provides meeting the specified requirements to the product, upon condition that the customer meets the operating procedures, including storage and transport regulations.

Operating life and performance period variations. There can be different variants of aircraft operating life, concerning the process of its operation delimitation and conditioned by the following provisions. If the assigned life and the warranty service, specified for a certain type of aviation vehicles, are not equal, then the operator is allowed to use this equipment within the limits of the assigned life.

If the assigned life is not specified yet, for some reason, then the aviation vehicles are used within the limits of the warranty service, after which the equipment is directed for a major repair or a special test.

Fail-safe performance is the product property to retain the performance capabilities within a certain performance period without forced interruptions.

For products, which are not repaired or replaced after the first failure, as well as for products, where failures are unallowable for reasons of safety, reliability characteristics can be such as, for example, “the fail-safe performance probability” and “the failure rate”. For those products, which are repaired, the reliability characteristics can be such as, for example, “the mean time between failures”, “the failure flux parameter”, and “the fail-safe performance probability”.

Opportunity to overhaul is product property, which lies in its ability to prevent, detect, and handle failures by maintenance and repair. To handle failures means to restore the performance. The overhaulability characteristics can be such as, for example, “the mean recovery time”, “the repair probability in the prescribed time”, “the average maintenance operations’ cost”.

Performance capability is the product state, when it can perform particular functions with the parameters prescribed by the technical specification requirements. Parameters which characterize the functions’ performance determine the operational characteristics of a product.

Endurance is the product property to retain its performance capabilities up to the limit behaviour with necessary interruptions for maintenance and repair works. The “limit behavior” of a product is determined by the impossibility of its further operation, or decrease in performance, or safety requirements, and is specified in the technical specification. The endurance characteristics can be such as, for example, “operating life”, or “performance period”.

Storage ability is the product property to retain operational characteristics during the storage period, as well as after the storage and transport period specified in the technical specification. The storage ability characteristics can be such as, for example, “the mean storage time” etc. Fig. 17.1 shows aviation equipment reliability factors.

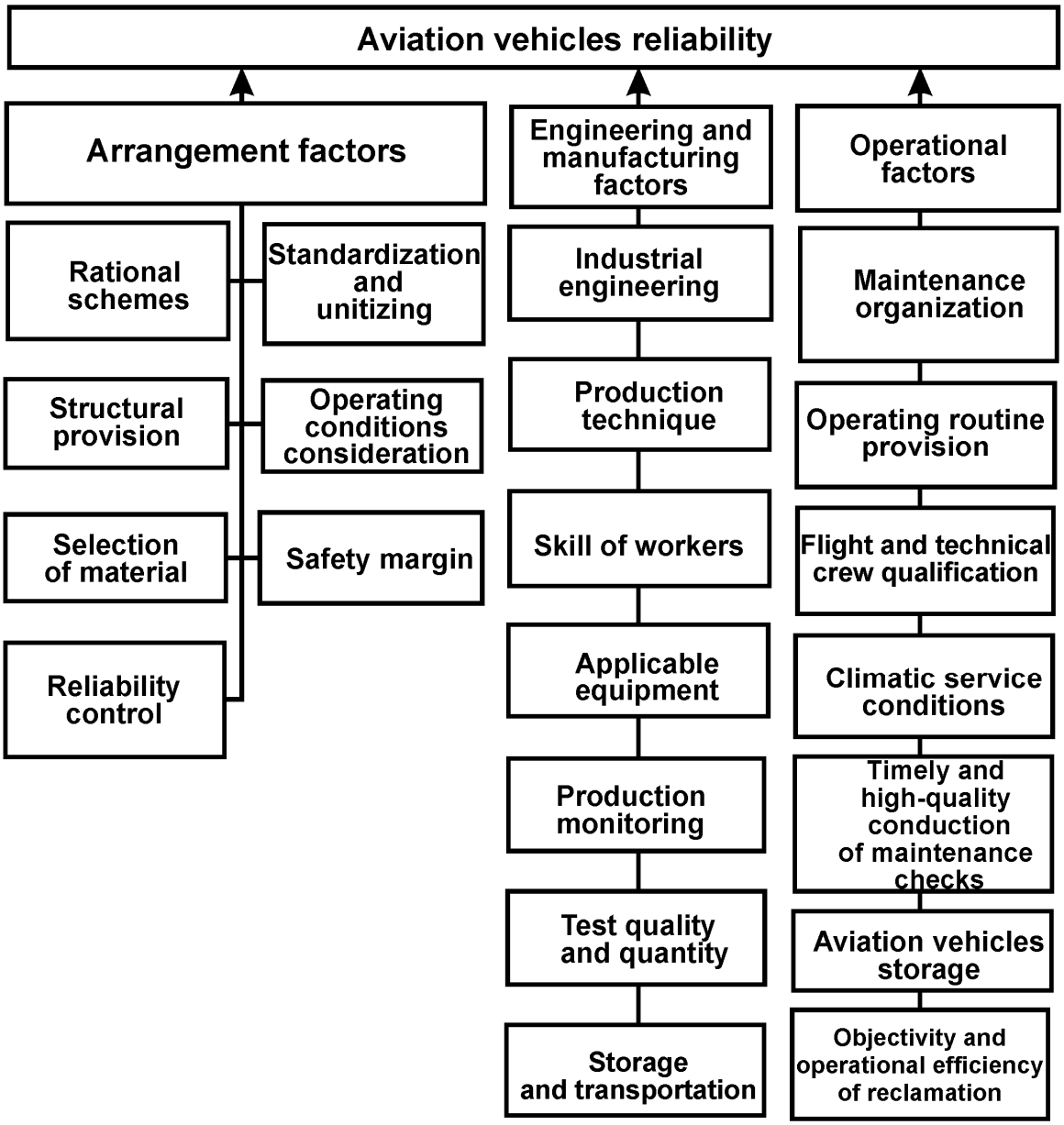


Fig. 17.1. Aviation Equipment Reliability Factors

17.1.1. Design Factors, Which Influence Reliability

The factors are the following:

– structure elements selection according both to their operational mode and conditions, as well as the operational conditions of the whole system;

– structure configuration is the number of elements and the way they are attached to each other;

– interchangeability and changeability of certain systems and elements;

– the degree of structure protection from various accidental atmospheric effects etc.;

– feasibility of monitoring the performance capabilities, inspections, adjustments,

and repair works;

– convenience of airborne systems management both in flight and on the ground, when monitoring the performance capabilities and detecting faulty parts.

17.1.2. Engineering Factors, Which Influence Reliability

Factors are the following:

– quality of materials used;
– degree of the manufacturing technique perfection;
– product test perfection degree.

17.1.3. Provision of Specified Reliability Requirements

The works on provision of specified systems and flight safety requirements during the engineering process are based on the following.

The engineering proposal stage. The key features, concept, and functional systems list are determined. For each functional system the list of functions is set, on the basis of which all possible functional failures are set. For each functional failure of a system possible after-effects for a helicopter are determined (by logical analysis, calculations, or modeling). For each after-effect of a functional failure its hazard level is set, according to the JAR classification.

Depending on the hazard level of the failure, an allowable possibility of its emerging is set. The designer must provide the necessary system fail-safe performance by introducing device redundancy or protection, which reduce the hazard level of the functional failure.

Basing on the list of functions performed by each functional system, the hazard level of functional failures, redundancy and protection devices requirements, a concept scheme of a system is created, which must provide a comprehensive picture of the system accessories composition, their operating principle, issues of control and providing the crew with the information about performance capabilities of technical state of the system.

The initial design stage. After creating the basic configuration of the functional system, the system's reliability is analyzed, as well as the influence of its failures on abnormal cases origination and the helicopter flight regularity.

This analysis aims to specify the list of functional failures, their consequences, and the hazard degree, control levels of their allowable probabilities, as well as establishing the logic connection between them and the accessories failure types, identifying the probabilities of a functional failure through the probabilities of system accessories failures. At the same time, the amount of necessary information about a failure is specified, as well as the crew's course of action, to accommodate the fault influence on the flight.

Based on the system reliability analysis, compliance with the requirements is monitored for the probability of special situations occurrence and flight regularity (determined by the corresponding control levels of reliability systems), in case of their fail-

ure, appropriate changes of concept system are carried out.

Based on the analysis of individual functional systems reliability, the helicopter safety in general is analysed, reflecting the interaction of each with the other systems and the impact of one system failure on the performance of others. The likelihood of exceptions is estimated due to the failure of several functional systems and the final control of performance requirements is carried out, to ensure the safety and regularity of the helicopter flight.

The detailed design stage. Reliability analysis is corrected concerning any changes in the configurations of functional systems and accessories' structures according to their arrangement in the helicopter.

The effects of functional failures is clarified on the basis of calculations, simulations, bench and flight tests, taking into account allowable range of expected operating conditions.

Based on the analysis of the reliability the following documents are developed:

– The list of potential functional failures;

– Evidentiary documentation in preparation for helicopter certification;

– A list of special situations in the air to include in the guide for years-exploitation;

– A list of failures to simulate to prepare flight crew;

– The basic principles and methods of helicopter technical maintenance.

One of the indicators that determine reliability of the helicopter accessories is its strength (fatigue life). Therefore, the reliability of accessories in operating conditions largely depends on the fatigue strength of parts. Fatigue tests and observing the parts in the operating conditions revealed the influence of the shape, size, condition of the surface layers of the material on the strength and durability of the parts.

To the present day, there have been practical ways to improve the fatigue strength and increase the service life of parts by the implementation of the structural, processing, metallurgical, operational and preventive measures.

Structural measures mean that the parts are shaped in such a way to promote rational, perhaps even more distribution of the internal forces.

The most common measures to improve the fatigue strength of parts that are used by the designer during the design process are the following:

increasing the area of the dangerous section and changing its geometry;

reducing the stress concentration;

changing the stiffness.

The **technological activities** include the following ones: finishing the parts' surfaces carefully; making the anticorrosive coating; creating the residual stresses in the parts, which being summed with the operating stresss, would provide favourable operating conditions for the parts (unloading and loading less stressed parts of the structure).

The metallurgical activities lie in choosing the most appropriate material and establishing the optimal mode of heat treatment, as well as preventing the cracks and shrinkage cavities, gas bubbles and inclusions. This may also refer to carburizing, nitriding and cyaniding of surface material layers.

The operational activities must include the protection of parts from corrosion (paint, grease); protection tense and critical parts from damage (scratches, nicks in the

assembly and disassembly, abrasion); the prevention of contamination of work surfaces and related phenomena of wearing and scuffing increase, etc.

17.2. TRANSMISSION. GENERAL

The helicopter transmission is a set of accessories, which transmit power from the engines to the main rotor and the tail rotor, as well as to the auxiliaries.

Transmissions of single-rotor helicopters with two gas-turbine engines are very similar in design and include the following main units:

- main gearbox;
- intermediate gearbox;
- tail gearbox;
- tail rotor drive shaft;
- fan drive shaft;
- freewheel clutch;
- spline couplings;
- main-rotor brake;
- main gearbox frame.

The main gearbox collects the engine power and distributes it to the main rotor and the drives of the tail rotor, fans, hydraulic pumps, the air system compressor, generators and other units, providing their optimum rotation frequencies.

The intermediate gearbox is designed to change the direction of the tail shaft axis in order to locate the tail gearbox and the tail rotor axes closer to the main-rotor plane of rotation.

The tail gearbox is used to alter the direction of the tail shaft end axis at an angle of 90°, reduce the rotation speed of the gearbox driven shaft, as well as to mount the tail rotor hub on this shaft.

The torque is transmitted through the **tail shaft** from the main gearbox to the tail rotor.

The fan drive shaft transmits the torque from the main gearbox to the fan wheel.

The freewheel clutch is designed to transmit torque from the free turbine of the engine to the main gearbox, as well as to disconnect automatically the free turbine shaft from the main gearbox in case of engine failure. This creates favorable conditions for continuing of the flight with one engine, as well as for the flight in the autorotation mode in case of failure of both engines. In addition, separate and alternate engine testing on the ground is possible.

Spline couplings of the tail shaft provide mutual swash and longitudinal displacement (within certain limits) of the shaft separate parts. This eliminates their loading with bending moments and longitudinal forces that may arise due to deformations of the rear and the end beams in flight, inaccuracies in individual lengths of shaft areas and the distances between its legs, the difference in thermal elongation of the shaft, as well as the tail and the end beams.

The main rotor brake is designed to reduce the time of the transmission stopping down after switching off the engines on the ground, eliminate the possibility of the main-rotor starting by means of wind in the parking, as well as transmission braking dur-

ing mounting work and routine maintenance.

The main gearbox frame serves to attach the main gearbox to the fuselage.

The transmission as a whole and its separate elements must not have any strength-threatening vibrations throughout the whole range of rotation permitted operating modes and frequencies of engines. It is essential that the critical values of transmission shaft speeds were at least 20 % higher than those highest possible in operation. To control the vibration characteristics of the transmission shafts and gears supports, as well as the state of clutches and other components and assemblies it must be possible to install test equipment periodically, check the separate transmission sections for misalignment, detect faults early, for example by the presence of metal chips in the oil.

Let us consider the impact of different **design solutions** on the fatigue strength of the helicopter transmission elements.

17.2.1. Shafts and Axles

Increasing the unsafe cross-sectional area can improve the fatigue strength of a part, but it increases its mass. For example, a shaft, operating in conditions of dynamic loads, is weakened with an opening for supplying oil to the bearings. In this case, to strengthen the shaft it is necessary to make its inner groove molded with increasing the cross-sectional area of holes, despite the fact that this complicates the shaft manufacturing and increases the shaft weight.

The fatigue strength of the axles and shafts decreases if bushings, bearing rings, or bosses are press-fitted onto them. Availability of mounting reduces the fatigue strength of the shaft with a press-fitted part compared to the fatigue strength of a free shaft. This is due to the following three reasons:

the concentration of stress as a result of the mounting;

the contact friction between the parts engaged with interference, occurring at variable stress;

influence of corrosion.

The contact friction at variable stress leads to deterioration of the shaft mounting surface, whereby the unprotected metal areas may be exposed to moisture and reactive chemicals. A significant reduction in the fatigue strength can be caused by burrs on the shaft surface, occurring as a result of pressing bushings cold.

When designing shafts, operating under the conditions of torsion and bending, it is necessary to take into account that the main stress concentrators are fillets, splines, edges of parts pressed onto shafts (bushings, bearing rings, etc.) and holes.

Fig. 17.2, a, shows the three forms of the shaft surface cut. All cuts are of the same depth t and input radius r, and differ in the output radius R. Local stresses in shafts having such forms differ slightly.

The main factor that affects the magnitude of stresses around the fillet, is the radius r of curvature. With the radius of curvature increasing, the stress concentration reduces greatly. However, in some designs it is not possible to increase the radius of curvature. Thus, Fig. 17.2, b, shows a shaft, which fillet radius is increased by means of the ring A. In the absence of such a ring it would be necessary to make a fillet radius r smaller than the radius r1 at the end face of the bearing ring.

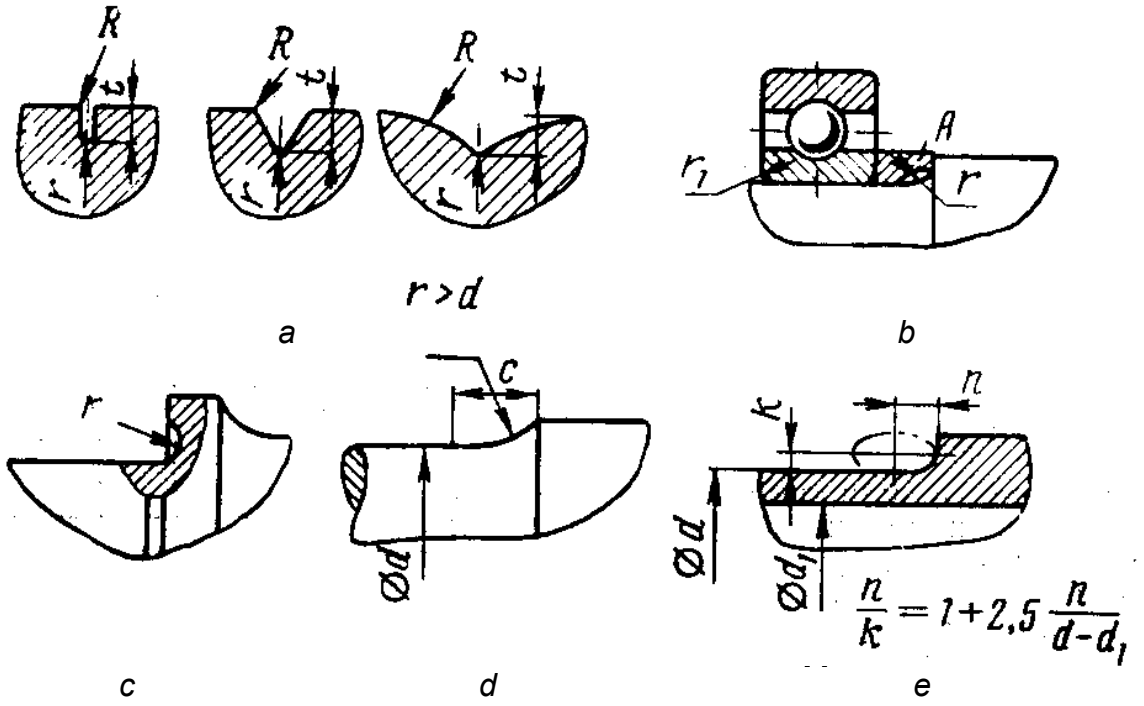


Fig. 17.2. Ways to Reduce Stress Concentration when the Shaft Diameter is Changed: *a* – impact of the cut shape on the local stress; *b* – spacer *A* increasing the fillet radius; *b* – undercut fillet; *d* – fillet with a large radius of curvature; *e* – elliptical fillet on a hollow shaft; t – cut depth; r – input radius; R – output radius: r_1 – radius of curvature of the bearing ring; c – length of the transition area; d – outer diameter of the shaft; d_1 – inner diameter of the shaft; n and k – ellipse semiaxes

Fig. 17.2, *c* shows how to reduce the stress concentration by performing a rounding entering the shaft protrusion (undercut fillet).

Fig. 17.2, *d* shows a smooth transition from a smaller diameter to a bigger one. Such couplings of two shaft parts eliminate the stress concentration almost completely, but often they are not structurally feasible, as in such case, the length of the transition area c is too high.

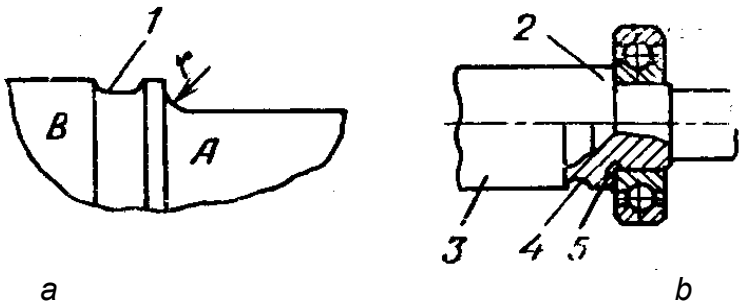
A transition with a gradual decrease in the radius of curvature of the fillet (elliptical fillet) gives a good result. Fig. 17.2, *e* shows the optimum ratio between the semiaxes of the ellipse formed and shaft dimensions. If for design reasons the size n must be as small as possible (n <0.1d), then the ordinary circular fillet is used as it is practically difficult to make the curvature of the desired shape over a length of less than 0.1d. If the transition is so mellifluous that n> d, then virtually at any waveform forming the fillet, it can be assumed that the concentration of stress is absent.

Small advantages of the elliptical fillet and technological difficulties in its implementation to use make us use such a transition only in particularly critical structures.

Where the rigidness of structure elements changes dramatically, the highest stresses occur. For example, the part A of the shaft is less rigid than the part B (Fig. 17.3, a). The reduction in rigidness of the part B, and hence some stress equalization can be achieved by creating unloading grooves 1. Unloading grooves are used when it is impossible for structural reasons to increase sufficiently the r fillet radius.

Fig. 17.3, *b* shows the usual shaft design where the bearing is mounted on it, and

the improved one with the undercut fillet 5 and the unloading groove 4.



a *b*

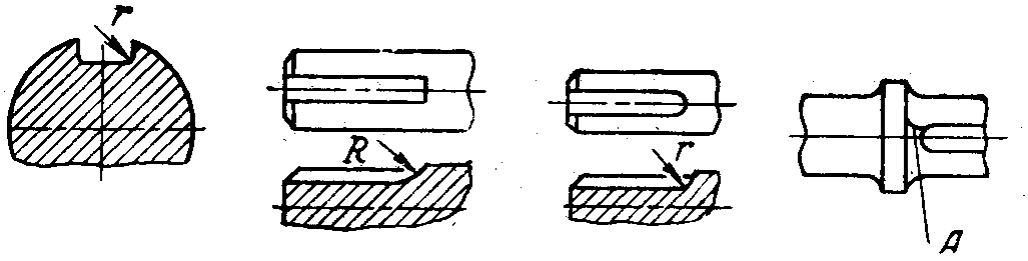
Fig. 17.3. Ways to Reduce Stress Concentration In Areas of Sharp Changes in Rigidness of Structure Elements:
a – using an unloading groove to reduce local stresses; 1 – unloading groove; f – fillet radius; *b* – normal (2) shaft structure where the bearing is press-fitted and an improved (3) structure (undercut fillet 5 with the unloading groove *4)*

Fatigue strength of shafts with fillets is increased by grit blasting or rolling the fillet with rollers. During the grit blasting strength increases sometimes even if the shape and dimensions of the concentrator does not allow grit particles to penetrate deep into the groove.

Polishing of rolled or grit-processed parts can provide good results if the layer removed is very thin.

Fig. 17.4, *a* shows the cross-section of the shaft with a key-groove. Local stresses depend greatly on the radius of curvature r.

For shafts operating at variable stresses incoming angles of the key-groove must be rounded.



a *b* *c* *d*

Fig. 17.4. Ways to reduce stress concentration when there is a key-groove on the shaft:
a – shaft key-groove; r – input radius; *b* – key-groove, made by peripheral milling; R – milling cutter radius; *c* – key-groove, made by face-milling cutting; f – ineffectual combination of the key-groove and the fillet; A – fatigue crack

The key-groove (see. Fig. 17.4 b), obtained by peripheral milling (radius R cutter) from the point of view of stress distribution should be considered more rational than the groove (see. Fig. 17.4, *c*) obtained by face-milling cutting (input radius r).

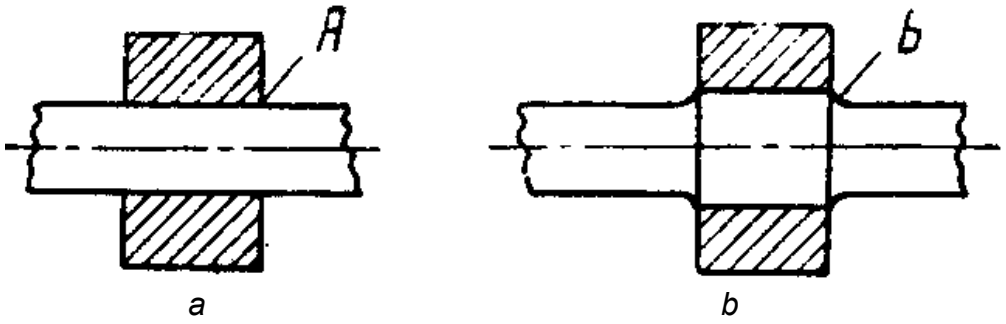


Fig. 17.5. Ways to Reduce Contact Stresses: *a* – shaft with bushing press-fitted; *b* – shaft thickening where bushing is mounted

Figs 17.6 and 17.7 show stress concentrators diversity and options for lubricating holes location

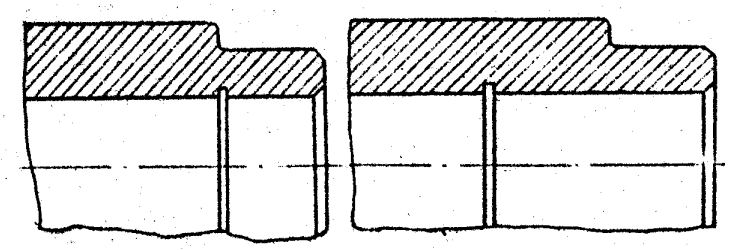


Fig. 17.6. Stress Concentrators Diversity

Around corners A of bushings, mounted on the shaft (Fig. 17.5, a), there occurs the concentration of stresses. Moreover, when the shaft is bended again, its surface is abrased with the bushing angles. This may cause fatigue cracks to appear. In order to eliminate abrasion and increase the shaft strength its diameter must be slightly increased where the bushing is press-fitted (see. Fig. 17.5, *b*). However, we must remember that in the transitions (points *B*), in turn, local stresses will appear.

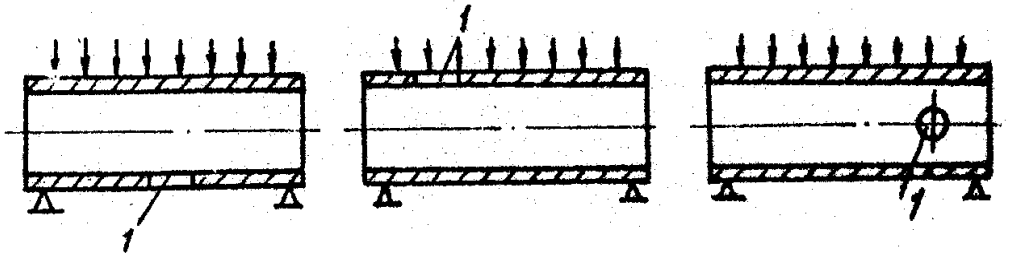


Fig. 17.7. Options for Lubricating Holes Location (1)

Despite the fact that the material of hollow shafts and axles near the inner surfaces is loaded less than that near the outside surfaces, the inner surfaces must also be processed thoroughly and hardened in order to avoid fatigue cracks around them.

Figures 17.8 and 17.9 show stress concentration factor dependence on shaft geometric parameters and one of structural solutions of reducing shaft stresses in support area.

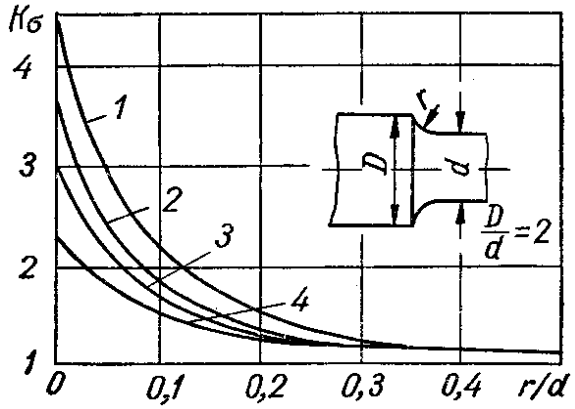


Fig. 17.8. Stress Concentration Factor Dependence on Shaft Geometric Parameters
1 – σ_s = 1200 MPa; 2 – σ_s = 1000 MPa; 3 – σ_s = 800 MPa; 4 – σ_s = 600 MPa

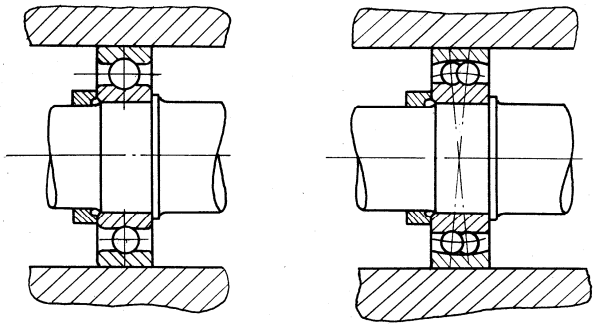


Fig. 17.9. One of Structural Solutions of Reducing Shaft Stresses in Support Area

Seizures that occur when pressing the bearing rings on shafts and mounting bolt connections greatly reduce strength. A good result can be reached when pre-processing the shaft with rollers where the bushing is press-fitted and using a special paste that eliminates seizures when mounting bearings and bolted joints.

17.2.2. Pivot Pins

During the tests, the destruction of the shaft pivot pin started at the mounting site of the drive piece connected by means of a strut with the swashplate. The crack propagated in a helical line (on a smooth surface) on which there was a significant work-hardening and friction corrosion (fretting corrosion) in the form of powdered metal oxide. Furthermore, the material degradation was observed in the most corroded area. Fretting corrosion was the main reason for the reduction in the shaft pivot pin fatigue strength.

17.2.3. Cardan Mechanism

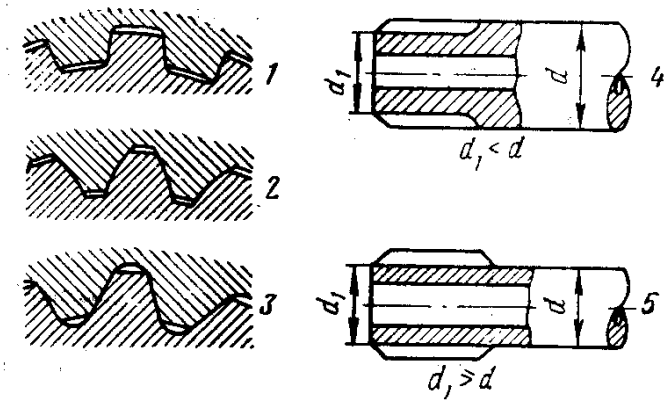
The cardan-joint fork was destructed during the tests on the conical part and started from the hole. The destruction is of the fatigue nature. The fatigue crack start co-

incides with a not rounded place of transition of the cylindrical part into the conical hole. The reason for the destruction of another cardan-joint fork is a poor-quality machine-processing of its inner surface.

17.2.4. Spline Coupling

Spline coupling is more efficient compared with key coupling and must be used in all critical joints. The spline form affects the strength of the spline coupling.

The highest increase in the local stresses occurs at the rectangular splines. The main reason for such a sharp increase in local stresses is a small transition radius in spline spaces. The spline coupling has the twice-less stress concentration, where the involute splines are made with the engagement angle α=30° and spaces are delineated along the radius (Fig. 17.10).



a *b*

Fig. 17.10. Ways to Reduce Stress Concentration in Spline Couplings:
a – spline shape: 1 – rectangular splines; *2* – splines with the engagement angle α=20° and a small transition radius in spline spaces; 3 – splines with the engagement angle α=30° and spaces, delineated along the radius; *b* – transition from splines to the shaft: 4 – spline spaces diameter is less than the shaft diameter(d_1<d),
5 – spline spaces diameter is equal to the shaft diameter or greater than it (d_1>d).

Fatigue tests have shown that the shaft (see. Fig. 17.10, b) the spline space diameter of which is less than shaft diameter, is less durable than the shaft which spline space diameter is equal to the shaft diameter or greater of it. This is because in the second case, there is almost no stress concentration at the transition from splines to the shaft. With proper sizing of d and d1 balanced strength of the shaft is provided in the sections close to the splines and remote from them.

The destruction of splines may occur from hardening. Hardening on the splines indicates either the irrational constructive solution during the design process or the low manufacturing and mounting quality during the production of the splined coupling, which results in the uneven load distribution on the splines.

Figures 17.11 and 17.12 show spline coupling structure and transmitting the torque by means of involute splines, operating under conditions of transmission shaft axles misalignment

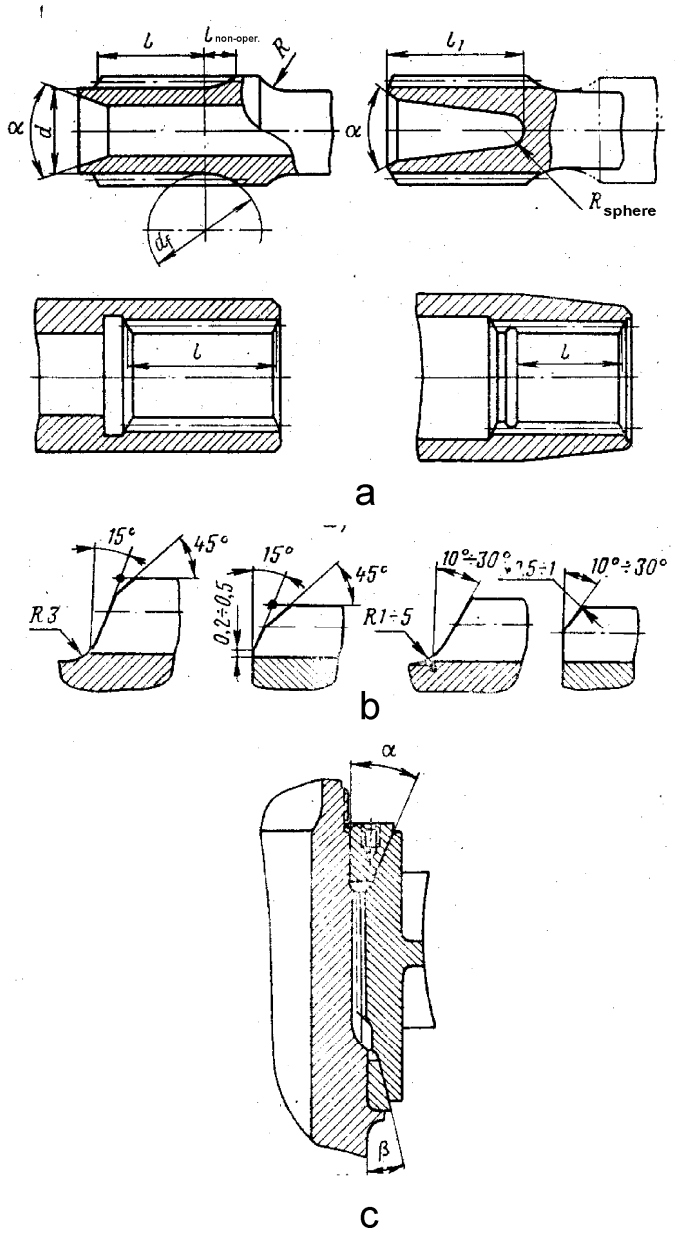


Fig. 17.11. Spline Coupling Structure:
a – spline areas of the part: operating spline length, shaft boring depth, non-operating spline area, milling cutter diameter;
b – spline end shapes recommended; c – centering parts according to two conical surfaces

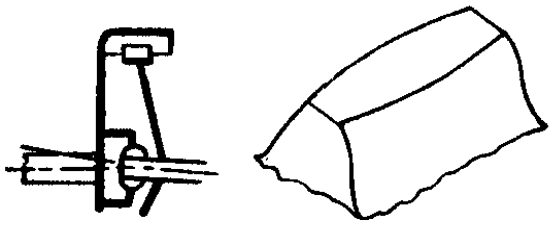


Fig. 17.12. Transmitting the Torque by Means of Involute Splines, Operating Under Conditions of Transmission Shaft Axles Misalignment

17.2.5. Threaded Couplings

The thread strength depends on the uniformity of the distribution of forces between the thread turns.

The load distribution on the thread turns is affected by the nut shape and height. In the standard design the lower turns of the nut are the most loaded. The approximate load distribution on nut turns is shown in Fig. 17.13. Increasing the nut height does not affect the load on the lower turns significantly, in this case the upper turns are loaded slightly.

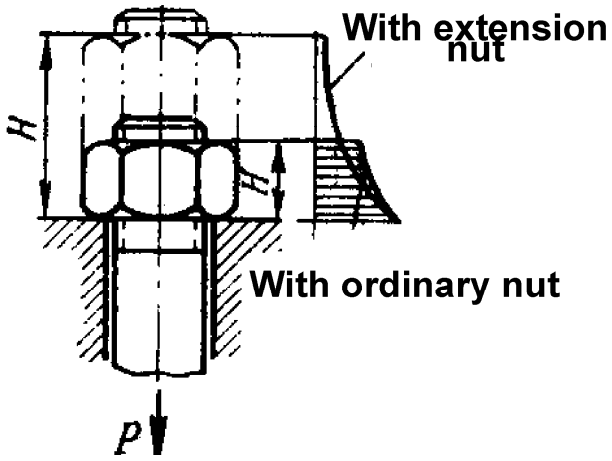


Fig. 17.13. Effect of nut H Height on the Load Distribution on Turns

Fig. 17.14 and 17.15 show the nut shapes providing more even load distribution on the thread turns.

In order to reduce the load on the lower turn, nuts with sunk thread are used. This structure provides some additional compliance of the lower turn.

To reduce the load on the lower turns countersink conical thread may also be used.

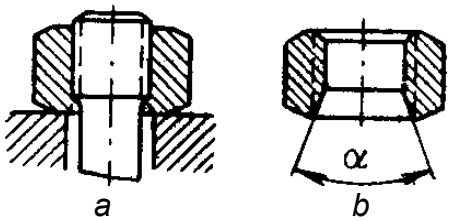


Fig. 17.14. Ways to Reduce Stress Concentrations in Threaded Couplings: shapes of nuts which provide more even load distribution on the thread turns: *a* – sunk thread nut; *b* – countersink conical thread nut; α – countersink angle

In a number of critical joints normal and standards for nuts are rejected. For example, you can make the load on turns more even if you change from the usual compressed nuts to stretched or stretched and compressed nuts (see. Fig. 17.15).

Stretched nuts provide better distribution on thread turns than compressed ones. Introduction of cones on the nut or simultaneously on the nut and the bolt end contributes to further improvement of the fatigue life of the coupling. The latter structure provides a more uniform load distribution, but it is technically more difficult than others.

Applying stretched and compressed nuts enables us to load the thread turns uniformly because the thread on the nut force is transmitted to the protrusion located in the middle of the nut.

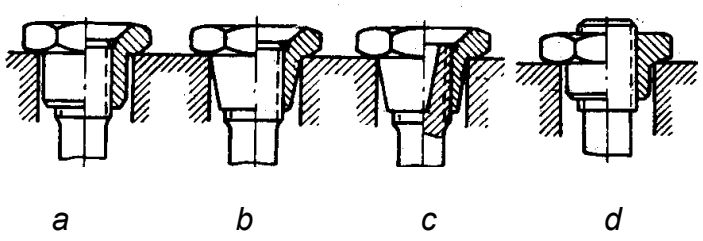


Fig. 17.15. Ways to reduce stress concentration in threaded couplings: shapes of nuts which provide more even load distribution on the thread turns: a – stretched nut; b – conical stretched nut; c – conical stretched nut and the cone on the bolt end; d – stretched and compressed nut

Proper selection of the radii of fillets leads to an increase in fatigue strength of the threaded coupling. To increase strength, it is better to use bolts and studs with the diameter of the rod equal to 0.85...1.0 of the internal diameter of the thread, as it makes them less rigid and equally strong (Fig. 17.16).

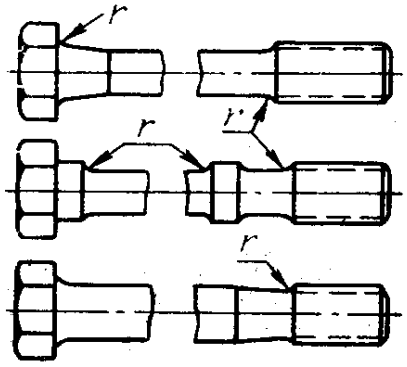


Fig. 17.16. Ways to Reduce Stress Concentration in Threaded Couplings: shape of the transition from rod to head, collars, and the thread of the bolt; r – fillet radius

When designing the special bolts and studs it is necessary to pay attention to the head of a bolt coupling with the rod, as well as the thread of the rod. It is recommended to make the radii r of curvature as follows when engaging the rod with the thread

$$r > 0.2\ d\ ,$$

where d is the outer diameter of the thread.

The radii r of curvature are recommended to make as follows where there is the rod transition to centering bands

$$r>0.5\ d.$$

Small radii of curvature lead to high stress concentration. Fillets composed of two arcs, elliptical fillet or fillet with conical sections, affect favorably the fatigue strength of bolt.

Stress concentration greatly depends on the radius of curvature in the tread spaces. The increase in radius of curvature r = 0.1S to r = 0.2S (where S is the thread

pitch) increments the fatigue strength by 100 %.

Usual metric thread, allowing radius r = 0.108S, is in many cases not strong enough for critical threaded couplings. In such cases, a thread with r = (0.15-0.2) S is used. The radii of curvature r> (0.2...0.25) S are disadvantageous because they reduce the degree of nut and bolt turns overlap and worsen the distribution of load between the turns of the thread due to the lower compliance.

The increase in strength of threaded couplings operating at variable loads, can be reached by introducing the running rollers thread base in areas of stress concentration. Especially favorable are the thread rolling small diameters. Heat treatment after manufacturing thread is always unfavorable (especially after rolling). When running-rollers having residual compressive stress, it is favorable to improve the fatigue strength.

17.2.6. Gears

Damage to gear teeth and splines often occurs due to cracks originating at the base of the teeth, where there is a concentration of stresses. The peaks of stress about the legs of the teeth can be reduced by reducing the rigidity of the gear groove grooves (Fig. 17.17, a). The groove (Fig. 17.17, *b*) simultaneously alleviates stress concentration at the transition from the shaft to the gear teeth and around the legs.



Fig. 17.17. Ways to Reduce Stress Concentration Near Gear Roots: *a* – reduction of the stress concentration near the gear roots by decreasing the gear rigidness: *A* – possible place for a fatigue crack to occur; *B* – grooves for reducing the stress concentration; *b* – reducing the stress concentration near the gear roots and in the place of transition from the shaft to the gear

The nature of the load distribution along the contact lines are affected by the **location of gears** with respect to the bearing supports the shaft. If, for example, cylindrical wheels are in the middle of the shaft span between the shaft bearings, the bending deformation does not have any significant effect on the distribution of load along the tooth length. In the case of unequal distances from the wheels to supports the unevenness of the specific pressures increases.

Especially unfavorable in this respect is the cantilever arrangement of the wheel on the shaft. Therefore, the arrangement of gears tends to be of such a manner that the support shaft located near the wheels or possibly symmetrically with respect to the wheels and their arrangement offset asymmetric high rigidity shafts. In those cases where it is technologically possible, highly stressed gears are manufactured integrally with the shaft. This reduces the number of connectors and connections, resulting in lower weight structures, simplify assembly and reduce manufacturing complexity wheel.

If the gear wheel and the shaft are separate, their assembly is carried out by means of splines and flange.

In the **design of flange couplings** the need to provide radial stiffness crown wheel and a uniform distribution of the load on the fasteners during assembly work are taken into account. Flange coupling must not be located on the circumference of a small diameter, since in this case the operating conditions deteriorate: loads are increased, the number of fastening elements which can be placed on the flange is reduced, and the influence of manufacturing discrepancies at the mating surfaces engaged gets greater. Flanged helical wheel is performed so that the axial force presses the wheel flanges and the shaft to each other.

A chamfer or rounding made by the faces formed by the intersection of the profile of the tooth surface with a cylindrical surface of its top also improve the reliability and durability of the gear. Burrs are removed, more favorable conditions for the formation of an oil layer are created, and the edging tooth contact is mitigated or eliminated. Chamfers are made on the contour and the end surface of the teeth, which is slightly evens the load distribution at the ends.

17.2.7. Holes

Part holes are the sources of stress concentration. There are different design solutions aiming to reduce the harmful effects of holes.

Fig. 17.18 shows some techniques of increasing the fatigue strength of parts with holes. The hole outside countersinking is desirable (if the shaft experiences high stresses, the other hole end in the inner wall surface should also be rounded). The concentration of stress decreases significantly if the flattened surface near the hole is removed.

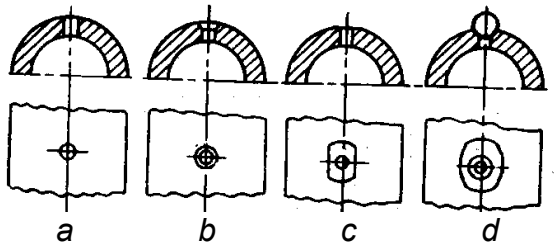


Fig. 17.18. Ways to Increase the Fatigue Strength of Parts With Holes:
a – the part hole is a source of stress concentration; *b* – the hole is countersunk from outside; *c* – the flattened surface from the outside of the part, where there is a hole, is removed; *d* – the hole surface and edges are clenched with a ball

Strengthening of the material surface at the hole by clenching it with hardened steel balls enhances the shaft fatigue strength. To round the hole edges and create favorable residual stresses in the hole a ball A is pressed into the edge. To clench the inner edge of the hole a special device is used.

To improve the fatigue strength of the hole area in parts burnishing is also used.

17.2.8. Bearings

The service life of the helicopter transmission depends largely on the durability of its bearing assemblies.

The durability of rolling bearings for general purpose can vary widely due to vari-

ous factors of metallurgical and technological nature. In this regard, in general engineering the necessary reliability of bearing assemblies is achieved by introducing appropriate safety factors, i. e. a certain overestimation of design loads.

For aircraft parts, where the increase of reliability must be achieved at the expense of design excellence without increasing the overall dimensions and weight of the bearing assemblies, this way is unacceptable, especially since aircraft bearings are made of advanced materials, have high accuracy and are subject to the most careful monitoring during production.

Bearings used in helicopter building must be calculated as accurately as possible, taking into account the peculiarities of their loading and operation.

Lubrication properties have a great impact on these bearings. Reliable operation of many critical helicopter bearing units is only possible when using special oils and greases. This applies especially to bearings, which receive considerable axial loads from centrifugal forces. The analysis of test results shows that for rocking motion bearing wear is largely determined by oxidative processes and inadequate lubrication in the areas of the balls (needles) contact rings. The iron oxidation products obtained, mixing with a lubricant form a kind of a polishing mixture, which causes rapid wear of the raceways.

Oil should be used instead of grease in all the bearing assemblies operating at rocking motion in all cases where it is possible for constructional reasons.

The durability of rolling bearings depends essentially on the quality of the seal bearing assemblies. In case of poor seals, allowing atmospheric oxygen to penetrate into the node, as well as small amounts of lubricants and large air volumes durability of these bearings considerably reduce.

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Chapter 18. TRANSMISSION RELIABILITY PROVISION IN OPERATION

18.1. OPERATIONAL RELIABILITY

Operational reliability is the property of the aviation vehicles to carry out the set functions during the established operating time, keeping values of parameters in the prescribed limits. Practically most often people understand reliability as failure – free, i.e. property of aviation vehicles to keep operating state during the established operating time in certain service conditions.

The reliability represents such technical characteristic, for quantitative assessment of which it is impossible to use usual ways of measurement of technical parameters. It is necessary to define a set of the most suitable indicators of reliability, to establish signs of serviceability (unserviceability) of equipment object, to regulate reference conditions of its operation, to measure the parameters defining reliability in numerous experiments which number is proportional to the required reliability calculation accuracy.

The main indicators of operational reliability of the aviation vehicles equipment are the following:

– the operating time between defects revealed on the ground and in flight; the less is value of this indicator, the worse is, on the one hand, the aviation vehicles in aspect of their reliability, but, on the other hand, the better the engineering and technical staff on identification and the prevention of refusals in flight with all other things being equal;

– the time between failures in flight; the more is the value of this indicator, the better is reliability of the aviation vehicles and work of engineering and technical staff characterized;

– the operating time between defects with consequences for which flight incident, compulsory landing, non-performance of a flight task, a departure delay, non-appearance in flight, flight with the failed engine or helicopter equipment object are considered;

– the fault pattern for the causes of their emergence for which generally mistakes of flying staff and engineering or technical staff in operation of aviation vehicles, low-quality repair, constructive or production shortcomings of aviation vehicles are considered;

– the fault pattern on groups of aviation vehicles (glider, engines, the aviation and radio-electronic equipment, aviation weapons) and glider systems (main rotor and anti-torque rotor, fuselage, transmission, etc.);

– the fault pattern in the place of their identification (in flight, when performing this or that preparation for flights, on scheduled tasks, at target inspection, under repair);

– percent of serviceability of aviation park (squadron, regiment).

Apparently, the basic concepts of operational reliability of aviation vehicles are refusal and nonoperation.

Refusal is the event consisting in violation of object operability of aviation vehicles object (full, partial, temporary), shown in flight or on the ground. The refusals which have come from the moment of helicopter motion start at take-off to the end of the movement when landing belong to failures of the aviation vehicles in flight.

Nonoperation is any discrepancy of helicopter system, unit, assembly, part to

specifications requirements and technical documentation irrespective of the fact whether this discrepancy has led or not to working capacity violation.

Refusal and nonoperation of aviation vehicles can be caused by following failures:

– constructive shortcomings;

– imperfection or violation of the production technology or repair;

– violation of the rules and norms of operation, storage, transportation;

– different damages by foreign objects;

– natural processes of wear and aging.

Refusals and nonoperations causes of occurrence of which are the violation of established rules and norms of operation and also unexpected damages owing to collision of the aircraft with ground objects, birds, hailstorms, etc., at assessment of reliability of aviation vehicles are not considered.

Reliability of the aviation vehicles is laid in the process of its designing, provided in production and maintained in operation.

Maintenance of high operational reliability of the aviation vehicles is reached by:

– exact implementation of service regulations and repair

– continuous improvement of quality of operation and repair on the basis of introduction of the best practices;

– development and deployment of preventive actions on the basis of systematic accumulation and synthesis of operating experience and repair; including the analysis of data on refusals and nonoperations of the aviation vehicles and mistakes of flight staff and engineering and technical staff during its operation;

– timely performance of modifications.

The analysis of the nonoperation causes of the aviation vehicles has to include:

– studying of circumstance of emergence of nonoperation of the aviation vehicles and conditions of their work with use of the recorded indications of devices, records of onboard and ground means of registration of its parameters in flight;

– correctness check of operation and repair;

– selection and study of the statistical materials concerning similar malfunctions;

– determination of nature of nonoperation on external signs and assessment of possibility of its detection by means of the test equipment;

– carrying out checks for determination of nature of nonoperation, dismantling of vehicles (if necessary) to check assembly and parts for condition;

– establishment of defect cause and development of actions for its elimination and prevention.

18.1.1 Operational Factors Influencing Reliability

Thereto:

– compliance of the actual modes and service conditions with set in the course of design;

– qualification of flight staff and engineering and technical staff;

– degree of perfection of the engineering management of the aviation vehicles' operation;

– the control and prevention methods applied in the course of preparation for flight, performance of scheduled tasks, airfield and stationary repair of the aviation vehicles;

– technical supply with spare parts, tool and expendables;

– perfection of technological and organizational methods of operation and repair;

– information objectivity and timeliness arriving from fleets, repair shops, technical warehouses and other organizations which repair, operate and store the aviation vehicles, the industries – to the manufacturer of aviation vehicles about all their refusals and malfunctions; the return timely flow of information from manufacturers to the organizations of the customer about all changes in a design and production of aviation vehicles, and about all restrictions and changes in the settlement modes and service conditions.

Maintenance of the set reliability of aviation vehicles in operation is reached by broad application of modern automation equipment and mechanization, its preparation and control of state, improvement of technological processes and methods, exact and timely implementation of the established rules and norms of operation, service and repair, accurate system of information of the operational organizations and the industries about shortcomings of the aviation vehicles and timely carrying out actions for their identification and elimination.

High reliability of the aviation vehicles has to be provided with the minimum labor costs for one flying hour, and the latest is defined by operational technological effectiveness and the organization of operation, service and repair of aviation vehicles.

Control and Research of Technical Condition. To obtain reasonable data for development of the actions directed to increase reliability of aviation vehicles the engineering and aviation service, the scientific organizations and design offices carry out systematically:

– control and research of technical condition of aviation vehicles at their various operating time;

– the accounting of all refusals and nonoperations revealed at operation, service and repair;

– deep studying and comprehensive analysis of the reasons and conditions of emergence of refusals and nonoperations;

– condition research of the aircraft having the greatest flying hours;

– the analysis of aviation vehicles' operation after carrying out modifications for assessment of their efficiency.

To define an initial stage of development of cracks, corrosion defeats, internal defects of material of parts and nonoperation of units widely it is used physical methods of defectoscopy: magnetic, luminescent, color, ultrasonic, electroinduction, x-ray, gamma raying, etc.

To define change of technical characteristics (parameters) and assessment of operability of units and devices of hydraulic, air, fuel and other systems tool inspections are carried out by means of the automated devices, stands, panels, test equipment, and various devices.

For assessment of technical condition of the radio-electronic and aviation equipment tool inspections are carried out by means of the built-in and not built-in systems of automatic and non-automatic control, test equipment of general and special purpose, and various devices.

The analysis of nonoperations and refusals of aviation vehicles is carried out in such sequence:

– study of circumstances of nonoperation emergence and refusals and working conditions of aviation vehicles (operating mode, operating time, a raid, the number of repairs, etc.);

– check of correctness of operation, service and repair;

– selection and study of statistical materials about similar cases of refusals and malfunctions;

– determination of the nature of the nonoperation and refusal be external signs and establishment of possibility of its test (check);

– conduction of test (check) of aviation vehicles for detection of nature of malfunction and refusal, then (after dismantling) analysis a condition of parts;

– establishment of presumable causes of defect and refusal;

– investigation of influence of the factors defining reliability of aviation vehicles, for check of the drawn conclusions and presumable causes of defect and refusal and establishment of their valid cause;

– development of actions for the prevention of similar cases.

The forecasting of refusals and nonoperations is an evidence-based prediction of their emergence on the basis of the carried-out tests, measurements or supervision. Using data of forecasting, it is possible to replace in due time unreliable units, preventing the emergence of refusals and nonoperations. However, forecasting can be generally applied to those elements, for which nature of accumulation of high-quality changes is revealed.

Comparison method is the method of refusals and nonoperation forecasting based on high-quality changes of units, systems of products by results of measurements of the parameters characterizing their work. For example, if as a result of these measurements the divergence are not found or they are small, the unit (system, product) can be considered reliable. If as a result of comparison of indications of two measurements essential divergences, the unit (system, product) are found they are considered unreliable. In this case, comparing key parameters at the output from the unit, at consecutive measurements it is possible to determine the speed of change of parameters and, extrapolating to determine the time of non-correspondence of the parameters to the rules of technical specifications or performance tolerance. If the parameters are not drastically different on two consecutive measurements, their adjustment is conducted (if possible).

Reservation is a method of reliability increase by introduction of the reserve parts which are superfluous in relation to the minimum functional structure of a product necessary and sufficient for performance of the set functions.

18.2. TRANSMISSION OPERATIONAL REABILITY

Basic constructive elements of units of transmission are:

– tooth gearings;

– shaft (springs);

– bearings.

18.2.1. Gear Train Operational Reliability

The gear train is a pair of meshing gear wheel (gears) designed to pass rotational motion. In some cases, a gear train is used to convert rotary motion into linear one (antitorque propeller pitch change mechanism). In helicopter transmission gearboxes cylindrical gearing (to connect parallel shafts) and conic (for connecting the shafts with intersected axis) are used.

The gear wheels mating is carried out either by their outer diameter (external engagement) or by interacting of one outer diameter wheel with an inner diameter of the other (internal engagement). In the first case the match wheels rotate in opposite directions, in the second – the rotation direction of the wheels coincide.

The gear trains, depending on the location relative to the teeth forming a wheel

are divided into spur, helical and curved (spiral) teeth. When transmitting from the spur gear to non-spur one a smooth operation increases, reduced wear and noise and grows a bearing capacity of engagement that allows higher speeds gearing and transmission of more power than that of spur gears. However, there arises an axial force in engagement, that requires the gears' amplification, setting the axial bearings, except the roller ones, and thrust bearings and, consequently, increases the weight of the structure.

The application of gear trains in transmission is related to their advantageous properties:

- constancy of gear ratio (the ratio of angular velocity or speed of driving and driven gears);
- a large range of transmission capacity;
- high efficiency (up to 0.995);
- compact design elements.

The preferential distribution in the mechanism, including the transmission units of helicopters, got tooth sections, outlined on involute. The involute, or involute of a circle – a curve that any direct point, tangential to the circle, circumscribes, if this line is rolled on a circle without slipping. The transfer of the rotational movement is possible by non-involute sections as well, to which, in particular, circular sections of M.L. Novikov's engagement system are related.

Each pair of meshing teeth cycle begins when the base of the drive gear tooth contacts with an apex of the tooth of the driven wheel. In the process of further joined tooth profiles rolling, the pressure transmits to a driven member. The engagement ends in a position when a drive wheel tooth pushes its top to the base of a driven gear tooth. As the tooth profiles interact with different areas of the involute, i.e., with the arcs of different lengths, their relative sliding motion occurs. The further from the pole is the contact point of sections, the greater is the difference in their respective arcs, and the greater is the slip (with no slipping in the pole), and direction of sliding is reversed, when passing through the pole.

This feature of kinematic castellation creates conditions for **working face teeth surfaces wear**, particularly at their heads and feet. Another feature of the gear – an **advanced level of specific pressure** in the teeth contact zone, because all power is transferred from the driving wheel to the driven wheel substantially through one tooth. As a result, in some areas of a contact it is possible to make something like metal particles welding due to the occurrence of molecular adhesion forces between the mating surfaces. This phenomenon is called **clutching**. The combination of high rotational speeds with high specific pressures leads to almost instantaneous significant increase in surface temperatures in a contact zone as well as an intense heat, which also contributes to clutch.

The effective measures to prevent wear and gearing clutch as well as ensuring its operability are:

– the right choice of gear wheels' material;
– an increase of the teeth working surfaces wear resistance;
– the application of an appropriate lubricant type.

The gear wheel materials must be necessarily resistant to ritting and sticking, work well to take up bending moment and allow to make the wheels teeth according to the required accuracy and surface cleanliness. The alloy steels 12X2H4A and 18XHB that meet all the requirements are used in helicopter gear wheels gearboxes. The increase of gear wheels wearing capacity is reached through a thermochemical

processing of their working surfaces that lie in the saturation of steel layer surface by various elements became for its hardening. The hardened gear wheels resist to heavier loads and thus enable creating assemblies with smaller dimensional specification weight.

The toothed coupling lubrication must have such a viscosity to save the oil film between the mating surfaces of parts that protects the teeth against wear and clutching.

Specific pressure in the contact zone of the teeth are large, and the thickness of an oil film, even providing friction of fluid lubrication, should not be less than the total value of the size of the irregularities and roughness of the mating surfaces. Therefore, the accuracy and purity of the gear teeth surfaces impose very high requirements.

A lubricant heat stability, its characteristics' stability, a well-organized heat abstraction from the contact area to the cooling elements by a lubricant, contribute to long-term reliable operation of gear connections. These requirements met in transmissions used in helicopters oil for hypoid gears, synthetic oil B-3V, a mixture of oil and hypoid oil AMG-10.

18.2.1.1. Side Clearance Between Teeth of Engaging Wheels Check

A sideway clearance is the shortest distance between profiled surfaces of the engaged wheels' related teeth in a transmission, that provide free rotation of one wheel with the other one being motionless).

Guaranteed side clearance. Special rules are set regardless to the wheels' and gearing accuracy rating. The basic rules are the normal pairing guaranteed clearance (marked by letter X), which provides compensation to reduce clearance from the gearing heat. It is allowed to change the rules of the guaranteed backlash and apply one of the following interfaces: C – with zero clearance, D – reduced and Sh – with increased one.

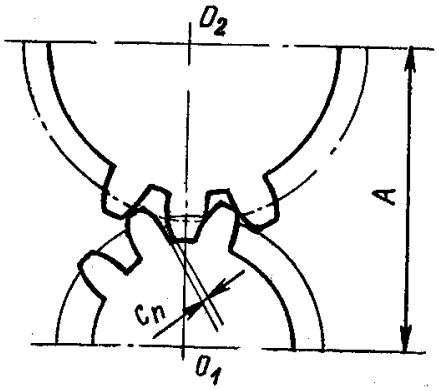


Fig. 18.1. Side Clearance Between Teeth of Engaging Wheels

A side clearance C_P for cylindrical wheels (Fig. 18.1) is determined in the cross section which is perpendicular to the teeth, in a flat tangential surface to the axes of main cylinders of two main match wheels, for bevel – along normal to the side surfaces of the teeth at the large base pitch cone.

The clearances of cylindrical and bevel gears are set (regulated) in various ways. A sideway clearance C_P between the cylindrical gears pair of teeth at all equal

conditions depends on the center distance A. The change of center-to-center distance ΔA may be due to a change in the ΔC_P clearance by the following ratio:

$$\Delta C_P = 2\Delta A \sin\alpha,$$

where α – gearing angle, determined in a ratio:

$$\cos\alpha = (r_{01}+r_{02})/A,$$

where r_{01} and r_{02} – base radius of mating wheels.

The clearance changing by wheel selection. As the center-to-center distance does not change with a few exceptions, so the only way to change the clearance is to select a pair of wheels, i.e. by using the variations of tooth thickness within tolerance of teeth geometrical parameters.

A sideway clearance should be measured nearly by a radius of a pitch circle with a feeler or indicator. To do this, the feeler of a desired thickness probe is inserted between the teeth of a meshing wheels pair in the pitch circle. This method is not accurate, as the curvilinearity of the tooth surface does not allow to set the probe in position. A more accurate way of measuring of the clearance is by an indicator of the gap.

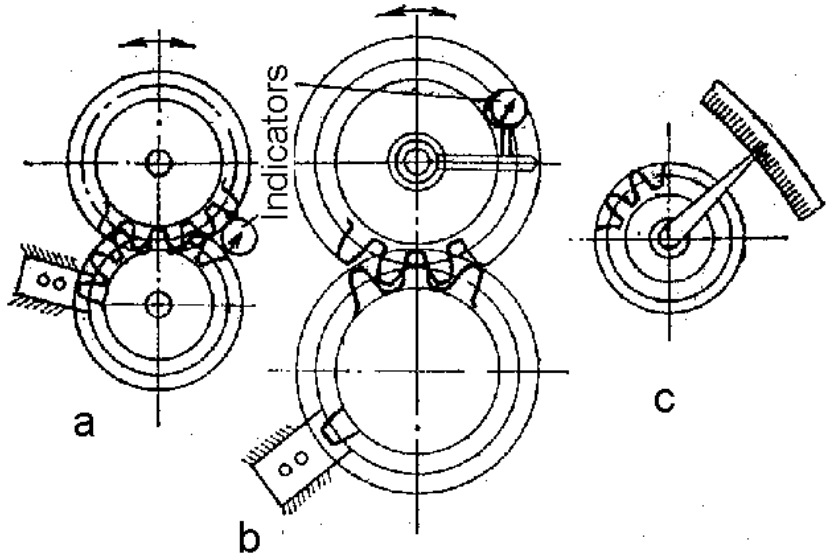


Fig. 18.2. Methods of Side Clearance Measuring: a – measurement by an indicator, b – remote indicator measuring method, c – arrow and scale measurement

A pin (leg) of the indicator is pressed against a tooth of one wheel at the point (Fig. 18.2 a) lying approximately on a radius of the pitch circle. The other wheel is fixed stationary.

By shaking a free wheel, the indicator value that will correspond to the value of a sideway clearance, is marked.

The measurement of a clearance with a driver pin or an arrow. If it is impossible to measure a clearance with an indicator directly on the gear, so a roller with a driver pin is applied (Fig. 18.2, b).

The roller is inserted into a central hole of a wheel and fixed (i.e. by a split terminal).

A mark corresponding to the radius of the initial circumference of the wheel is applied on a driver pin.

Then a pin of an indicator is put to a mark and the clearance is measured as described above.

In the absence of an indicator, you can use the arrow and the scale (Fig. 18.2, c). In this case, the scale is calibrated in linear terms, considering the scale, or the ratio of the radius of the initial circumference of the wheel and the circle of scale divisions.

When regulating a side clearance the feature of the bevel gears is that this clearance is changed by moving the wheels along their axes. However, moving the wheels disrupts the engagement, which, as known, is considered to be correct if the initial vertex cone wheels coincide. Practically one or both wheels move. The wheels are positioned by using the shims of the required thickness. In some cases, the shim is set under the gear bearing housing flange. The thickness of the shim is determined by the selection method.

A side clearance in the bevel gears pairs are measured with an indicator in the same manner as in the cylindrical wheels.

The incorrectly matched clearances lead to increased wear of the teeth, noise, vibration, errors in the adjustment mechanism, driven by the wheels, and can cause damage to the teeth.

The values of the side clearances for various products are indicated in the task cards (drawings), regulations or instructions.

18.2.1.2. Determination of Wheels Engagement Accuracy on the Contact Spot

The correctness of the contact of teeth surfaces of the gearing wheels is determined on the contact spot, which is part of the flank of tooth of the wheel, on which the traces of its fit to the teeth of the doubled wheel after rotation of the assembled transmission at lightly braking are presented.

The relative sizes of the spot. Both for cylindrical and bevel wheels the correctness of the contact is determined by the relative sizes of the contact spots (which can be solid or with a gap) in percent (Fig. 18.3, a):

on the length of the tooth – the dependence of the distance between the end points of traces of the fit, excluding breaks, exceeding the value of the module to total length of the tooth, i.e.

$$[(a\text{-}c)/B]100\%;$$

on the tooth height – the dependence the average height of the fit spot along the entire length of the tooth to the working (active) the height of the tooth, i.e.

$$(h_{AV}/h_T)100\%.$$

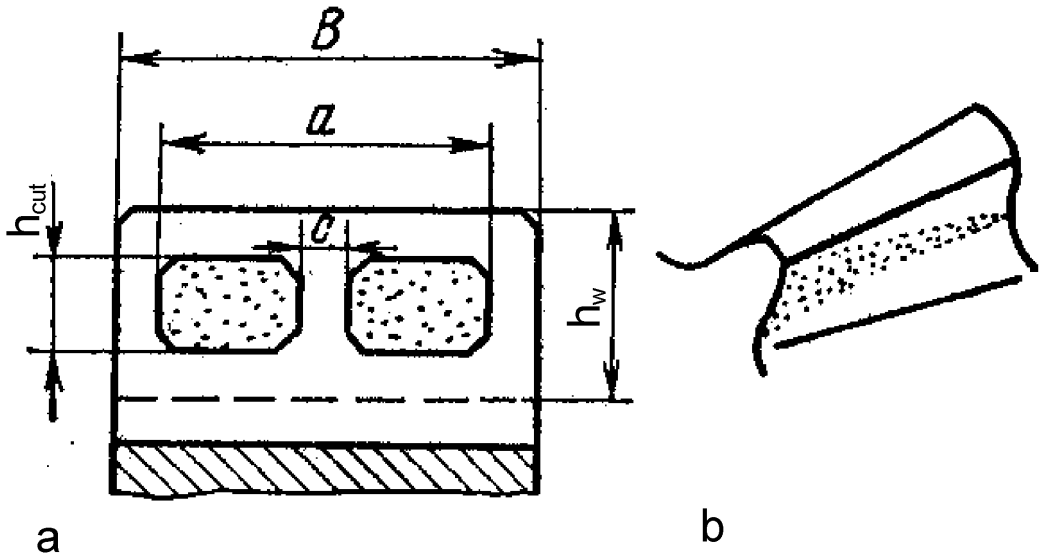


Fig. 18.3. Contact spots: a – with a gap on tooth of cylindrical wheels, b – correct form of the spot on the conical wheel

Rates of the size of the contact spot for the screw wheels of different degrees of accuracy are set by the standards.

Spots of conical teeth as opposed to cylindrical wheels where the permissible contact spot is located symmetrically along the length of the tooth, in the bevel wheels the spot can be shifted in the direction of more subtle parts of the tooth (Fig. 18.3, b). This considers the deformation of the transmission during operation. To facilitate inspection of spot fit the special control cards are created, these cards have the allowable distortion of the shape and position of the contact spot.

Almost gear pair engagement is controlled on the paint as follows. A thin layer of paint (Berlin blue) is applied to the surface of the wheel teeth with fewer numbers of teeth. At complete revolution of the large wheel the paint is transferred to all of its teeth, forming an imprint on them. Instead of one of the wheels it is possible to use a model wheel, and then paint is applied to this model. For the best print the second gear wheel (driven) is slowed down slightly to create a small pressure on the teeth for a more distinct imprint when you rotate the wheels.

With good engagement of teeth the mark on the tooth of the driven gear must be located on the line of engagement (Fig. 18.4, *a*), and its length should be 60-70 % of length of tooth. When the distance between the axes of gears is bigger than permissible the mark moves to the head of the tooth (Fig. 18.4, *b*). When the distance between the axes of the toothed wheels is less than the allowable one the mark moves to the foot of the tooth (Fig. 18.4, *c*). In both cases, the fit of the teeth is unsatisfactory. The one-sided location of the mark relative to the length of the tooth indicates the misalignment of axes of the gear wheels.

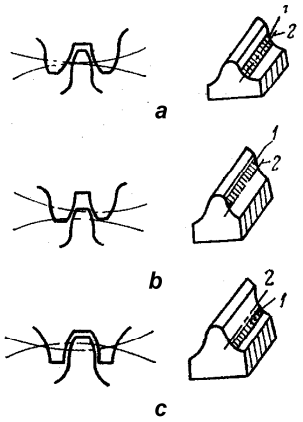


Fig. 18.4. Determination of Quality of Cylindrical Wheels Engagement on Print:

a – right engagement, *b* – the distance between the centers is bigger than normal, *c* – the distance between centers is less than normal; 1 – paint mark, 2 – axis

To eliminate errors characterized by improper marks, in some cases is possible by selection of pairs of wheels, the separate errors of which cancel each other out. Removal of errors by means of the metal fitting is unacceptable, and therefore, in such cases it is necessary to change the whole part or its separate elements.

Check by touch. Smooth engagement and rotation of toothed wheels also is checked by touch when rotating one of the wheels. When smooth running the toothed wheels should be run without jamming and jerks approximately with constant torque moment and they should work without noise.

Gears transmissions used in gearboxes of helicopter transmissions present delay mechanisms, significantly reducing rotational speed and increasing torque moment of the output shafts of motors to ensure optimal modes of operation of main and tail rotors. It is required to implement this in the transmission by large reduction ratio, reducing frequency of rotation of the motor shaft 50 to 70 times. However, in one pair of toothed wheels from the point of view of overall dimensions it is unreasonable to do ratio bigger than 5.

Also, when $I > 5$, the teeth of small wheels get a significantly larger number of loading cycles than the large teeth, it can result in a faster wear out. Therefore, in those cases, when the given gear ratio exceeds the value that is appropriate for one pair of wheels, it is reasonable to use the mechanism, comprising a series of toothed wheels. Such a mechanism in helicopter transmission is the main gearbox.

18.2.2. Main Gearbox Operational Reliability

The main gearbox is the most complex, large-size and heavy transmission assembly. It transmits high torque moments to units-consumers and serves as a central load-bearing element of the helicopter, i.e. directly takes up all the forces and moments generated by the main rotor.

The application on helicopter gas turbine engines with a much higher frequency of rotation of the output shafts than piston engines required for maintenance of optimum performance of the main rotor to increase substantially the degree of reduction (increasing of output shaft frequency of rotation) of main gearbox.

For heat removal and lubrication of bearings and gear teeth the main gear box has an **autonomous oil system**, including oil pump (one pressure and two scavenge stages), oil filters and cooler. In the oil system they use synthetic oil B-3V, receptacle for which is the lower part of casing of the main gearbox – pallet. The magnetic plugs for trapping steel particles from oil are fixed in the pallets this enables to control the technical condition of the main gearbox parts in operation. Cooling oil radiators and casing of main gearbox is maintained by a flow of air from the fan.

The main gearbox is provided with the **means of monitoring** of its serviceability

and technical condition. Thus, the degree of refilling of gearbox by oil is determined visually through two sight glasses located on the left and right surfaces of the body of oil reservoir. To check the status of parts of the gearbox the magnetic plugs subjected to periodic removal and inspection are mounted. Temperature and oil pressure control is carried out with the help of instruments. There is indication of the chip presence in oil. The recommended oil temperature can be maintained in flight by changing the angle of setting of the turning fan blades by means of the electromechanism, which is controlled from the workplace of flight technician.

The main gearbox frame is used to mount it to the fuselage and takes up static and variable dynamic loads transmitted from the main rotor. The most common design frame consists of eight struts made of high strength steel and are pairwise connected by four Y-shaped fork. Each fork is attached to two units on a crankcase of the main gearbox and to one units on the fuselage. The development of this scheme is the frame of the main gearbox of helicopter Mi-26, each of the eight struts which is independent and attached in two points on the crankcase of the gearbox and on the fuselage.

In consideration of the high level of loads transmitted to the frame of the main gearbox, and an opportunity of development of fatigue cracks, the welded joints of the fastener assembly of the main gear box to the frame and the frame to the fuselage should be inspected periodically with a magnifying glass, mirrors and illumination from portable lamps. If it is necessary use a portable magnetic flaw detector or device for search of cracks by the method of color radiography.

Operational control of technical condition of gearbox has its own characteristics. Since visual inspection and methods of direct inspection of parts and assemblies, located inside the gearbox are impossible in operation, the condition of the oil filling in the gearbox is regularly monitored during the assessment of their technical condition. During operation of the gearbox the oil washing all the gears, bearings and parts of their attachment adsorbs the products of wear and thus is a data carrier about the character of the processes of wear in the gearbox and the technical state of the elements of its construction.

To detect the metal particles trapped in oil the following work is periodically carried out:

– drain a small amount (2...3 l) of oil from the gearbox through a funnel with mesh, and visual inspection of the purity of the oil;

– inspection of magnetic plugs;

– inspection of oil filters.

When mounted main gearbox on the helicopter instead of taken off due to the discovery of the metal particles in oil all the pipelines of the oil system of the main gearbox are pre-washed and the oil cooler is replaced. Another indirect method of checking the serviceability of the gearbox and transmission in general is cranking of main rotor on the ground by hand and assessing the easiness of transmission rotation (should not be extraneous noise and jamming).

Instrument control of gearbox serviceability is carried out by two parameters – oil pressure and temperature, their deviations from set parameters for the values of relevant flight modes inform the crew about fault conditions of the gearbox.

Thus, a simultaneous increase in temperature and falling oil pressure happens at the tightness of the reducer or the destruction of its parts.

Specified emergency character of the change of parameters can occur with quite good gearbox as a result of failure of the thermo-valve of oil cooler of the oil system of

the main gearbox. As a result of failure of the thermo-valve hot oil passes by the cells of oil cooler, and does not cool. As a result, oil temperature rises above the permissible in the main gearbox and pressure of the oil drops.

When the temperature of the oil increases the removal of heat from bearings and other parts of the main gearbox considerably reduces, the viscosity of the oil decreases and its lubricating properties deteriorate. All this can lead to bearing damage.

In such cases, the helicopter crew must stop the execution of the flight task.

18.2.3. Freewheel Clutches' Operational Reliability

Freewheel clutches are mechanisms that perform their functions automatically throughout the period of operation and does not require any maintenance, adjustments or special maintenance, except for preheating before starting the engine at low outside temperatures.

In the conditions of inclusion and jamming of the freewheel clutches a significant influence of the **oil viscosity** has great influence due to the following circumstances:

– with the increase in the viscosity of the oil the strength of the oil film separating the contact surfaces of jammed rollers, holder and sprockets of the freewheel clutch increases; at insufficient pressure force on the rollers the slippage of the clutch during the period of jamming is possible due to the lack of adhesion of the rollers with the working surfaces of the holder and sprocket;

– with oil viscosity growth the viscous friction force between the separator with an sprocket increases, as a result the working conditions of activation device make worse and the value of hold down pressure on the rollers from the side of the separator decreases.

Used in main gearboxes and the freewheel clutches of helicopters the B-3V synthetic oil has a significant advantage over mineral oils – ensures engine starting without preheating the oil in the gearbox when outside temperatures up to –40° C, due to its best viscosity-temperature characteristics, low temperature properties and lubricity. In the temperature range from – 60 to +100 °C viscosity oils B-3V in average two times smaller than the viscosity of mineral (petroleum) oil mixture used for the lubrication of main gear boxes and freewheel clutches and requiring preheating before starting at low temperatures.

The reliability of the freewheel clutches largely depends on the following:

– perfection of their design;
– quality workmanship;
– the dynamics of transmission;
– modes of operation of the power plant.

As a result of this diversity of factors influencing on the reliability, the nature of the manifestations and consequences of failures of the freewheel clutches of different helicopters have both general regularities and individual peculiarities.

The main types of failures of the freewheel clutches:

– switching failure;
– impact clutch;
– disengagement.

The switching failure of free-wheel clutches is refusal, which concluded in short-term or long-term absence of contact of the rollers with the working surfaces of the sprocket and the holder when starting to move the sprocket relative to the holder in the

direction of torque moment transmission. The switching failure is the result of unreliable operation of clamping devices, in the result of which the contact and the clutch of the rollers with holder assembly and the sprocket does not come at all or come with some delay. In the latter case, the wedging of the rollers begins at different angular speeds of sprockets and holders and is accompanied by impact, i.e. the impact clutch happens.

The contact fault of the rollers with the working surface of the holder or the sprocket platform in the initial time period of clutch (engagement failure of freewheel clutches) can occur as a result of the following causes:

– action of the dynamic loads on the rollers in the wedging direction;
– the existence of a stable oil film on the contact surfaces of the parts;
– wedging action of the hydrodynamic forces in the lubricant layer by relative motion between working surfaces of parts;
– malfunction of the clamping device (breakage of the spring, jamming of the separator, etc.).

The main condition for reliable engagement of freewheel clutches is the presence of a sufficiently large clamping force, ensuring the overcoming of all the forces that prevent the formation of the contact of roller with holder and sprocket.

Feature of switching failure is that on contact surfaces of coupling parts any damage is not formed. Damages arise only in cases when, as a result of not including short impact clutch and slipping the clutch happens.

In operation considering **failure of freewheel clutches** is accompanied by the following external things:

– the lack of rotation of the main and tail rotors after start of the engine;
– intense increase in the rotor speed of the engine free turbine;
– the availability of high frequency engine noise;
– growth of high-frequency vibrations of the helicopter, followed by a rattle and blows in the region of the main gearbox,
– sudden jerks of propeller blades in the plane of rotation (with a short un-switching and subsequent impact clutch of the coupling with a slippage of the rollers);
– automatic engine shutdown by protection turbine screw system from spin-up;
– the emergence (in cases of severe manifestations of failure) of metal chips on magnetic plugs and oil filters of main gearbox indicating the damage of the contact surfaces of the parts freewheel clutches with impact clutch and slippage of the coupling.

The risk of failure of the freewheel clutches, which consists in the **possibility of rapid spin-up** with the destruction of the rotor of the free turbine of the engine by centrifugal forces in the case of a breach in the process of rigid connection of the engine with the main gearbox made necessity of (among other activities) development and installation of the automatic system of screw turbine (free turbine) protection from spin-up on the helicopters.

The principle of operation of this system is that when failure of freewheel clutches and the subsequent spin-sup of a free turbine rotor an automatic shutoff of the fuel supply line to the engine happens. In this case re-starting is impossible without the implementation of special measures to bring automation to its original position.

Impact clutching of the clutch is, essentially, the second phase of short-term disengagement of the coupling. At impact clutch as a rule the slippage of the clutch occurs due to the deformation of its parts and the contact surfaces of the parts, in the result of which the formation of craters on platform of sprocket and the working surface

of the holder, and edges – on the cylindrical surfaces of the rollers is possible in zones of contact of the rollers with the slip and the sprocket. The percussive coupling of the clutch is accompanied by sharp jerks of main rotor blades in the plane of rotation, grinding sound and blows in the region of the main gearbox, growth of helicopter vibrations.

A characteristic **feature of declutching** is that it occurs not at transient conditions and at steady state, when the clutch is already engaged and a tight coupling is made between the engine and the main gearbox. The failure is accompanied by a sharp increase in the frequency of rotation of the free turbine rotor, beats and screeching in the area of the main gearbox, the actuation of the turbine protection system of screw turbine from spin-up.

The main described types of failures of the freewheel clutches are prevented both constructively by **improving their production** and establishment of **special rules and restrictions** in operation aimed at improving the reliability of coupling and providing for the following basic preventive measures:

– oil heating in the main gearbox and the freewheel clutch before starting engines at low ambient temperatures;

– exclusion or change of engine operation modes, which can create conditions for the decoupling of the freewheel clutch;

– the limited resources of the free-wheel clutches, which are the main cause of failure is the premature wear of parts.

Thus, the application of oil B-3V in the main gearboxes of Mi-6 has had an adverse impact on the process of clutching of the freewheel clutches; the B-3V oil was replaced by a mixture of mineral oils MK-22 and MK-8.

In cases of unreliable operation of the freewheel clutches on the Mi-6 in the process of different regime operation of the engines and the dynamic jamming of the coupling in simultaneous rotation of the sprocket and the holder in operation were introduced the following **restrictions**:

– prohibited was the testing of one engine while the second engine in the idling mode;

– prohibited was the execution of the planing of the helicopter when working on engines idle; the mentioned airplane mode was performed at increased idle, that has eliminated the mode of engine operation in which the rotational speed of the free turbine being less than the rotational speed of the holder of the freewheel clutch and fail (wedging) of clutch happened;

– introduced control of freewheel clutch operation using two indicated pointers when starting and warming up engines and taxiing and in flight.

Because of increased wear of freewheel clutches on helicopter Mi-2 the following measures to prevent their failure are taken:

– periodic (every 200 hours of operating time) disassembly of the clutches to assess the technical condition of their components;

– mandatory replacement of freewheel clutch after 200 hours of experience in helicopters, operating in the training units;

– limit of the cumulative operating time of the clutches at the alternate modes of testing engines, flying with one engine and flying on main rotor autorotation.

However, the constructive improvement of freewheel clutches has been carried out:

– to improve the reliability of engagement of the rollers;

– to reduce the impact of the dynamic loads from the propeller shaft connecting

the engine to the main gearbox on performance of the clutch.

In the result a new clutch, having the following main design features has been created:

– increased the number of support bearings of the drive shaft (two to three);

– increased the distance between the supports, which excluded the console connect drive shaft with driven shaft with a clutch, it is substantially increased the accuracy of centering and reduced the possibility of mutual misalignments of the shafts during the clutch operation;

– increased accuracy of centering of the separator relative to the drive shaft by means of two sliding bearings, which increased the synchrony of jamming of the rollers in the clutch and contributed to a more even load distribution along their length;

– uniform stiffness of the holder in the area of the rollers, which increased the uniformity of loading of the rollers along the length;

– improved the organization of the supply of lubricant to the points of rollers' contact with the sprocket of drive shaft (lubricant is provided in the vicinity of the contact line), which increases the durability of the freewheel clutch parts.

18.2.4. Tail Transmission Units' Operational Reliability

The intermediate gearbox provides change of the torque transmission direction from the main gearbox to tail without change or with little change of frequency of rotation.

The gearbox is installed in a joint of tail and trailer beams and consists of one couple of conic gear wheels. The case of intermediate gearbox is cast from a magnesium alloy, gear wheels are produced from the high-alloyed steel, for ensuring reliable work of support the steel holders are installed in them.

Gearbox lubrication is splash one, i.e. one of gear wheels is partially shipped in oil and at rotation greases the second gear wheel at the time of teeth contact. At the same time splashes of oil create oil fog in a case, providing hit of oil on a gearbox parts.

Besides, splashes of oil get to the special pockets cast in a gearbox case and on the available channels oil spontaneously arrives to its bearings. In intermediate gearboxes of average and heavy helicopters greasing is carried out by compulsory supply of oil by means of the gear oil pump put in action from a shaft of the conducted gear wheel through a special spring. At the same time also splash – the duplicating oil system is provided for gearbox greasing in case of refusal of the main oil system. For check of amount of the oil which is filled in a gearbox the control of oil level by means of oil-level gage, on which risks of maximum and minimum admissible levels are caused, is provided. Pressure and temperature of oil in a gearbox are controlled on devices.

Tightness of the driving and driven shafts is reached by installation of shaft labyrinth, protection of bearing blocks against hit of dust is carried out by the felt washer impregnated with graphite greasing.

Cooling of an intermediate reducer is blown by the running stream of air or its forced blowing by means of the special fan (on Mi-6 helicopter). The Mi-26 helicopter is provided with the external oil system with an oil assembly for cooling of intermediate gearbox .

The **tail gearbox** is a final link in system of power transfer on the tail rotor, provides the necessary frequency of rotation of the rotor and management of the angle of installation of its blades. Transfer of power to tail gearbox is carried out on majority of helicopters, as well as in intermediate gearbox, by means of one couple of conic gear

wheels, however with bigger degree of a reduction (rotation frequency of the driven shaft of a tail reducer decreases in two or three times in comparison with the rotation frequency of the main shaft).

Greasing of a tail gearbox is carried out similar to greasing of intermediate gearbox – forcibly, by means of an oil pump, and splash lubrication. Behind the level of oil it is controlled by means of oil-level gage. The available devices allow to control pressure and temperature of oil in gearbox. Gearbox is cooled from the running stream of air and inductive stream from the tail rotor. The Mi-26 helicopter has an external oil system for more intensive cooling of tail gearbox oil.

Operational control of technical condition of intermediate and tail gearboxes are carried out as well as the main gearbox.

The **tail shaft** serves for torque transmission from the main gearbox to the tail rotor. The shaft consists of the separate pipes connected either rigidly (by flange joints), or by means of restrictedly mobile elements (the spline coupling, the cardan and sliding spline joints). Such design has sufficient stiffness and at the same time provides a possibility of small distortions and axial movements of separate sites of a shaft, unloading it from the additional tension connected with inaccuracies in lengths of pipes and distances between shaft bearing and also with a bend of tail shaft in flight.

The tail shaft is subjected **in operation** to regular inspections and tool control. During inspection they check whether there is no twisting of a shaft (along a strip applied to it) and weakening of nuts of fastening of conical bolts. Tool control of tail shaft consists in check of its beating by means of the dial test indicator.

The tail shaft is installed on the support fixed by bolts on frames of a fuselage, tail and trailer beams. Each support has a seat in which the shaft radial ball-bearing is placed on a rubber holder. The rubber holder is intended for compensation of distortions and not coaxiality of support, and also for depreciation of cross fluctuations of a shaft.

The distance between tail shaft bearing is chosen such to prevent resonant fluctuations of a shaft. The rotating shaft possesses a large supply of kinetic energy which at certain combinations geometrical and the stiffness properties of a shaft and speed of its rotation can turn into energy of cross fluctuations. At the same time rotation becomes unstable, there are big deflections of a shaft on sites between support. Frequency of shaft rotation with which there is a stability loss is called critical.

According to requirements imposed to transmission at design of tail shaft the margin on rotation frequency excluding a possibility of loss of stability by it in operation is created.

Tail shaft bearings need to be examined regularly for identification of cracks and weakening of rivets of fastening the support to frames of tail boom. At the same time it is necessary to check technical condition of bearings of support very carefully.

Besides a traditional set of units the tail transmission of the Mi-24 helicopter has one more gear mechanism – **a drive gearbox**. It carries out function of intermediate (between the main and tail gearboxes) stages of reduction and is used for the drive of two generators of alternating current. The box of drives is a two-stage gearbox in which are established six spur gear wheels on four shafts. The torque from the main gearbox is divided in a box of drives into two flows: on shaft of the drive generators and on flange to which the middle part of a tail shaft joins.

For cooling of a box of drives the special fan and internal oil system with compulsory supply of greasing to gear wheels and bearings are used. For periodic control of serviceability of parts of box of drives a magnetic plug is installed in it.

The **main rotor brake** is intended for a compulsory stop of the main and tail

rotors after disengagement of engines, and for fixing of transmission during the parking and at towage of the helicopter. The drum brake is installed, as a rule, in the place of connection of the tail shaft to the main gearbox.

Basic elements of a brake – the drum which is rigidly connected to a tail shaft and the frictional blocks connected by a control system to the handle of a brake located in a crew cabin. Braking is carried out by pressing of blocks to a drum cylindrical surface. In the released state between surfaces of a drum and blocks the small gap by means of the coupling spring connected to blocks is supported. Brake control is blocked with system of start of engines in such a way that it is possible to start engines only at completely released transmission.

18.2.5. Transmission Spline Joints' Operational Reliability

Loading the spline joints by torque on condition of its uniform distribution between all splines and on all length of their interface is rather small. At the correct production the spline parts, geometrically exact assembly of couple, absolutely rigid parts there shouldn't be wear of surfaces of splines to separation of metal particles from surfaces of contact.

Material of splines in such connection has to work only for crumpling. Therefore their wear can happen only in the form of process of plastic deformation until there is a crumpling of edges of roughness and equalizing of pressure on a contact surface. Upon termination of process of running-in the spline joint has to work very long time practically without increase in a gap between splines.

However, in real designs the situation is a little different. Owing to inaccuracies of production and adverse combination of deviations, admissible at production, in sizes of parts in spline couple inevitably there is some not coaxiality – a distortion (break) of axes or their radial shift.

At **distortion of axes** all loading is transferred only through two opposite splines, and not along all their length but only along the small sites located on the ends of connection. Contact of one of splines is at one end of connection, and opposite – at another. At rotation of parts other couples of splines serially come into contact. For one turn of a shaft every spline comes into contact with spline of the mated coupling part twice (the first time contact occurs at one end of mating end, the second time – at another one).

At the **radial shift** of axes of the coupling parts all loading in not run-in spline connection is taken up only by one spline (at rotation of a shaft every spline is serially loaded), i.e. at disengagemtn from gearing one spline engages the following, etc.

Besides, discrepancy of axes of parts – the cause of emergence of movement under loading (sliding) of surfaces of splines from each other that directly creates necessary conditions for wear of parts. Thus, discrepancy of axes of the spline mated parts causes sharp reduction of the area of their contact and, therefore, increase of pressure in places of the actual contact, and the high pressure and sliding in a zone of contact create conditions for wear of splines.

In such conditions the importance is got by **greasing** of the rubbing surfaces of parts of the spline coupling, as well as in usual frictional units. The choice of a type of greasing for spline – low-viscous or jellied – is carried out proceeding from the analysis of operating conditions of individual spline couple and nature of influence of the specified types of greasing on wear process.

Thin oils at a small speed of movement of the contacting parts do not eliminate their direct contact, however, washing contact surfaces, delete from them the formed

firm particles of oxides of products of wear and thus protect them from abrasive destruction.

Thickened lubricants usually well protect spline couplings from wear during the certain period of work. In process of gradual accumulation of abrasive particles in a layer of greasing – products of wear and loss of greasing to fast prevent oxidation the wear of connection surfaces begins.

For lubrication of spline couplings of helicopter transmissions the oil for hypoid gears is used. Regular replacement of the oil, provided by instructions and maintenance regulations, ensures removing of wear products and recovery of oil lubricating properties.

However, the lubricant reduces only the rate of wear but does not eliminate the wear of splines, if any mismatch of geometric axes of coupling parts has a place. Hence, **the quality of their manufacturing and assembly is of great importance f**or the reliable operation of spline couplings. It should provide such mutual position of the contact parts of the coupling at which the mismatch of their geometric axes is minimal. This can significantly increase the actual contact area of the parts in the coupling, approximate contact pressure values closer to calculated ones, minimize sliding of spline surfaces relative to each other. Spline couplings of tail shaft of helicopters rather reliably complete the set time between repairs, however need periodic checks of a state the spline joint.

Degree of wear of splines in operation directly can't be measured therefore a lateral gap between splines of the mated coupling parts is checked. The way of check involves transmission slow down by means of MR brake and by applying loading (effort of a hand) to the blade of the tail rotor a tail shaft is turned serially in both parties to a complete elimination of gaps in spline connection. At the same time a gap is measured by an adaptation with the indicator which is installed serially on each checked coupling.

It is regularly necessary to control the **degree of misalignment** of the coupled parts of the spline coupling. For this purpose similar adaptation is used which during checking is installed on the case of the spline coupling, and at rotation of transmission by main rotor blades the indicator measures misalignment (a break of axes) of coupling parts.

18.3. PREDICTION METHODS OF RELIABILITY RATES

As previously stated, the main reliability rate is the probability of failure-safe operation. Any other reliability rate of a non-repairable item is evaluated through it. The *reliability rates* of non-restorable items can also be applied to restorable items when considering their work to the first failure.

The main indicator of restorable items' reliability is an assessed failure rate or mean time before failure. Depending on the definition of failure, the same system can be regarded as restorable or non-restorable.

During the flight, an abnormal case, which reduces the safety of the flight, is possible. As prescribed in aviation rules, all abnormal situations are divided into four groups in order of danger: catastrophic, emergency and difficult situations, as well as the complexity of the flight conditions.

The increasing complexity of the flight conditions is a special situation, which is characterized by a slight deterioration in performance, in consequence of which the

flight plan can be changed or the slight workload of the crew can increase.

Complex situation is an abnormal situation, which is characterized by a noticeable deterioration in performance and (or) the release of one or more flight variables of the operating restrictions, but without reaching the extreme limits or reducing the ability of the crew to cope with adverse conditions (the arisen situation) both because of the increased workload, and of conditions, reducing the effectiveness of crew actions.

Emergency is an abnormal situation, characterized by a significant deterioration in performance and/or by an achievement (excess) of limit restrictions or physical fatigue or it can be characterized by such cockpit workload, that is no longer possible to rely on the fact that they will perform their tasks accurately or completely.

The catastrophic situation is an abnormal situation for which it is assumed that when it occurs, the prevention of people loss of life is practically impossible.

The limits on the probabilities of occurrence of specific situations for 1 hour flight were set, according to the safety requirements of Aviation Rules for transport aircraft of commercial aviation. Therefore, the main *flight safety indexes* are the following probabilities:

– the catastrophic situation q_{cs},
– emergency q_{es},
– complex situation q_{cxs},
– the increasing complexity of the flight conditions q_{fcc}.

The following factors, which evaluate the level of flight safety, are connected with the probability of a catastrophic situation happening:

K_1 – the number of catastrophes per 10^5 of flight hours;

K_2 – the number of catastrophes per 10^5 of flights;

K_3 – the number of catastrophes per of flight distance in km;

K_4 – the number of fatalities per 10^8 passenger-kilometers.

The structural modification and the improvement of maintenance conditions and the repair of airplane during the initial operating period – it all increases the flight safety.

The dependence of coefficient K_1 from the aircraft flight hours is shown in Fig. 18.5.

One of the reliability rates is the durability, which in its turn, may be characterized as a service life. As for the most of the helicopter major units (both blades and hubs of the main and tail rotor, rotor control systems, gearboxes, frames under gearboxes, etc.), for the transmission service life is assigned under the terms of the fatigue strength.

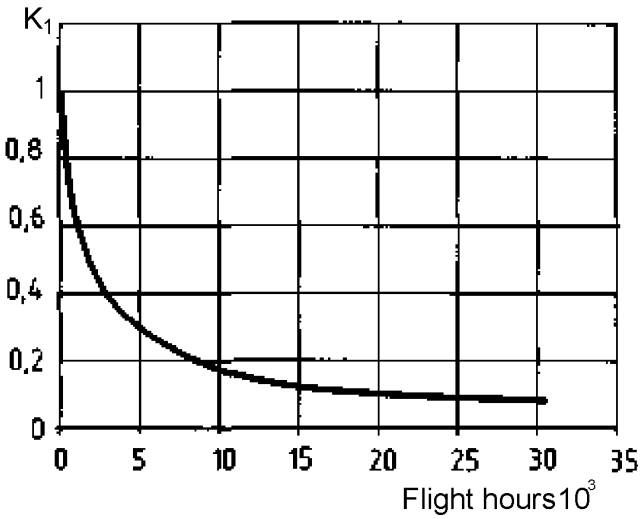


Fig. 18.5. Dependence of Catastrophes Coefficient K_1
from Aircraft Flight Hours within Assigned Life

Source under the terms of the fatigue strength is the allowed time of operating under conditions of practical absence of fatigue damage. As a rule, it is calculated in flight hours of helicopter. During operation the crew takes into account the total flight hours on a given helicopter exemplar. When the flight hours reach the specified service life, the unit is removed and replaced with a new one. When necessary, while calculating the flight hours, the work of the helicopter on the ground is also taken into account.

Source under the terms of the fatigue strength should not be confused with the general technical resource and calendar time, although these concepts are related.

General technical resource is established not only in terms of strength, but also from more general considerations, taking into account other factors that determine the structure suitable for further use (e.g., secondary structural member deterioration, runout, and so on).

Calendar time is set not in flight hours, but as general useful time of unit, regardless of the operation intensity during this time. Factors, such as adhesive joints aging, corrosion and so on are also taken into account. The calendar time affects on the resource under the terms of the fatigue strength.

In the future, a resource under the terms of fatigue strength will only be considered; further it is named just a resource.

To determine the safe structure resource under the influence of alternate loads it is necessary to know the load rate and the number of cycles, and the fatigue characteristics. At different stages of the creation and operation of the helicopter at the disposal of the designer there is a variety of data on its accuracy.

At the **planning phases** there are usually estimated data, the test results of models in air tunnels and materials, which were obtained during the tests of similar structures. On the magnitude of alternate loads these data in many cases have a low accuracy.

That is why the resource on data, available at the design stage, can not be determined accurately. The rough assessment of a resource with error correction is made and these corrections go to the safety margin. The magnitude of these corrections is determined on the basis of available experience. Such assessments allow us to estimate the prospects of this design and the possibility (for security reasons from

fatigue failures) to start flight tests of prototype helicopter.

Since the first flight of a **helicopter prototype**, stresses in most units are measured. These measurements make it possible to clarify the magnitude of loads. Concurrently, the tests of separate construction items are conducted and it allows us to refine fatigue characteristics. With measurements of flight strain of this helicopter and fatigue tests of real design units, one can have more reliable determination of a resource. The resource fixed in such a way is called a **preliminary resource**.

It is found that even minor mechanical tolerance can significantly change the fatigue characteristics. That is why, during batch manufacturing, the control tests on fatigue are performed. The resource, set due to test results of batch manufacturing units fatigue, is called the **initial**. It represents the largest resource that can be installed without taking into account the operating experience.

Thus, the same unit, as far as the accumulation of relevant data may have different resource, even without any structural modifications. At first it's small and set with a large margin, and then, due to the level of knowledge and increasing experimental materials, it gradually rises.

Typically, this process of resource refinement is accompanied by structural and technological modifications aimed at reducing stress and improving the fatigue characteristics.

The value of alternate loads significantly depends on the flight mode. If the helicopter in actual use flies a relatively longer time in mode with a small load level, the defect will be smaller than if most of the time it flew with a high level of load conditions. Therefore, when calculating the resource one need to know the relative time of the helicopter being at a particular mode in real operation.

The most proper distribution of the relative time of flight modes can be obtained by analyzing the experience of helicopters operating in real conditions. In various embodiments of helicopter operation the relative time of the considered mode may be different. Therefore it is necessary to consider several options of allocating time on the flight modes (for each variety of helicopter use).

On the basis of experience, you can create tables of the relative time allocation of helicopter operating at different modes. They are used, if there are no measurements in the operating conditions. Such tables for helicopters of various purposes listed in the norms of safety. Sometimes a mode can be characterized by particularly high variable loads. In this case, the residence time in this mode may be limited. Such restriction is made for pilots in instruction for helicopter piloting control.

Steady flight modes are characterized by a process of changing alternating stresses in the primary members of its design, this process is very close to a periodic. At the same time, the fatigue test is usually carried out under a load, which changes in a sinusoidal manner. For measuring the process actual form the approximate representations are used, they depend on the features of the alternating stresses change law over time in specific case.

In transient conditions (for example, during helicopter braking) the maximum value of the amplitude mode is determined. Further, with some margin it is considered that stresses with such amplitude value exist during the mode. A more accurate method is a damage assessment made by each load cycle. On the totality of all amplitudes at this mode the lumped voltage or total defect is determined.

Experimental determination of fatigue characteristics is rather labor-consuming. It is necessary to examine the full-scale objects, made on a particular technology. A combination of various components of forces and moments with a variable part and a

static component must be applied. There is a number of methodological difficulties.

To examine a large number of copies is virtually impossible. Typically, tests of limited design copies are made.

Calculation of the resource of helicopter designs is made according to the following formula:

$$N=, \frac{N_{min}}{3600 n_s \eta_\Sigma \eta'_\sigma \eta_N} \left(\frac{\Delta\sigma_t}{\Delta\sigma_e} \right)^m \qquad (18.1)$$

where N_{min} is the lowest durability according to test results; n_s is a number of load cycles per second during the flight; $\acute{\eta}_\Sigma$ is a safety coefficient for the hypothesis of damage linear summation; $\acute{\eta}_\sigma$ is a safety coefficient according to the level of alternating stresses; $\acute{\eta}_N$ is a safety coefficient for the scatter of durability; $\Delta\sigma_t$ is the amplitude of alternating stresses at the fatigue tests; $\Delta\sigma_e$ is an equivalent stress (determined by stress during the flight).

Multiplier $(\Delta\sigma_t / \Delta\sigma_e)^m$ takes into account the differences in levels of alternating stresses during fatigue tests and in flight. If the tests were performed before the measurements of flight stresses, than $\Delta\sigma_t$ was determined according to any other preliminary data. $\Delta\sigma_t$ is often chosen obviously more than $\Delta\sigma_e$ in order to reduce the time of fatigue tests. That is why usually variables $\Delta\sigma_t$ and $\Delta\sigma_e$ differ from each other.

To determine the durability under the influence of stresses $\Delta\sigma_e$ due to durability N_{min} calculated as a result of fatigue tests, which corresponds to the other alternating stresses ($\Delta\sigma_t$), we use the formula 18.3. On the basis of this formula we got the considered multiplier.

$$N_2/N_1=(\Delta\sigma_1/\Delta\sigma_2)^m.$$

The value of the exponent m can approximately be accepted:

when $N<10^8$, than m=6;

when $N>10^8$, than m=10.

The value of equivalent stress $\Delta\sigma_e$ is usually determined by the formula

$$\Delta\sigma_{\acute{Y}} = \sqrt[m]{\Sigma \overline{n_i}} \Delta\sigma_i^m \qquad (18.2)$$

The values of relative duration of the flight modes $\overline{n_i}$ are substituted in this formula, according to the standards of strength, depending on the helicopter purpose.

Since the formula (18.2) does not exclude some errors due to possible inaccuracies of linear addition hypothesis, the formula (18.1) provides a safety coefficient for the linear addition hypothesis $\acute{\eta}_\Sigma$. Based on strength standards, if the fatigue tests were carried out on the same level of alternate loads, than $\acute{\eta}_{\Sigma r}=2$. For multi-stage routine tests $\acute{\eta}_\Sigma=1$, since in this case the linear summation hypothesis is not used.

Some specific features of each construction unit, caused by abnormalities during its manufacture, lead to the fact that loads differ a little bit on the same flight conditions. During total operating of this helicopter its load history has personal character associated with the peculiarities of this particular helicopter application with the particular crew for carrying out particular works. That is why the formula (18.1) provides a safety factor η_σ. This factor also serves to accommodate inaccuracy measurements of flight stresses. In accordance with the norms of strength, the values of stresses, measured during the test flight in order to take into account the indicated factors, should

be increased by 1.2 times. In conversion to durability it equals $(1.2)^6 \approx 3$. That is why $\eta_\sigma = 3$.

Finally, we must take into account the random variation of durability both during the fatigue tests and from one helicopter copy to another. This measurement is based on the use of mathematical statistics methods. For this purpose there is coefficient $\acute{\eta}_N$ in this formula (18.1). The selection of coefficient $\acute{\eta}_N$ is carried by using the strength standards.

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Chapter 19. METHODOLOGY OF INTEGRATED DESIGNING AND MODELLING OF HELICOPTERS AND THEIR TRANSMISSIONS

19.1. CONCEPT AND PRINCIPLES OF INTEGRATED DESIGNING OF HELICOPTER ASSEMBLED STRUCTURES

Nowadays, no aircraft manufacturing company can compete if it isn't able to provide high quality of produced samples of the aviation vehicles, fast upgrade or model range change. At the same time, it's norm if the modern market has a large number of modifications of a basic model. It is very difficult to provide high work pace with quality assurance of the final product and its wide range without using computer integrated CAD/CAM/CAE systems that allow to integrate processes of design, engineering analysis and production preparation.

Process of the aviation vehicles development and their modifications is followed by development of design methods. The stages of the statistical, analytical, optimal, automated and system methods of design have already passed. Helicopter design methodology is based on the method of optimum design on the basis of integrated criteria of quality of the modern helicopters chosen from a condition of the Customer (Buyer) requirements' satisfaction to the helicopter and Aviation Rules.

Currently, the conventional quantitative criterion of civil helicopter quality evaluation is the cost of transportations and flight safety. The designers who project a helicopter and its airframe reach conceptually set quantitative aims of quality criteria by:

- reduction of frame weight as the main factor which reduces direct operational costs due to a possibility of payload increasing;
- increasing of a frame operating life and its service life when ensuring reliability and flight safety as the factors which decrease expenses on depreciation, maintenance and repair.

The main criterion in the basis of modern design methods of aviation construction is the requirement of development and functioning of reliable and safely damageable frame with a nominal service life. At the same time, the possibility of damage detection before reaching the admissible critical value should be ensured and sufficient residual frame durability should remain.

It is obvious that manufacturing of safely damageable design is provided by strength standards and aviation rules, but design with the set service life and minimum weight reflects economic problems.

The program of the aviation industry development of Ukraine involves manufacturing of new light-weight multi-purpose helicopters with a wide range of the functions which differ by:

- modern technical and operational level of development, exceeding the development level of the 20th century which is reached based on the new concepts, scientific and technical decisions, and inventions in the field of aerodynamics, design, construction, durability, weight perfection, power plants, helicopter systems, equipment, materials, production technologies and its preparation, operability, reliability and safety;
- compliance to modern standards of the air worthiness and to the aviation rules harmonized according to the structure and requirements with FAR (JAR), to the quality standards and perspective environmental standards;
- high degree of construction, technological and operational unification and succession of modern helicopters;
- economic efficiency caused by lower price of similar helicopters at comparable

operational indicators than of the other manufacturers;

- applying of strategy of technical operation according to the condition;
- introduction of the integrated technologies of design, production preparation, engineer analysis, tests, certifications, information support during life cycle of aviation complexes by means of the CAD/CAM/CAE/PLM and ERP systems (Fig. 19.1).

Development of the integrated systems of quality, durability, reliability and service life assurance, also certification of the aviation vehicles and their manufacturing, scientific and technical capacity creates a background for aviation vehicles improvement of the next generations with the use of integrated computerization. It is carried out during design, construction, technological preparation of production, mass production, flight tests on the basis of continuous information support during the life cycle of a product (CALS-technologies) and it is an important task to release aviation vehicles in the conditions of the modern market.

Information technologies together with progressive aviation technologies of design and production with common information space allow to increase significantly labor productivity, the quality of the manufactured aviation vehicles under considerable reduction of delivery time for production and release of the new, more modern helicopters corresponding to buyers' requirements.

To organize a common information space, it is important to arrange integration of design, production and operational databases into the uniform database.

The idea of the uniform information environment formation and its integration in all links of product maintenance during the life cycle also promotes performance of the main objective of Ukrainian aviation to ensure safety of transportations at minimum costs of transportation of ton-kilometer or one passenger-kilometer, decreasing of a helicopter life cycle cost.

According to the tasks of the product maintenance during life cycle, the uniform database has to contain data about the enterprises which manufacture aviation vehicles and service centers with the description of the on-going organizational, design and technological processes. Now methods and ideas of maintenance of the aviation vehicles during life cycle and integrated information technologies based on them find more use in all aviation firms of the world.

Development of information technology allows to intensify processes of creation of technical publications, design and technological preparation of production, production management and logistics of product, and the most important, to arrange information support of product life cycle, the scheme of which is shown in Fig. 19.1.

Data for information support are collected and put in order in the uniform database distributed by organizations with open access for all participants of life cycle maintenance.

During creation of a new product, its design and technological production preparation with the integrated CAD/CAM/CAE/PLM systems in the integrated information system, the structure of a product, its structure, and all components are described: parts, joints, assemblies, component parts, materials.

Use of the integrated information technologies in the process of the helicopter design allows to reduce innovation, production cost and maintenance during life of the helicopter. It increases productivity and ergonomics of work for 30 % that finally will increase quality and competitiveness of the production and engineering quality.

Information technology of the integrated helicopter design includes application of the parametrical analytical frame standard created in the CAD\CAM system in calculations of: aerodynamics and strength; service life and survivability; helicopter weight and its CG position; dynamics and safety of design and its functioning, also

during technological preparation of production, quality management, operation and repair.

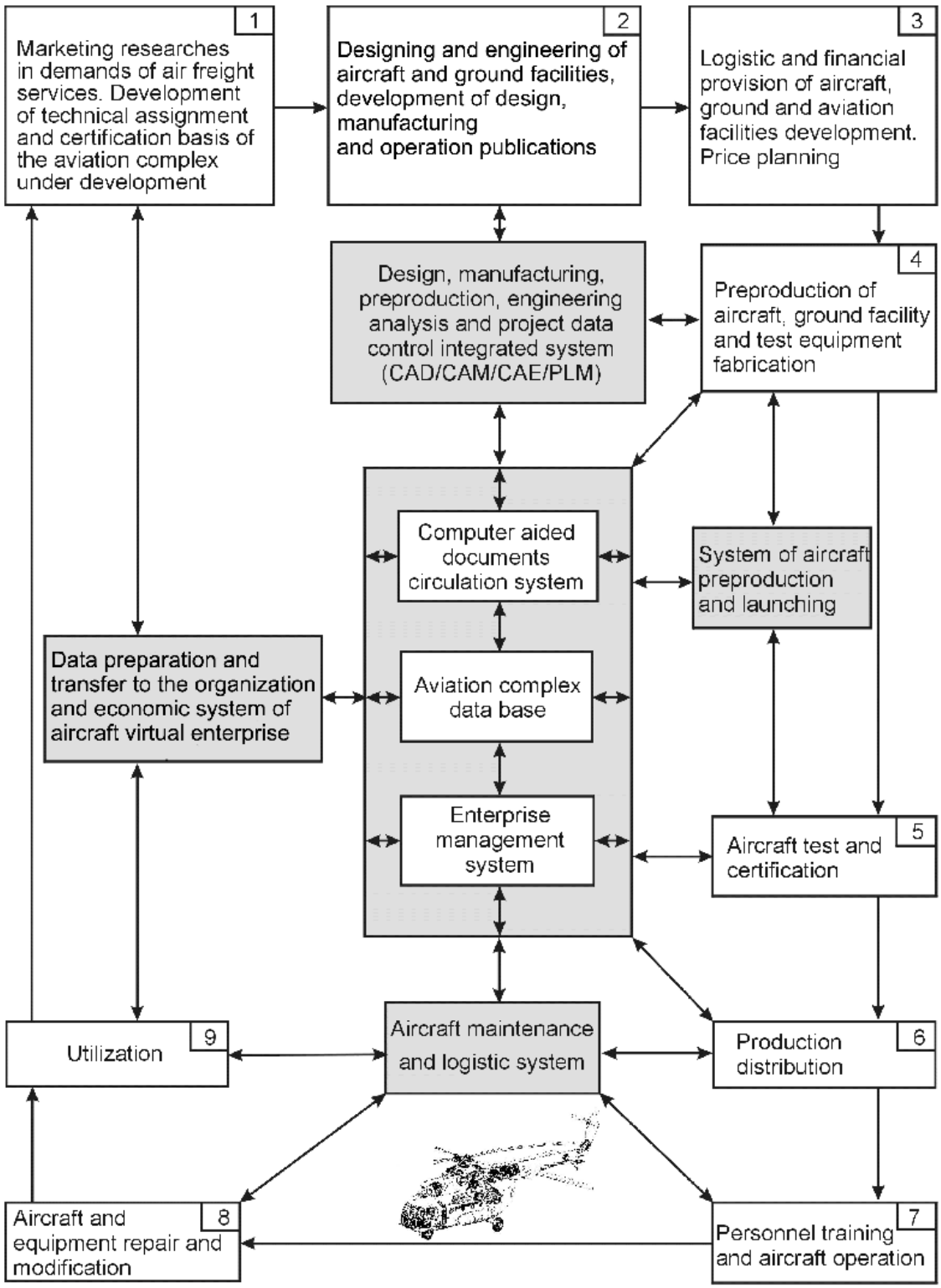


Fig. 19.1. Components of Helicopter Life Cycle and its Information Support Maintenance

The method of integrated design covers design and parametrical three-dimensional computer modeling of a helicopter frame in general and particularly with separate parts. The helicopter is a set of parts, assemblies, joints and units interconnected by various types of detachable and permanent joints, the quality of which ensures mass, service life, aerodynamic and esthetic characteristics properties of the helicopter.

The former methods of helicopter assembled structures were based on two-dimensional models and their mold loft alignment that didn't allow to consider all design and technological features. It has resulted in need to create the integrated design method.

The new concept of the integrated design of helicopter assembled structure is shown in Fig. 19.2.

The developed concept of the integrated design of helicopter assembled structures is a methodological basis for the creation of airframe primary member joints with set static strength, service life, pressure-sealing and quality of external surface (with the minimum assembly mass).

The integrated engineering of assembled helicopter structures can be divided into stages which interrelation is shown in Fig. 19.2:

1. Creation of the integrated information environment, technical and software complex for helicopter creation project, production and experimental base, group of experts (Figs 19.3-19.5).

2. Creation Development Concept of the new helicopter or modification which is already existing with application of the computer integrated systems of design CAD/CAM/CAE/PLM.

3. Development of helicopter master geometry in the CAD/CAM/CAE/PLM system.

4. Definition of the design loads, acting on helicopter units, and standard flight loads, the allowed tension in a regular zone for ensuring the regulated durability.

5. The integrated engineering of assembled helicopter structures' joints.

6. Creation of analytical standards of assembled helicopter structures (Fig. 19.6).

7. Development of design, technological and operational publications.

All works on the integrated engineering of assembled helicopter designs are performed in the uniform database of the designed helicopter with application of design and technological databases.

On the basis of the offered concept the principles of the integrated engineering of assembled helicopter structures have been developed:

1. Creation Principle of Analytical Standards of Helicopter Assembled Structures

Three-dimensional computer models of master geometry, space distribution, analytical elements' standards of helicopter assembled structures are created by methods of analytical geometry by means of the integrated CAD/CAM/CAE/PLM systems in the uniform information environment of helicopters life cycle support.

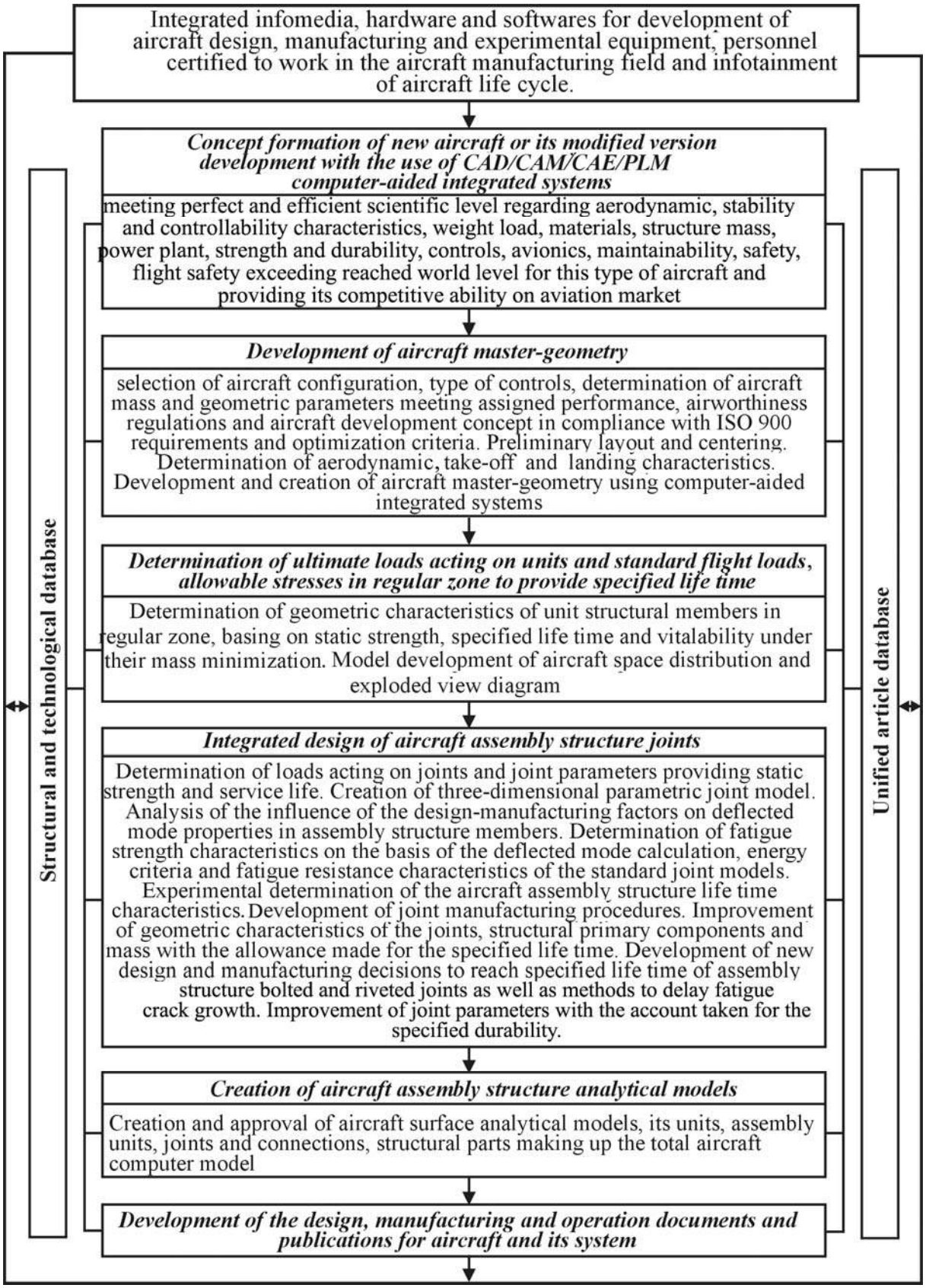


Fig. 19.2. New Concept of Integrated Designing of Assembled Helicopter Designs

2. Creation Principle of Shape Master Geometry of Helicopter

Shape parameters of the new helicopter with minimum weight and the regulated durability have to meet the set perspective tactical technical requirements, Aviation rules, the creation concept of the new helicopter and to be defined from ratios:

TTR, AR $\rightarrow$ *scheme of helicopter* $\rightarrow$ $\rightarrow$

$$m_0 = \frac{m_c + m_c + m_{pl}}{1 - \left[\bar{m}_к(p, n_p, N_{reg}, ОГП) + \bar{m}_{pp}(p, \bar{N}_0, \gamma_e, R, N_e) + \bar{m}_T(p, C_T, k, L)\right]} \rightarrow$$

$$\rightarrow \quad m_{0\min} \rightarrow p_{opt} \rightarrow \bar{N}_{opt} \rightarrow N_0 \rightarrow \sigma_i \rightarrow \text{blade} \quad \text{airfoil} \rightarrow (R, \lambda_1, \eta_1, D_f, L_e, L_{TU}) \rightarrow$$

$$\rightarrow (\mathrm{tg}\varphi_{cm}) \rightarrow$$ *analytical standard of helicopter surface.*

3. *Design Principle of Helicopter Assembled Structures' Regular Zones*

Design data and technology of regular zones performance (r.z.) of helicopter designs have to provide taking up of design ultimate loads, the regulated durability at the loads equivalent to loads of standard flight in the operational environment, the set fatigue quality coefficient (K_y), the set quality of external surface, degree of tightness and to satisfy the following inequalities: $P_d \geq P_e(DD_{r.z}, \sigma_{dr.z.}(N_{r.r.z}))$;

$N_s \leq N_{d.r.z}(DD_{r.z}, \sigma_{0e}, \sigma_s, T)$;

$\Delta_s < 0$ *at* $P = P_o$; $\Delta h \leq 0.05\text{mm}$; $K_f \leq 3$.

4. *Design Principle Of Helicopter Assembled Structures' Irregular Zones*

Design data and technology of irregular zones performance (i.r.z) have to provide taking up of design loads in irregular zone at static loading, the regulated durability, the quality of external surface and tightness equal to characteristics of a regular zone or exceeding them and to satisfy the following inequalities:

$P_b \geq P_e(DD_{i.r.z}, \sigma_{di.r.z}(N_{s.i.r.z}))$; $\Delta h_{i.r.z} \leq \Delta h_{p.f}$; $\Delta_{3i.r.z} < \Delta_{3r.f}$;

$N_e \leq \min(N_{d.i.r.z}(DD_{i.r.z}, (\sigma_{0e} \cdot \varepsilon_e), \sigma_s, T)$; $N_o(DD_{i.r.z}, \sigma_0, \sigma_s, T))$.

5. *Survivability Maintenance and Achievement Principle of Helicopter Assembled Structures with Fatigue Cracks*

Design data of safely destroyed helicopter assembled structures have to provide a possibility of critical places control, detection of fatigue cracks and application of progressive ways of their growth delay, restoration of the bearing ability and tightness of the damaged design and to satisfy the following inequalities: $(N_{r.WFCGD} / N_{r.st}) > 1$;

$\Delta_{F,WFCGD} < 0$.

Evidence-based methods development of the integrated design is necessary for realization of these principles.

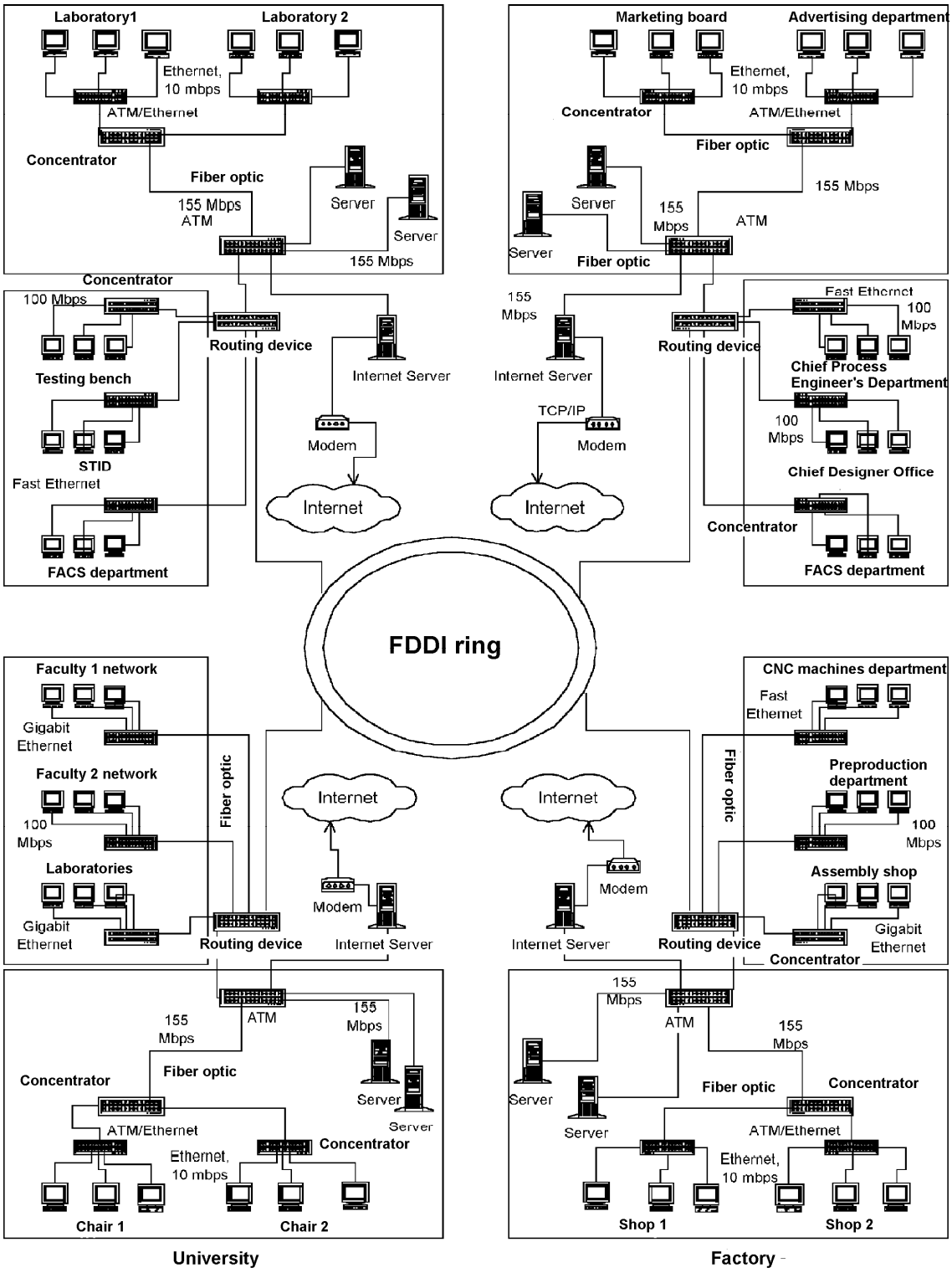


Fig. 19.3. Design Solution of Information Network Structure of Aviation Complex Virtual Enterprise

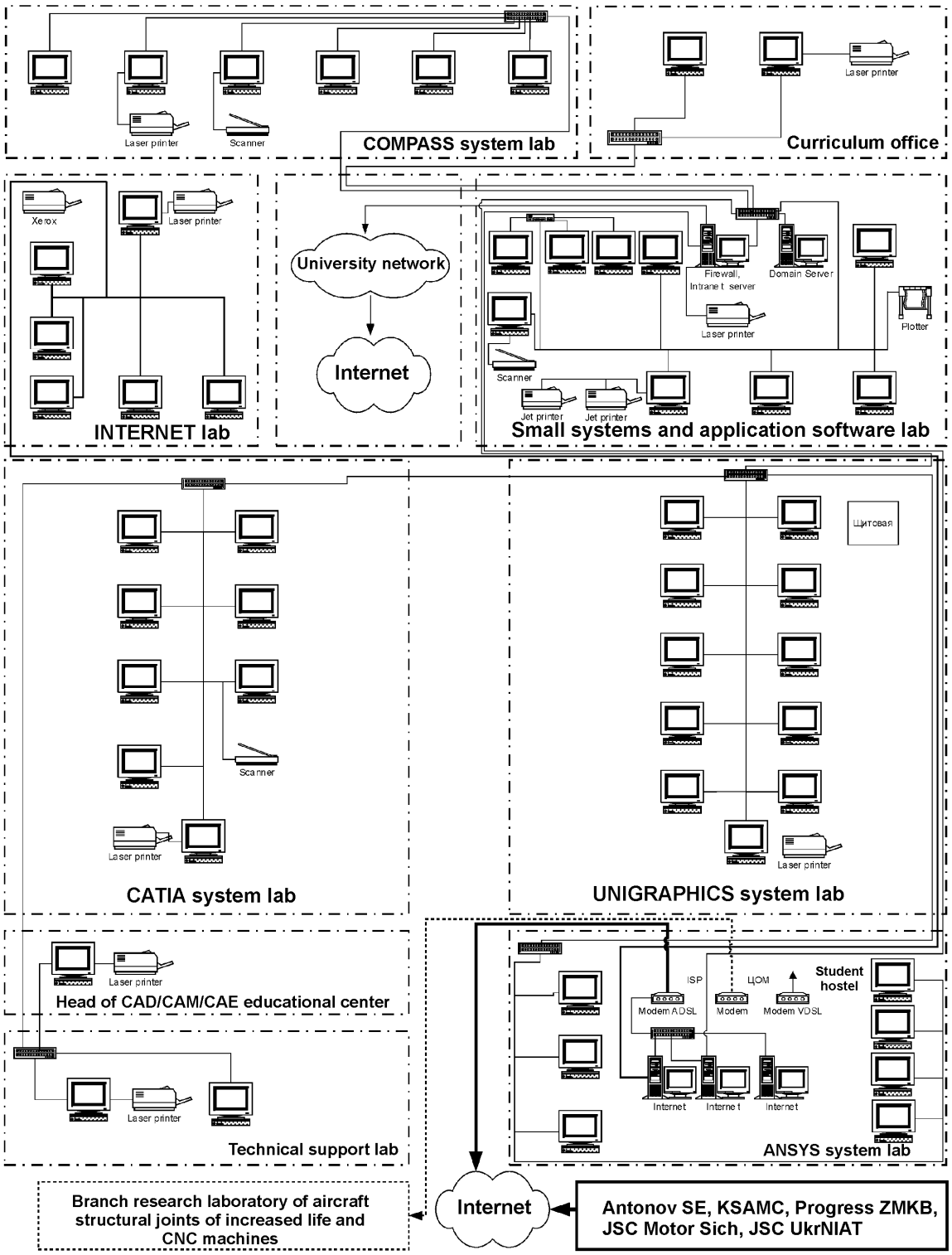


Fig. 19.4. Topology of Local Network of CAD/CAM/CAE Educational Center of KhAI

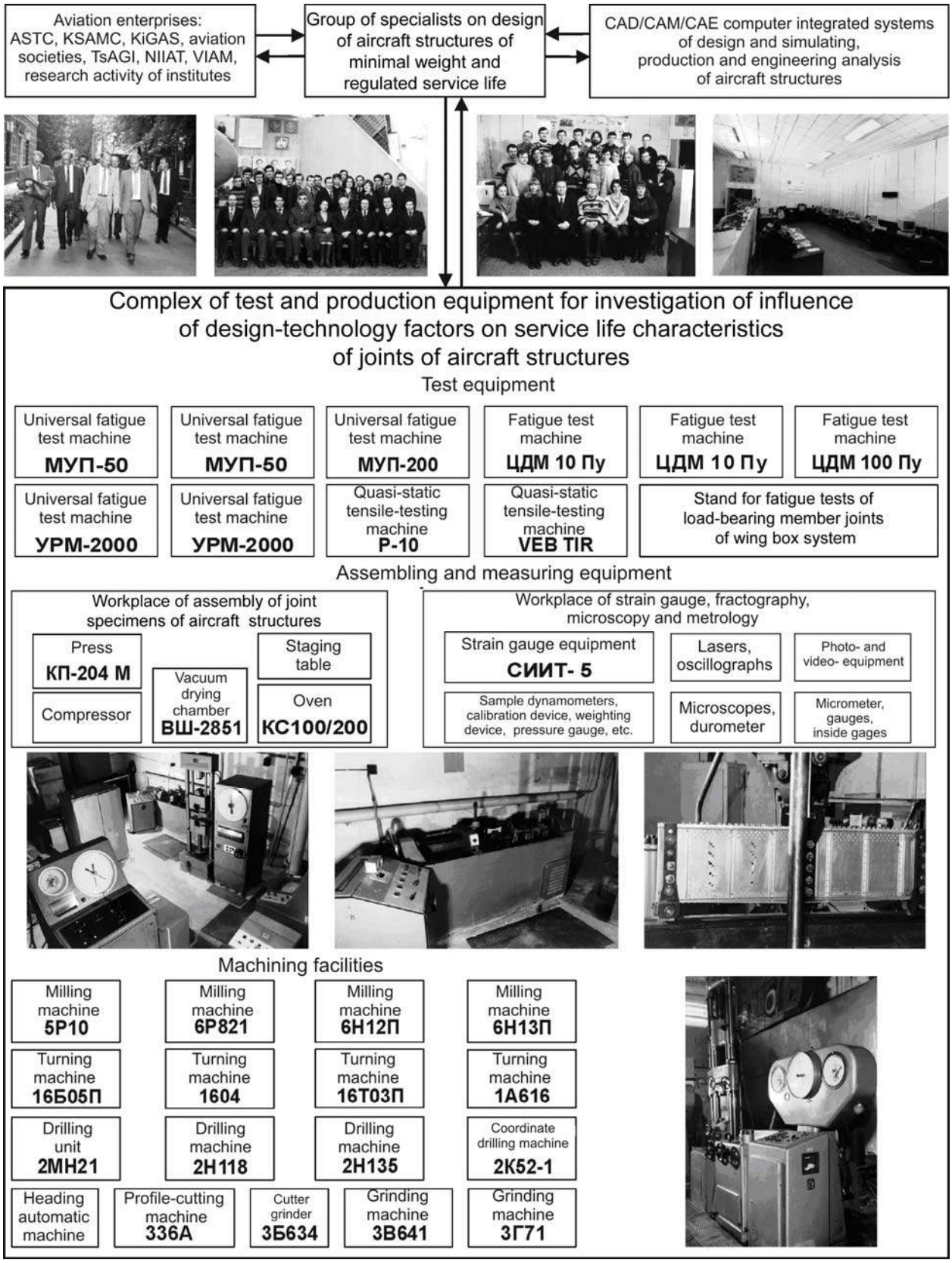


Fig. 19.5. Structure and Composition of Means for Integrated Designing of Aviation Structures of Assigned Service Life

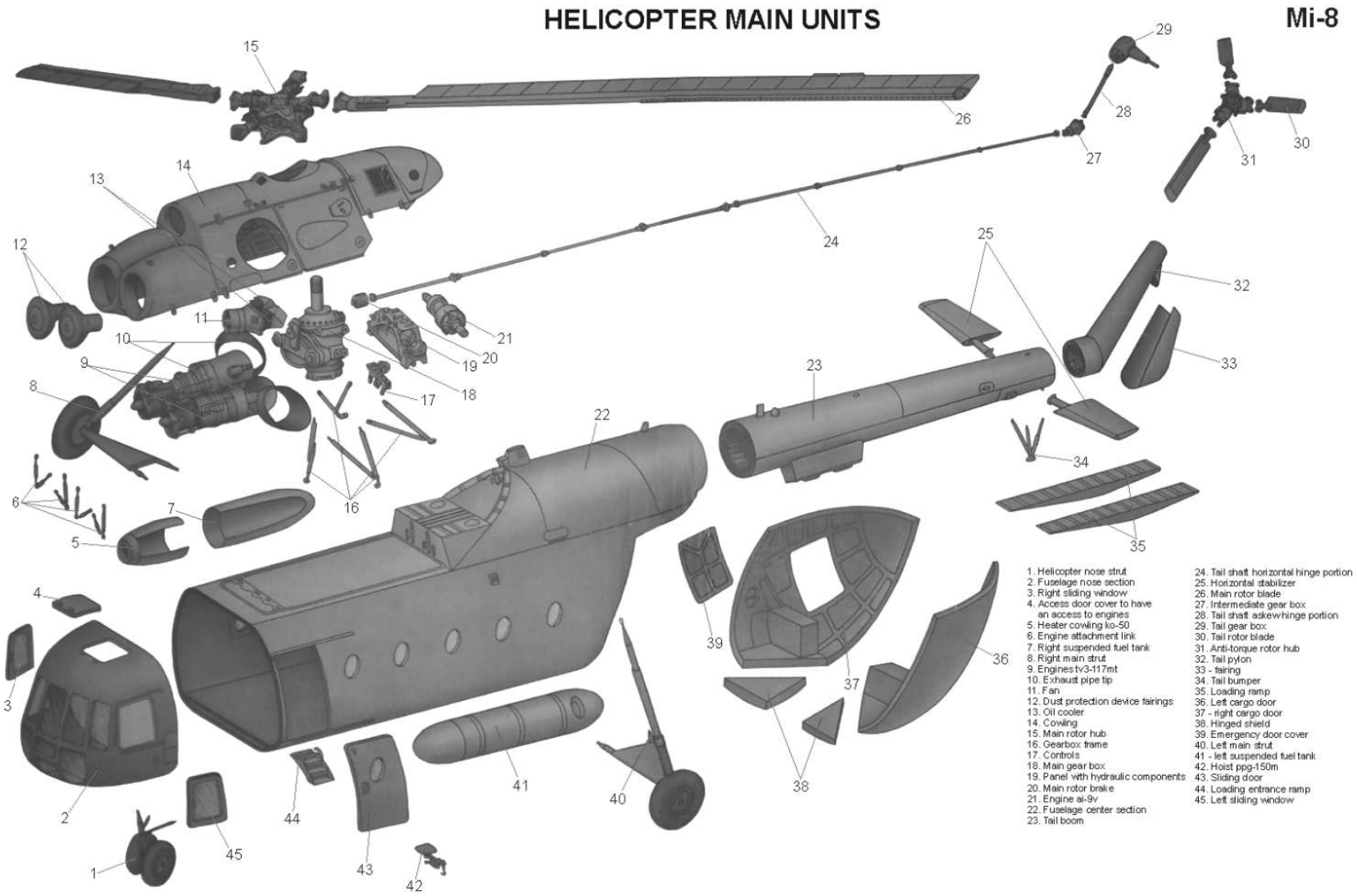


Fig. 19.6. Structural Joints

19.1.1. Methods of Master Geometry, Space Distribution Models and Standards of Helicopter Assembled Structures Creation

Limited design opportunities in descriptive geometry language, the big labor input of works caused by need of mould loft creation and parts lofting, assemblies, units, and a helicopter as a whole can be overcome by use of analytical geometry language. Development of the computers and appearance of the computer assistants to the engineer working at this language such as CADDS-5, the CATIA, EUCLID, UNIGRAPHICS, have given such opportunity. In Fig. 19.7 as an example structure of the computer integrated CAD\CAM\CAE\PLM UNIGRAPHICS system is shown by means of which it is possible to carry out the integrated design of the helicopter.

However, use of such systems significantly changes both a traditional form of design-engineering documentation, and process of parts and assemblies design, doing it more evident and certain. However, use of design-engineering documentation in a computer form generates also the new difficulties connected with easiness of computer models' modification and low reliability of long information storage on computer carriers.

The part geometry model in a computer form (further – an analytical standard, or anet of a part) is basic, primary element of design at computer designing of a new aircraft. It comprises reference coordinates of all points of a part surface in the set system of coordinates and represents a computer project basis of helicopter geometry.

Today we understand system of design, prediction and engineering models, and data for certification, quality management, maintenance in operation, utilization, i.e. management of helicopter life cycle as the computer project.

The computer project of the helicopter includes the following models:

– model No. 1. Helicopter master geometry (or the model of the helicopter surface defining all points lying on the surface of the helicopter);

– model No. 2. Model of helicopter space distribution;

– model No. 3. Models of joints and connections on production breaks;

– model No. 4. Model of all product geometry (analytical standards of all parts, assemblies, units and helicopter in general), i.e. model of complete computer definition of the helicopter.

Let's consider creation process of each listed helicopter model.

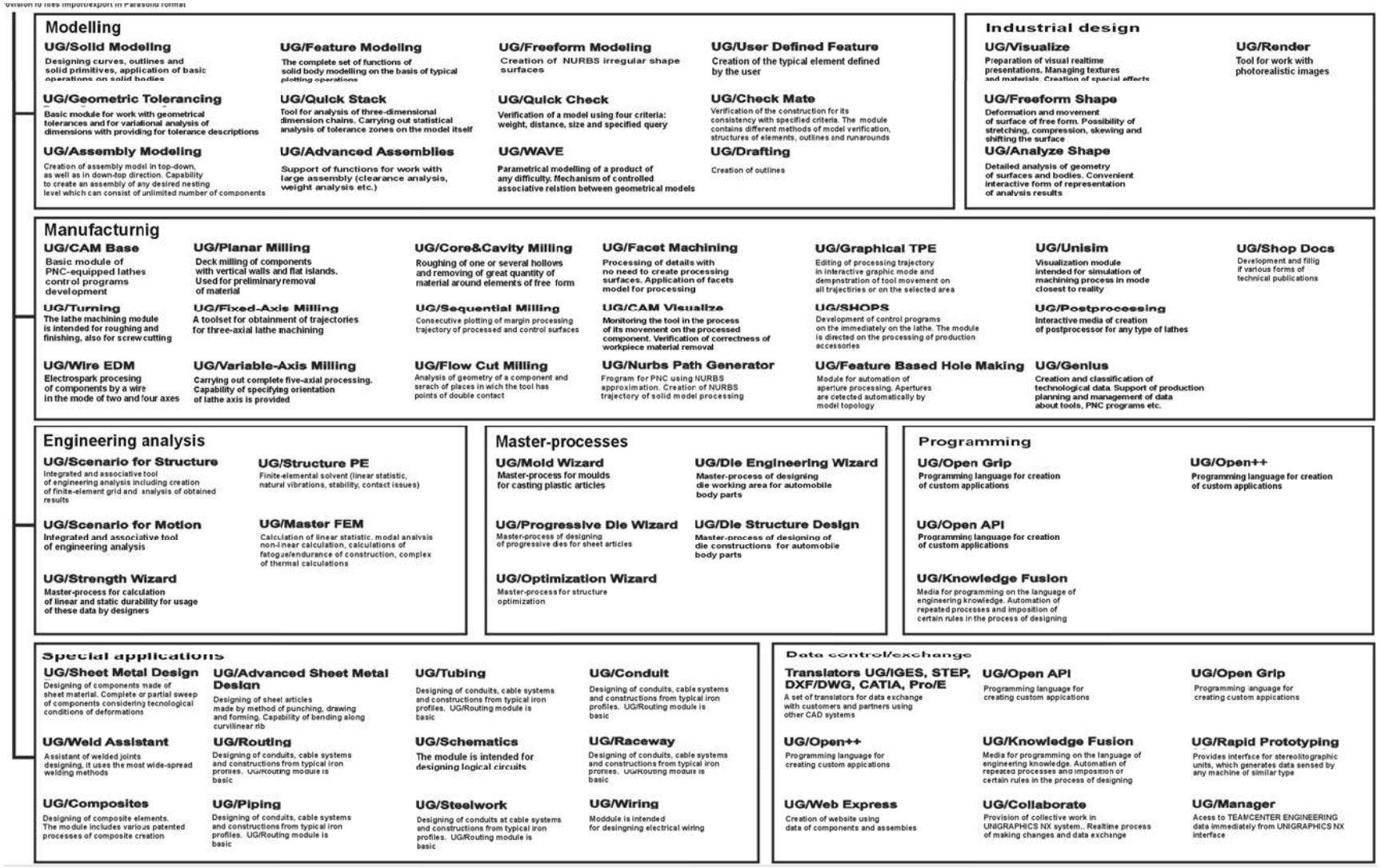


Fig. 19.7. Structure and Composition of Computer Integrated System CAD\CAM\CAE\PLM UNIGRAPHICS Modules

Model No. 1. Helicopter master geometry.

Creation process of model No. 1 can be divided into the following stages:

1) development of helicopter mathematical model;

2) development of theoretical drawings of units;

3) creation of unit surface models, their formation in model of a helicopter surface;

4) framework creation (drawing traces of basic surfaces of the structural framework (SF) in volume of the theoretical drawing (TD) and the load-carrying structure (LCS)).

Model No. 2. Model of helicopter space distribution.

Creation process of model No. 2 can be divided into the following stages:

1) development of constructive and technological partitioning;
2) panelling;
3) creation of structural framework;
4) creation of design elements;
5) placement of the equipment, devices, purchased products, etc.;
6) configuration of systems;
7) configuration of a crew cabin;
8) configuration of a passenger cabin (cargo cabin);
9) joining of units and systems;
10) creation of helicopter project tree.

Model No. 3. Models of connections and joints on production breaks.

Development of model No. 3 includes the following stages:

1. Complete definition of design connections and joints.
2. Purpose of interconnected system of allowances for joint elements.

Model No. 4. Model of all product geometry.

Development of model No. 4 includes the following stages:

1. Zone modeling:

– dimensional and exact components of a design with all connections and interfaces;

– systems with structural elements of fastening;

– outlines of units and instruments with an exact positioning of fastening elements, and check of design elements for definition of mutual penetration and allowances, assemblability.

2. Sectional modeling:

– analytical standards of all design elements;

– databases of drawings;

– fillings by attributive information.

3. Aggregate modeling:

– of the systems passing through units without technological partitioning;

– collecting and control of all information on the project.

As basic data for development of parametrical computer models of the helicopter the specification obtained as a result of the general automated design of the helicopter: scheme of the helicopter, weight and geometrical data, flight technical characteristics is accepted. These data for the Mi-8 helicopter is given below.

19.2. GENERAL CHARACTERISTIC AND SPECIFICATION OF HELICOPTER

19.2.1. Modifications of Mi-8 Family Helicopters

Mi-8T (HIP-C) – the main transport modification (Figs 19.8, 19.9).

Mi-8TV – the upgraded version.

Mi-8TVK – the export Mi-8TB version.

Mi-9 – the flying command helicopter.

Mi-14PL – the main anti-submarine modification.

Mi-14PLM – the modified Mi-14PL version.
Mi-14PS – the search and rescue option.
Mi-14PW – the Mi-14PL in the Polish operation.
Mi-8MT – the transport and combat helicopter on Mi-8TB base.
Mi-8MTV – the transport and combat helicopter, next development of Mi-8MT.
Mi-17 – the new version on a basis of Mi-8MT.
Mi-17-1BA – option of the flying hospital, is equipped with TV3-117VM engines.

19.2.2. Main Options of Helicopter Use:

– transport;
– landing;
– sanitary;
– fighting.

The fighting option for the helicopter isn't priority, and in some cases for ensuring forcible entry operations as a part of fighting groups the fire support and covering action helicopters – Mi-24 are supposed to be used.

Besides, the helicopter can carry out a number of special tasks, such as rescue of the crews hit and aircraft in distress, installation of various designs, statement of mines, towage of mine trawls and barges, conducting of air reconnaissance, ensuring communication, correction of gun-fire, rendering of radio-electronic counteraction.

19.2.3. Specification

19.2.3.1. Weight:

Normal take-off weight, kg ……..11100
Maximum take-off weight, kg …… 13000
The maximum weight of the transported freight
at full fueling of the fuel tanks (without AFT), kg…… 4000
Suspended load gross weight, kg.....3000
Normal operating load, kg 2000
Q-ty of the transported soldiers 24
Q-ty of the transported stretcher cases 12

19.2.3.2. Geometry:

Helicopter length, m:
– without MR and TR.....18,4
– with MR and TR.....25,4
Helicopter height, m:
– without TR 4,76
– with TR 5,55
Wheel track, ...4510
Wheel base, mm.....4280
Helicopter clearance (along Fr. 14), m 0.445
Helicopter static ground angle .....4°10'
Distance from the blade tip to the tail boom on the ground, m.....0.45
Main rotor diameter, m 213
Antitorque propeller diameter, m 3.9
Original dimensions, m……5.34x2.34x1.8

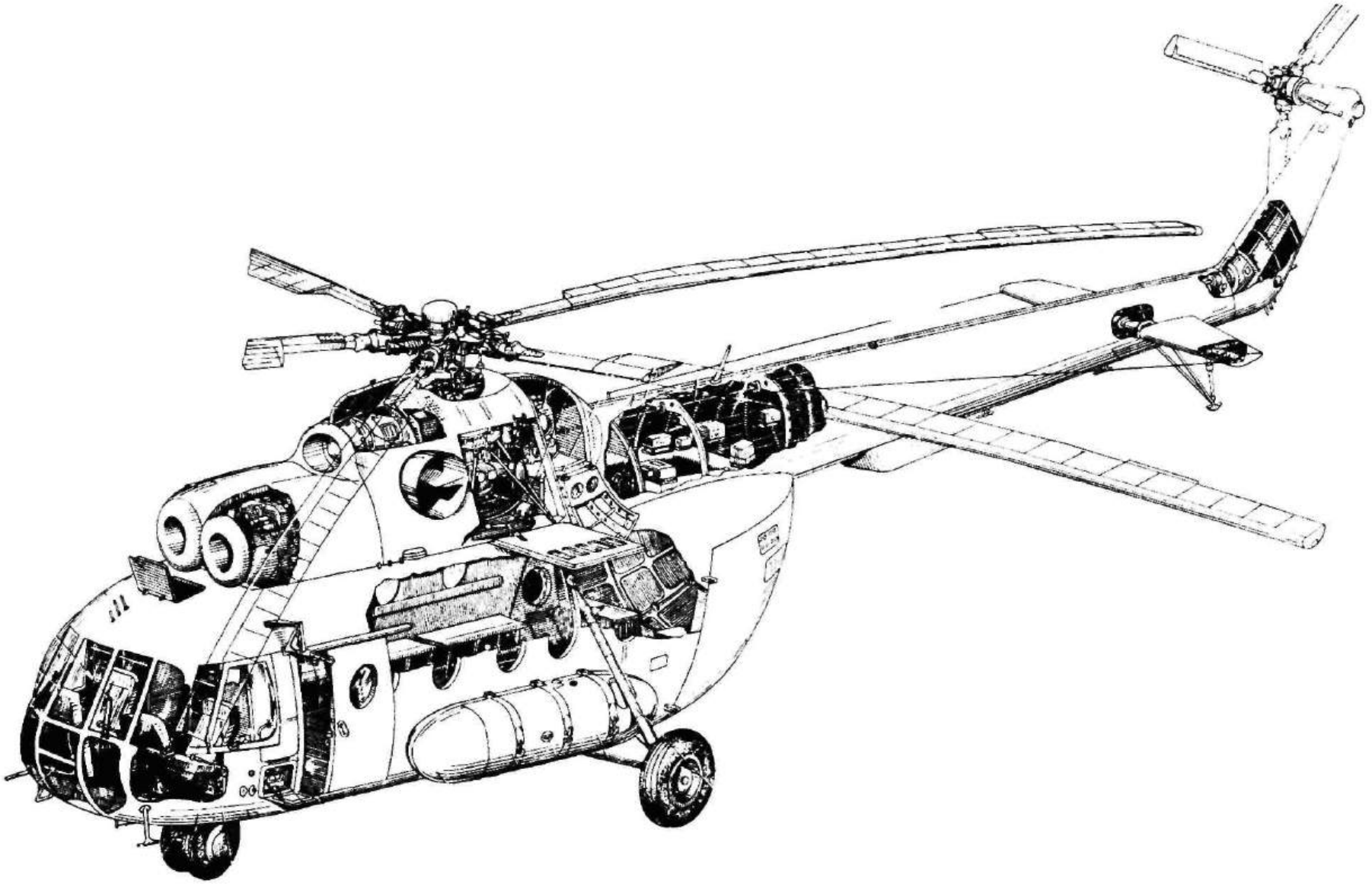


Fig. 19.8. MI-8T Helicopter Overall View

19.2.3.3. Performance:

Maximum speed in level flight (H = 0...1000m), km/h:

– with normal takeoff weight.....250

– maximum takeoff weight 230

Cruising speed (H < 1000 m), km/h

– normal takeoff weight.....220...240

– maximum takeoff weight .....205...215

Static ceiling (G_{norm}, AFCS, out of ground effect), m 1760

Operating ceiling, m:

– normal takeoff weight.....5000

– maximum takeoff weight 4100

Climb time («NOMINAL», $V_{opt.}$ = 120 km/h, A/ICE OFF), min:

	with G_{norm}	with G_{max}
– 1000 m	1.8+0.5	2.4+0.5
– 3000 m	6+1	10.9+1
– 4000 m	9.5+2	

Operating range (H = 500m, $V_{cruis.}$, without AFT, c m_f = 5%), km

– at G_{tr} = 2117 kg.....495

– atG_{tr} = 4000 kg.....465

– at 1 AFT 725

– at 2 AFT 950

Note:

1. When A/ICE is on H_{oper} decreases by 200...300 m.

2. EHS decreases H_{oper} by 150...200 m.

Center-of-gravity limits:

– fore, mm.....+300

– aft, mm.....-95

Attention. When the helicopter crew is provided with an armored protection the forward center-of-gravity limit is +330 mm, with full feeder tank, otherwise the shooter of the aft gun set must not leave his position.

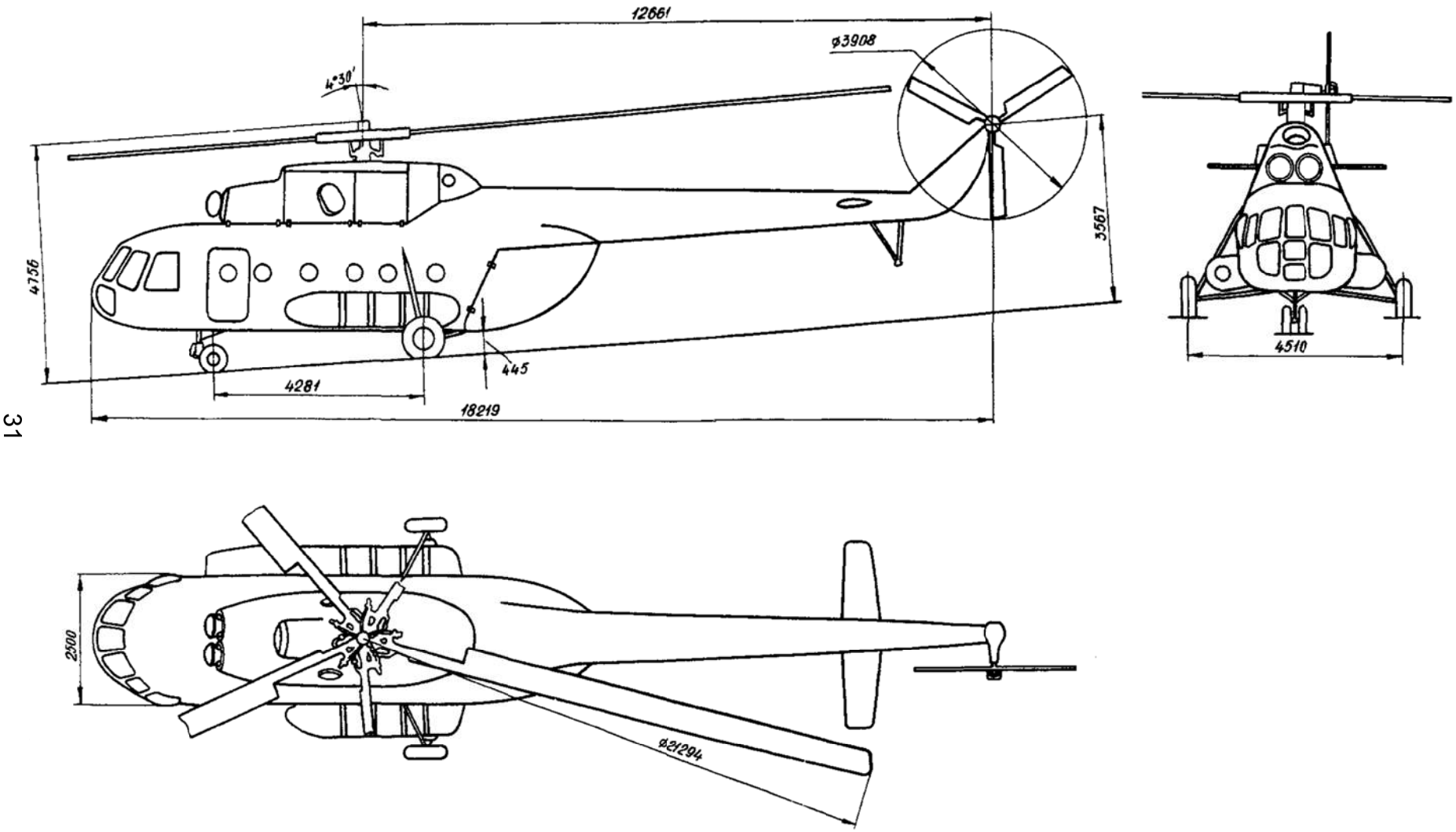


Fig. 19.9. Helicopter Main Dimension

19.3. Helicopter Fuselage. Emergency Escape Means

19.3.1.Fuselage. General Description

Fuselage is main helicopter load-bearing framework that is a full-metal semimonocoque of the riveted structure and variable section with smooth stressed skin.

Fuselage has three technological breaks and includes: (Figs 19.10, 19,11)

– nose section (frames 1N...5N) – 2;

– center section (frames 1...23) – 21;

– tail boom (frames 1...L7) – 10;

– tail fin (TF) (frames 1...9) – 14 and 13.

Fuselage main parts are connected at frames 1 and 23 of center section; 1 and frame 17 at tail boom.

Thus, the fuselage has a production break and two maintenance breaks.

The fuselage strong members include frames, stringers and longitudinal beams. The stressed skin, which thickness is 0.8...1.2 mm forms a stiff structure.

The main structural materials are: D-16AT; V-95; AK-6; AK-8; ML-5T; 30KhGS(N)A.

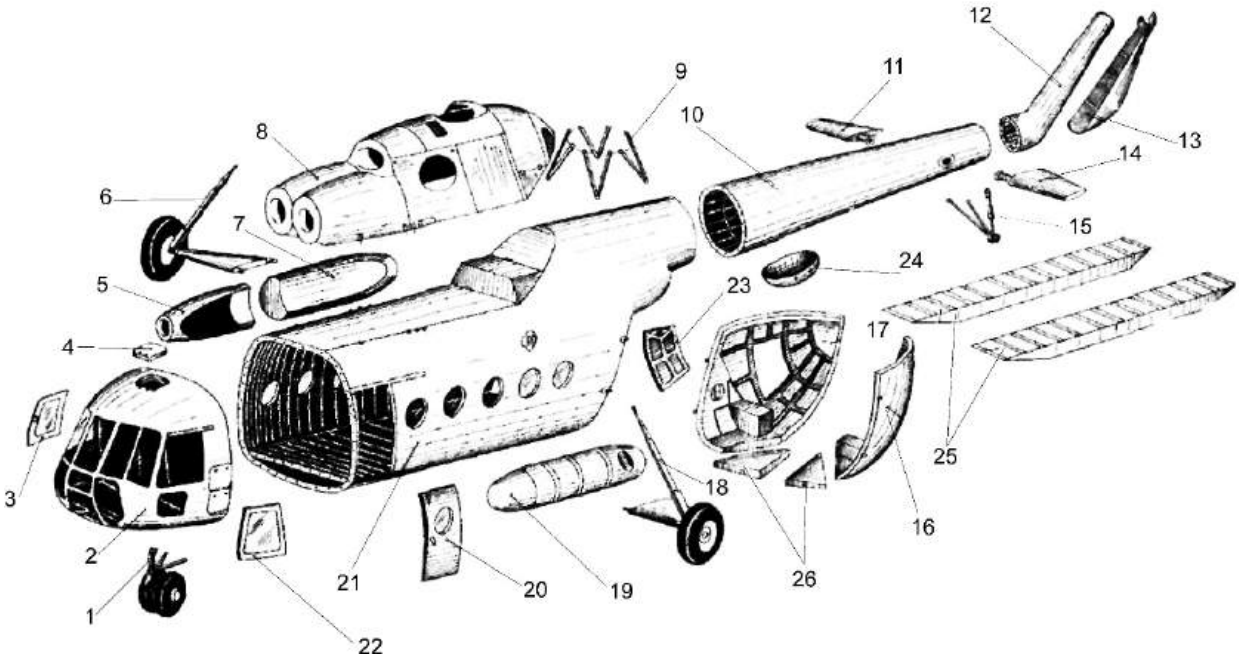


Fig. 19.10. Airframe Structural Joints:

1 – nose LG; 2 – fuselage nose section; 3, 22 – movable blisters; 4 – engine unit door; 5 – KO-50 heater fairing; 6, 18 – main LG; 7, 9 – external fuel tank; 8 – fairing; 9 – main reducer framework; 10 – tail boom; 11, 14 – left and right tail unit halves; 12 – tail boom; 13 –fairing; 15 – tail bumper; 16, 17 – cargo doors; 20 – sliding door; 21 – fuselage center section; 23 – emergency escape door; 24 – radome of DIV-1 item; 25 – loading ramps; 26 – panels; 27 – aft door

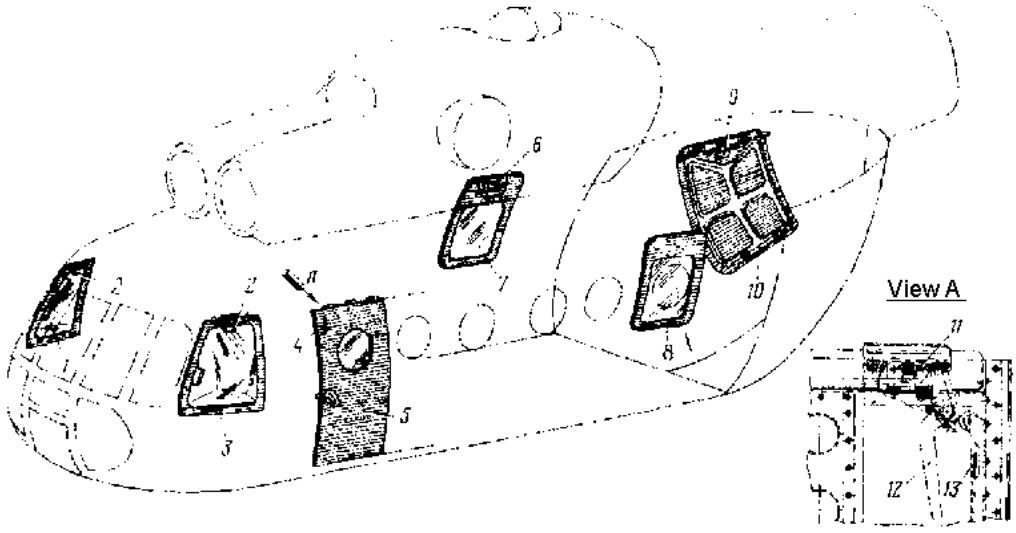


Fig. 19.11. Door, Blister and Emergency Exit Arrangement

1 – right blister; 2, 6, 9 – escape assist handle; 3 – left blister;
4 – outer escape jettison handle; 5 – left sliding door;
7, 8 – emergency hatches (passenger helicopter);
10 – emergency hatch in right cargo door (transport helicopter); 11 – locking pin;
12 – inner jettison handle; 13 – door middle lock rope.

19.3.2. Fuselage Primary Units Construction

19.3.2.1. Fuselage Nose Section

Fuselage nose section (Fig. 19.12) is an independent section, that accommodates the cockpit, helicopter and engines controls, equipment, etc. (L = 2150 mm).

Cockpit is situated between frames 1N...5N and separated with bulkhead 5N with a door.

Cockpit accommodates left and right pilot seats, technician seat, controls, instrument panels, two battery cases, and radio equipment racks. Cockpit may be equipped with a shooting system.

The right and left sides are fitted with sliding blisters 5.

The left side has plug connectors SHRAP 400-3F and SHRAP-500K for AC/DC sources between frames 3N...5N.

The right side between frames 4N 5N is equipped with rectifier units VU-6A, cooled with airflow from the maintenance panels shutters.

The cockpit overhead panel 3 accommodates hatch 2 to give access to the engines.

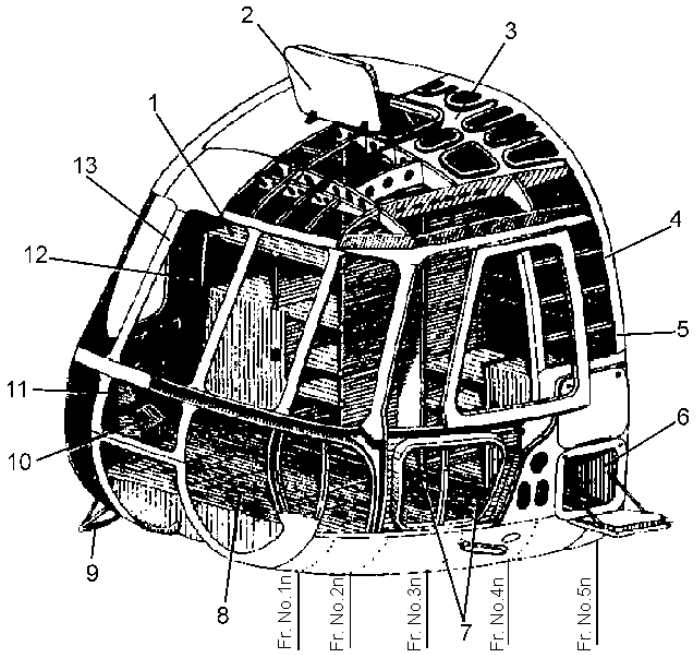


Fig. 19.12. Fuselage Nose Section:
1 – radio equipment rack; 2 – engine unit hatch; 3 – stamped out ceiling stiffeners;
4 – frame No. 5N web; 5 – sliding blister; 6 – accumulator case ;
7 – cockpit seat mounting brackets; 8 – cabin floor; 9 – outer footrest ;
10 – inner footrest case; 11 – grounding link dome; 12 – console; 13 – canopy

19.*3.2.2. Fuselage Center Section*

Fuselage center section (Fig. 19.13) is located between frames 1...23 and it length is of 8740 mm. It consists of duralumin stressed skin structure. Cargo cabin is situated in the fuselage center section with aft cargo door having leaves 16, and sliding door 2 on the left side.

Original dimensions of the cabin are – 5.34 × 2.34 × 1.8 m.

Loading ramps providing a track of 1300...1700 mm can be mounted in the aft cargo door opening.

Fuselage cargo door dimension are: height – 1.62 m, width at the center line – 2.29 m.

The engines, main reducer and airborne auxiliary power unit (APU AI-9V) are situated above the cabin. Feeding tank and tail drive shaft are placed in superstructure 7 behind the center section.

Fuselage center section accommodates the following units:

– between frames 1...13 – cargo cabin;

– between frames 13...21 – aft compartment with cargo doors;

– between frames 2...7 – engines;

– between frames 7...10 – main reducer;

– behind Fr. 10 – superstructure 7;
– between frames 10... 13 – feeding fuel tank;
– between frames 16...21 – radio equipment bay.

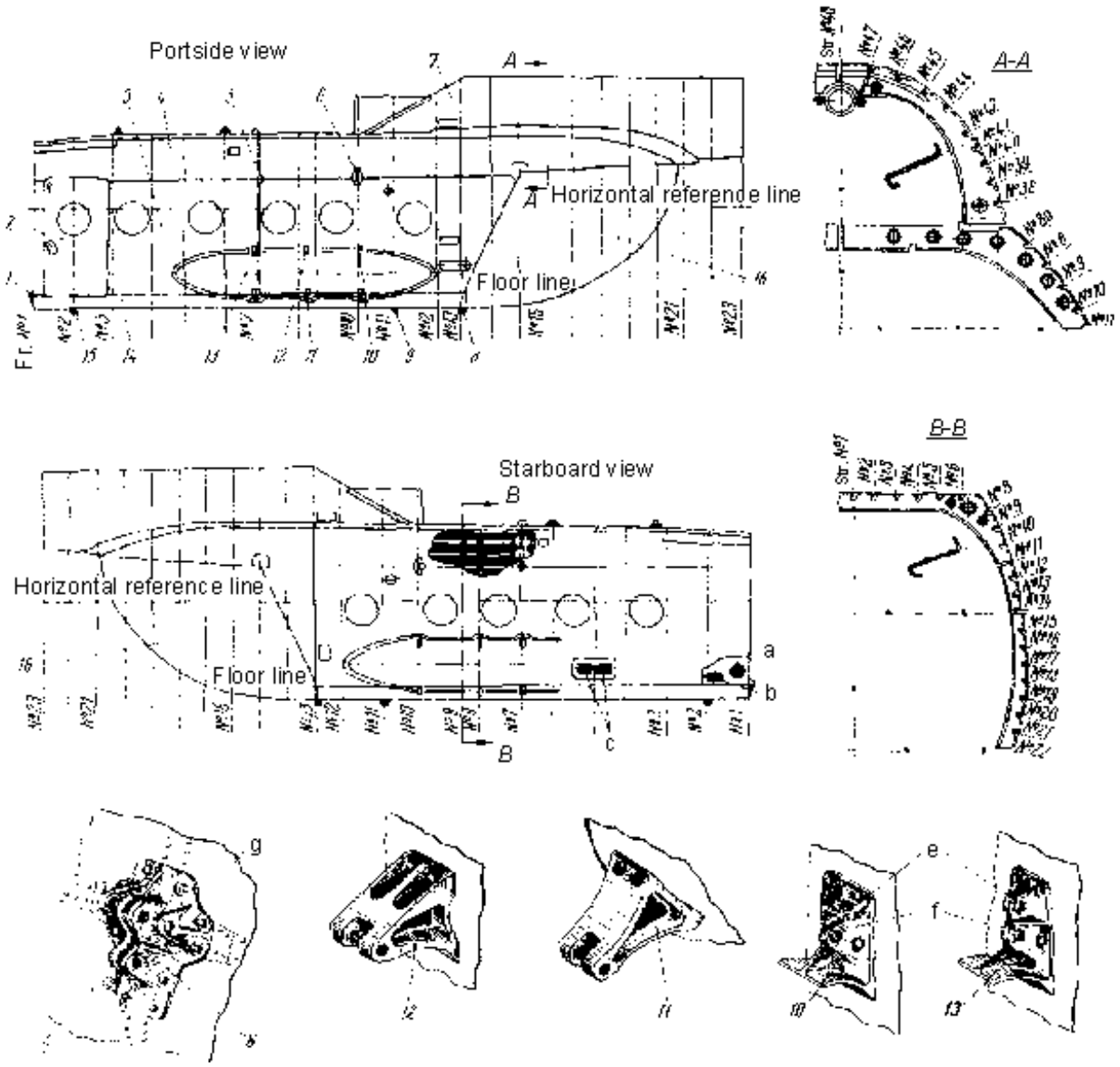


Fig. 19.13. Fuselage Center section:

1 – nose LG shock strut mount ; 2 – sliding door; 3 – aircraft panel; 4 – overhead panel; 5 – nose upper bracing attachment fitting of special unit truss; 6 – main LG shock strut attachment fitting; 7 – superstructure; 8 – main LG leg shock strut attachment fitting; 9 – main LG radius rod attachment fitting; 10 – upper aft attachment fitting of feeding fuel tank; 11 – lower mid feeding fuel tank attachment fitting ; 12 – upper mid attachment fitting of feeding fuel tank; 13 – upper fore attachment fitting of feeding fuel tank; 14 – cargo floor; 15 – nose LG leg radius rod attachment fitting; 16 – cargo doors; a – cargo opening for cargo cabin air intake connection; b – warm air duct inlet; c – kerosene-combustion heater connection inlet; d – lugs for bolts attaching lower braces of special unit truss; e – lugs for bolts attaching fixing straps of external fuel tank; f – lugs for bolts attaching upper brace of special unit truss and mooring device.

19.3.2.3. Tail Boom

Tail boom (TB) (Fig. 19.14) is a unit of riveted beam-stringer structure, made in a form of truncated cone 5440 mm long. TB consists of airframe and smooth stressed skin. The frame includes 17 frames and 26 stringers.

The following units are placed in the tail boom:

– DISS-15 fairing between frames 2...6;

– cutout for flashing beacon between frames 3...4;

– cutouts for formation lights between frames 7...8 and 15... 16;

– cutout (view B-B) for the stabilizer spar between frames 13... 14.

Tail bumper (TB) attachment fittings 1 and 2 are placed at frames 15 and 17. Outside the tail bumper there are access door 6 and 7 to examine the tail shaft. The tail drive shaft gear are located in fittings 5 inside the tail shaft. Chocks 4 with antitorque rotor control cable rollers are situated on the bracket 3.

19.3.2.4 Tail pylon

Tail pylon (Fig. 19.15) places the antitorque rotor spinning axis to the main rotor plane of rotation.

The tail pylon is of riveted structure, consists of beam 8 and fairing.

The tail pylon axis is deflected up to the 43°30' degrees relatively to the tail boom axis.

The basic structure consists of 9 frames, spar and stringers.

The intermediate reducer is attached to frame 3, and tail reducer is attached to frame 9.

On star board between frames 2...3 in the mid part there is an access door, to check oil level on the sight gauge. Another access door is used to provide flow about intermediate reducer with oncoming airflow. Access door for examination of the tail drive shaft is situated on the top of tail pylon between frames 3...4 and 8...9.

Fairing forms the tail pylon aft profile. The fairing consists of lower removable part 6 and upper fixed part 9 and is fixed aerodynamic control surface.

19.3.2.5.Stabilizer

Stabilizer (Fig. 19.16) provides the helicopter pitching stability both in normal operating mode and in autorotation.

The stabilizer area is of 2 m^2.

It is set at fixed angle of -6°degrees relatively to the tail boom (for helicopter Mi-8MTV the angle is of 3°). Stabilizer setting angle may be changed on the ground within +9°...-9° depending to the helicopter purpose.

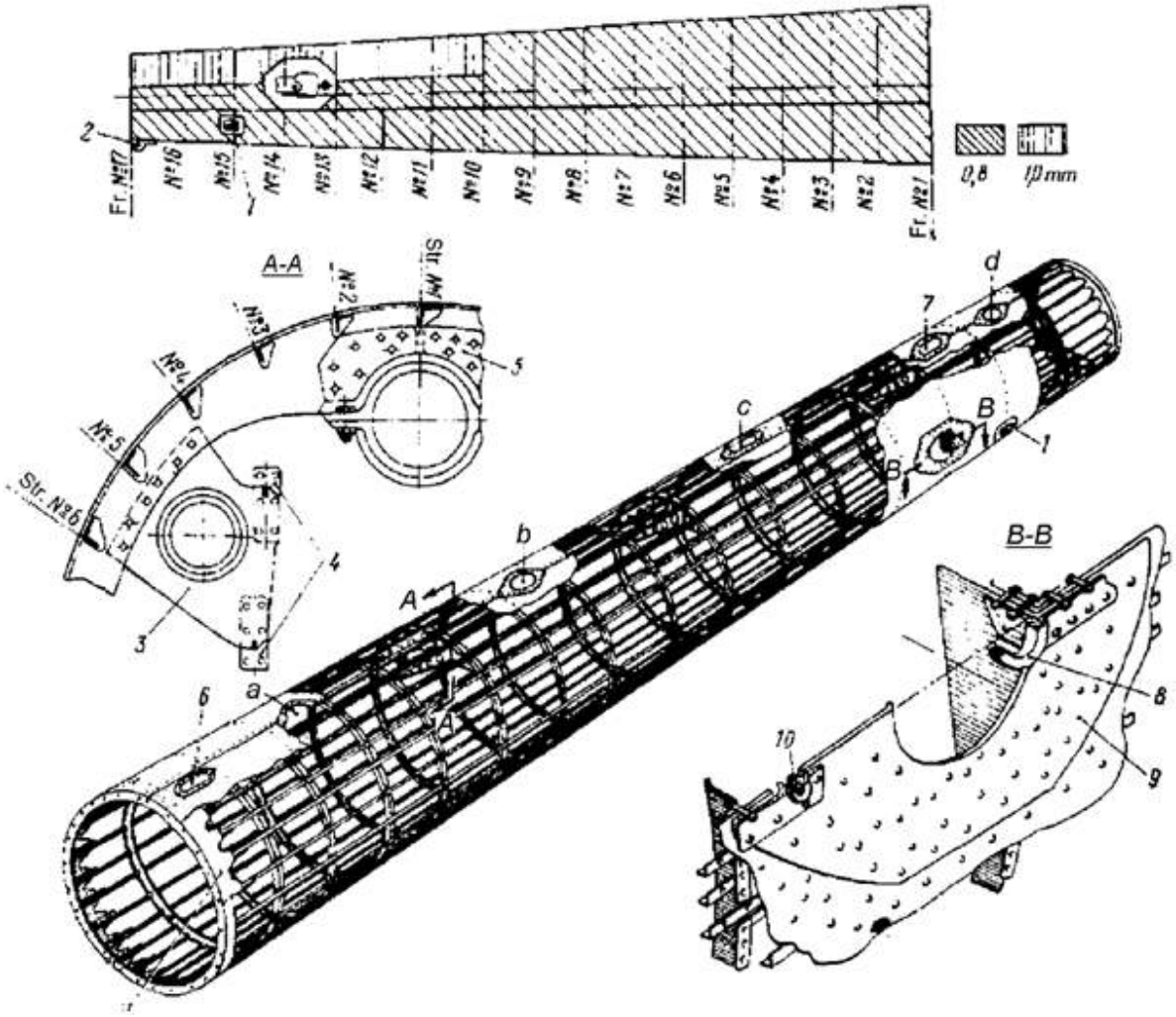


Fig. 19.14. Tail Boom Structure:

1 – tail bumper bracing attachment fittings;

2 – tail bumper shock absorber attachment fitting;

3 – pad attachment bracket ; 4 – tail rotor control cable textolite pads;

5 – tail shaft gear fitting;

6, 7 – hatches for inspection and lubrication of tail shaft spline couplings;

8 – stabilizer attachment bracket; 9 – cover plate;

10 – bumper; 11 – cog belt;

a – cutout for flash beacon;

b, d – cutout for formation lights;

c – cutout for compass system transducer.

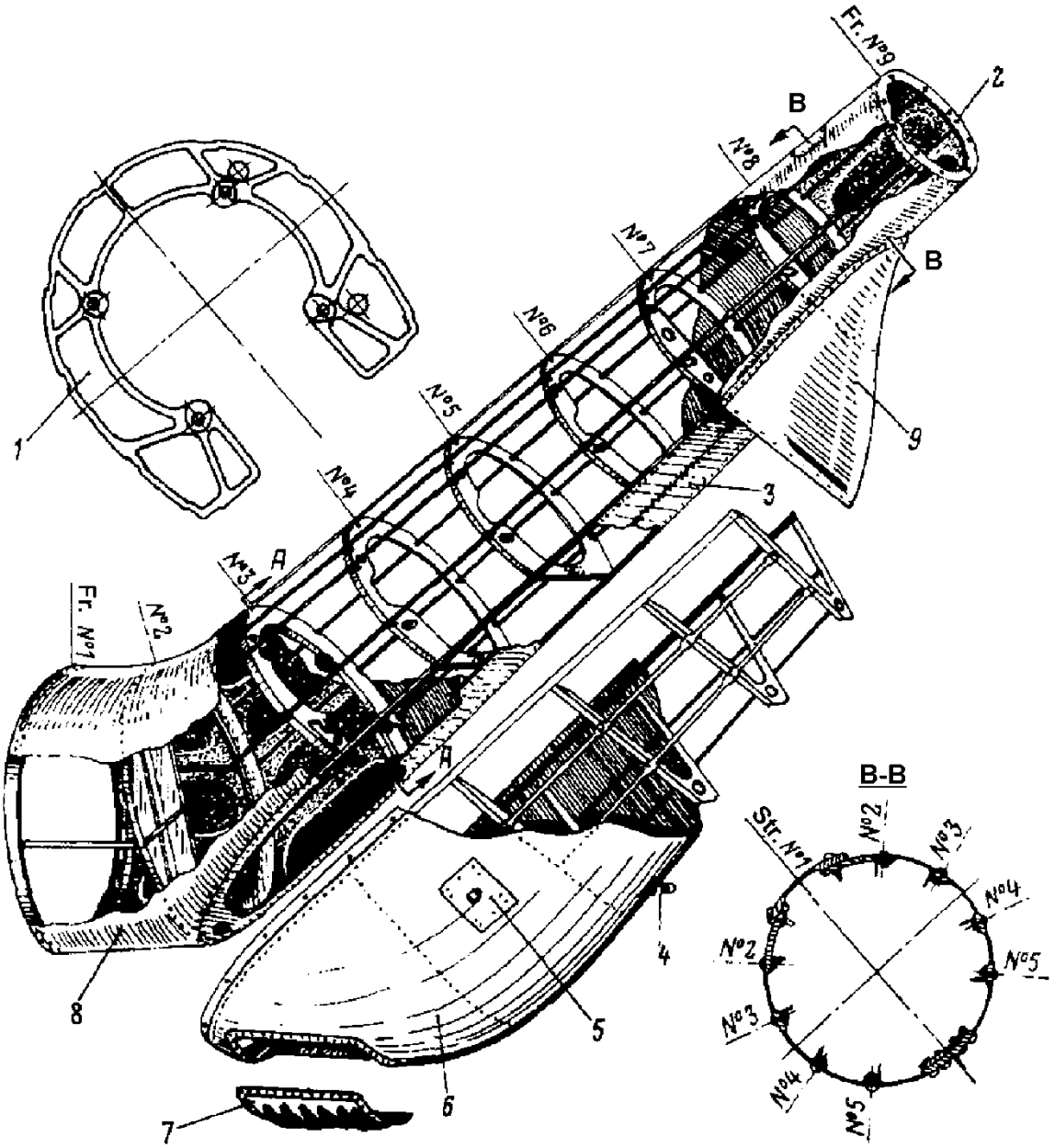


Fig. 19.15. Tail Pylon Structure:
1 – frame No. 3 with intermediate reducer attachment fittings;
2 – frame No. 9 with tail reducer attachment flange; 3 – spar; 4 – tail light;
5 – antenna of item 020M; 6 – fairing detachable part; 7 – cover with gills;
8 – fin beam; 9 – fairing upper (fixed) part

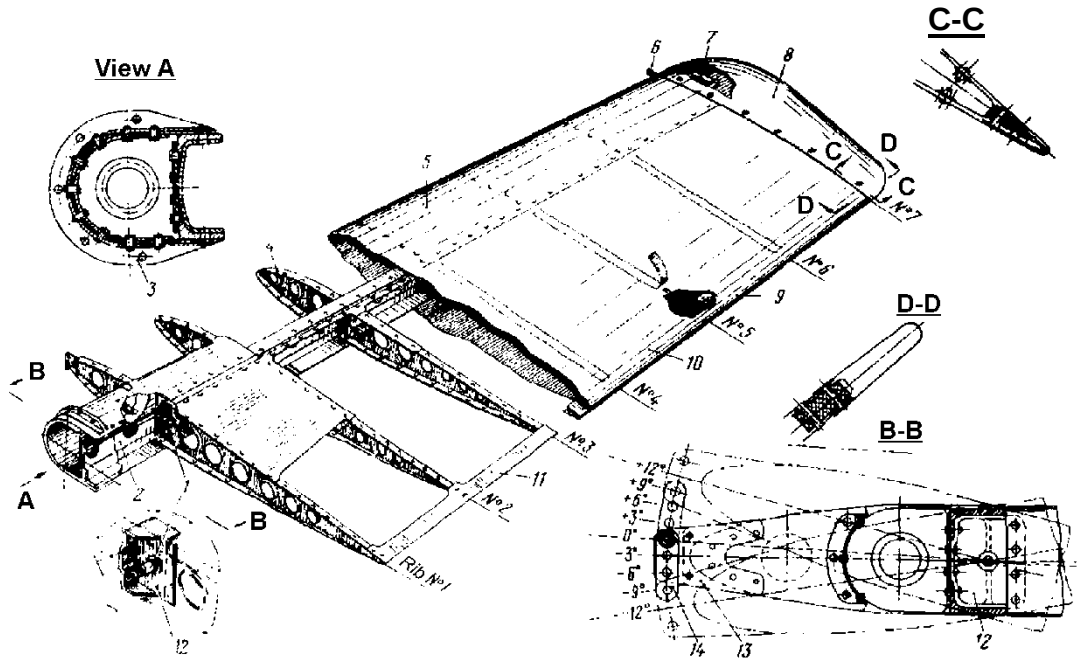


Fig. 19.16. Structure of Stabilizer Right Part:

1 – stabilizer hinge pivot; 2 – spar; 3 – coupling flange; 4 – rib; 5 – duralumin skin; 6 – beam antenna attachment lug; 7 – balancing weight; 8 –tip fairing; 9 – disk with drain outlet; 10 – fabric skin; 11 – tail stringer; 12 – fitting; 13 – adjusting bracket; 14 – adjusting lug

Stabilizer has asymmetrical airfoil NACA-0012 and consists of the right and left parts of the tapered planform shape.

Each part consists of:

– spar 2;

– 7 ribs 4;

– tail stringer 11;

– tail fairing 8.

Both parts are connected to the hinge pivot 7, provided with setting angle control unit (view B-B).

Stabilizer nose part is covered with 0.8 mm D16T duralumin sheets, and the tail part is covered with AM100-OP canvas. Stabilizer must be replaced in case of the skin breaks exceeding 50 mm.

Balancing weight 7 of 0.2 kg is riveted to nose of rib 7. Attachment lugs 6 of RST «KARAT» antenna cable are riveted outside the nose of rib 7.

19.3.2.6. Windows

Helicopter windows are intended for view, lighting and inspection. They are made of plexiglas and silicate (laminated) glass.

The right and the left pilot windshields are made of silicate glass. They are flat, applied with film heaters and electric wiper blades.

Controls are placed on the anti-icing system panel of the left console of control board. They are “W/S & WINDOWS HEATING MANUAL-AUTO” switches.

The windshield wiper switches «OFF – RESET – START – SPEED 2 – SPEED 1» are placed on the left and right panels of the control panel.

Other windows including the blisters are made of cambered plexiglas of 3 mm thickness that is blown over with a warm air from the heating system.

The windshields are installed into the canopy frame cast of magnesium alloy and fixed with sealing compound VITEhF-1NT and duralumin planks and screws.

Cracks in windows not exceeding 100 mm are allowed with further drilling of its ends.

There are 11 round windows, 6 on the port side and 5 on the star board of cargo cabin, which are open inwards and made of cambered plexiglas.

Separate windows are flat. Some helicopter models has one more glass in the cockpit entrance door.

19.3.2.7. Doors and Access Doors

There are entrance doors (see Figs 19.11, 19.17), operational doors, emergency doors and service doors.

Entrance doors:

– Sliding door (on port side)

– Cockpit door;

– Cargo doors.

Operational doors:

– Engine compartment assess doors;

– External load cable release opening;

– Service doors (Fig. 19.17).

Escape doors:

– Cockpit blisters (see Figs 19.11, 19.18);

– Emergency exit (right side);

– Access door in the right leaf.

Entrance door (Fig. 19.19) is situated on the portside between frames 1...3. Sliding door is 1405×825 mm and is fitted with key lock with outer handles 4 and 10 and inner handles 3 and 7, plug in chap lock and window 6.

Cargo door leaves (Fig. 19.20) are situated between frames 13...21 of the fuselage center section, mounted on two hinges to frame 13.

Door opening dimension – 2340×1800 mm.

The door leaves are mounted with folded ramps 13 and are fixed from inside with longitudinal locks 10 and lateral locks 11.

Right door leaf has an emergency opening closed with cover 8, and access door 6 to connect the heater sleeve.

Cargo doors position control is provided by red annunciator “CARGO DOOR OPEN” on the left console of control panel, that is switched on by microswitch A802V, mounted under the left leaf. If annunciator lights this indicates that cargo doors are not locked.

To transport MR blades or other long cargo the leaves are slightly opened and fixed in this state with the additional bars. The angle between open leaves mustn’t exceed 15°.

Cockpit door is made of duralumin plate or other material (plywood). The door has a lock with 2 handles, and 2 latches from the side of cockpit. Upper part has an eyesight or a window, from the inner side there is the technician seat backseat. The door is attached to frame 5N post by means of 2 hinges.

Engine compartment access door (Fig. 19.21) is mounted on the cockpit overhead panel. The door of the compartment is locked mechanically with inner handle 4, which has release button 3. Lock control linkage is fitted with adjustment coupling 9.

Right hinge is fitted with open position stop.

The door opening is sealed with molded rubber sealant.

To examine the engine air intakes the access door is provided with oblong plexiglas windows.

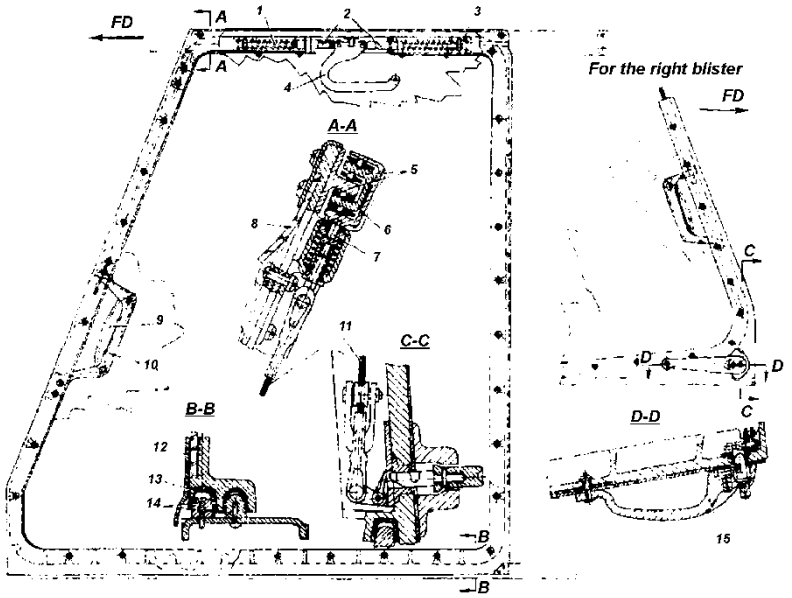


Fig. 19.17. Sliding Blister:

1 – spring; 2 – locking pins; 3 – bracket; 4 – blister jettisoning handle; inner guide sections; 6 – outer guide section; 7 – lock pin; 8 – plexiglas ; 9 – horn; 10 – handle; 11 – cable; 12 – rubber joint; 13 – felt washer; 14 – bottom channel; 15 – outer handle.

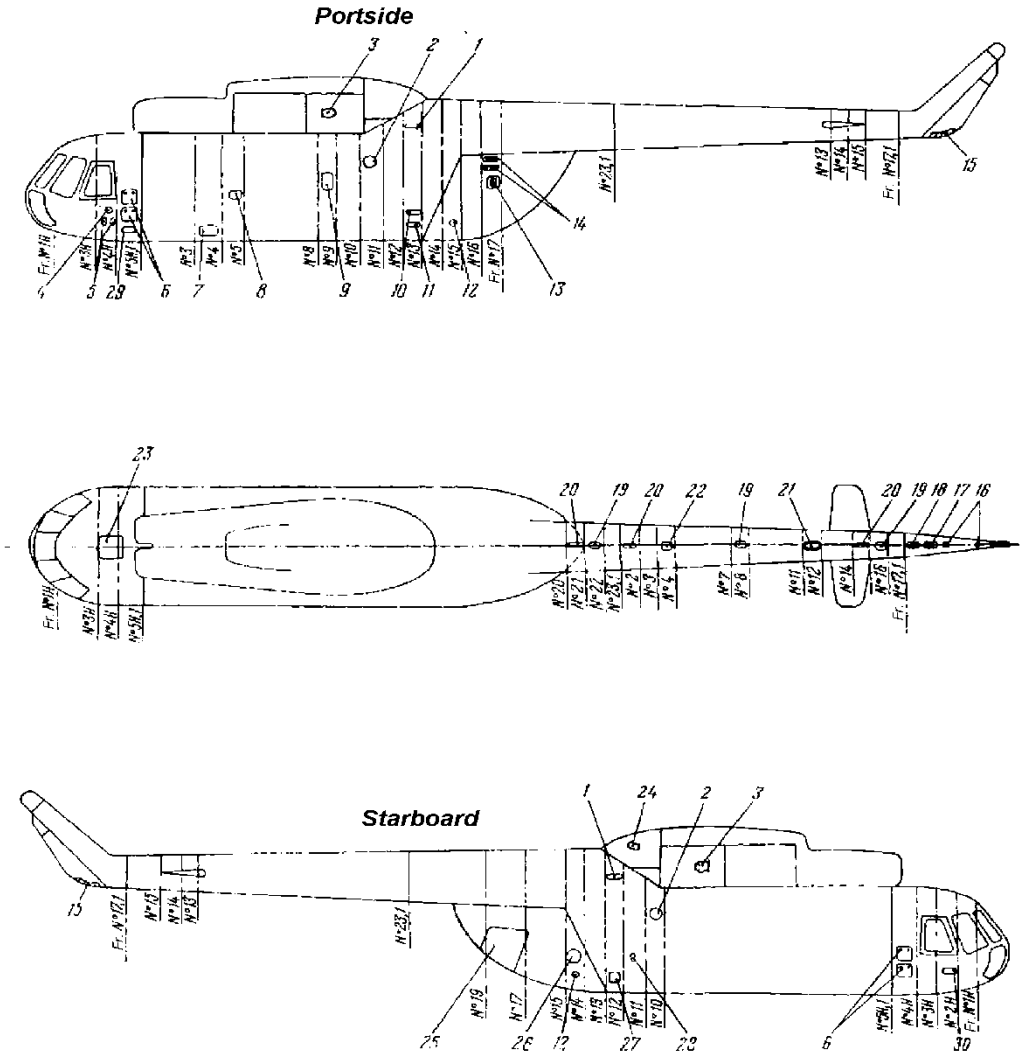


Fig. 19.18. Arrangement of Access Doors in Fuselage:

1 – access door (right – for the fuel flow system pipe fitting, left – for the hydraulic power system fitting); 2 – doors to the tank breathing pipes; 3 – doors for sleeve attachment when reducer heating; 4 – DC SHRAP-500 door ; 5 – AC SHRAP200LK door; 6 – batteries access doors ; 7 – long-range tank drain valve access door; 8 – drain tank drain cock; 9 – extra-long-range fuel tank fill access door; 10 – air system panel access door; 11 – hydraulic power system access door; 12 – transported equipment exhaust gases discharge doors; 13 – access door with ventilation gills; 14 – access doors for signal pistols; 15 – access door with ventilation gills of the intermediate reducer cooling system; 16 – access door for tail shaft inspection; 17 – access door with ventilation gills of the intermediate reducer cooling, oil supply and tail shaft torsion angle fitting; 18 – access door with ventilation gills of the intermediate reducer cooling, tail shaft splined couplings lubrication and inspection attachments and tail shaft torsion angle fitting; 19 – cut out for signal pistols; 20 – access doors for the tail shaft splined coupling inspection and lubrication; 21 – cut out for the compass system transducer; 22 – cut out for the anti-beacon light; 23 – door to the engine compartment from the cockpit; 24 – access door for filler tank service; 25 – emergency exit door; 26 – access door for the conditioning sleeve attachment; 27 – fuel drain valve access door; 28 – oil drain valve access door; 29 – access door for the grounding cable; 30 – footstool access door

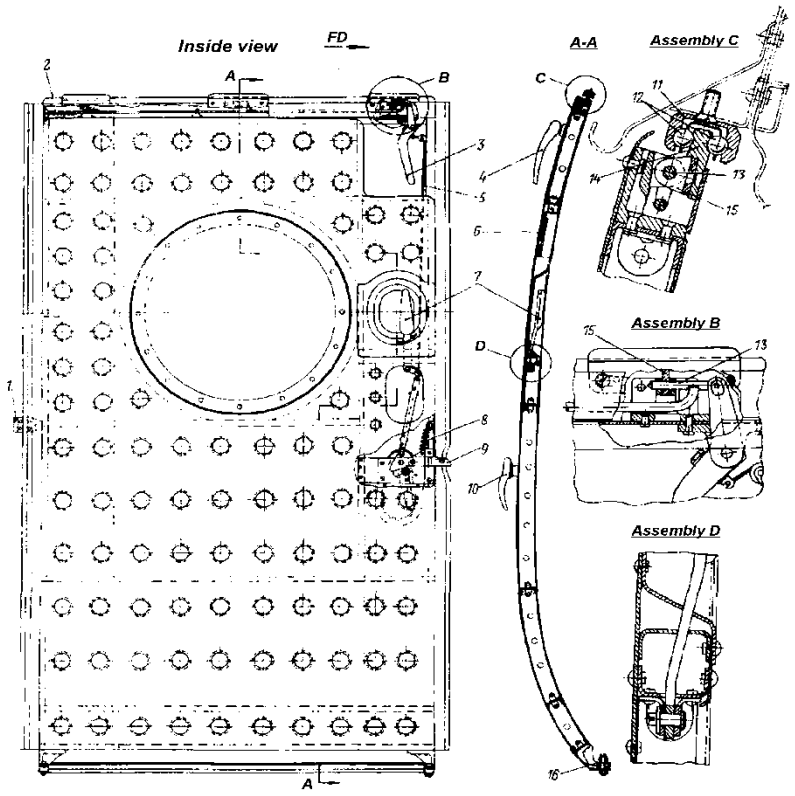


Fig. 19.19. Sliding Cargo Door:

1 – retaining pin; 2 – locking pin spring; 3 – inner door jettisoning handle; 4 – outer door jettisoning handle; 5 – cable; 6 – plexiglas; 7 – inner handle; 8 – pawling spring; 9 – paw; 10 – outer handle; 11 – top guide track; 12 – balls; 13 – locking pin; 14 – runner; 15 – bracket; 16 – roller.

External load cable release opening is made in the cargo compartment floor between frames 7 and 8. The opening is closed with double cover fitted with moulded rubber strips. The upper cover is mounted on two hinges, and the lower one is connected to upper cover with two adjustable control rods.

The upper cover is fitted with two locks and handle for opening.

Cockpit blisters (see Fig. 19.18) are the frames made of mg-alloy attached to the cambered plexiglas windows with planks and screws.

The size of opening is 750×750 mm.

Blisters are moved backward over guide channels 6 made of D-16T with felt packing and fixed in every 100 mm and in extreme positions.

Blister handle lever 9 is connected with cables 11 to locking pins 7.

The blisters have the emergency release springs, with red handles situated upside the door opening.

The blisters are conditioned with the air from the ventilation and heating system.

The left blister has visual ice detector outside.

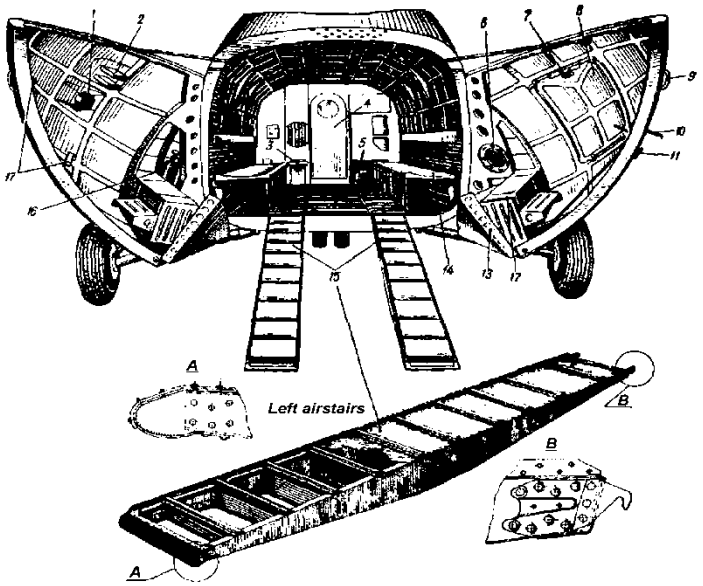


Fig. 19.20. Loading Ramps and Cargo Doors:

1 – exhaust ventilation shutter; 2 – signal pistol; 3 – flap seat; 4 – cockpit door; 5 – electric winch LPG-2; 6 – hose admission access door; 7 – emergency exit door jettison handle; 8 – emergency exit door; 9 – handle; 10 – lateral lock; 11 – longitudinal lock; 12 – tool box; 13 – panel board; 14 – troop seats; 15 – stairs; 16 – portable fire extinguisher; 17– attachment brackets of support assemblies for stretchers

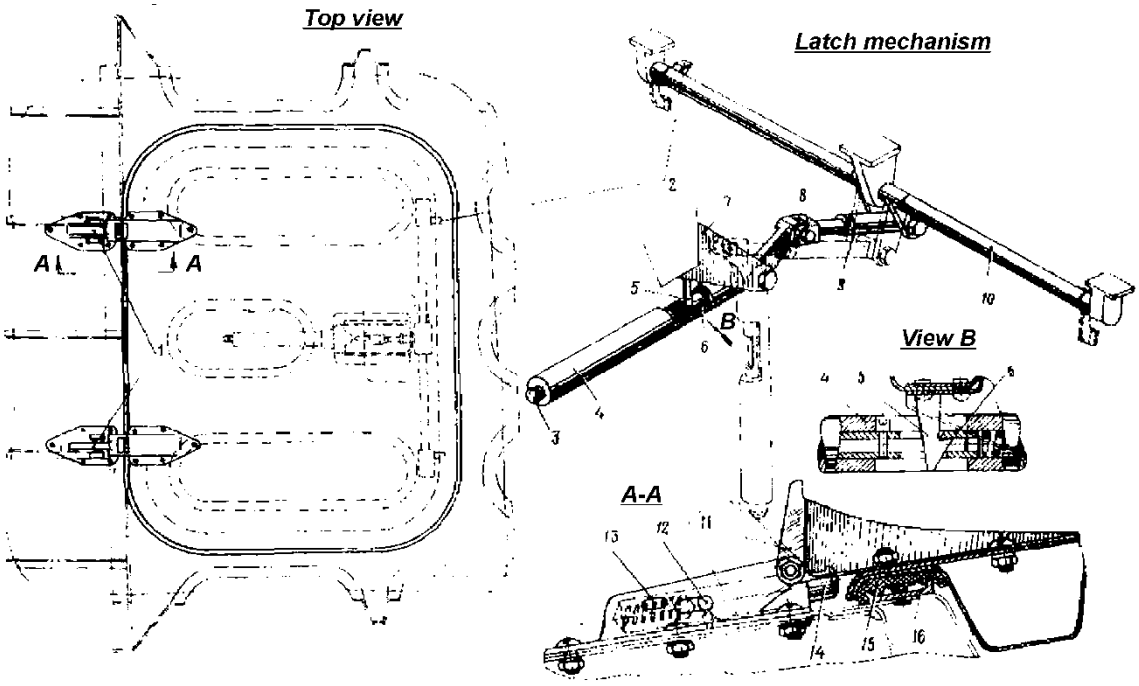


Fig 19.21. Engine Compartment Access Door in Overhead Panel Construction:

1 – hinge; 2 – lock lug; 3 – release button; 4 – lock handle; 5 – claw; 6 – handle retainer; 7 – bracket; 8 – fork; 9 – adjustment coupling; 10 – shaft; 11 – profiled rib; 12 – retainer pivot; 13 – retainer spring; 14 – retainer; 15 – moulded rubber strip; 16 – duralumin section

Right cargo leave hatch (see Fig. 19.11) has an opening of 700×1000 mm with a cover.

The cover 10 is fitted with emergency release spring assembly. The release handle 9 is situated overhead near the door opening. After pulling the handle the cover is pushed outward by the springs.

In the door opening a shooting unit can be mounted.

Right emergency exit on starboard is situated between the frames 3...4 of the fuselage center section and protected with a cover equipped with non-openable round window.

The door is opened outward, equipped with inner and outer handles and emergency release plug-in type mechanism.

The size of the opening is of 460×700 mm, a shooting unit can be mounted there.

19.3.2.8. Cockpit Equipment

The cockpit accommodates crew workstations, controls, monitoring devices, indication systems, furnishings and crash axe. Another crash axe is in the cargo compartment. If needed the cockpit has an armor protection.

According to the crew manual the panels in the cockpit have the following names:

– left CB panel;
– right CB panel;
– electrical panel left side panel;
– electrical panel left panel board;
– electrical panel left panel;
– electrical panel mid panel;
– electrical panel right panel;
– electrical panel right panel board;
– electrical panel right side panel;
– power distribution panel;
– left cockpit panel;
– center pedestal;
– right cockpit panel;

Furnishings

Furnishing include the equipment providing necessary flight conditions, including cockpit stores, cargo compartment touching-up and equipment, and emergency escape equipment.

Cockpit furnishings include:

– pilot's seats;
– attendant seat;
– instrument flying blind;

– glint blind;
– suspension hook FUP-2 for Flight Manuals' bag;
– med kit;
– light filters
– blister door opening handles;
– correction chart cartridge;
– mat on floor;
– heat-sound t insulation;
– carrier system observing mirrors;
– retrospect mirror.

Cargo compartment furnishings include:
– decorative trim panels on ceiling and sides;
– radio bay door blind;
– frame No. 13 heat-insulating blind;
– black-out flaps;
– clothes rack;
– equipment and weapons safe;
– temperature indicator TV-45 and a lamp KLRSK – 45.

Pilot's seats (Fig. 19.22) are of the same structure and consist of the pan, backrest, guide tracks and a rack.

The seats can be regulated in high (by 160 mm) and be fixed in 5 positions. The fixator handles are on the side, behind the seat. The seat goes up influenced by the springs, and down under the pilot's weight influence. It is not regulated laterally.

Attendant seat is throw-back. All the seats have safety belts.

19.3.2.9. Cowl Panel

Cowl panel (Fig. 19.23) consists of:
– engine air inlet 2;
– engine compartment doors 16;
– fan air inlet 7;
– cooling compartment doors 8, 15;
– gear compartment doors AI-9 10, 13, 14;

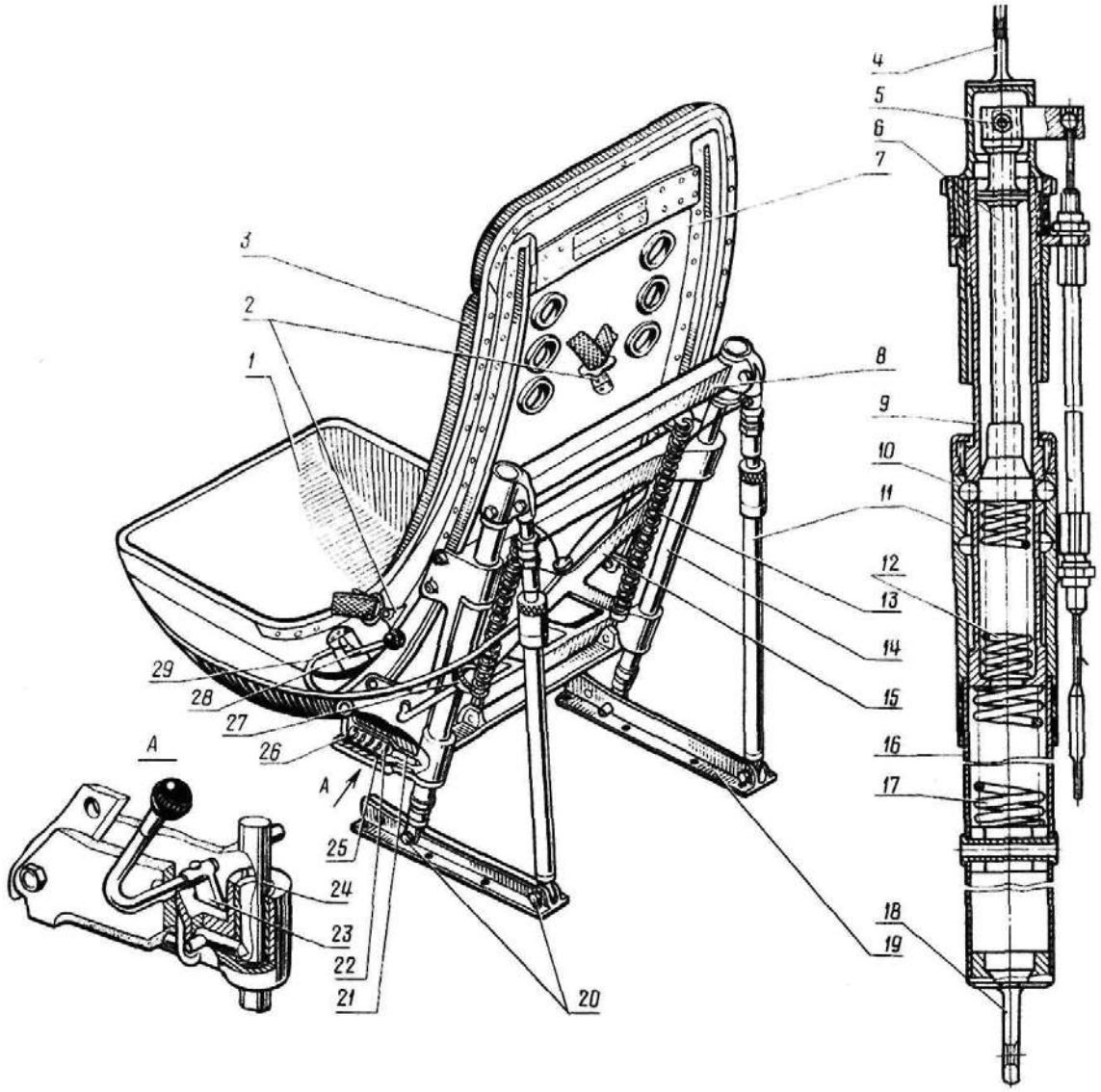


Fig. 19.22. Cockpit Seat:

1 – pan;
2 – safety belt;
3 – back cushion kit;
4 – sit bracing tag;
5 – crowbar;
6 – core
7 – back rest;
8 – spreader bar;
9 – ball-point lock case;
10 – lock ball;
11 – seat frame jib with defection mechanism;
12 – ball point lock spring;
13, 22 – springs;
14 – guide track;
15 – rigid-jointed frame
16 – duct;
17 – brace strut spring;
18 – bracing to the cockpit floor bracket fitting lug ;
19 – bracket;
20 – slave bolts with fixing spins;
21 – retaining pin;
23 – driver;
24 – head;
25 – jar;
26 – locking pin;
27, 28 – handles;
29 – control cable

– frames 11, 18;

– longitudinal and lateral fire barriers 16, 18.

The engines compartment doors and main reducer are work platforms to serve the power plant and when opened are kept with two cylinders 20, those are air dampers, preventing the fuselage from damage when opening.

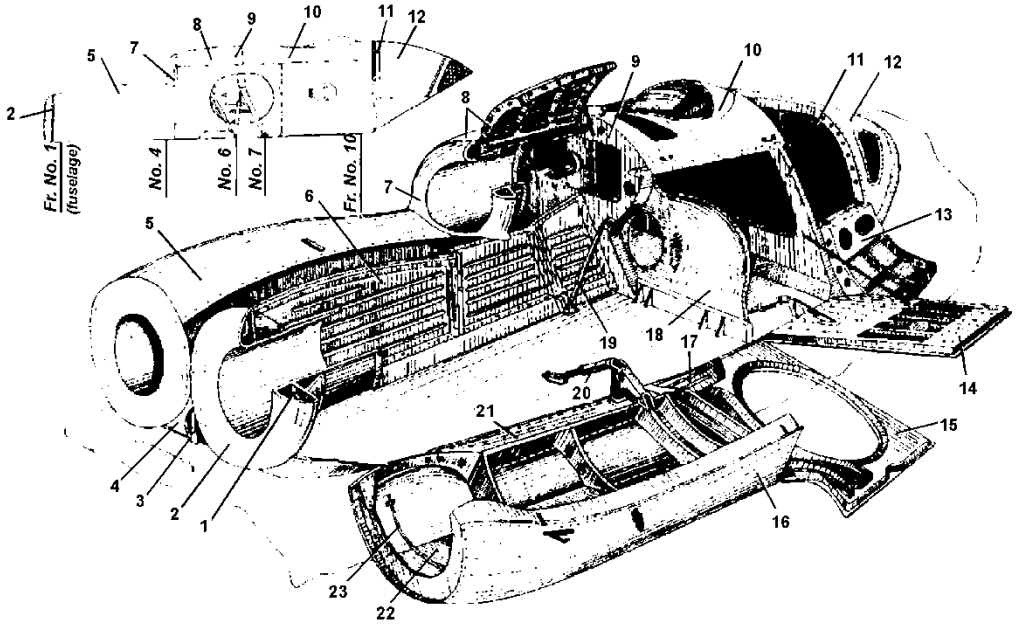


Fig. 19.23. Cowl Panel:

1 – antiicing system bellmouth, supplying warm air; 2 – air feed to engine duct; 3 – rail; 4 – leg; 5 – engine compartment cowl panel; 6 – lengthwise fire protection; 7 – fan air delivery duct; 8, 15 – ventilation compartment cowl panels; 9 – frame No. 1К; 10 – main reducer compartment cowl panel; 11 – frame No. 2К; 12 – cowl end compartment; 13 – cowl compartment side panel; 14 – reducer compartment cowl panel; 16 – engine compartment cowl panel; 17 – footrest hatch access door; 18 – lateral fire web; 19 – brace strut; 20 – air vibration damper ; 21 – upper site; 22 – lower site; 23 – cable.

19.3.2.10. Engines and Gearbox Attachment

The engines are fixed:

– in the front part – behind the volumetric compressor case to the assemblies of the overhead panel with 4 stand bars of normal engine cross-section.

– in the aft part – to the fulcrum block, attached to the case VR-14.

Drawing bars and forks are made of steel 3OKhGSA and 18X2N4VA. Fork protrusion is allowed no more than 28 mm.

The gear is fixed with a steel structure to the assemblies on frames 7 and 10, this provides gear mast tilt forward by 4°30'.

The geared frame consists of four welded struts, each of them consists of the main and additional braces.

19.3.2.11. Fan

Intended to jet the air to the oil assembly boxes and accessories, attached to the main reducer for cooling.

Specification:

Type.....8A-8311-00 IV series with season adjustment
Wheel diameter, mm.....400
Direction of rotation.....left
Rotation speed, rpm..... .....6000±60
Capacity (in mode «Summer» and t_{amb} = +40°C), m^3/s….....> 4.61
Water pressure (in mode «Summer» and t_{amb} = +40°C), mm.....≥ 495
Consumed power (in mode «Summer» and t_{amb} = +40°C), hp…....≤ 60
Maximum M_t of cardan shaft, kgm.....12
Weight, kg.....16.8

The unit consists of:

– axis fan;

– diffusor shroud;

– air ducts and stacks.

Ventilation assembly (Fig. 19.24) is an axle single-staged low-pressure compressor of high capacity, mounted between the engine exhaust ducts. The fan consists of the vane ring 3, working wheel 5, and an outlet straightener. The working wheel shaft 2 has 2 ball-bearing assemblies and gets M_t from main reducer through the cardan. The lubricating oil goes to the bearings through the push-button oiler 7. Fan coaxiality to the actuator flange on main reducer is regulated at the plant, and require no regulations when working.

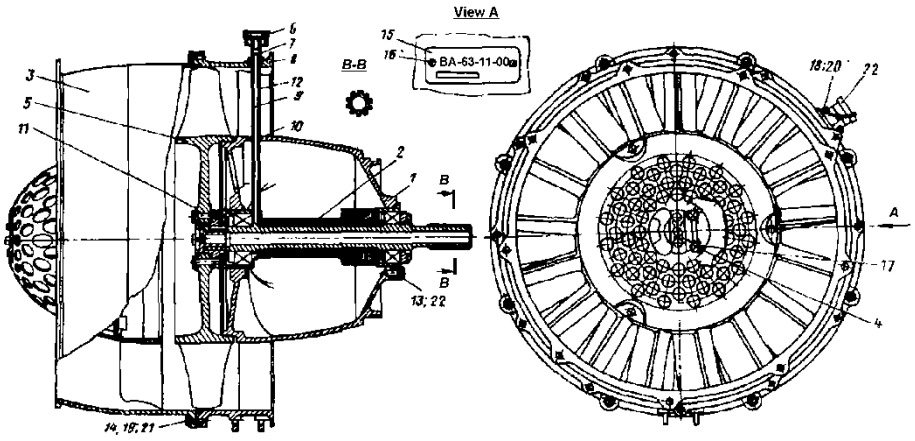


Fig. 19.24. Fan of IV series:

1 – assembled straightener blades; 2 – assembled straightener blades shaft; 3 – vane ring;; 4 – copying plane; 5 – fan spacer; 6 – lubricator cap; 7 – lubricator case; 8 – flange adaptor; 9 – standpipe; 10 – space block 11 – securing nut; 12 – resealing spacer; 13 – nut; 14 – locking pin; 15 – logo; 16 – screw-shaped clinch; 17 – male screw; 18, 19 – screw nuts; 20, 21 – rivets; 22 – locking wire

The oil from the engines cooling is provided by the air conditioning through the air-oil coolers. The coolers, hydraulic pumps and air compressor are ventilated by the air coming from the fan through the flexible ducts.

The fan gets the air from the access duct. Part of that air goes to the oil coolers, and then heated air goes outside through the welding nipples in the upper part of the main reducer compartment.

Another part of the air goes to frame 1K core space, that is an air collector through the intake screens. From there air through the flexible ducts goes to cool the reducers, hydraulic pumps and air compressor. The fan has a device, that allows to change the water pressure and the capacity. That is possible due to the flexible diaphragm turns at the season operation. In winter three windows of the unmovable diaphragm are opened, in summer they are closed. The fairing is supplied with a placard with “W” and “S”.

19.4. LIFTING SYSTEM. TRANSMISSION

19.4.1. Lifting System. General

A lifting system is the propulsor of helicopter and it is intended to maintain and move the helicopter in air.

The type of the system is classical (single-rotor) with tail rotor.

The system includes main and tail rotor as well as control devices.

19.4.2. Main Rotor

19.4.2.1. General Information

Main rotor is intended to create lifting power required to implement vertical ascent and forward flight of the helicopter. Main rotor consists of five blades and a head.

Specification:

Diameter, m.....21.3
Rotor disk area, m^2.....356
Space factor..... .....0.0777
Direction of rotation (bottom view).....left
Rotor weight, kg.....1285

19.4.2.2. Main Rotor Head

The main rotor head transfers rotation from main gearbox to blades. It is also used for taking up and transfer of the aerodynamic forces emerging in the main rotor on the fuselage.

A scheme of the head (Figs 19.25 – 19.28) – five-bladed, with spaced drag hinges (DH), with spaced and rotated flapping hinges (FLH) and feathering hinges (FH).

Specification:

Spaced FLH, mm.....220
Spaced DH, mm.....507
FLH middle ear displacement, mm.....45
Pitch-flap coupling value..... .....0.5
Flapping angle.....25°±30'

Droop angle:
– on the bracket support.....3°40'...4°10'
– on the travel limiter of the centrifugal limiters of blade droop..... 1°40'...2°
Turn angle relatively to DH:
– forward rotation..... .....12°45'...13°15'
– backward rotation..... .....10°50'...11°10'
Frequency of rotation of the main rotor with which centrifugal limiters of blade droop works, rpm:
– at acceleration.....105...111 (52...55%)
– at braking.....92...98 (45.5...48.5 %)
Angle of forward tilt of the main rotor axis .....4°20'...4°30'
Head diameter, mm.....1744
Head weight, kg.....610
Lubricant:
– FLH, DH and FH of the head..... Appendix No. 2
– other joints.....TSIATIM 201
Hydraulic dampers refuelling.....AMG-10

The design of the head provides reduction of blade angle of installation Δφ at its flap relatively to FLH on an angle of β according to the law: Δφ =-kβ which is implemented by displacement of a fastening point of rod from the swashplate to a FH driver with FLH axis.

The coefficient of k is equal to 0.5. It is called a pitch-flap coupling coefficient and is expressed as a tangent of hinge displacement angle or as b/c relation.

To reduce a blade droop and create necessary gaps between blades and tail beam under small frequency of rotation of the main rotor the centrifugal limiters of blade droop were installed in the head structure.

Basic Elements of Head (Fig. 19.29)

A. Body

The body 5 of the head is attached to main gearbox shaft with splines and is fixed to it with a nut. Tightening of a nut is made with a special hydraulic tong unit. The body has five eyes located in one plane with angle 72° to each other.

B. Flapping Hinges.

Five brackets of head 6 in joint with body 5 eyes form FLH by means of studs 67 and needle bearings. Displacement of FLH eyes is chosen so that at the main modes of flight the resultant force R of aerodynamic Q and centrifugal forces F_{cf} of blades (Fig. 19.26) is directed approximately on the middle part of FLH. Such design provides more uniform distribution of loading between the FLH needle bearings and significantly increases their durability.

On a filler plug 4 a pressure compensator is installed.

C. Drag Hinges.

FH five operating pins 9 in connection with bracket eyes form DH. They are structurally similar to FLH.

D. Feathering Hinges.

There are five cases of FH 29 on the head mounted on pins 9. The FH body 29 can be turned relatively to pin 9 on three bearings. Two ball bearings 24 and 30 take up bending moments from the blade, and a roller bearing 27 – centrifugal forces.

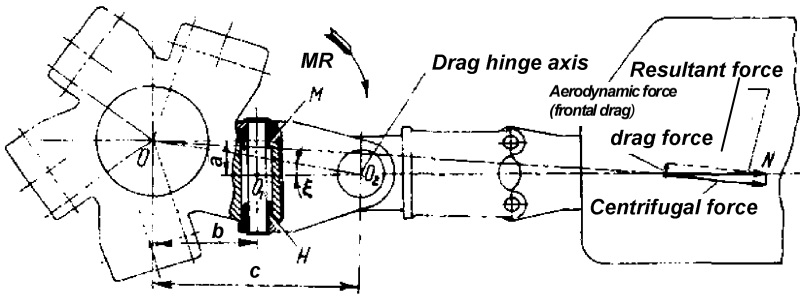


Fig. 19.25. Arrangement Scheme of Head Hinges

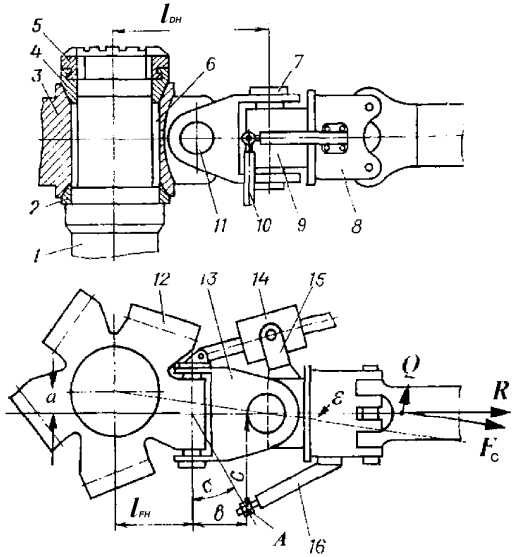


Fig. 19.26. Layout Scheme of Main Rotor Head:

1 – gearbox shaft; 2 – lower ring; 3 – head body; 4 – top ring; 5 – nut; 6 – splines; 7 – drag hinge stud; 8 – case of the feathering hinge; 9 – pin of the feathering hinge; 10 – feathering link; 11 – flapping hinge stud; 12 – eye, 13 – bracket; 14 – drag hinge damping device; 15 – damper bracket; 16 – push rod

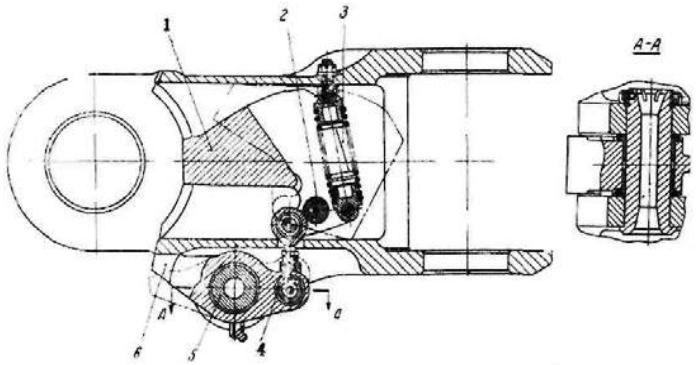


Fig. 19.27. Scheme of Centrifugal Droop Stop:

1 – counterweight; 2, 5 – studs; 3 – spring; 4 – link; c – travel limiter

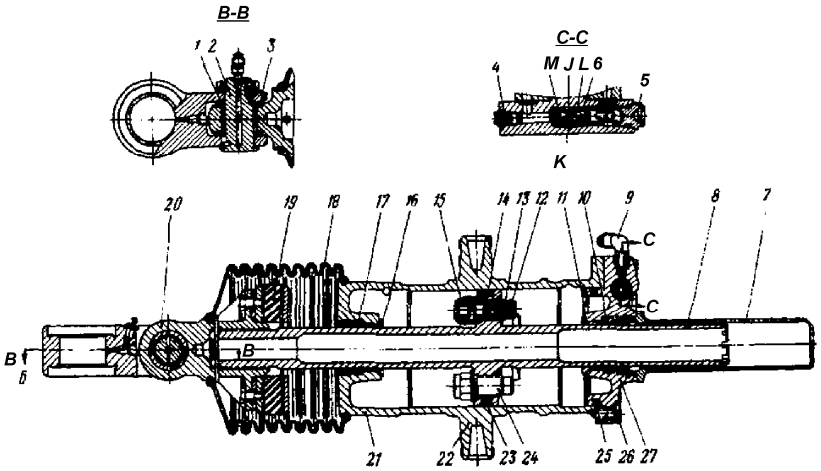


Fig. 19.28. Hydraulic Damper of Main Rotor Head:

1 – bronze head; 2 – stud; 5 – sealing ring; 4 – filler; 5 – compensating valve; 6 – stopper; 7 – body; 8 – rod; 9 – square; 10, 17 – sealing rings; 11, 16 – bronze heads; 12 – case of the return overflow valve; 13 – cone; 14 – spring; 15 – nut; 18 – cover; 19 – shock-absorber; 20 – support case; 21 – cylinder; 22 – operating pin; 23 – piston rings; 24 – return overflow valve; 25 – sealing ring; 26 – bolt; 27 – cover

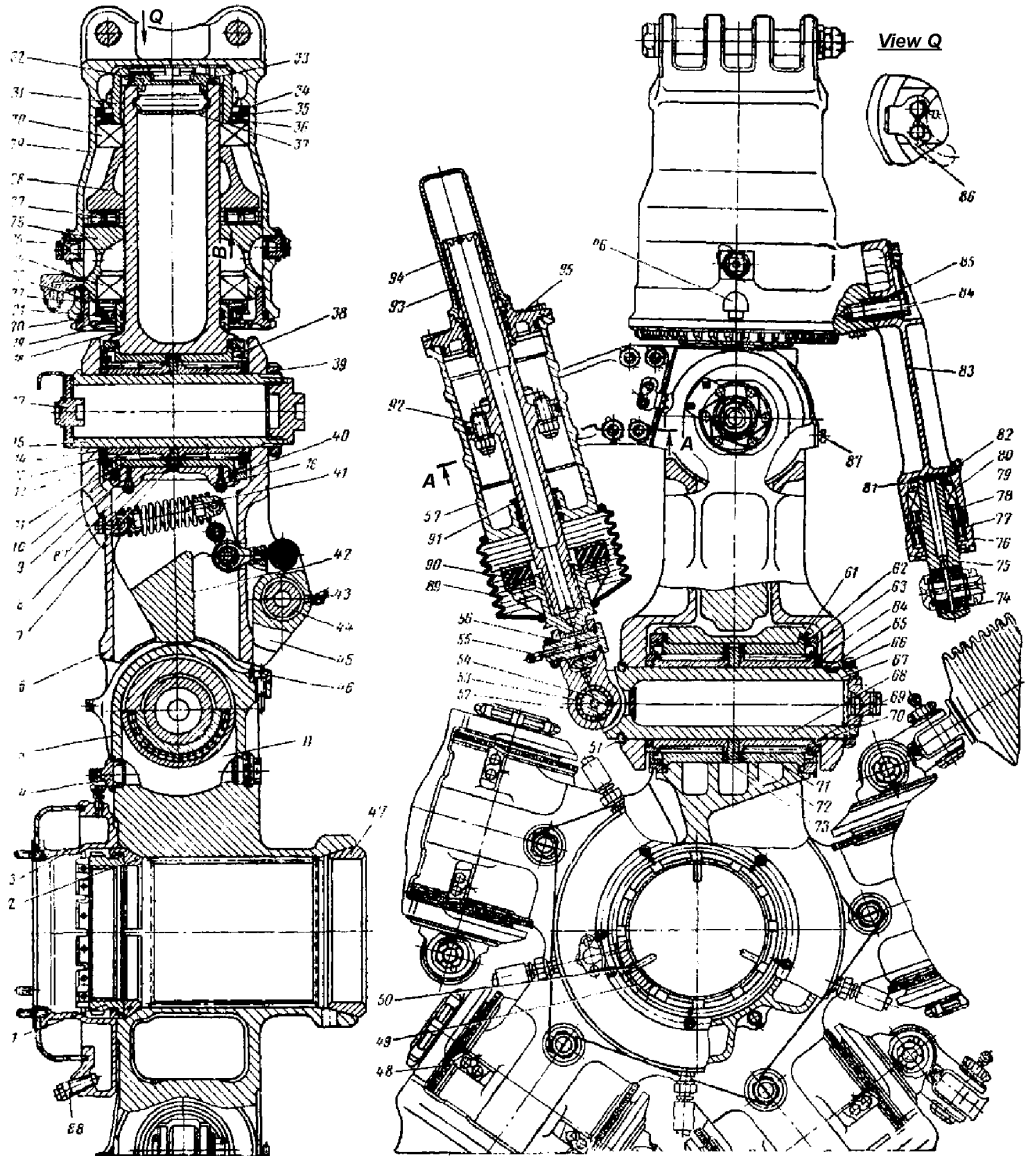


Fig. 19.29. Main Rotor Head:

1 – shaft nut; 2 – top cone; 3 – tank of hydraulic dampers; 4 – stopper; 5 – head case; 6 – bracket; 7 – thrust washer of drag hinge; 8 – washer; 9 – operating pin of feathering hinge; 10 – nut; 11 – external ring of the bearing; 12 – internal ring of the bearing; 13 – ring; 14, 20 – sealing rings; 15 – stud of drag hinge; 16 – lubrication fitting; 17 – stopper; 18 – ring; 19, 20 – collars; 22 – nut of feathering hinge case of; 23 – oil thrower ring; 24, 30 – radial ball-bearings; 25 – stopper; 26 – distance sleeve; 27 – roller bearing (separator); 28 – thrust ring; 29 – body of feathering hinge; 31 – nut; 32 – stopper; 33, 34 – adjusting rings; 35 – dish-shaped spring; 36 – washer; 37 – plug; 38 – collar; 39 – nut of stud of drag hinge; 40 – sealing ring; 41 – spring; 42 – counterweight; 43 – lubrication fitting; 44 – axis of travel limiter; 45 – travel limiter; 46 – support; 47 – lower cone; 48, 49 – lock plates; 50 – lock pin; 51 – sealing ring; 52 – ear; 53 – needle roller; 54 – stud; 55 – lubrication fitting; 56 – earring stud; 57 – hydraulic damper; 58 – nut; 59 – needle bearing; 60 – bracket; 61, 69 – sealing rings; 62, 66 – nuts; 63 – collar; 64 – ring of flapping hinge; 65 – key; 67 – stud of flapping hinge; 68 – internal ring of needle bearing; 70 – roller of needle bearing; 71 – external ring of needle bearing; 72 – stop ring; 73 – thrust washer; 74 – ball-bearing; 75 – roller; 76 – cover; 77 – roller bearing; 78, 79 – distance sleeves; 80 – ball-bearing; 81 – nut; 82 – lubrication fitting; 83 – push rod; 84 – bolt; 85 – bushing; 86 – lock plate; 87 – return overflow valve; 88 – filler neck; 89 – cover; 90 – shock-absorber; 91 – sealing rings; 92 – piston valve; 93 – rod; 94 – sleeve; 95 – cover; 96 – safety valve

The attach fitting with eyes to fasten the blade is available at the bottom of FH sleeve. The hinge is equipped with a drain magnetic plug with an oil inspection cup. Oil in the hinge has to be transparent (the opposite wall of a glass is visible).

A pressure compensator is installed on a filler plug. Due to a membrane deflection increasing the volume at pressure increase in the hinge.

E. Push Rods.

Push rods 83 are mounted on body of FH and connect them with links to the swashplate.

F. Centrifugal Droop Stop.

Centrifugal droop stop of blades (see Fig. 19.27) is mounted in eyes of brackets of the bushing. It limits a droop of blades at the stopped main rotor and with its small frequency of rotation.

The counterweight 1 of the centrifugal droop stop is suspended to a bracket on a stud 2 and through a link 4 is connected to one end of the travel limiter 6. Axis of rotation of the travel limiter is the stud 5 passed through bracket eyes. The second end of the travel limiter serves as a support which limits a droop of the blades.

At rotating of the main rotor to 105±5 rpm the spring holds a counterweight and travel limiter (the angle of the blade droop is equal to 1°40').

When the main rotor reaches rotation of 105±5 rpm, the counterweight under the influence of centrifugal force turns, stretches a spring and turns the travel limiter against the stop of a fork in a bracket. At the same time between a support of the case and travel limiter the gap is formed, and blades droop is limited only by constant support of a bracket which allows it to deviate for 3°40' down. When the rotation of the main rotor decreases to 93±5 rpm the travel limiter comes to the position corresponding to an angle of the blade droop of 1°40'.

At $t_{mr} < – 25$ °C it is necessary to remove lubricant from axes of the travel limiters of droop limiters for the guaranteed operation of the centrifugal droop stop.

G. Hydraulic dampers

Hydraulic dampers (see Fig. 19.28) serve to clear fluctuations of blades relative to DH.

Hydraulic dampers are attached to DH cases, and each of them consists of the cylinder in which the piston with a rod is placed. Eight spring valves, four of which in one party, and four – in another, are installed in the piston. To protect against dust and dirt the rod on the one hand is closed by a duralumin case, and on the other – with a rubber corrugated cover.

To soften the blow against the DH back limiter at the time of moving of the blade during main rotor rotation the support with the rubber shock-absorber is fixed on a rod.

With a small frequency and amplitude of fluctuations of the blade liquid (AMG-10) flows from one cavity of a hydraulic damper into another through openings in valves. With increase of frequency and amplitude of blade fluctuations the pressure of liquid increases, and valves open.

Thus, the hydraulic damper transforms the energy seeking to deflect the blade to work for pushing the liquid through openings and forces the blade to fluctuate near neutral situation.

To withdraw the foamed liquid, to compensate temperature expansions and replenishment of leaks the compensation tank (one for all dampers), connected by pipelines to hydraulic dampers, is installed on the main rotor head. The AMG-10 level in

a tank is controlled through a transparent cap from organic glass and has to be not higher the level, applied to a cap, and not below the lower edge of a cap.

To avoid liquid flow from cylinder cavities to a tank during the work of hydraulic damper, the compensation valve consisting of two big and one small balls is installed in a damper cover. During the work of a damper the big ball is pressed under the liquid pressure to a seat, separating a cavity of high pressure with a tank, and through a small ball wrings out the second big ball from a seat, connecting a cavity of low pressure with a tank. Such design provides passing of bubbles to a tank and a sediment of liquid in it.

19.4.2.3. Main Rotor Blade

Specification:

Blade form in the plan..... .....rectangle
Blade chord, mm.....520
Geometric twist.....from +5° in section 1...4 to 0° in section 22
Profile in section 0...1.....NACA-230
in section 2...3.....NACA-230-12
in section 4...22..... .....NACA-230
Blade weight, kg.....135
Primary members (Figs 19.30–19.33):
– spar;
– 21 tail sections;
– tip;
– wingtip.
Besides, the blade is equipped with:
– pneumatic system of the spar damage alarm;
– electrothermal anti-icing system (A/ICE);
– end-outline marker lamp;
– counterweight;
– static balancing joint.

Spar (Fig. 19.31, pos. 6) – basic primary member of the blade. It is made by a pressing method of the AVT-1 alloy and is a hollow beam to which cap and a back wall 21 tail sections are pasted by the VK-3 glue. Spar surface is cut for 3/4 according to a theoretical profile.

Rear section (Figs 19.31, 19.32) includes:
– skin 8 (avial 0.3mm);

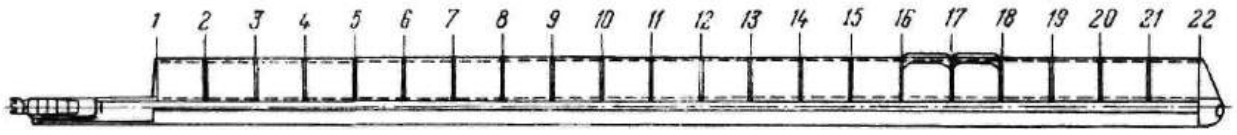


Fig. 19.30. General View of Blade In Planview:

1 – 22 – blade sections

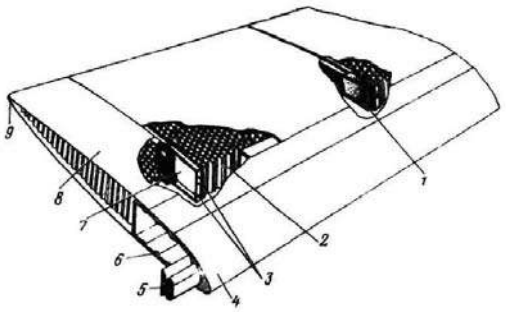


Fig. 19.31. Standard Section Of Blade:

1 – intersection insert; *2* – honeycomb core; *3* – rib tab; *4* – antiicer; *5* – counterweight; *6* – spar; 7 – rib; *8* – skin; *9* – tail stringer

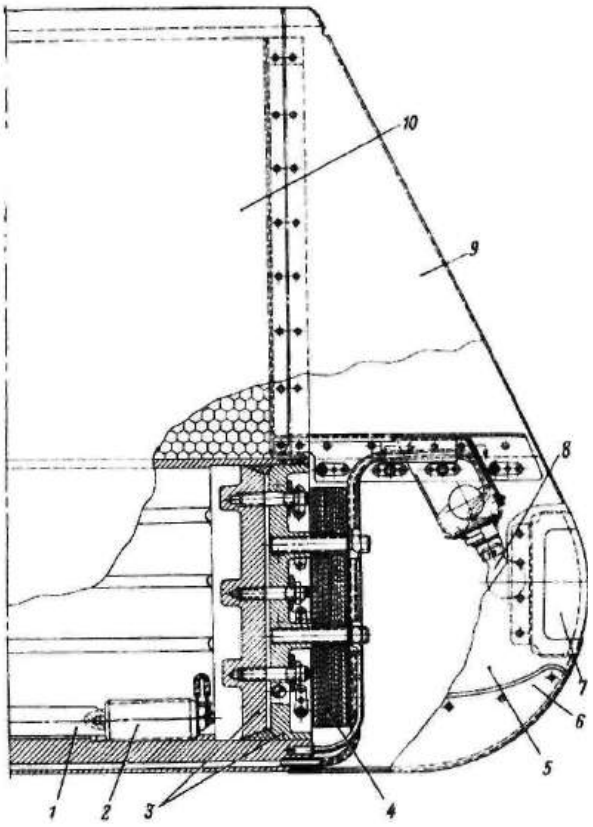


Fig. 19.32. Trail Part of Blade:

1 – counterweight; 2 – helistop; 3 – plugs for sealing of spar cavity; 4 – plates of balance weight; 5 – removable part of tip; 6 – heating plate; 7 – end-outline marker lamps' glass; 5 – end-outline marker lamp; 9 – fixed part of blade tip; 10 – tail part of section No. 21

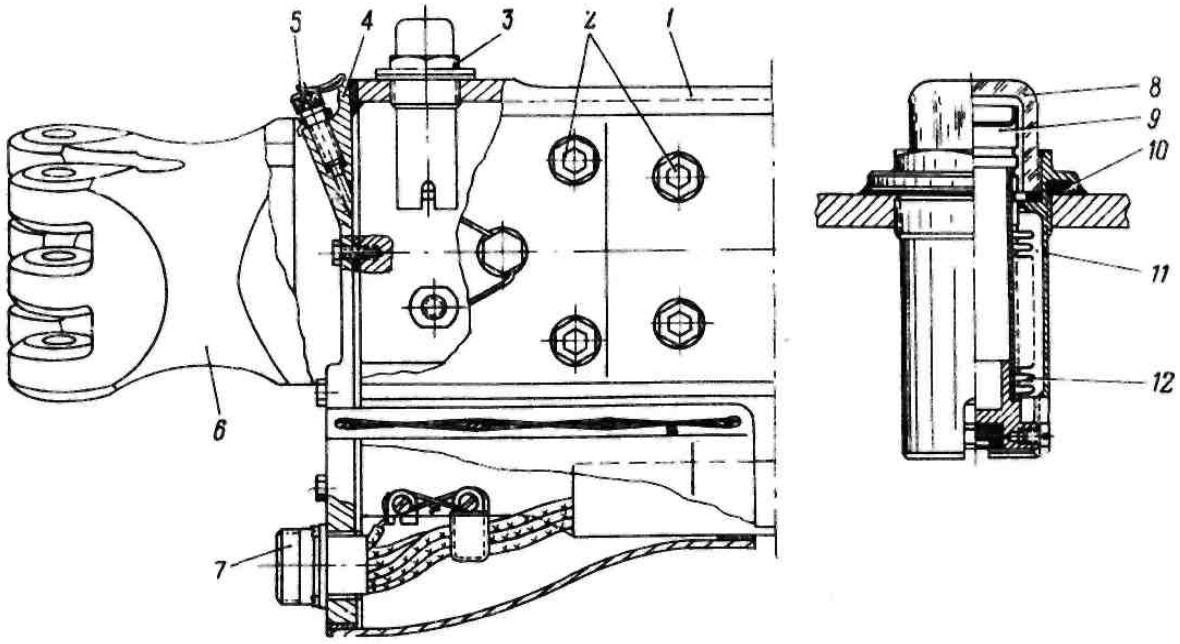


Fig. 19.32. Root Part of Blade With Air Pressure Signaling Device:

1 – spar; 2 – bolts of tip fastening to spar; 3 – air pressure signaling device; 4 – spar cavity cover cap; 5 – charging gate with valve shuttle and cap key; 6 – end; 7 – plug socket; 5 – transparent cap; 9 – bellow valve cap with red side surface; 10 – rubber ring; 11 – case; 12 – bellows-sealed sensitive element

– cellular filler 2 (aluminum foil 0.04 mm, honeycombs side 5 mm);

– two side ribs 7 (avial 0.4 mm);

– tail stringer 9 (textolite).

Rubber sealing gaskets are set between sections.

In sections 16 and 17, there are bent off trim plates of 40 mm wide serving to change moment characteristics of the blade at elimination of a no in-track condition of MR.

End (Fig. 19.33, pos. 6) is made of the alloyed steel and fixed in the root part of a spar with MPF-1 glue and 9 through bolts 2 with nuts on heads.

Tip (Fig. 19.32) is closed by the trail cowling consisting of removable 5 and fixed 9 parts. Removable part is set on screws and closes static balancing joint 4 and end-outline marker lamp 8.

Spar fault indication pneumatic system (Fig. 19.33) includes:

– bottom 4 and trail (Fig. 19.32, pos. 3) caps;
– charging valve with a cap key 5;
– bellows-sealed signaling device of pressure 3.

All elements are sealed with sealant. Air pressure in a spar has to be equal to pressure of a signaling device operation plus 0.15±0.02 kgf/cm^2 to that.

Counterweight (Fig. 19.32, pos. 1) are installed to reach necessary cross centering of the blade and increase critical speed of a flutter. In internal cavity of a spar, closer to the nose portion, eight rubberized steel whetstones of 400 mm long and weighing about 1 kg each are inserted.

Centrifugal force from counterweight is taken up by a helistop 2 of trailer caps of a spar.

Static balancing joint is intended to adjust longitudinal centering of the blade (position of the center of weight and obtaining the identical static moment of all blades relative to the center of rotation).

It is a set of plates (Fig. 19.32, pos. 4), which are fixed on pins to an end plug of a spar.

Each blade is equipped with electric anti-icing system and end-outline marker lamp. There is a hole for electrical wiring in a spar nose portion, and heating plate are pasted on the surface.

To protect against abrasive wear, external electric isolation has cheek-plates and a rubber tape of 0.5 mm thick or a layer of polyurethane 0.8...1 mm thick over.

The surface of the blade is covered with AG-3A primer coating and perchlorovinyl enamel KhV-16 of gray-blue color with 2 % aluminum powder. Trail part is enameled with KhVEh-31.

NOTE. Before designation of the blade modification figure "1" or "2" is put. It indicates a blade that belongs to the first or second weight group which has the identical static moment relatively to rotation axis.

19.4.3. Anti-Torque Rotor

19.4.3.1. *General* Characteristic

TR serves for equilibration of the jet moment from MR and for helicopter directional control.

TR consists of the hub and three blades and is installed on a flange of output shaft of TGB.

Specification:

Rotor type..... tractor
Direction of rotation.....forward-up back
Rotor diameter, m..... 3.908
Filling coefficient..... 0.135 (0.149)
Mass of the rotor, kg..... 121(122.8)

Note. In brackets different data for rotor of helicopter Mi-8MTV is specified.

19.4.3.2. *Anti-Torque Rotor Hub*

The hub (Fig. 19.34) is intended for torque transmission to the blades, taking up of aerodynamic forces from blades and their transfer through TGB to the fuselage.

Specification:

Hub type.....with the combined flapping hinge of "universal link" type
Pitch-flap coupling coefficient.....1
Range of angles of rotation of blades relatively FH:
– minimum..... 6 °35'... 8 °05'
– maximum..... 20 °10'... 20 °50' (22 ° 40'... 23 ° 20')
Blades deflection angle in universal link.....±11 °
Primary members of the hub:
– hub group;
– universal link assembly;
– feathering hinges;
– scissor assembly.

Hub group of 3rd hub is fixed to HR by flange.

On a hub group the limiter of a wave 4 and universal link is fixed.

Universal link consists from traverses 35, universal link cases 34 and an external ring 53. Which external ring is the hub body.

Universal link provides a hub body inclination from the rotation plane on a corner 11 ° in any direction. For a single whole with the case pins are executed.

On each pin on three roller bearings the FH case 52 is installed. On the case there is a comb for fastening of blades and inflow in which on needle 29 and ball 27 bearings the stud 28 is installed.

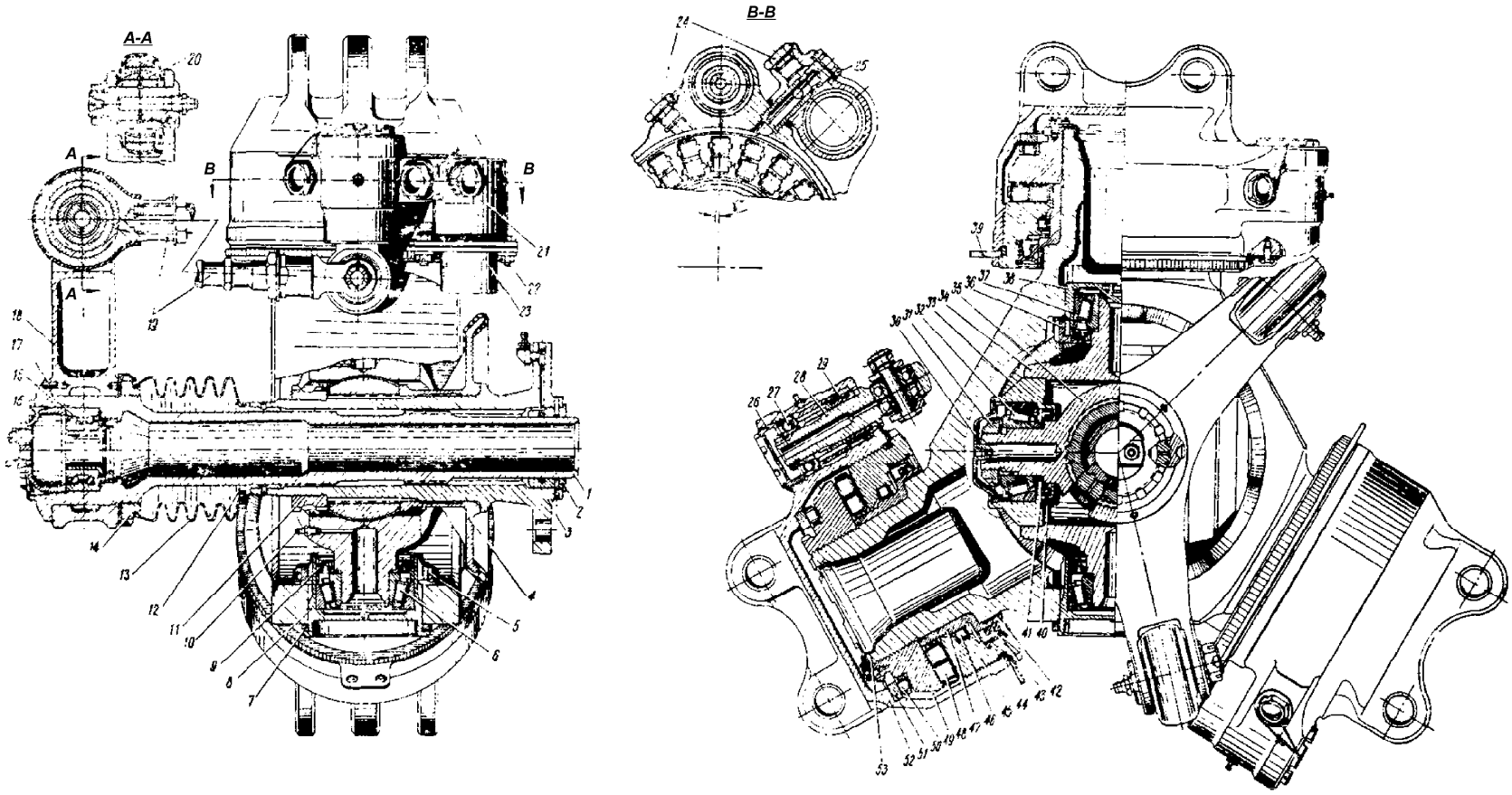


Fig. 19.34. Anti-Torque Rotor Hub:

1 – slide block; 2 – lower hub; 3 – hub group; 4 – flapping stop; 5, 11 – nuts; 6 – roller conic bearing; 7 – adjusting ring; 8 – final drive outer bearing cage; 9 – cuff; 10 – grease packing gland; 12 – top hub; 13 – cover; 14 – slide block nut; 15 – cover; 16 – ball two-row bearing; 17 – pin; 18 – driver; 19 – rod; 20 – spherical hinged bearing; 21 – alemite fitting; 22, 25 – bolts; 23 – cap; 24 – plugs; 26, 31 – nuts; 27 – ball-bearing; 28 – roller; 29 – needle bearing; 30 – cover; 32 – roller conic bearing; 33 – drive outer bearing cage; 34 – universal link body; 35 – traverse; 36 – nut; 37 – cup; 38 – adjusting ring; 39 – washer; 40, 43 – cuffs; 41 – adjust ring; 42, 44 – sealing rings; 45 – nut of the case of feathering hinge; 46 – roller; 47 – thrust ring; 48, 50 – roller bearings; 49 – nut; 51 – ring of the thrust bearing; 52 – case of the feathering hinge; 53 – hub case

Through **rod** 19 pin is connected to a three-port **driver** 18. The driver is pressed on the **slide block** 1 will also be tightened by nut 14. For protection against dust and moisture the slide block is closed by a rubber cover 13.

In a hub group there are evolvent vents on which the slide block moves. In boring of the slide block head the two-row ball bearing 16 closed by a cover is installed. The internal ring of the bearing is connected to a rod of the of TR pitch change mechanism. The rod moves an internal ring of the bearing which through balls moves the rotating external ring and the slide block along a hub axis. The slide block through a driver and rods turns the FH case, changing the blades' angles of installation.

The amount of oil in FH is controlled on level in the transparent plexiglass caps 23 installed on bodies of hinges. Oil level in control glasses has to be between fillets at position of the blade vertically down, oil has to be transparent.

19.4.3.3. Blade of Anti-Torque Rotor

Specification:

Blade type..... squared, flat, without twist
Airfoil..... NACA-230M
Chord of the blade, mm..... 270 (305)
Mass of the blade, kg..... 13.5 (14.1)

Blade Design

The TR blade (Fig. 19.35) is of the mixed design also includes:

The **spar** 4 – the primary element of the blade, is made of aluminum AVT-1 alloy by a pressing method.

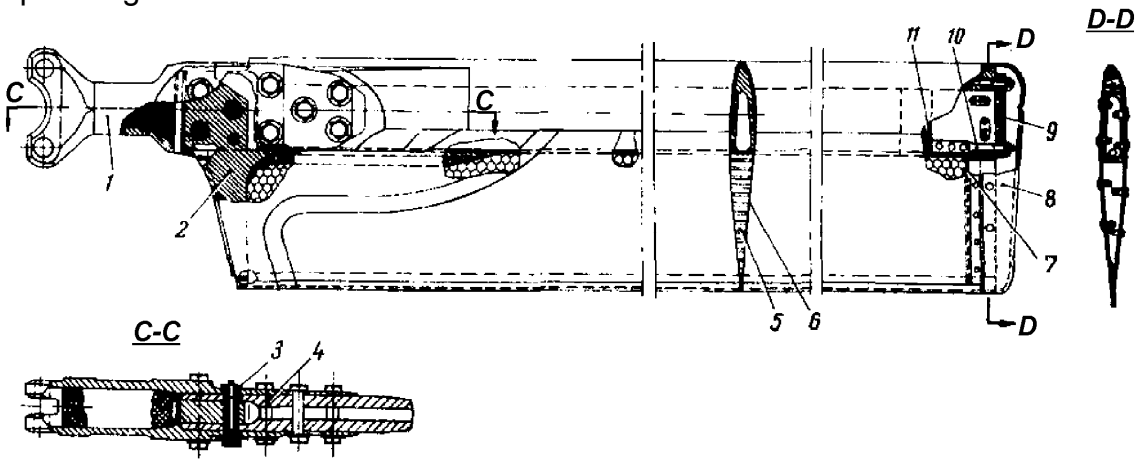


Fig. 19.35. Blade of Tail Rotor:

1 – steel tip; 2 – bracket; 3 – bolt; 4 – spar; 5 – cellular block; 6 – skin; 7 – end rib; 8 – fin tip; 9 – plates of balance weight; 10 – pin; 11 – cap

On a nose part of spar for 20 % chordwise two sections of the A/ICE heating element of electrothermal action are pasted. The heating plate from above is protected by erosion shield of stainless steel from mechanical damages and a layer of abrasion-resistant rubber of the VR-3v.

Tail part is pasted to a back wall of a spar of the blade by VK-3 glue film and consist of cellular filler 5 (same, as in MR blades) and a two-layer fiberglass covering of 0.4 mm thick. On a back edge in a tail part the fiberglass plate is pasted.

End part 1 is made of steel 18X2H4MA. It is fixed to a spar by means of eight bolts with bushes and MPF-1 glue film.

Fin tip 8 has removable and fixed parts. Under a removable part the static balancing assembly 9 is fixed to a spar.

19.4.4. Helicopter Transmission

19.4.4.1. General information

Transmission of the helicopter is intended for transfer of power of engines to MR and TR with the necessary frequencies of rotation corresponding to the most favorable operating conditions of rotors and to the units installed on MGB.

The main units of transmission (Fig. 19.36) are:

– main gearbox VR-14 2;
– intermediate gearbox 11;
– tail gearbox 9;
– tail shaft of transmission 8, 12;
– fan drive shaft;
– main rotor brake 3;
– control devices.

19.4.4.2. Intermediate Gearbox

IGB is intended to change the direction of transmission tail shaft axis on an angle of 45 ° according to the direction of trailer beam bend without change of rotation frequency. It is reached by application of couple of conic cogwheels with identical quantity of teeth in a gearbox.

Specification:

Gear ratio..... 1:1
Direction of rotation (a look from MGB).....left
Nominal speed of shaft, rpm..... 2594
Oil temperature:
– minimum for start without heating
(oil is diluted), °C..... 30
– the most admissible, °C..... +110
Principle of lubricantion..... splash
Dry mass of a gearbox, kg..... 23.9... 24.9

Primary Elements of Gearbox (Fig. 19.37):

– case 11;
– assembly of the drive gear wheel 3... 9;
– assembly of the driven gear wheel.

Besides, the gearbox is equipped by:

– oil-level gage 15;
– breather 10;
– magnetic plug with the gate-type valve;

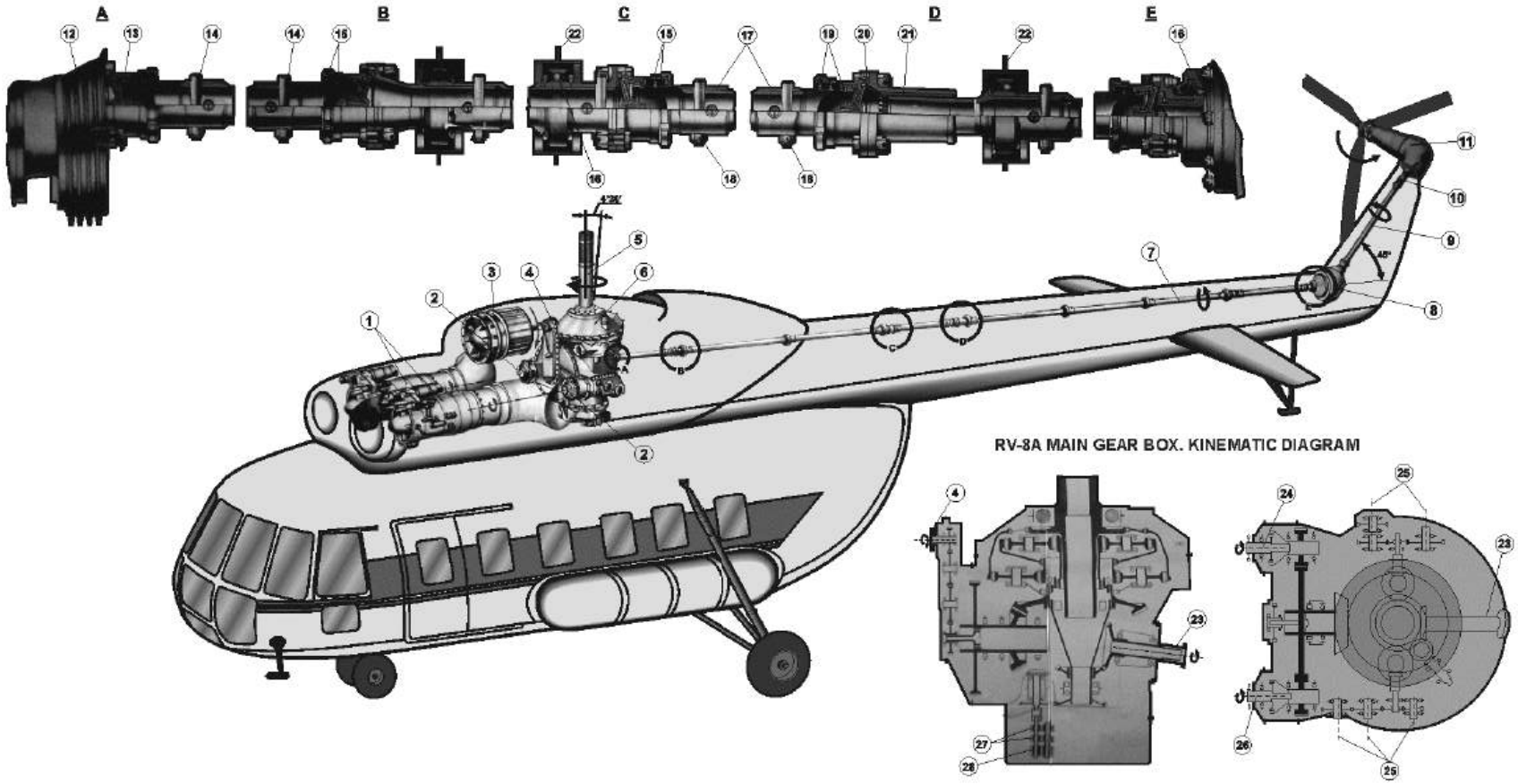


Fig. 19.36. Transmission of Helicopter Mi-8:
1 – gas-turbine engine; 2 – free turbine drive shaft; 3 – fan; 4 – fan drive; 5 – main rotor shaft;
6 – gearbox VR-8A; 7 – intermediate shaft; 8 – intermediate gearbox; 9 – anti-torque rotor shaft;
10 – drive shaft of tail gearbox; 11 – anti-torque rotor gearbox; 12 – brake drum; 13 – spline cup;
14 – hexagon fit bolt; 15 – rubber ring; 16 – spherical bearing; 17 – clutch neck; 18 – nut; 19 – semi-ring; 20 – flange of clutch;
21 – splined end lug; 22 – shaft support; 23 – anti-torque rotor drive; 24 – input of right engine; 25 – accessory drives of helicopter; 26 – input of left engine; 27 – oil suction pump; 28 – oil supply pump

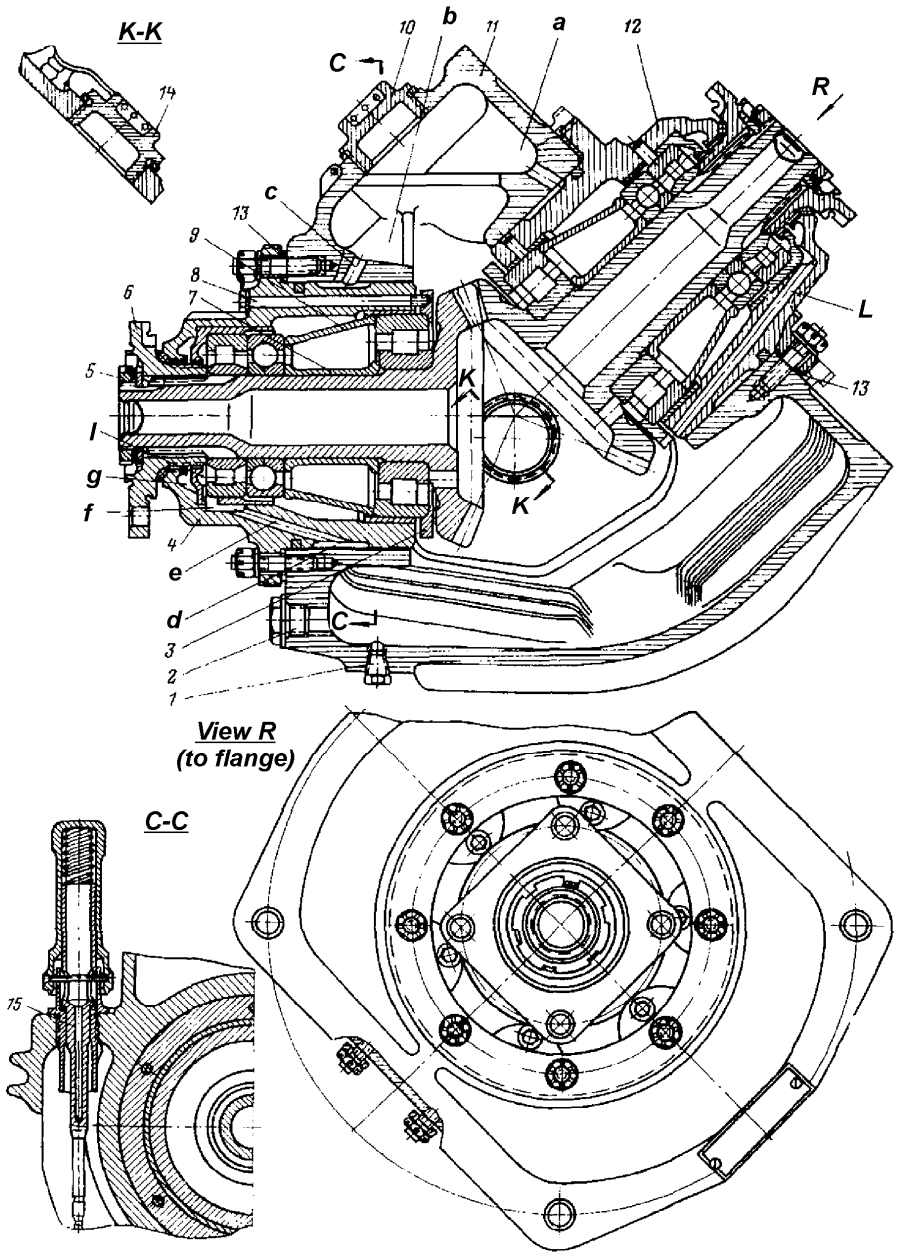


Fig. 19.37. Intermediate Gearbox:

1 – plug; 2 – cap; 3 – flange of fastening of the roller bearing; 4 – driver gear; 5 – nut; 6 – multiple-splined flange; 7 – internal distance sleeve; 8 – bolt; 9 – external distance sleeve; 10 – breather; 11 – case; 12 – cup of the driven cogwheel; 13 – adjusting ring; 14 – plug; 15 – oil gauge; a, b – pockets; c, e, l – channels; d – ring bore; f – ring cavity; g –interlabyrinth cavity; i – cavity of bearings.

– oil-level gage;

– temperature probe P-1T.

The gearbox is fixed by bolts to Fr. 3 of trailer beam.

Excessive pressure from a case is relieved through the breather's opening located in his top part (at the left). In a case there is an opening for a magnetic plug. Oil is filled in a gearbox via the union of the oil-level gage. Control of gearbox filling with oil is conducted on oil-level gage with notches "V" and "N" and by means of the probe.

19.4.4.3. Tail Gearbox

HR is intended for transfer of rotation from a tail shaft to TR with the set rotation frequency.

Specification:

Gear ratio..... 19/44 = 0.4318 or 1:2.316
Direction of rotation of
– the driving shaft (from the drive's view)..... left
– the driven shaft (if to take a detached view TR).....right
Nominal speed, rpm
– driving shaft..... 2594
– TR shaft..... 1120
Temperature of oil, °C
– minimum for start without heating (oil is diluted).....-30
– maximum admissible..... +110
Principle of lubrication..... splash
Full speed of rod, mm..... 67.75... 69.15
Dry weight, kg..... 57.9... 59.2

Primary Elements of Gearbox (see Fig. 19.38):

– case U;
– cover of case 14;
– assembly of the leading gear wheel with shaft in support 2... 10:
– assembly of the driven wheel with shaft in support 11... 13, 16... 21;
– assembly of rod 15 with sprocket 26, screw couple and a splined coupling 22.

Besides, TGB is equipped:
– two oil-level gages;
– two magnetic plugs with gate-type valves;
– breather;
– two openings under P-1T (one is involved).

The gearbox is fixed by bolts to Fr. 9 of trailer beam.

Transfer of power to TR is carried out by couple of conic wheels which angle between axes of rotation equal to 90 °.

In a gearbox case except conic wheels the assembly of TR variable pitch control link is placed. In the driven shaft a rod 15 is placed by means of which TR change pitch. Progress of a rod is carried out by means of the actuating screw 25 which rotates from sprocket 26 with screw female fitting.

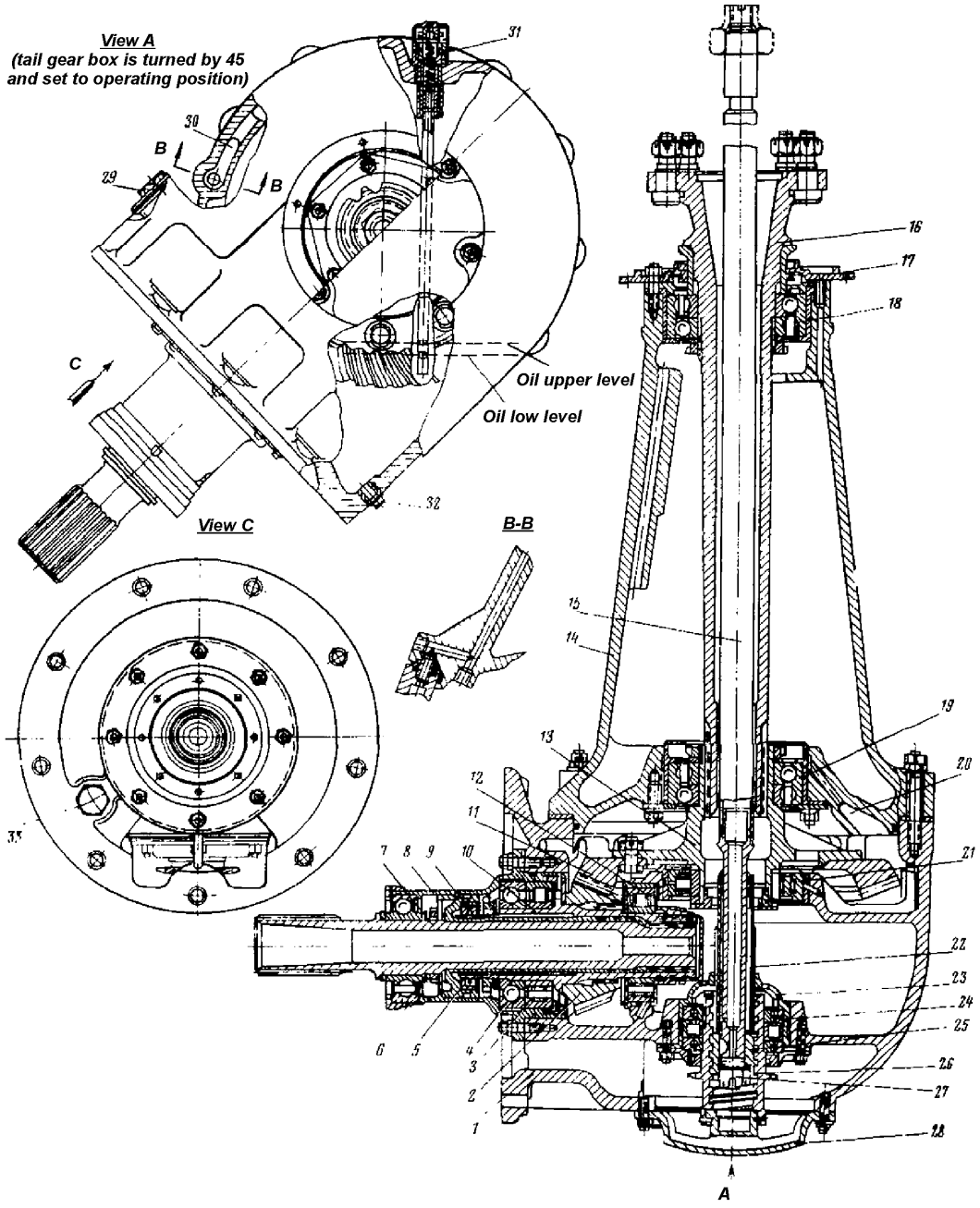


Fig. 19.38. Tail Gearbox:

1 – case; 2 – leading cogwheel; 3 – roller bearing; 4 – ball bearing; 5 – sealing sleeve; 6 – ball bearing with bilateral seal; 7 – leading shaft; 5 – cup of bearings of the leading wheel; 9 – the rubber reinforced cuff; 10 – plug of the leading cogwheel; 11 – driven cogwheel; 12 – the roller bearing 13 – hub group of the driven cogwheel; 14 – case cover; 15 – rod of control of tail rotor; 16 – driven shaft; 17 – cover of labyrinth seal; 18, 19 – two-row ball bearing; 21 – roller bearing; 22 – splined sleeve; 23 – cup of rod bearings; 24 – ball bearing; 25 – worm gear of rod; 26 – case of sprocket of tail rotor control; 27 – nut of fastening of rod worm gear; 28 – cover; 29 – plug; 30 – oil catcher pocket; 31 – oil-level gage; 32 – drain plug; 33 – opening cap for oil thermometer sensor

Filling with oil – through the breather's opening, control of filling – on oil measure device with notches "V" and "N".

Existence of two magnetic plugs and two the oil measure device provides application of TGB and on other types of helicopters (with the pushing TR).

Note. FGB and TGB of the MI-8MTV helicopters can be equipped with the oil chip signaling devices PS-1 closed on an annunciator "CHIPS. MGB" on the CPU.

19.4.4.4. Tail Shaft of Transmission

The tail shaft (TS) of transmission is intended for torque transmission from MGB through FGB and TGB to TR.

TS has horizontal 12 and inclined 8 sites and consists of six portions: two rigid (front and back) and four hinged ones.

All parts of the shaft are the pipes made of the alloyed steel 40HN2MA-Sh.

Rigid parts

The front portion is installed in a fuselage superstructure, consists of two pipes tightly pressed on conical bolts and rotates on three supports.

Back one is installed in TB, consists of three pipes and rotates on four supports 14. Supports have the closed ball-bearings 13 which are pressed-in fittings of the fuselage and are filled by NK-50 lubricant on all service life of a shaft.

Hinged parts

Are installed at MGB, in the place of TB-to-fuselage joint, near FGB and in TB.

Parts of a shaft are connected by means of hinges with rubber rings 5 and flanges. Cavities of hinges are filled with oil for the hypoid gear in volume of 25... 35 ml.

Middle and trailer hinged parts have the mobile splined connections intended for compensation of deviations in linear sizes of the fuselage, TB and TR and to provide shaft change of length in the process of beam bending in flight. These connections are greased with the NK-50 lubricant.

The mass of the shaft is 56...58 kg. The shaft break in the splined couplings shouldn't exceed 1.2 mm on the measuring instrument indicator. The runout in any aperture has to be no more than 0.445 mm.

19.4.4.5. Fan Driveshaft

The fan drive shaft (Fig. 19.39) is connected to the fan shaft with a forward splined fork 1, and to MGB shaft – with a back flange fork.

The shaft is supplied with two universal cardan hinges on needle bearings 4 compensating assembly distortions.

Specification:

Nominal speed, rpm.....6000
Maximum torque, kgm.....12
Oil for lubrication of hinges.....TS gip
Outer diameter of shaft, mm..... 28
Mass of shaft, kg.....3.24... 3.44

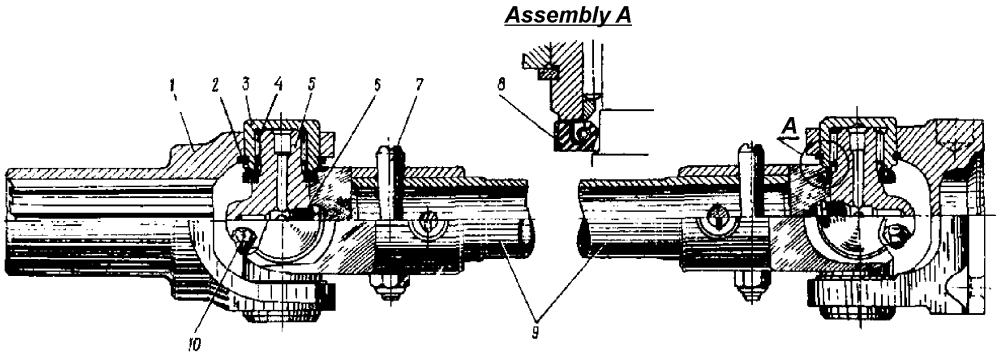


Fig. 19.39. Fan Driveshaft:

1 – splined fork; 2 – ring; 5 – external holder of bearing; 4 – needle bearing; 5 – crosspiece; 6 – relieve valve; 7 – conical bolt; 8 – rubber cuffs; 9 – shaft; 10 – oil fitting

19.4.4.6. Main Rotor Brake

MR brake (Fig. 19.40) is intended for reduction of time of MR stop after cut off of engines, and for latching of transmission in parking of the helicopter.

The brake is of drum type, plate type of brake, with mechanical control, is mounted on the TR drive case on MGB.

Basic elements are:

– bracket 10 (brake case);

– shoe 7 with facings 8 of "ferodo";

– expanding lever 9;

– dwang 5;

– drum 1.

The gap between shoes and a drum should be 0.2... 0.5 mm.

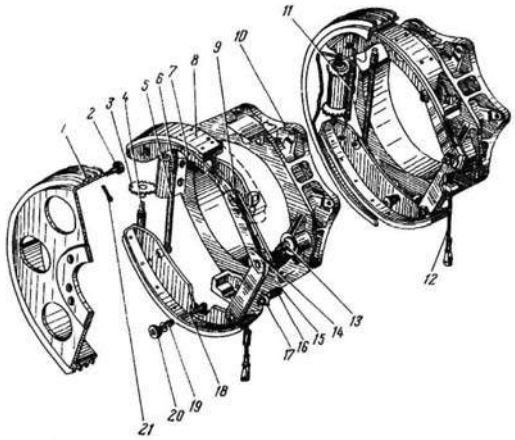


Fig. 19.40. Main Rotor Brake:

1 – brake drum; 2 – screw of brake expanding lever; 3 – hand crank of brake adjustment; 4 – adjust screw; 5 – dwang; 6 – core spring; 7 – brake shoe; 8 – frictional facing; 9 – expanding lever; 10 – brake bracket; 11 – coupling spring; 12 – cable of brake control; 13 – thrust stud of shoes; 14 – brake shoe link; 15 – screw fastening link to shoe; 16 – bushing; 17 – figured washer; 18 – core of clamping cup; 19 – core spring; 20 – cup; 21 – cotterpin; 22 – spring of brake control cable

19.5. LANDING GEAR

19.5.1. General Information

Alighting gear (AG) is designed to absorb shock loads during helicopter landing and to move the helicopter on the ground during taxing and take-off.

Alighting gear (Fig. 19.41) includes:

– wheel landing gear (non-retractable in flight), equipped with two main landing gears (MLG) 2 and nose landing gear (NLG) 7;

– tail bumper 3.

Specification:

Parameter	NLG	MLG	TB
Wheel type	К2-116	КТ-97/3	
Tire dimensions, mm	595x185	865x280	
Initial pressure in tire, kg/cm^2	4.5+0.5	5.5+0.5	
AMG-10 volume, cm^3	2080	3510	300
including:			
– in low-pressure shock-absorber	1110		
– in high-pressure shock-absorber	2400		
Initial pressure of nitrogen in shock-absorbers, kg/cm^2	32±5 0.5		27+1
including:			
– in low-pressure shock-absorber	26±0.5		
– in high-pressure shock-absorber	60±0.5		
Full damper piston rod stroke, mm	165		200
including:			
– low-pressure shock-absorber			120±2
– high-pressure shock-absorber			240-±2
Piston rod extension, m			
including:			
– at G_{trld} = 0			240
– at $G_{take\ off}$ = 1110 kg			90±20
– at $G_{take\ off}$ = 13000kg	≤150		68±20
Tire deflection, mm	≤43		≤68

19.5.2. Nose Landing Gear

NLG (Fig. 19.42) is a beam/braced type landing gear provided with the castoring wheel levered suspension providing the best operating conditions of the shock-absorber when taxing on uneven surface and the built-in shock absorber.

It consists of the following members:

– absorber levered suspension;

– forked welded strut;

– two wheels without brakes.

Attached to Frs 1 and 2 of fuselage center section.

19.5.2.1. Shock Absorber Strut

Primary members are:

– shock-absorber;

– swivel 15;

– piston rod 21;

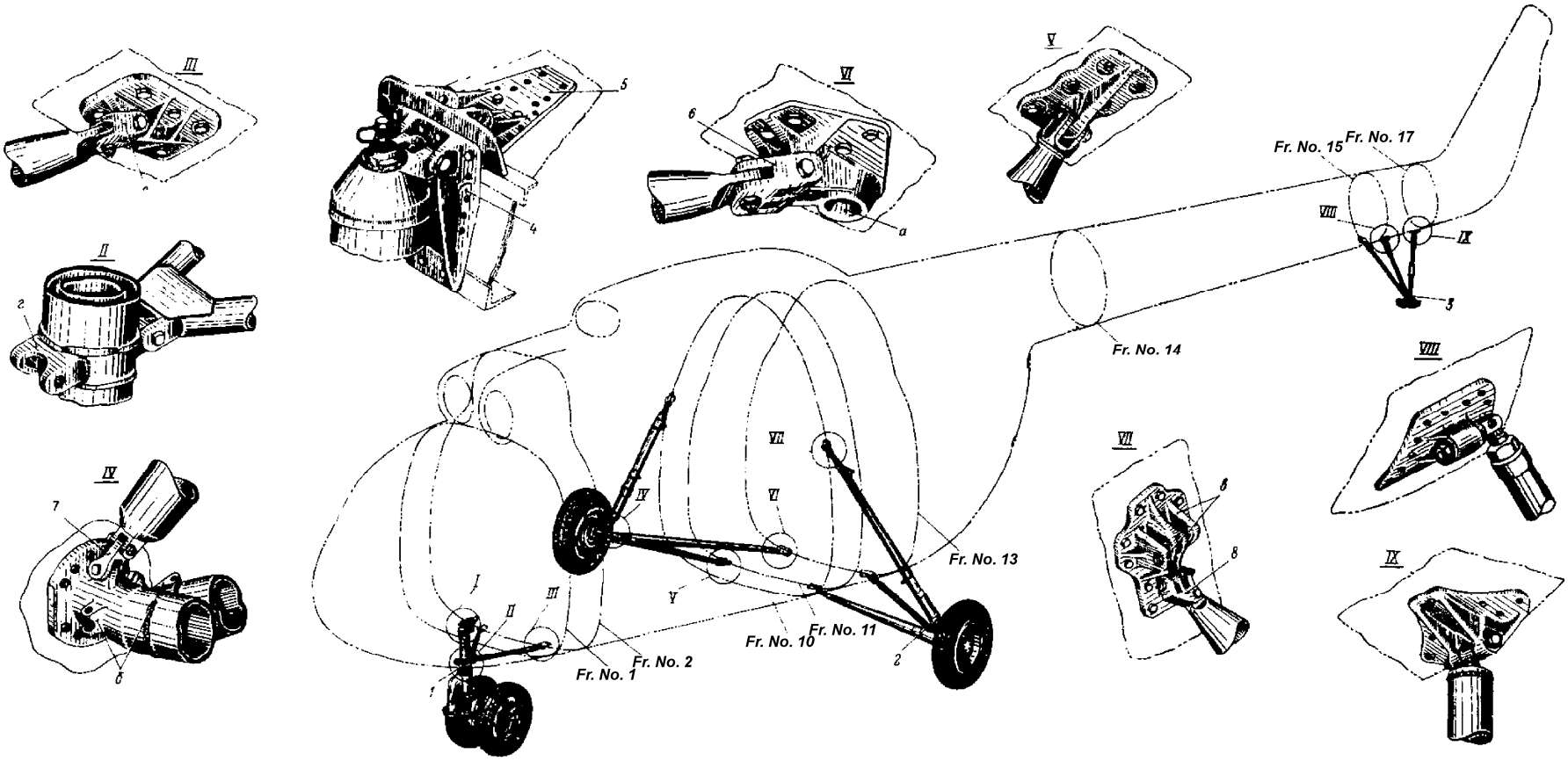


Fig. 19.41. Alighting Gear:

1 – forward leg of landing gear; 2 – main leg of landing gear; 3 – tail bumper; 4 – bracket for fastening of forward shock strut; 5 – fitting; 6, 7, 8, 9 – universal link; I – shock strut-to-forward leg attachment fitting; II – fork strut-to-shock strut attachment fitting; III – fork strut-to-fuselage attachment fitting; IV – shock-absorber-to-axle-arm of MLG attachment fitting; V – axle-arm of MLG-to-fuselage attachment fitting; VI – MLG strut-to- fuselage attachment fitting; VII – MLG shock-absorber-to-fuselage attachment fitting; VIII – tail bumper strut-to- tail beam attachment fitting; IX – shock-absorber of tail bumper-to-tail beam attachment fitting; a – spherical seat for jack support; b – eyes for towing; c – eyes for mooring device and fastening of back upper strut of special installation truss; d – eyes for mooring device

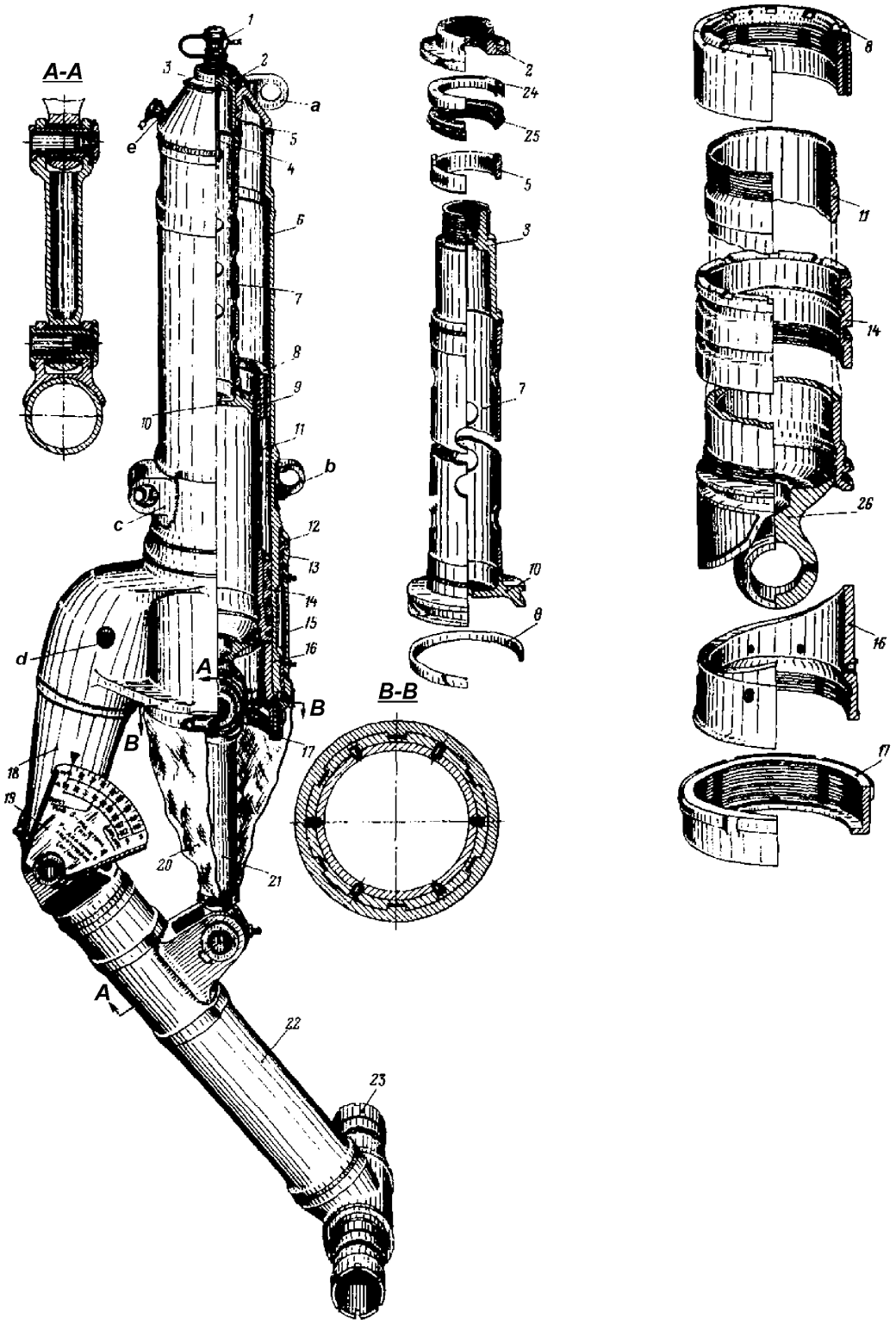


Fig. 19.42. Lever Shock Strut:

1 – charging valve; 2 – nut of fastening of plunger; 3 – plunger shaft; 4 – charging tube; 5 – lock plug; 6 – cylinder; 7 – plunger; 8 – top axle box; 9 – piston ring; 10 – plunger piston; 11 – rod with top clamp; 12 – thrust ring; 13 – plug; 14 – lower axle box; 15 – rotary bracket; 16 – lower clamp; 17 – nut; 18 – arm of rotary bracket; 19 – indicator of pressure in cylinder and shock strut compression value; 20 – cover; 21 – rod; 22 – lever; 23 – wheel fastening nut; 24, 25 – friction rings; 26 – rod head; a – top eyes for shock strut attachment; b – lower eyes for fork strut-to-shock strut attachment; c – eye for mooring device; d – bushing for tug; e – drain adapter

– lever with axis of wheels 22.

The primary members of the shock-absorber:

– cylinder 6;

– piston rod 11;

– inner cylinder 7 with a diffuser;

– upper 8 and lower 14 axle-bearings;

– wheel steering mechanism.

When landing the helicopter, the force acting on wheels is transmitted through the lever of a wheel suspension and a connecting rod to the shock-absorber rod which strokes forward.

The fluid from the lower hollow of a rod is forced up in the upper hollow of the cylinder of the shock-absorber by a plunger piston, compressing nitrogen, and further to the hollow between the rod and the cylinder. At the same time energy dissipation emerges due to friction in axle boxes, compression of nitrogen, and liquid deceleration through the central calibration orifice in the diffuser and longitudinal openings in the upper axle box.

At reverse motion the nitrogen compressed during a forward stroke pushes out the liquid from a cavity between the rod and cylinder, and further from the upper hollow of the cylinder into the lower cavity of a rod.

Braking of the rod is achieved by reduction of the total flow passage for liquid by means of the so-called ring snubber valve which is forced against the upper axle box by the fluid stream, blocking a part of openings.

According to the scale of non-loaded helicopter compressing of the shock-absorber rod has to be within 55...75 mm, if helicopter weight is 11100...11200 kg – 120...140 mm.

The **steering wheel mechanism** is intended to turn forward wheels to the flight line at the unloaded forward shock strut.

It consists of a cam plunger pin, the counterparts of which are mounted below on the internal surface of the cylinder (lower clamp 16) and on external surface of the rod correspondingly.

When load acts on forward wheels, the rod of the shock-absorber moves up and cams of the steering mechanism are separated.

When forward wheels are unloaded, the rod of the shock-absorber moves down under gas pressure and the top cam contacts with the lower cam of the mechanism, steering thereby forward wheels to the flight line.

19.5.2.2. Wheel K2-P6

The wheel (Fig. 19.43) includes:

– hub 6;

– two half-flanges 5;

– two protective shields 7, 9;

– chamber air tire.

Besides that, during installation of wheel on the axle-arm the following is mounted:

– snap ring 13;

– scraper ring 10;

– roller bearer;

– distance sleeve 11;

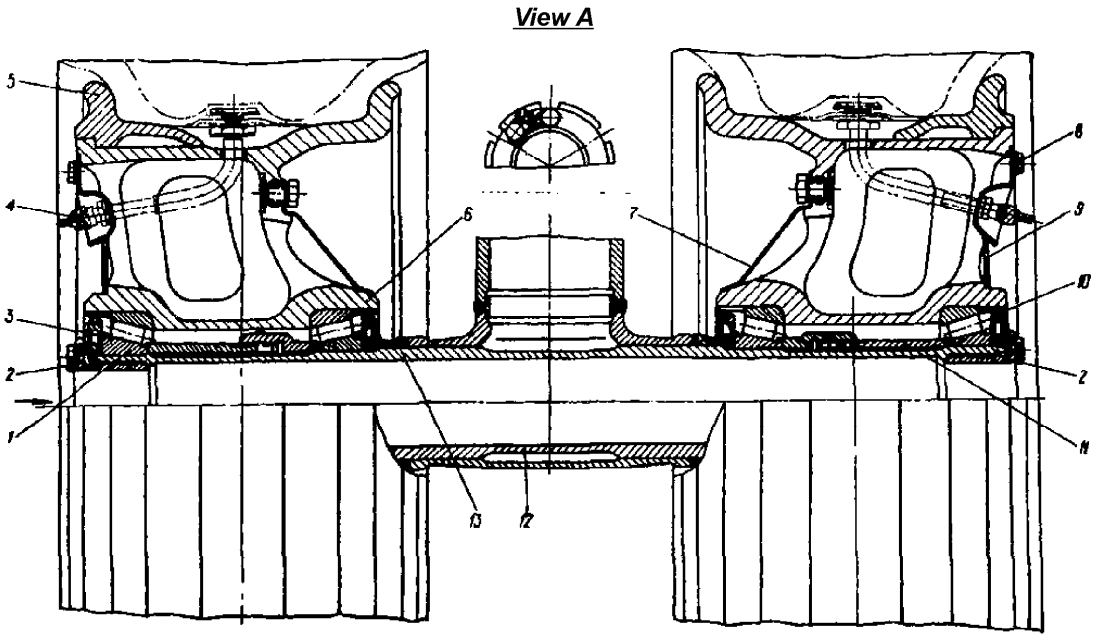


Fig. 19.43. Wheel Attachment of Nose Landing Gear Leg:
1 – roller bearing; 2 – wheel fastening nut; 3 – cover; 4 – valve; 5 – semi-flange; 6 – hub; 7 – shield; 8 – shield fastening bolt; 9 – shield; 10 – felt rings; 11 – distance sleeve; 12 – axis of wheels; 13 – friction ring

– roller bearer 1;
– scraper ring;
– nut 2 with lock screws.

Bearings are filled with graphite lubricant NK-50 (Black sea roach).

Attention!

All parts of wheels are marked. Dekitting and use of parts of other wheels is forbidden!

19.5.3. Main Landing Gear

Main landing gear includes:
– twofold shock-absorber;
– axle-arm;
– brace strut;
– wheel;
– fairing.

Struts are attached to Frs 10 and 11 of fuselage central part.

19.5.3.1. Shock Absorber

Shock-absorber (Fig. 19.44) is intended to absorb shock kinetic energy emitted from the helicopter on the ground during landing time, and unlike the main landing gear, the shock-absorber participates in dissipation of cross fluctuations like "terrestrial resonance".

The method of shock energy dissipation in the shock-absorber is similar to the described one above, however there is a number of features:

1. A twofold shock-absorber (in literature – dual-chamber), consists of low-pressure and high-pressure shock-absorbers, which work coaxially.

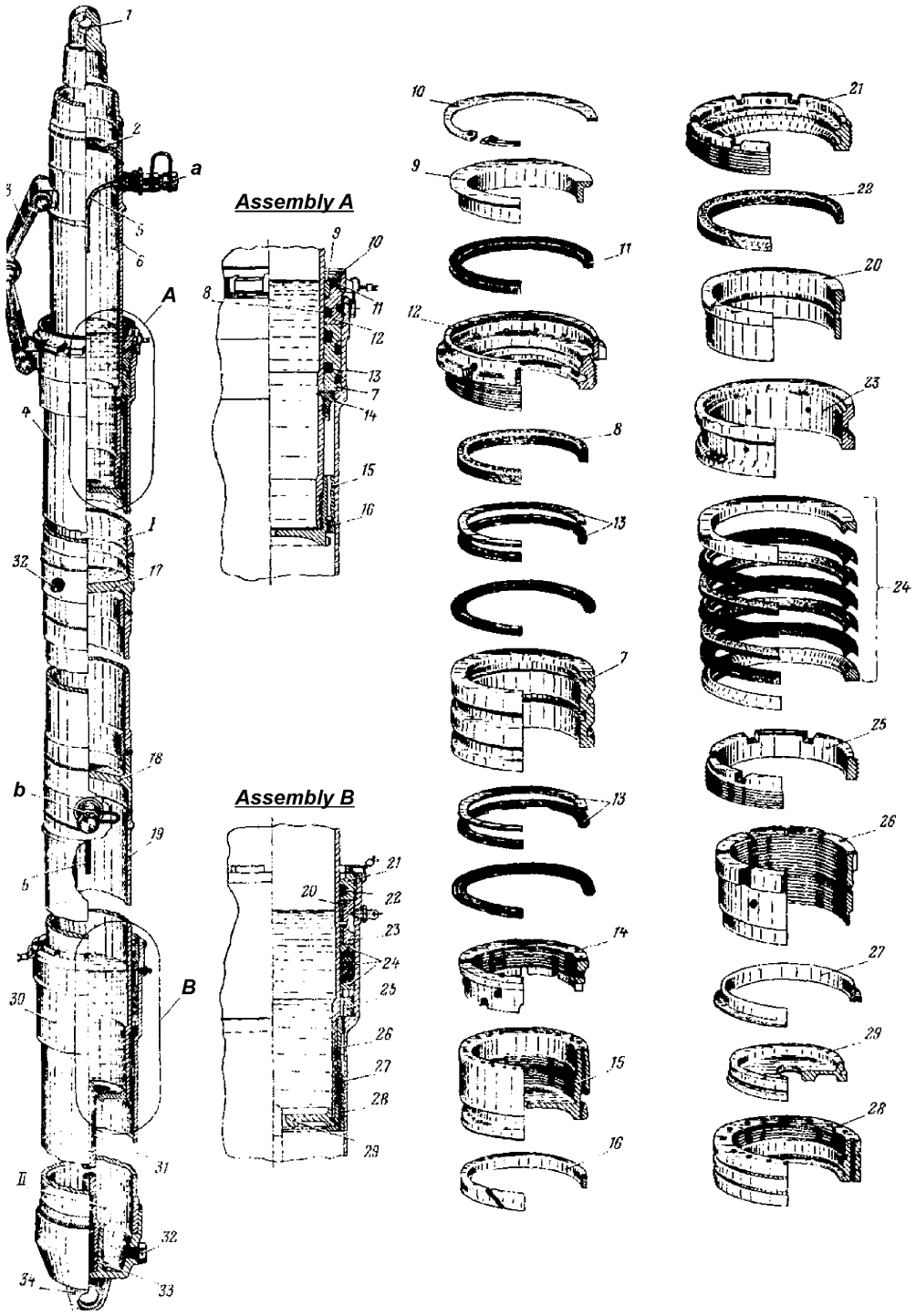


Fig. 19.44. Double-Chamber Liquid-Gas Shock Absorber of Landing Gear Main Leg:

I – low-pressure chamber; II – high-pressure chamber; 1 – lug tip for fastening to fuselage assembly; 2 – adapter with bottom; 3 – torsion links; 4 – cylinder of low-pressure chamber; 5 – liquid filling restriction tube; 6 – rod of low-pressure chamber; 7 – axle-box of low-pressure chamber; 8 – felt ring; 9 – bushing; 10 – expanding locking ring; 11 – buffer rubber ring; 12 – nut; 13 – sealing rings; 14 – nut; 15 – lower axle-box of the low-pressure chamber; 16 – valve (ring); 17, 18 – adapters with bottom; 19 – rod of high-pressure chamber; 20 – plug; 21 – nut; 22 – felt ring; 23 – upper axle-box of high-pressure chamber; 24 – sealing (filler) rings; 25 – nut; 26 – thrust bushing; 27 – valve (ring); 28 – lower axle-box of high-pressure chamber; 29 – diffuser; 30 – cylinder of high-pressure chamber; 31 – metering pin; 32 – stopper of drain adapter; 33 – locking pin; 34 –fork tip for fastening to axle-arm; a, b – charging valves

2. Low-pressure shock-absorbers are situated in the upper part and the high-pressure one – in the lower part.

3. Shock-absorbers ensure the "inverted" scheme (functions of plungers are performed by rods).

4. The rod of high-pressure shock-absorber goes to the cylinder of the low-pressure shock-absorber.

5. Rod and cylinder of low-pressure shock-absorber are connected by torsion link to prevent rotating of the cylinder.

6. Metering pin reducing flow passage of squeezing way is installed in the cylinder of high-pressure shock-absorber.

7. When landing, the low-pressure shock-absorber responses the first, and after full squeezing of its rod responses high-pressure shock-absorber.

8. Reverse motion of the rod of the low-pressure shock-absorber can occur only at bending vibration of the helicopter on the landing gear or during take-off when shock-absorbers are unloaded. The rods of the low-pressure shock-absorber have to be squeezed out during parking.

9. A microswitch VM-800K is fixed on a rod of the low-pressure shock-absorber over torsion link. It is designed to turn on the MS-61 tape recorder and system of a hydraulic thrust roller in longitudinal control channel. (The gap between the upper shoulder of tappet lever and rod of microswitch with the compressed strut has to be within 1±0.5 mm).

19.5.3.2. Brace and Axle-Arm

The brace and axle-arm are of a welded design, made of steel pipes (grade of steel for primary elements of all struts is 30HGSA).

The brace is used as a cylinder for compressed air. It is equipped with charging fitting from the pneumatic system, pumping of wheel tires and condensate draining. The brace has attachment fittings to the fuselage and axle-box on its ends.

The axle-arm has eyes to be fastened to the fuselage, brace and towing cables. A brake flange is fixed on the axle-box, and the spherical seat for a jack is welded below.

19.5.3.3. Wheel KT-97/3.

Each of wheels (Fig. 19.45) of the main landing gear leg consists of such elements:

– hub;
– detachable flange;
– tire inner tube;
– braking device.

Besides that, each wheel is mounted on an axle-arm by:

– two roller bearers;
– distance sleeve;
– two carburetors;
– wheel fastening nuts with lock screws.

The **hub** of the wheel – a single part with a rim, a hub group and one flange (the second – detachable) – is cast of Mg-alloy.

A **tire** is mounted on a cylinder rim and consists of the inner tube with a stop air valve and casing.

A **brake device** (drum type) with a pneumatic actuator and includes:

– brake body 9 (attached to the axle flange);
– two brake shoes 13 on anchor rollers 20;
– two release springs 15;
– two expanding levers 14 with cores and adjusting hand wheels;
– two air cylinders 12;
– guard with openings A for measurement of the clearance;
– brake drum 10.

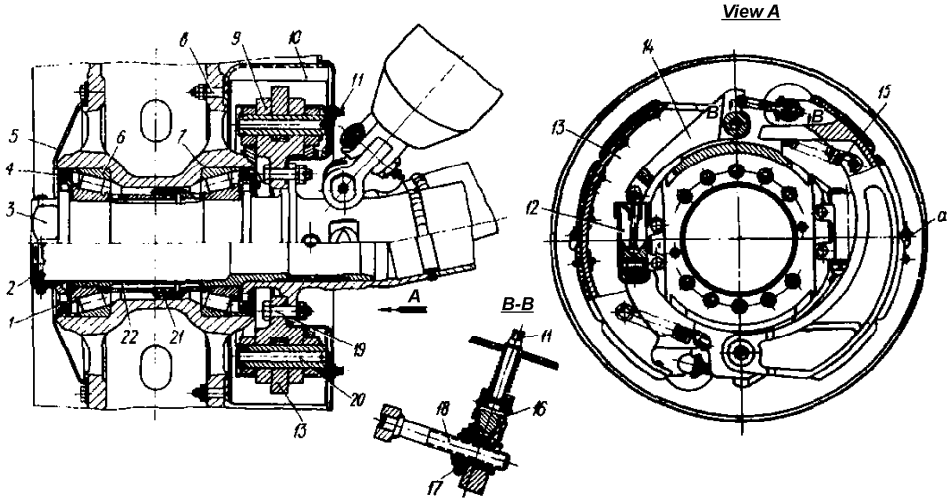


Fig. 19.45. Main Landing Gear Leg Wheel Attachment Fitting:

1 – cover; 2 – safety bolt; 3 – wheel fastening nut; 4 – dustproof felt ring; 5 – guard; 6 – roller bearing; 7 – wheel drum; 8 – bolts of fastening of brake drum; 9 – brake case; 10 – brake drum; 11 – regulating roller; 12 – braking air cylinder; 13 – brake shoe; 14 – lever; 15 – pull-off spring: 16, 17 – conical gear; 18 – adjusting screw; 19 – brake fastening bolt; 20 – anchor roller; 21 – distance sleeve; 22 – axis; a – hole for clearance measurement

19.5.3.4. Fairing

The fairing provides necessary aerodynamic form of the lower part of MLG between an axle-arm and a brace. It is made of duralumin sheets, angular sections and polyfoam inserts.

The fairing is fixed to an axle-arm and a strut with collars. There is a cover on the fairing from above. It is used to get access to pipelines of air system. There is a placard "DO NOT STEP" on the cover.

19.5.4. Tail Bumper

Tail bumper (TS) (Fig. 19.46) is intended to protect tail rotor blades from damage and tail beam from destruction when landing the helicopter with a big angle of a pitch-up and overload.

Tail bumper consists of:
– shock absorber 32;
– two brace struts 31;
– bumper 30;

The liquid-gas shock absorber has a reverse scheme and consists of the cylinder and the rod.

The principle of operation and structure of the shock absorber are similar to the one described above.

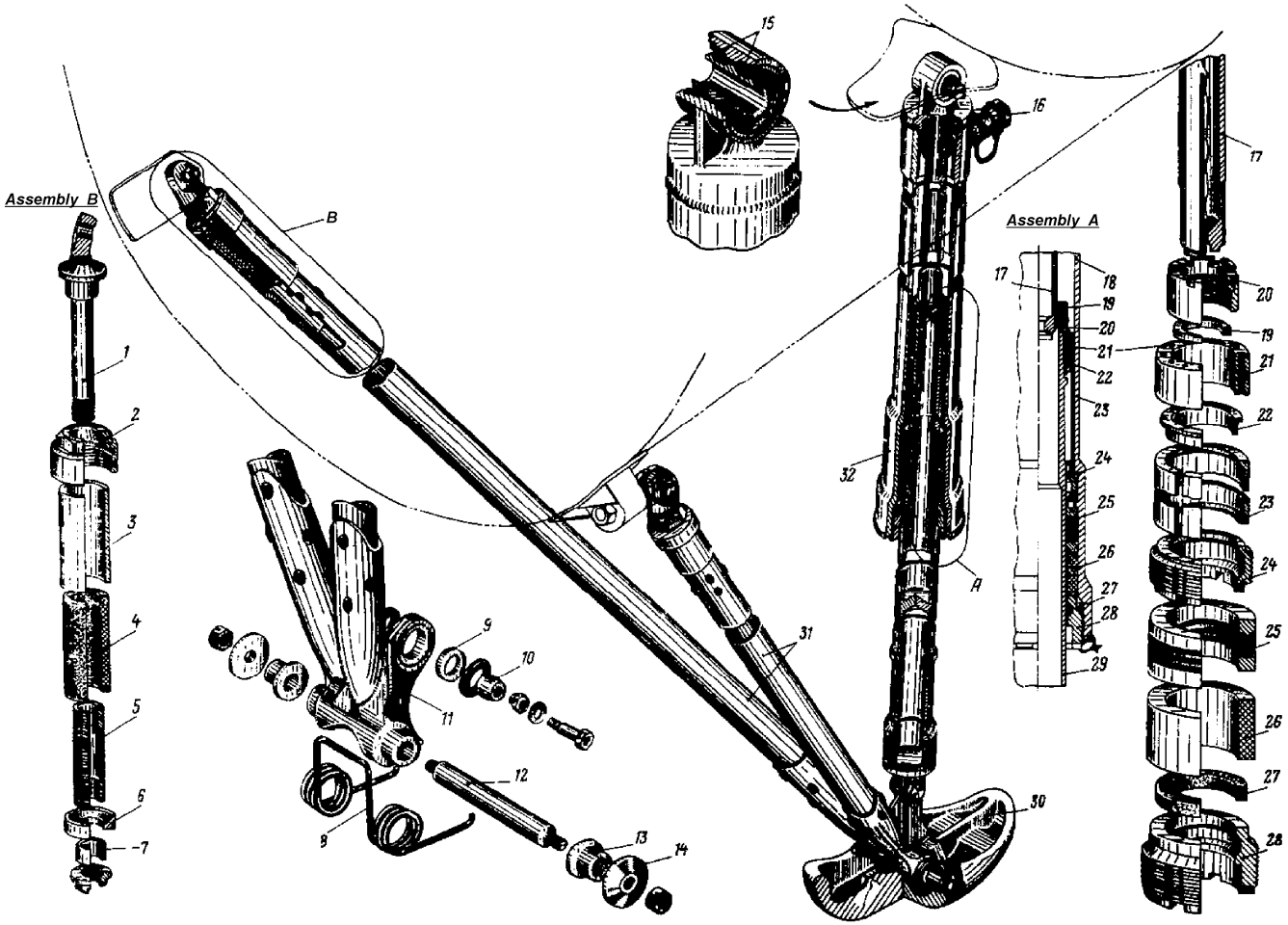


Fig. 19.46. Tail Bumper Structure:

1 – lug bolt; 2 – nut; 3 – external holder; 4 – damping rubber plug; 5 – internal holder; 6 – ring; 7 – bushing; 8 – spring; 9, 13 – bushings; 10 – ball-and-socket bearing; 11 – fork fitting; 12 – roller; 14 – washer; 15 – damping rubber rings; 16 – charging valve; 17 – plunger; 18 – cylinder; 19 – piston ring; 20 – nut; 21 – top axle box; 22 – valve; 23 – bushing; 24 – nut; 25 – filler ring; 26 – lower textolite axle box; 27 – felt ring; 28 – nut; 29 – rod; 30 – bumper; 31 – brace struts; 32 – shock absorber

Brace struts are made of duralumin pipes and provided with steel tips.

The bumper is stamped of Al-alloy AK-6, and fastens to a fork joint 11 by a stud 12 on bushings 9, 13. It is set to avoid digging into soil during ground contact.

The rod of the shock absorber and braces are attached to the tail beam ribs by dampers which are steel holders, between which rubber plugs 75 are cured in.

19.5.5. Landing Gear Operation

19.5.5.1. Checking of Helicopter Conditions Before Flight

Preflight Survey of Helicopter

During survey the flight mechanic must:

– make sure of correct charging of a forward shock strut wheel tires; compression of NLG rod (according to the index scale) at various combination of weight and centering of the helicopter shouldn't exceed 150 mm, compression of tires of the nose wheels shouldn't exceed 43 mm;

– make sure of correct charging of the right MLG: whether there is no leak in the rod, in correctness of charging wheel tires (a rod exit on a mirror). Depending on helicopter weight it has to be: if no assault aircraft load – 240 mm, with helicopter weight of 11100 kg – 90±20 mm, with helicopter weight of 13000 kg – 68±20 mm; tire compression – no more than 68 mm;

– check a condition of joints fastening to the tail bumper;

– inspect the left main landing gear in the sequence described above.

At external inspection, the captain of the crew must check:

– landing gear – conditions of the shock strut (whether the rods are polluted or there is any working fluid dribble);

– correct charging of the shock strut, state and compression of wheels tires.

19.6. HELICOPTER CONTROL SYSTEM

19.6.1. General Information

Control is a process of changing of forces and moments applied to the helicopter in time.

Helicopter control relative to three axes is performed by change of force intensity and direction of thrust of the main rotor and thrust force change of the tail rotor.

These force changes take place at turn of blades relatively to feathering hinge and at change of the torque applied to rotors.

Thus, the control system (Fig. 19.47) is a system of kinematic connections of the pilot with feathering hinges of blades and throttle devices of engine.

Direct effect on feathering hinge of main rotor blades is carried out by the swashplate, to which three control channels are attached.

Helicopter control system includes:

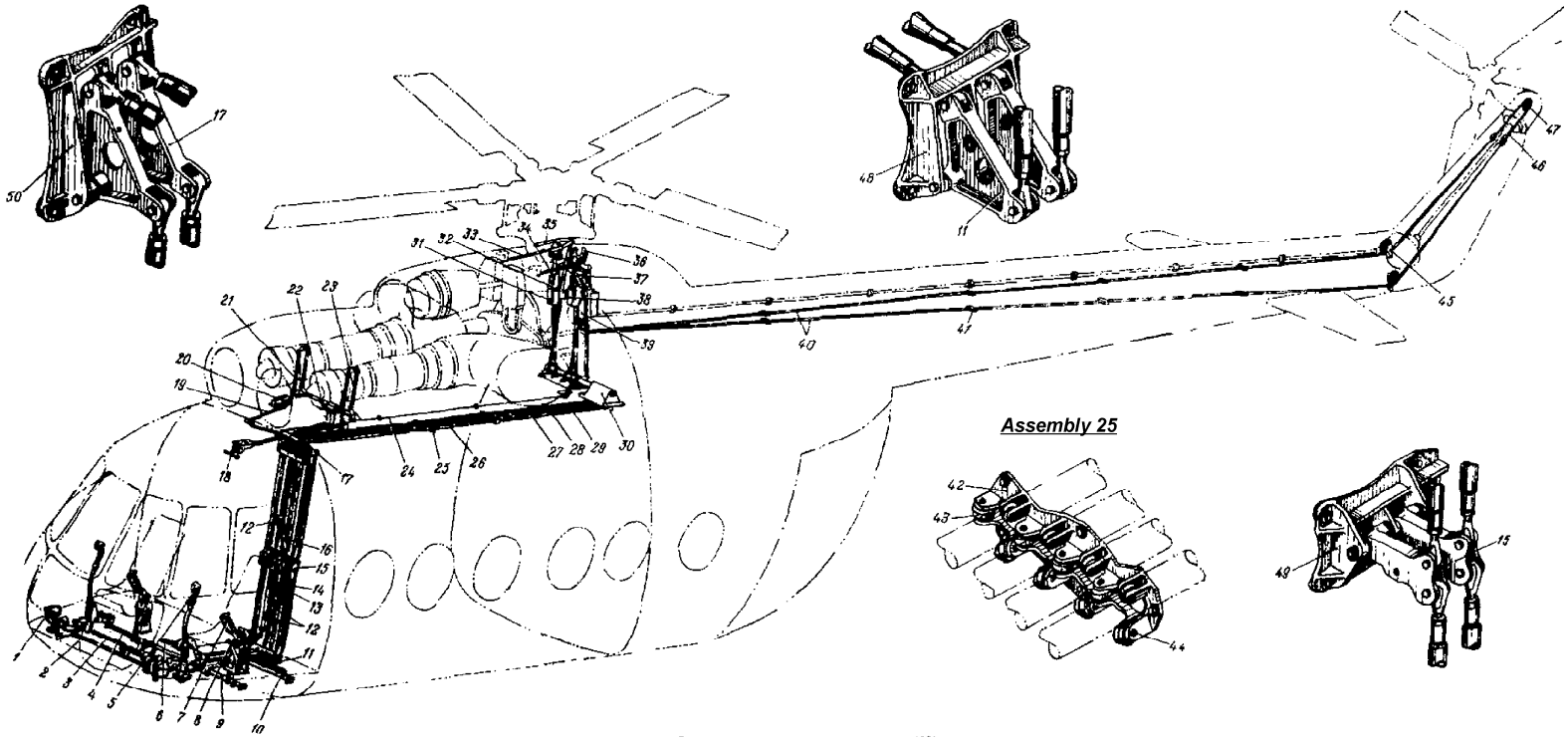


Fig. 19.47. Helicopter Control Chart:

1 – rudder control pedals; 2 – down-lock actuating rod connecting arms of pedals; 3 – lateral control rod; 4 – longitudinal control rod; 5 – cyclic pitch control stick; 6 – control handle of main rotor brake; 7 – collective pitch control stick; 8 – separate engine throttle control levers; 9 – locking shaft; 10 – differential joint; 11 – lower angular arms; 12 – electromagnetic brakes of EMT-2M; 13 – feel spring of lateral control; 14 – feel spring of longitudinal control; 15 – intermediate arms; 16 – feel spring of directional control; 17 – upper angular arms; 18 – engine shutdown handle; 19 – engine shutdown control cables; 20 – bracket with rollers; 21 – actuating rod of engine shutdown; 22 – shaft block; 23 – actuating rod of engine control; 24 – control cable of main rotor brake; 25 – roller guides; 26 – actuating rod of aileron control; 27 – actuating rod of main rotor collective pitch control; 28 – actuating rod of rudder control; 29 – actuating rod of longitudinal control; 30 – unit of longitudinal, aileron, rudder and collective pitch control; 31 – hydraulic booster of aileron control; 32 – hydraulic booster of collective pitch control; 33 – actuating rod of aileron control; 34 – actuating rod of longitudinal control; 35 – lever of the swashplate of main rotor collective pitch control; 36 – quadrant; 37 – bracket fastening the hydraulic boosters; 38 – hydraulic booster of rudder control; 39 – hydraulic booster of longitudinal control; 40 – control cables of tail rotor; 41 – textolite cables guides; 42 – bracket; 43 – roller; 44 – eccentric axis; 45 – rollers; 46 – bush roller sequence; 47 – chain gear of tail gearbox; 48 – bracket of lower angular arms; 49 – bracket of intermediate arms; 50 – bracket of upper angular arms.

1. Double cyclic pitch control in which two cyclic pitch control sticks (CPCS) are kinematically connected among themselves and with the swash plate.

2. Double directional control in which pedals are kinematically connected among themselves and with the TR pitch change mechanism.

3. The double joint control of the collective pitch and engines in which two collective pitch levers (CPL) are kinematicallly connected among themselves, and also with the swash plate slide block and levers of throttle cranes of engines pump-controllers.

Besides, these are is installed on the helicopter:

– separate engine control without the collective pitch change in which the levers of separate engines throttle control levers (LSCE) are kinematically connected with shutoff levers of pumps-controllers of engines;

– engine shutdown control in which shutoff levers are kinematicaly connected with shutoff levers on engines pump-controllers;

– MR brake control;

– MR speed change-over control;

– feel spring mechanisms control.

Control in all main channels is dual, push-pull control linkage is not duplicated and is a set of rods and arms. Exception is made for MR (and partially TR) brake control linkage where cables are applied.

19.6.2. Lateral-Longitudinal Control

19.6.2.1. System Structure

Dual lateral-longitudinal control of helicopter is made with push-pull control linkage, power-assisted, combined with the autopilot (AP).

Parts of push-pull control linkage (Fig. 19.48):

– two steering columns (cyclic pitch control sticks) 5;

– two load spring mechanisms 13, 14 with electromagnetic brakes EMT-2M 12;

– longitudinal, lateral, directional control and collective pitch control ("shaft of shaft") 30;

– two combination control hydraulic actuators KAU-3OB 31, 39;

– swash plate;

– elements of push-pull control linkage (system of rods and arms);

– thrust roller system in longitudinal control channel;

19.6.2.2. System Units

Control columns (Fig. 19.48, view A, B) are identical on design which basis is the steel pipe in the case fixed pivotally to a longitudinal beam of the floor panel. From below the case has two levers to which control rods are connected.

On the cyclic pitch control stick are mounted:

– ICS-RADIO button trigger;

– AP shutdown button;

– trigger button;

– control button of electromagnetic brakes of load spring mechanisms (TRIMMER button);

Lower under the stick a wheel brake trigger is provided.

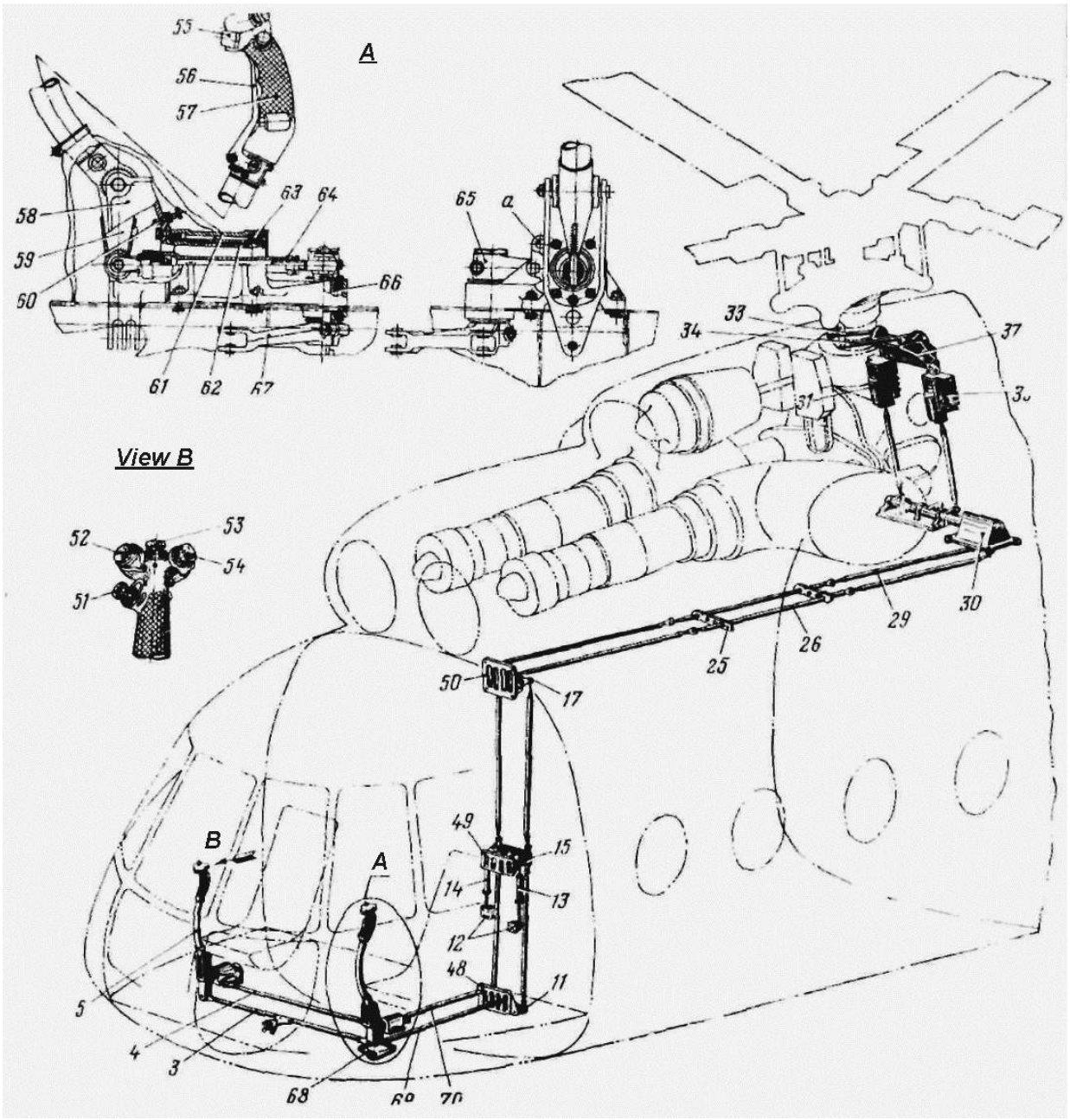


Fig. 19.48. Scheme of Manual Control:

3 – aileron control rod; 4 – longitudinal control rod; 5 – longitudinal-lateral control stick; 11 – lower angular arm; 12 – electromagnetic brakes EMT-2M; 13 – load spring mechanism of aileron control; 14 – load spring mechanism of longitudinal control; 15 – intermediate arm; 17 – top angular arm; 25 – roller guides; 26 – rods of aileron control; 29 – rods of longitudinal control; 30 – unit of longitudinal, lateral, directional and collective pitch control; 31 – hydraulic booster of aileron control; 33 – rod of aileron control; 34 – rod of longitudinal control; 37 – fastening arm of hydraulic boosters; 39 – hydraulic booster of longitudinal control; 48 – arm of lower angular arms; 49 – intermediate arm; 50 – arm of top angular arm; 51 – button (trimmers) of control of electromagnetic brakes of EMT-2M of load spring mechanisms; 52 – ICS button; 53 – radio button; 54 – button of autopilot shutdown; 55 – special purpose button; 56 – lever of wheels braking; 57 – helicopter control handle; 58 – case; 59 – lever; 60 – restrictive bolt; 61 – arm; 62 – cup; 63 – ball-bearing; 64 – hinged rod; 65, 67 – arms; 66 – axis; 68 – arm of longitudinal control; 69 – rod of aileron control; 70 – rod of longitudinal control; a – opening for the handle pin fixing in center position in the cross direction

On the handle of the right stick there is no fighting button and trigger of wheels brake control.

Load Spring Mechanisms with Electromagnetic Brakes of EMT-2M.

In total combination control hydraulic actuators are included in the helicopter control system according to the irreversible scheme (forces from control surfaces aren't transferred to control levers). Therefore, for development of forces on levers, and for removal of such forces at the set flight mode load spring mechanisms with electromagnetic brakes of EMT-2M are installed in systems of longitudinal, lateral and directional control (Fig. 19.49).

They are the spring cylinders 2, 3 connected to control linkage in parallel and arranged so that at any signs of movement of linkage spring 10 of mechanisms work for compression.

Cylinders are connected by the top tips to intermediate arms, and by lower ones – to leads of electromagnetic brakes 1 which are rigidly fixed at the released TRIMMER button.

The forces developed by springs are taken up on governing bodies, are proportional to deviation of these bodies and increase under the linear law.

The maximum efforts at the end of control levers stroke shouldn't exceed:

– in longitudinal control 8.6 kg;
– in lateral control 6.0 kg;
– in directional control 13 kg.

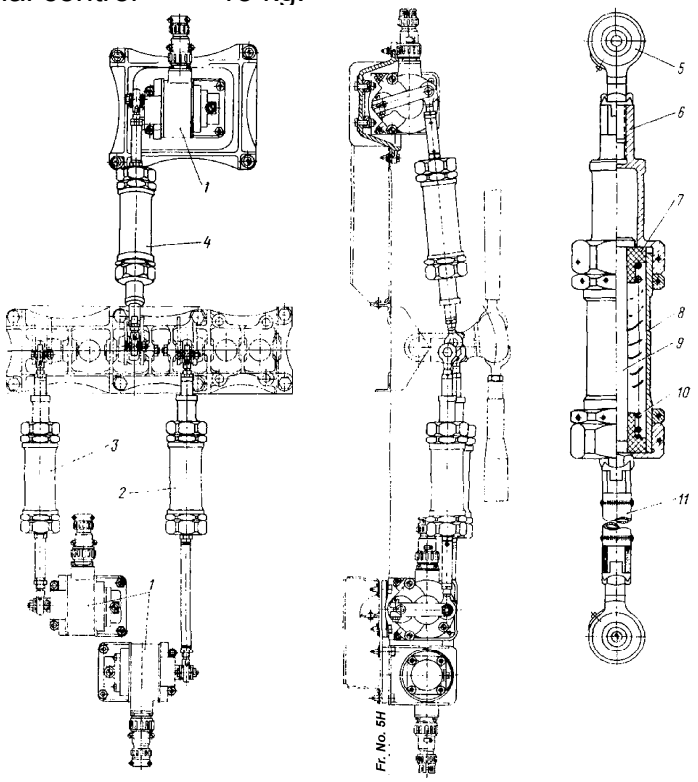


Fig. 19.49. Feel Spring Mechanisms. Installation Scheme:

1 – electromagnetic brakes of EMT-2M; 2 – feel spring mechanism of longitudinal control; 3 – feel spring mechanism of aileron control; 4 – feel spring mechanism of directional control; 5 – bearing with threaded cap; 6 – cover; 7 – bushing; 8 – cylinder; 9 – rod; 10 – spring; 11 – bushing

The electromagnetic brake of EMT-2M is intended for removal of the load from LLCS and pedals on the set flight modes.

Consists of:

– gearbox;
– electromagnetic coupling;
– the centrifugal brake grouped in one block fixed on frame 5N web.

By pressing the TRIMMER button electromagnetic couplings are powered and centrifugal brakes are disconnected. Forces of the compressed springs turn the

released scissors, and forces from LLCS and pedals are removed instantly and completely.

The movement from the handle in the longitudinal direction is transmitted through system of rods and arms to the shaft of shaft located in a geared compartment of the helicopter which is the kinematic link providing rational configuration of the control system links.

The unit comprises four coaxially located shafts 1, 3, 5, 7 made of 30HGSA steel which rotate independently from each other on bearings in the arm 9 cast from ML-5PCh alloy. Shafts have pair of levers each 2, 11; 4, 12; 6, 13 and 8, 14 on the ends (see Fig. 19.50).

Output levers of this unit are connected to input links of the combination control hydraulic actuators. The unit in general functions as angular arm.

KAU-3OB are boosters which power rods are connected to swash plate arms. Will be in detail considered in chapter "Hydraulic System".

The swash plate is intended to change value and direction of MR total aerodynamic force and ensure longitudinal and lateral controllability of the helicopter with that and change the modes of its flight.

Change of the MR collective pitch, i.e. increase or reduction of installation angles of all five blades by the same size, leads to change of total aerodynamic force in value (to change the helicopter flight mode).

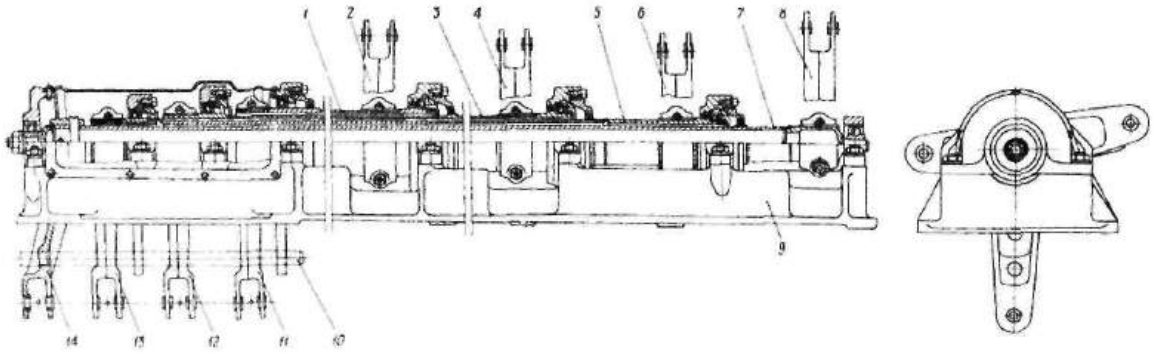


Fig. 19.50. Unit of Longitudinal, Lateral, Directional and Collective Pitch Control:

1 – shaft of longitudinal control; 2, 11 – levers of longitudinal control; 3 – shaft of directional control; 4, 12 – levers of directional control; 5 – shaft of MR collective pitch control; 6, 13 – control levers of MR collective pitch control; 7 – shaft of aileron control; 8, 14 – levers of aileron control; 9 – bracket; 10 – pin for fixing of control stick in middle position

By corresponding plane inclination change of plate rotation of the swash plate there is a cyclic change in installation angles of each blade therefore the direction of full aerodynamic force of MR changes and longitudinal and lateral controllability of the helicopter is provided.

The unit is a kinematic link in structures of longitudinal, aileron control and the collective pitch channel and is fixed by a guide flange by means of hairpins to the MGB case.

The basic device of the swash plate is shown in Fig. 19.52.

The members of the swash plate:

– ram guide;
– slide block;
– slide block arm;
– MR collective pitch change lever;
– two arm assemblies of longitudinal and aileron control;
– external and internal rings of the universal hinge (Cardan);
– plate;
– five rods of blades' turn;

– scissor assembly.

Specification.

Plate tilt angles:

– forward..... 6 °48'... 7 °6'

– back.....4 °48'... 5 °6'

– left.....4 °... 4 °24'

– right.....4 °... 4 °24'

Maximum stroke of slide block, mm.....46... 48

Mass of the unit, kg.....118

Slide block guide (Fig. 19.54, pos. 9) is the steel cylinder with a flange for fastening to MGB.

The slide block 77 is executed in the form of the steel cylinder with bronze bearings riveted from within which slide on a guide. In the central part of the slide block there is a flange to which an bracket 78 of slide block is fixed. In the top part of the slide block two openings in which the studs 55 connecting the slide block to an internal ring of Cardan 56 are installed on bearings.

The common axis of the studs 55 connecting an internal ring to the slide block is located perpendicular to the general axis of the studs 63 connecting an internal ring 56 Cardan with external 46.

At such connection an external ring 46 and the plate 59 mounted on it can bend in all directions relative to the slide block.

Two studs 64 which are connected to rods 14 of longitudinal and aileron control are cantilever fitted in an external ring at an angle 90 ° to each other.

On a cylindrical surface in the top part of an external ring of Cardan on the radial-stop two-row ball-bearings 39 the plate 59 of the swash plate is installed. The plate is made in the form of a five-pointed star with a massive central ring. Steel cups 32 for installation of trailer hinges of a plate are pressed in the star beams (in plate lugs) located at an angle of 72 ° to each other.

The plate is brought into rotation by the scissor representing the kinematic link consisting of a collar 47, a lug 58 and the lever 67 pivotally connected among themselves. The scissor provides synchronous rotation of a plate and the MR head at any plate inclinations of the swash plate and its forward movement along a guide axis. The scissor collar is mounted on the case of the MR head and fixed in a certain angular position by a pin.

The plate of the swash plate is inclined by means of arms of 1 and 15 of longitudinal and aileron control mounted in lugs of the arm fixed on a slide block flange with shift on 21 ° against MR rotation to ensure independent longitudinal and aileron control.

Scales 2 and the noniuses 83 installed on longitudinal arm assemblies and aileron control allow to carry out adjustments of linkage of the corresponding control channels within 6' on a swash plate tilt angle without application of goniometers and adjustment of the helicopter.

Swash Plate Collar Shift Limiter

The swash plate collar shift limiter (Fig. 19.53) prevents an escape of the deformed collar from the MR head seat, and also signals about emergence of deformation of collar arm.

The limiter consists of two half rings 1 and 2 which are pulled together by bolts.

The collar shift limiter is mounted on the MR head case. At the same time the gap between end faces of the limiter of shift and the lower plane of the MR head on a surface R isn't allowed, and the facet K of a half ring has to be below from a scissor side.

On a half ring there is a boss which covers a swash plate scissor collar edge at installation.

Nuts of coupling bolts 4 are tightened by moment of 2+0.5 kg·m. The tightening is carried out evenly from two sides, at the same time in joints of half rings the difference of the sizes J more than 1.0 mm isn't allowed.

On two opposite flat surfaces of the half ring plates 3 which close the sites painted in red (orange) color are fixed by means of two brass screws 7.

At deformation of the collar arm the deformed part of the arm is displaced up and presses a plate edge. At the same time brass screws are cut off, and the plate falls down, having hanged on a locking wire. The bared site painted in red (orange) color indicates about arm deformation presence.

Hydraulic Thrust Roller in Longitudinal Channel

For prevention of shock of MR blades against TB during landing time and when taxing the helicopter in longitudinal control system the hydraulic thrust roller (Fig. 19.51) limiting a plate inclination of the swash plate back of 2°12' is mounted. After liftoff the system is disconnected.

The members of the system are:

– hydraulic cylinder (HC) – a roller 2 (under the top angular arm of the longitudinal channel on Fr. 5N);

– (EHV) GA-192 electric hydrovalve (on Fr. 5N);

– microswitch VM-800K (on MLG shock absorbers).

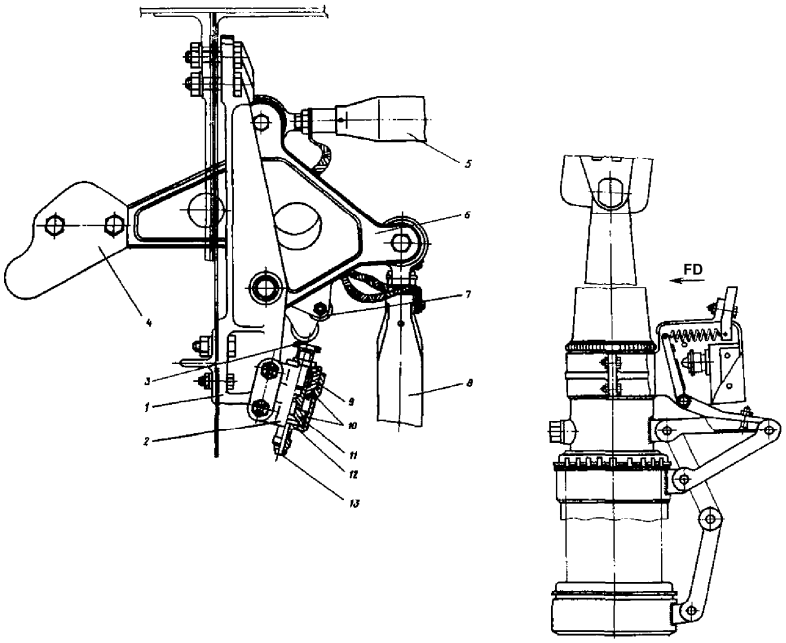


Fig. 19.51. Hydraulic Thrust Roller in Longitudinal Control:

1 – arm;
2 – hydraulic thrust roller;
3 – adjusting screw;
4 – balance weight;
5 – rod;
6 – arm;
7 – roller;
8 – rod;
9 – axle box;
10 – sealing rings;
11 – piston;
12 – hydraulic thrust roller case;
13 – connection of working liquid supply

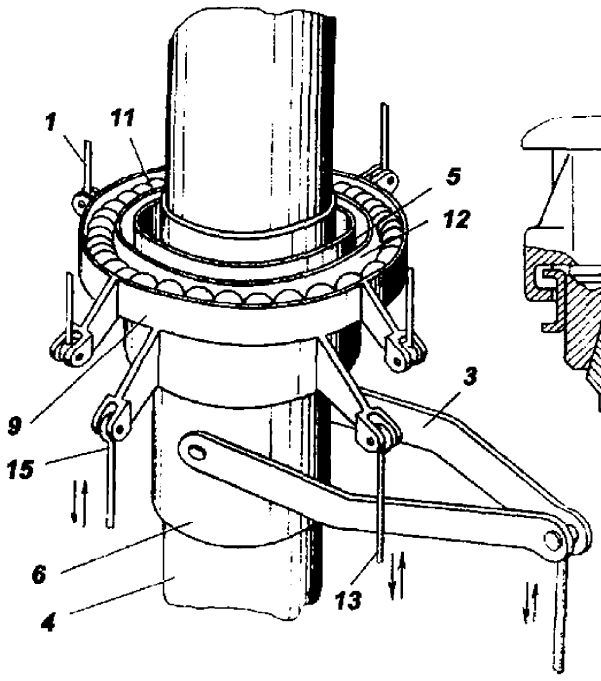


Fig. 19.52. Principal Arrangement of Swash Plate:

1 – adjustable rods; 2 – collar; 3 – collective pitch lever; 4 – slide block guide; 5 – internal ring of Cardan; 6 – slide block; 7 – arm; 8 – lever; 9 – plate; 10 – ear; 11 – bearing; 12 – external ring of Cardan; 13,15 – rods

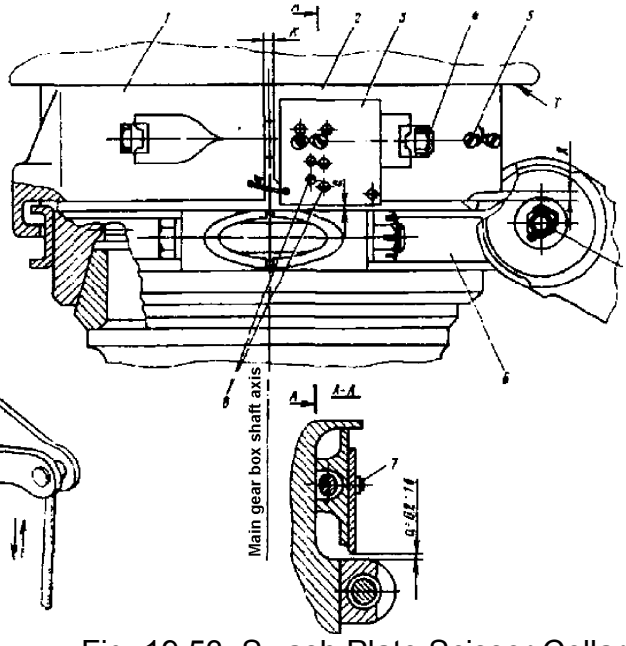
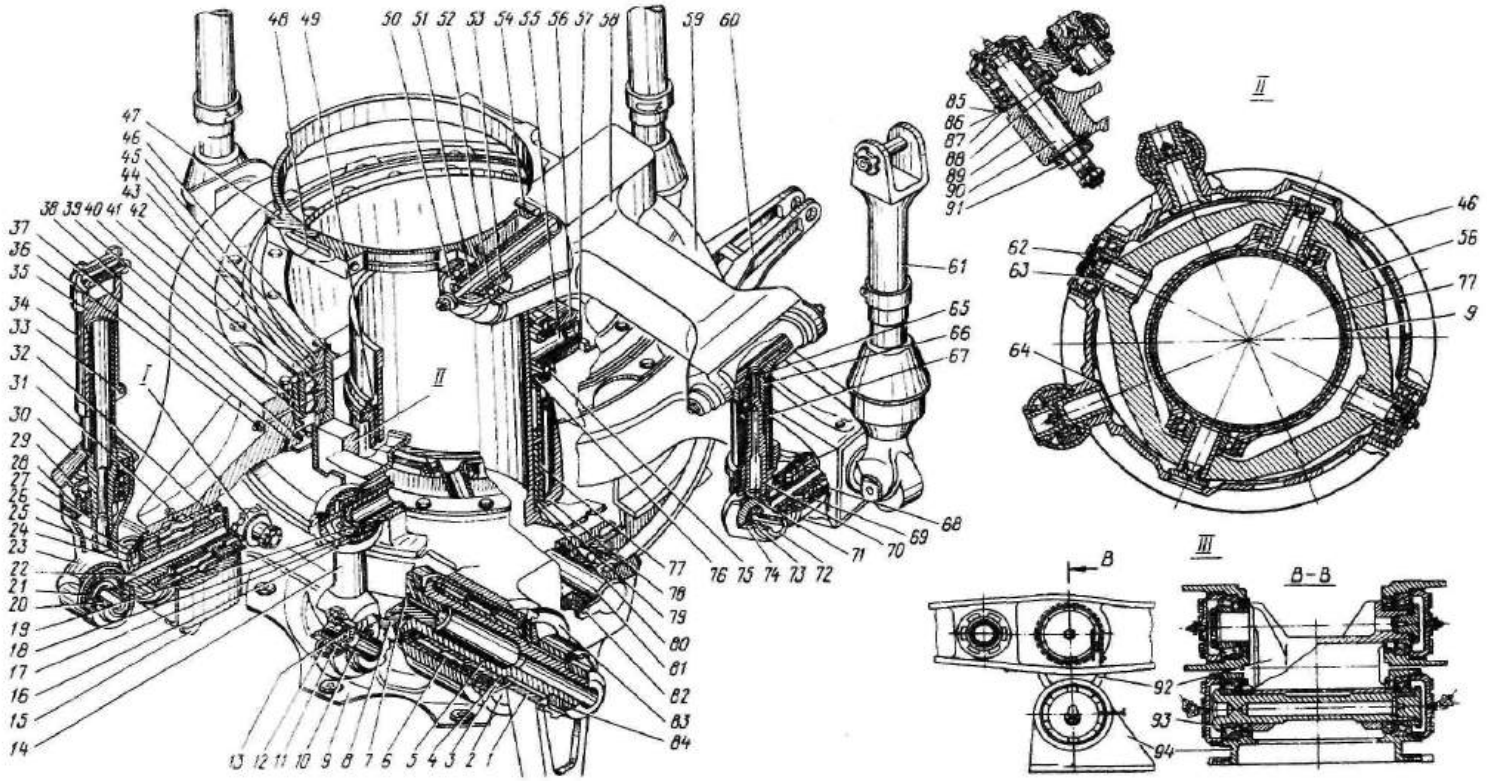


Fig. 19.53. Swash Plate Scissor Collar Shift Limiter:

1 – half ring; 2 – half ring; 3 – plate; 4 – coupling bolt; 5 – spare brass screw; 6 – swash plate scissor collar; 7 – brass screw; 8 – openings

Fig. 19.54. Swash plate:



1 – fork arms;
2 – scale;
3 – nut;
4 – washer;
5 – roller;
6 – bushing;
7 – screw;
8 – arm lever of longitudinal control;
9 – ram guide;
10 – stud;
11 – spherical bearing;
12 – cover;
13 – oil fitting;
14 – rod;
15 – cross control arm fork
16 – rubber cover;
17 – nut;
18 – spherical bearing;
19,20 – studs;
21 – ball bearing;
22 – roller;
23 – lower fork of rod;
24 – ring;
25 – rubber ring;
26 – cover;
27 – nut
28 – ball bearing;
29 – nut;
30 – rubber cover;
31 – oil fitting;
32 – cup;
33 – bolt;
34 – rod core;
35 – top fork of rod;
36 – oil fitting;
37 – case;
38 – cuff;
39 – bearing;
40 – bushing;
41 – flange;
42 – cuff;
43 – ring;
44 – shield;
45 – nut;
46 – external ring of Cardan;
47 – scissor arm;
48 – bolt;
49 – cuff;
50 – nut;
51 – pin;
52 – cover;
53 – axis;
54 – pin;
55 – stud;
56 – internal ring of Cardan;
57 – nut;
58 – scissor ear;
59 – plate;
60 – lever;
61 – turn blade rod;
62 – cover;
63, 64 – studs;
65 – oil fitting;
66 – nut;
67 – scissor lever;
68 – nut;
69 – case;
70 – fork;
71 – nut;
72 – stud;
73 – needle bearing;
74 – roller;
75 – ball bearing;
76 – bronze bushing;
77 – slide block;
78 – slide block arm;
79 – bronze bearing;
80 – cuff;
81 – stud;
82 – bolt;
83 – nonius of longitudinal control;
84 – nut;
85 – scale of aileron control;
86 – disk;
87, 88 – pins;
89 – plug;
90 – axis;
91 – nut;
92 – strap;
93 – stud;
94 – arm;
I – on a arm of aileron control;
II – according to Cardan plate;
III – lever support of the collective pitch

With the compressed landing gear struts the top links of torsion links of shock-absorbers press micriswitch pushers, GA-192 power-supply circuits become isolated, hydromix comes to hydraulic cylinder.

The hydraulic cylinder rod moves forward, and at LLCS deflection to its side on the angle corresponding to a tilt angle of the plate 2°12', the top angular arm 7 rests the roller against the hydraulic cylinder rod.

At further deflection of LLCS to its side the force will increase up to 12±3 kg.

Members of push-pull control linkage in all control channels are presented by rods and arms.

Rods and arms in not power portions (to CLU) are made from duralumin, and after CCHA actuator – of steel. Tips of rods are steel, spherical plain bearings are pressed in them.

To increase stiffness of linkage and decrease vibrations on the portion on Fr. 5N intermediate arms 15 (see Fig. 19.47), and on the portion of the ceiling panel of cabin – roller guides of rods are installed (assembly 25). One of three rollers of each guide is installed on the eccentric roller which turn allows to adjust a gap between rollers and rod.

19.6.3. Directional Control

19.6.3.1. System Structure

Helicopter dual directional control (Fig. 19.55) has mixed design linkage, power-assisted, combined with the AP.

Members of directional control:

– pilot and pilot-navigator pedals 1;

– the load spring mechanism 16 with electromagnetic brake EMT 2M 12:

– shaft of shafts 30:

– control unit RA-60B 38;

– quadrant 36;

– control linkage:

a) push-pull control linkage (system of rods and arms);

b) cable linkage (cables 40 and bush roller sequence 46):

– pitch limit system SPUU-52.

19.6.3.2. System Units

Pedals (Fig. 19.56) are made in the form of the separate units fixed to longitudinal beams of the crew cabin floor panel. The mechanism which allows basic platforms *12* to make the plane-parallel movements for convenience of control is the basis for a design.

Pedals of the pilot and the pilot-navigator are blocked among themselves by rod of synchronization. Extreme provisions are limited to stop screws.

Angular levers of 8 pedals are connected among themselves by adjusting screws 10 to hand wheels which provide longitudinal adjustment of footboards in the range of ±32.5 mm. Basic platforms of pedals are equipped with microswitches 13 to shutdown AP “DIRECTION” channel.

RA-60B is the booster which power rod is connected with linkage quadrant. Features of its design will be considered in the chapter "Hydraulic system".

The quadrant (view A, Fig. 19.55) finishes the push-pull portion of linkage and transfers the movement from a rod 38 of RA-60B to control cables of TR. It is pivotally attached to CCHA arm against MGB.

Cable linkage

It is presented by doubled (for reliability) cables in a superstructure of the fuselage, TR and TB, and a bush-roller chain to TB.

The ends of cables by means of bolts and dowels are fixed on quadrant, and from other side (view C) are connected to a bush-roller chain. The chain is put on HR sprocket. To change the direction the cables rest against the guide rollers (type $D)_9$, installed on CCHA arm and in TB. A part of rollers is textolite, another part is duralumin (for bonding). For prevention of fluctuations cables and chain pass through textolite guiding shoe (type B). To increase survivability according to the bulletin cables are spread on sides of a tail beam.

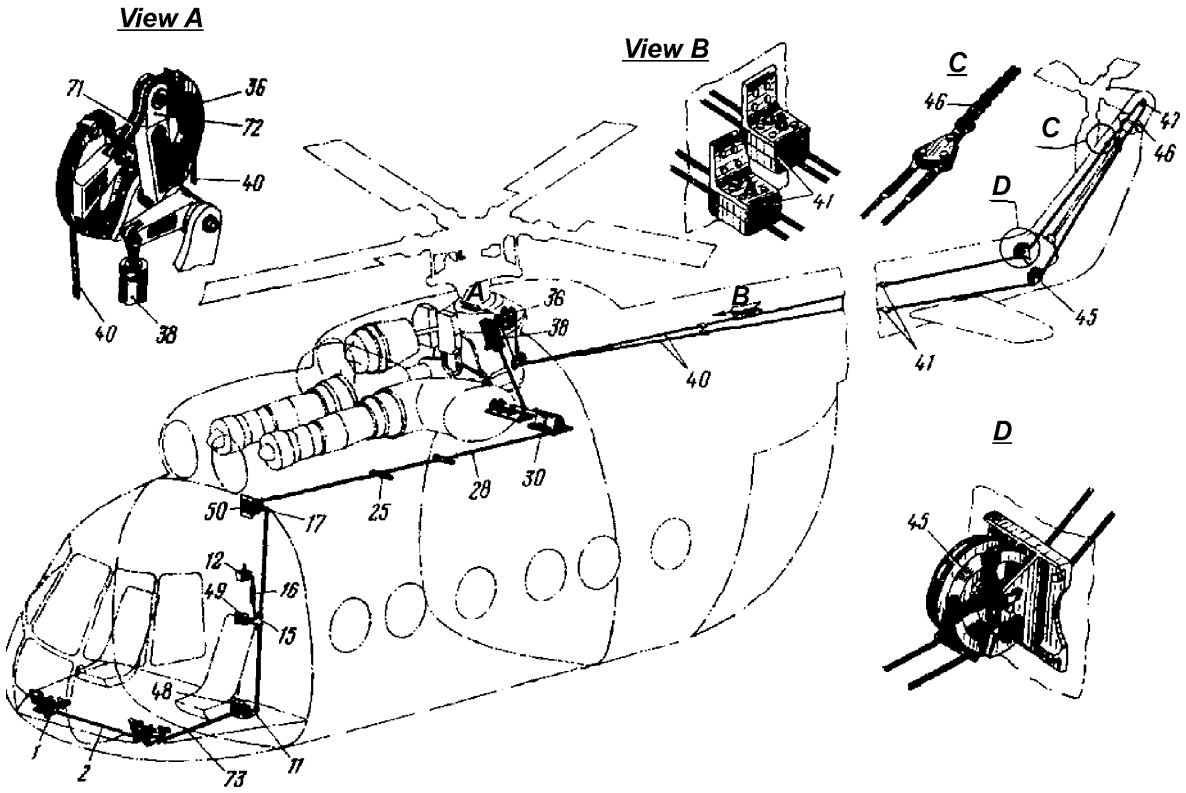


Fig. 19.55. Scheme of Foot Control:

1 – pedals of directional control; 2 – rod connecting arms of pedals; 11 – lower angular arms; 12 – electromagnetic brake of EMT-2M; 15 – intermediate arm; 16 – load spring mechanism of directional control; 17 – top angular arm; 25 – roller guides; 28 – rods of directional control; 30 – unit of longitudinal, lateral, directional and collective pitch control; 36 – quadrant; 38 – hydraulic booster of directional control; 40 – control cables of tail rotor; 41 – textolite guides of cables; 45 – rollers; 46 – bush-roller chain; 47 – sprocket of tail gearbox; 48 – arm of lower angular arms; 49 – arm of intermediate support; 50 – arm of the top angular support; 71 – link; 72 – arm; 73 – rod

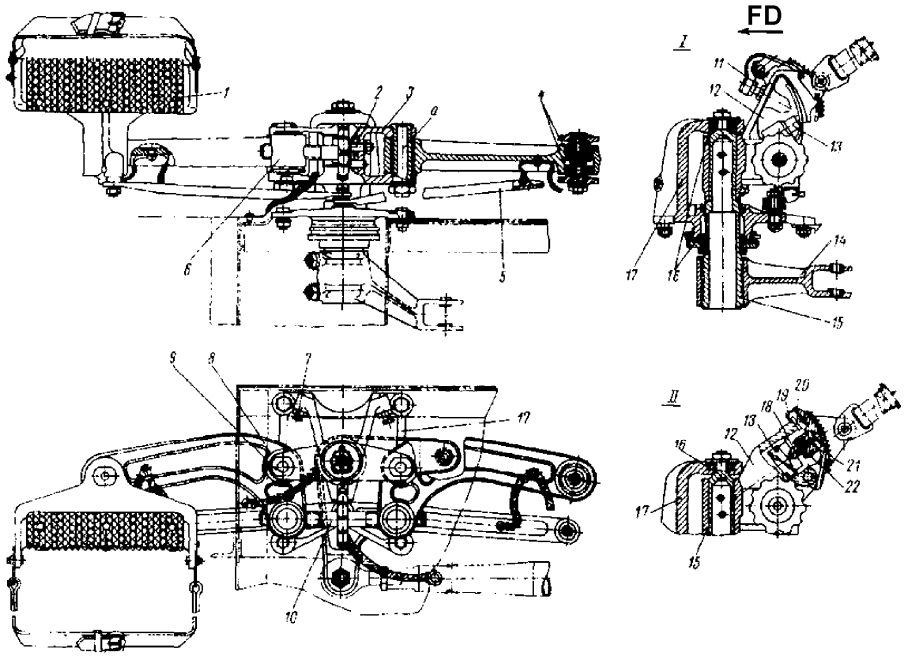


Fig. 19.56. Foot Pedals:

1 – for initial release helicopters; 11 – for latest release helicopters, 1 – button switch; 2 – hexagon fit bolt; 3 – bracket; 4 – ball bearings; 5 – rod; 6 – lining; 7 – restraint bolt; 8 – angular lever; 9 – hollow bolt; 10 – adjusting bolt with hand wheel; 11 – adjustable screw-stop; 12 – rest pad; 13 – limit switch; 14 – arm; 15 – shaft; 16 – ball bearings; 17 – pedal base; 18 – stop; 19 – limiter; 20 – plate; 21 – spring; 22 – holder

Tail Rotor Maximum Pitch Limit System SPUU-52-1

Tail Rotor maximum pitch limit system (MPLS) is intended for automatic change of stop position, limiting maximal value of tail rotor blades angle depending on density of air in attempt to prevent the rotor and transmission from overload.

As the density of air is reduced, the maximum angle of tail rotor blades enhance, when density of the air enhances, the angle reduces, wherein the necessary directional control reserve is provided.

Specification (for helicopter Mi-8MTV)

Maximal pitch setting of tail rotor:

a) sliding cylinder of limit system is full-up..... .....22°45'...23°30'

b) sliding cylinder of limit system is fully extended.....16° 55'...17° 45'

Full stroke of limit system sliding cylinder, mm.....41±1

The system (Fig. 19.57) consists of:

1. Actuating mechanism of the system located on the feeder tank capsule wall in gear compartment and comprised of the next members:

– movable stop 9, that is designed as arm with microswitch AM-800K 3;

– electric actuator MP-100MT 6;

– rod 19, going from shaft of shaft to CCHA slide valve with screw nut 12 and spring bushing 10;

– feedback transducer 16 DOS-6;

2. Temperature receiver 1 P-1, set in front of engine air intakes.

3. Altitude pressure transmitter 2 IKD-27DA, set left on Fr. 3N (sw.3H)

4. Control unit BU-32, on the front panel of it are located:

– zero indicator, that shows position of movable stop (move of bug to the left

corresponds to range extension of the right pedal or the minimal air density);

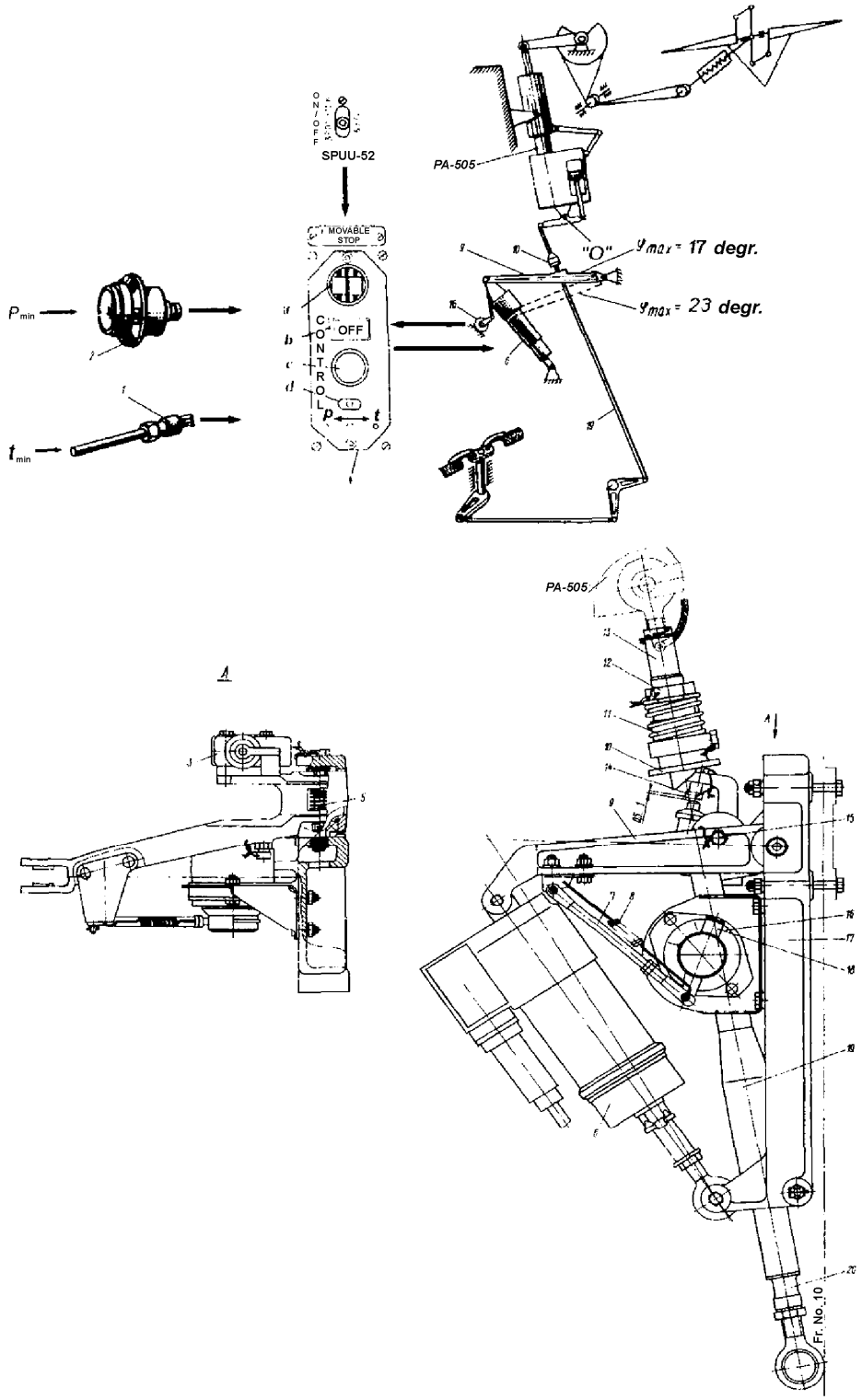


Fig. 19.57 Scheme and Mechanisms of
Tail Rotor Maximum Pitch Limit System SPUU-52:

1 – temperature receiver; 2 – pressure transmitter; 3 – control unit; 4 – spring; 5 – electric actuator; 6 – electric actuator; 7 – turnbuckle; 8 – spring; 9 – arm-stop; 10 – hub; 11 – spring; 12 – screw nut; 13 – hub; 14 – adjustment screw; 15 – eccentric screw; 16 – feedback transmitter; 17 – bracket; 18 – arm; 19 –rod; 20 – cup

– red switch-light “OFF”, indicating about system shutdown and its check;
– centering handle, responsible for line circuit system of control unit operability check;
– push-to-activate switch of signal imitation on temperature and pressure for system check;

5. “SPUU-52” switch of forced retraction (shutdown) of stop is located on the left electropanel.

To shut down the system it is necessary to turn on circuit breaker (CB) “SPUU-52” on the right CB panel wherein 27V voltage direct current is applied to actuating mechanism, its sliding cylinder pulls in, and stop completely retracts.

At the same time 36V voltage alternate current is applied to control unit and switch-light “OFF” lights up, and zero indicator takes extreme left position.

When setting the “SPUU-52” switch on the left electropanel to position “ON” the BU-32 unit is also powered with direct current, switch-light “OFF” turns off.

When the density of air changes, signals from temperature and pressure transmitters is applied to control unit. Signal from the feedback transmitter applies to the same unit. All the signals sum up in the control unit, and if the position of movable stop doesn't correspond to density of the air, the actuating signal that appears applies as voltage on electropanel MP 100, that sets the stop in position corresponding to given conditions, is formed.

On the heights close to sea level, at medium and low temperatures of open air hand of zero indicator moves right. At low values of density hand can be in extreme left or close to it position.

During the system work from AP in “RUNNING” mode (move of the right pedal forward) microswitch on the movable stop works when it is pressed by link spring and turns off AP “DIRECTION” channel. The movement of pedals stop.

If the pilot used pedals (“DIRECTION” channel is off), then at the further rod movement the stop reaches overhang on the arm and stops, providing overtravel of the microswitch of 1.8...2 mm after actuation; wherein rod passes a way about 1 mm.

Further movement of rod, for approximately 5 mm more, leads to spring compression on rod. At the moment of face of hub contact with overhang on screw nut the possibility of rod movement goes away and the rod takes position on dead stop.

19.6.4 Integrated Control of MR Collective Pitch and Engines

Integrated control of main rotor’s collective pitch and engines is meant for thrust force of main rotor change with simultaneous engine's power change to speed of main rotors maintenance without significant alterations.

To provide manual constant speed regulation of main rotor (at failure of ACS) and for idle mode set on land the throttle is installed on collective control stick, that is kinematically connected only with NR throttle valve crowbars of engines.

Control is dual, with push-pull control linkage, power-assisted, combined with AP (Fig. 19.58).

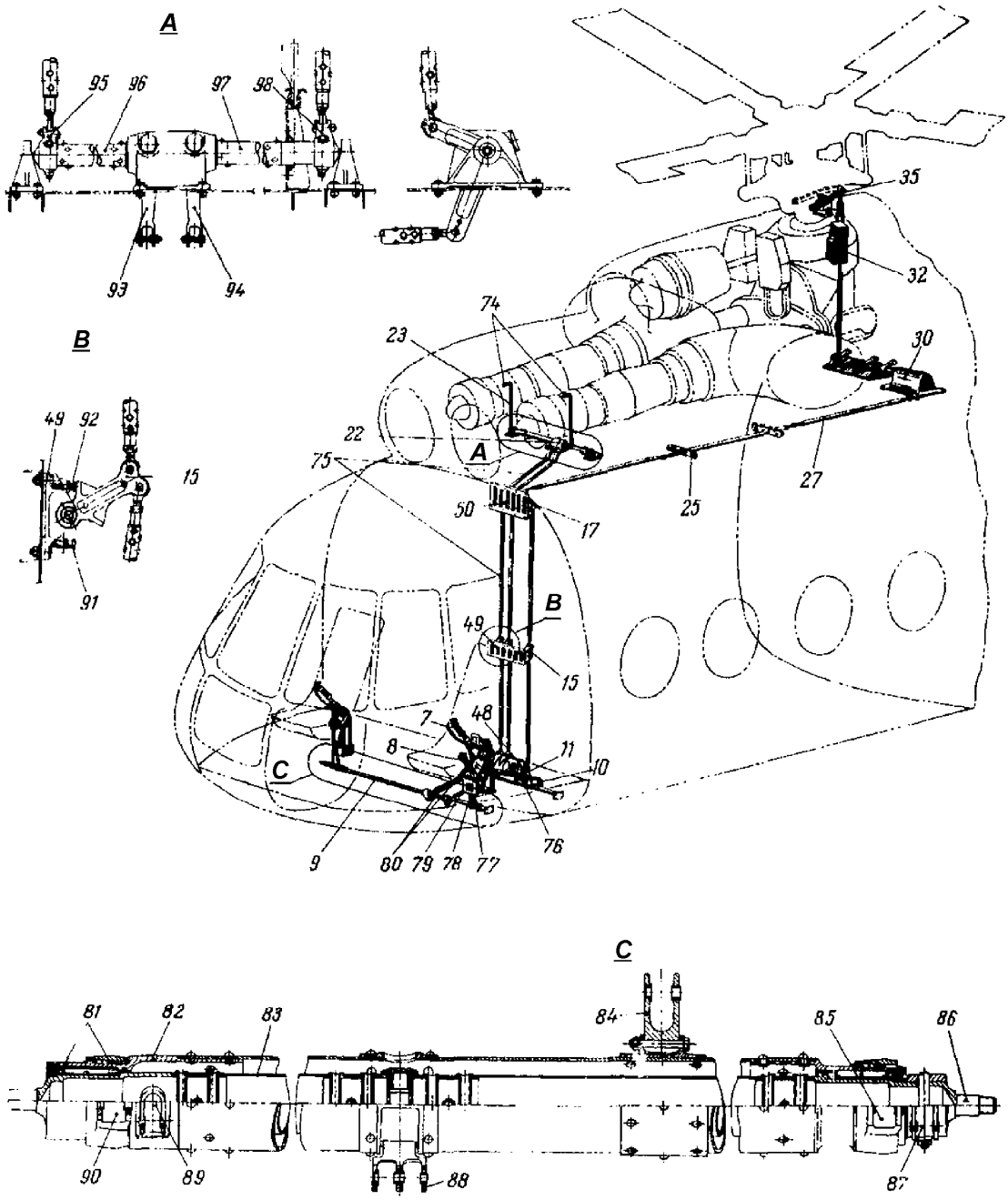


Fig. 19.58. Scheme of “Collective Pitch” Integrated Control:

7 – “Collective pitch” integrated control stick; 8 – levers of engines decoupled control; 9 – closing shaft; 10 – differential assembly; 11 – lower angular arms; 15 – transitory arms; 17 – upper angular arms; 22 – shaft block; 23 – engine actuating rod; 25 – roller fairlead; 27 – collective pitch control rod of main rotor; 30 – accessory of longitudinal, lateral and directional cyclic control and collective pitch control; 32 – collective pitch control hydraulic boost system; 35 – swash plate of main rotor collective control crowbar; 48 – lower angular arms arm; 49 – transitory arm arm; 50 – upper angular arms carrier; 74 – fuel control unit NR-40VR installed on engines; 77 – engine actuating rod; 78 – collective pitch control rod; 79 – collective pitch control rod connecting closing shaft with lower angular arm; 80 – engine actuating rods connecting crowbar of closing shaft with differential assembly; 81 – ball bearings; 82 – outer shaft; 83 – inner shaft; 84, 85, 90 – collective pitch control rod hook-up arms; 86 – operating pin; 87, 88, 89 – engine control rod hook-up arms; 91, 92 – adjustable retainer bolts; 93, 94, 95,98 – arms connected to engine control rods; 96, 97 – drive shafts

19.6.4.1. System Structure

Let’s split control linkage in two lines:

– collective pitch control;

– engine control;

Common accessories for these lines are:

– collective control stick 7;

– closing shaft 9.

Then lines are divided, and the next accessories are installed on lines of collective pitch control;

– shaft of shafts 30;

– CCHA KAU-30B 32;

– swash plate;

– members of push-pull control linkage.

Engine control lines include:

– differential assembly 10;

– shafts block 22;

– members of push-pull control linkage.

19.6.4.2. ***System Aggregates***

Collective control sticks.

A) **Left-hand** (Fig. 19.59) represents steel tube, on pivotally shared carrier with engine throttle control lever (ETCL) on the cockpit floor and is made as a separate unit.

On the lower part of arm, disk friction is assembled on its axis, on the upper part throttle and the case of buttons with a safety clip.

The next elements are located on the case of buttons:

– friction turn button;

– floodlights control button;

– tactical store release from external load button;

– emergency store release from external load button;

– push-to-activate switch of main rotor rotation reconfiguration.

Disk friction (view A-A) with electrohydraulic actuation safely holds collective control stick in any position.

The tightening force of friction is regulated by fly wheel 21, installed on the axis of arm. When rotating a flywheel clockwise (from the pilot's side) spring force decreases. At normally tightened and activated friction movement forces on collective pitch control lever has to be within 20...25 kg.

When pressing friction shutdown button EhGK GA-192 opens and the liquid under pressure goes to cylinder of friction, made together with arm 15 axis. Piston 77, compresses spring while moving and releases disks 27. Friction shuts down and provides stepless misalignment of collective control with force up to 1.5 kg.

Collective control lever range of motion is 56°. Extreme positions are limited by adjustable stops 22, 23.

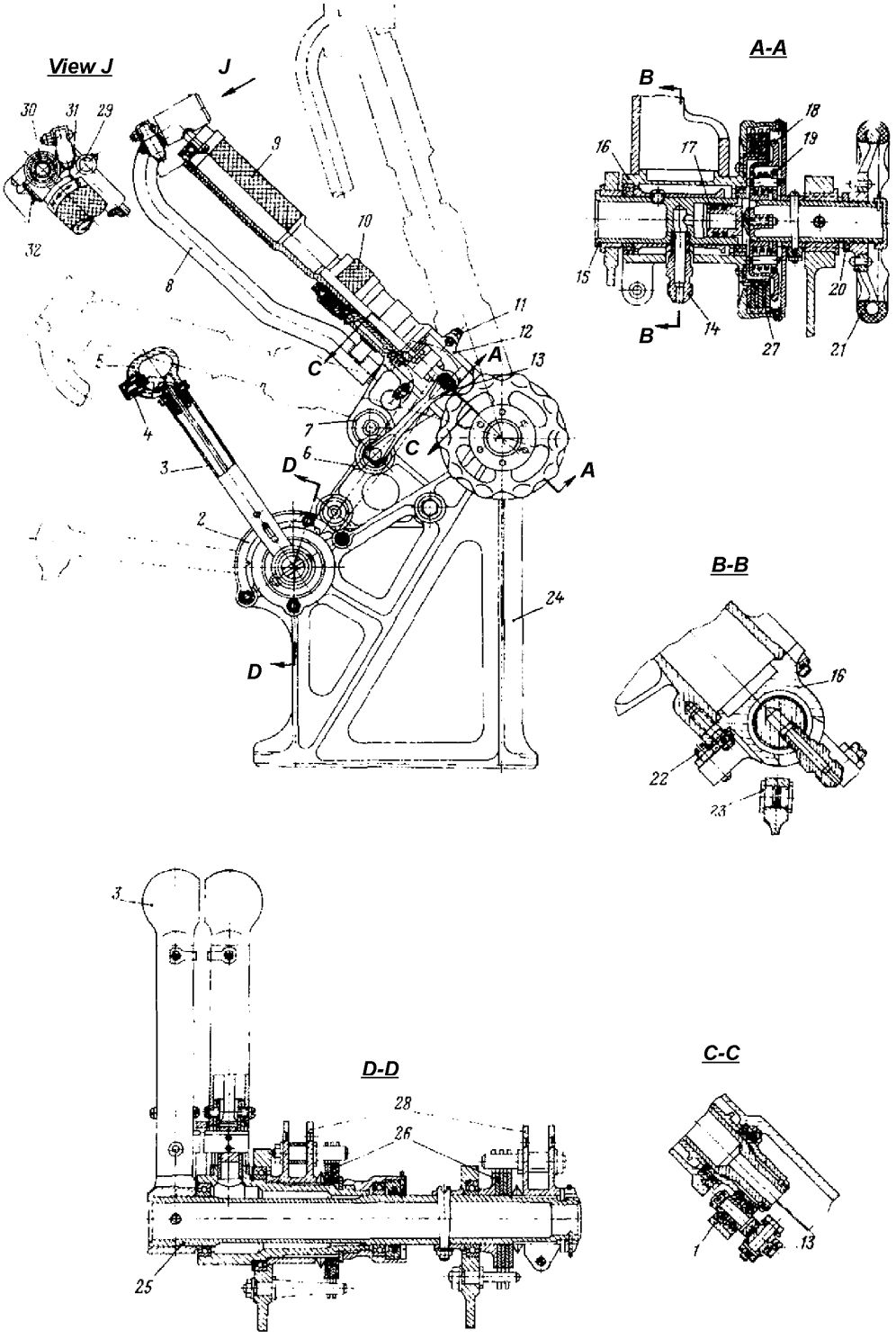


Fig 19.59. Left Stick of “Collective Pitch”:

1 – scissor; 2 – rake; 3 – engine decoupled control crowbar; 4 – release button; 5 – grip handle; 6 – arm; 7 – body ear; 8 – button mounting case; 9 – correction turning handle; 10 – clutch of friction; 11 – limit screw; 12 – handle body; 13 – link; 14 – reducing nipple; 15 – axis; 16 – handle base; 17 – piston; 18 – retainer; 19 – bearing disc; 20 – closing sleeve; 21 – handwheel; 22 – limit screw; 23 – restraint; 24 – arm; 25 – lever shaft; 26 – shaft frictions; 27 – friction discs; 28 – crowbars; 29 – “Collective pitch” friction handle shutdown button; 30 – floodlights control button; 31 – tactical store release from external load button; 32 – emergency store release from external load button;

Correction throttle lever 9 is installed on the upper part and kinematically connected only with throttle valve crowbars of main rotor engines. Full angle of lever rotation is 90° and limited by two screws. Locking lever has friction composed of duralimin discs, springs and clutch 10. When tightening the clutch, friction spring lock discs and fixes correction throttle in the necessary angular position. It is **FORBIDDEN** to turn lever to the left in flight (enter the left position), as it will lead to shut down of Automatic Control System (ACS) and push the limits of main rotor from set values. Extreme left position of lever is set only on the land, during the work of control system (CS) in idle mode, as in this position of lever main rotor throttle valve crowbars will reach idle stop.

B) **Right-hand** is toughly connected with left-hand and unlike it, doesn't have friction and store release from external load buttons.

Closing shaft (view C, Fig 19.58) is located under the cockpit floor and connects both collective pitch controls levers.

Composed of two coaxially located shafts 82 and 83, put together one in another and capable to turn independently from each other.

These shafts are kinematic elements in control linkage of collective pitch (outer shaft) and engines (inner shaft).

Differential assembly (view D, Fig. 19.60) is meant for engine decoupled control lever connection to control linkage of engines from collective control and their movement summation.

It is a kinematic element in engine control linkage from decoupled control lever and collective control lever. The assembly allows changing an operating mode of two engines simultaneously with collective pitch change, and operating mode of both or any engine at a rigid pitch of main rotor. Accessory represents two coaxial shafts 6 and 8 with double-arm levers 4 and 12. It is mounted under cockpit floor to the left from 5N.

Shaft block (view A, Fig 19.58) is a transferring link in engine control linkage, it consists of two kinematically not connected shafts 96 and 97 with a pair of levers 93, 94 and 94, 98 each at the ends. Besides, the unit accomodates control linkage in engine compartment tightly. Shaft rotates on ball-bearings in the arms casted from Mg-alloy, and installed on ceiling panel near Fr. 3 of fuselage central section under engine compressors.

Another linkage elements, such as arms and rods, shaft of shafts, CCHA and swashplate are discussed above.

19.6.5 Separate Engine Control

Separate engine control allows to change an operating mode of each engine individually without changing main rotor collective pitch at any collective pitch control position and correction throttle lever.

19.6.5.1. System Structure

Control is performed by two engine decoupled control levers, installed on left collective control carrier. Movement from individual throttle lever is transferred to differential assembly and then from rods and shaft block to levers of engine NR.

As a result, next members are installed in Engine Throttle Control (EDC) system:

– pair of engine throttle control levers;
– differential assembly;
– shaft block;
– arms and rods.

19.6.5.2. Accessories

Engine decoupled control arms (see Fig. 19.59, C-C) are provided with friction couplings 26, that requires a force of 3...4 kg to be applied.

Engine decoupled control crowbars are stopped by special gates in neutral position. The gates' lock-on buttons are located on the lever heads.

Note. Kinematics control specialty is autogenous creeping left of the correction throttle lever when turning on the takeoff mode. That is not a fault, that is why the corrective turn to the right is not required.

Fig. 19.61 shows hydraulic boost system installation diagram.

19.6.6. Engine Shutdown Control

Engine shutdown control (Fig. 19.62) is provided by two levers, located on the cockpit ceiling bracket, on the left from the helicopter axis.

System consists of:

– two fuel shutoff valve levers 98;
– two drive rollers 102 with drivers in the arms 100;
– two drive rollers 110 with drivers 111 in the arms 108;
– members of push-pull control linkage 21;
– cable linkage (cables 19 with drive rollers 106).

Shutdown controls can stop with the springs and fixators in positions "OPEN" and "CLOSED". To shut the engine down change the lever back and down to the maximum, turn in push the lever forward after the "ON" button is pressed.

19.6.7. Main Rotor Brake Control

Main rotor brake control (Fig. 19.63) is performed by the pilot in command with help of the cable linkage, that consists of:

– brake controller 6;
– cable 4 with a turnbuckle;
– drive rollers 112;
– textolite pads 113;
– unload spring 114.

Main rotor brake control is mounted to the cockpit floor, on the right from the left seat. It is hinged on the pin to the carrier and applied with a cog sector fixator 123, which gate is controlled by the button 115 on the threshold of the grip. Main rotor brake control is interlocked with engine starting system with microswitch 122, that closes the electric circuit of the engine starting system when the transmission is disengaged. The microswitch is mounted in the controller carrier.

The unload spring is meant to protect the cable linkage and the brake lever expander from the overloading and destruction at the abrupt deceleration.

The spring, mounted before the brake lever expander in the gearbox compartment, restricts the load on the cable up to 150 kg.

The helicopter arresting must be carried out by smooth brake actuation with a rotation speed reduction to the 15 %.

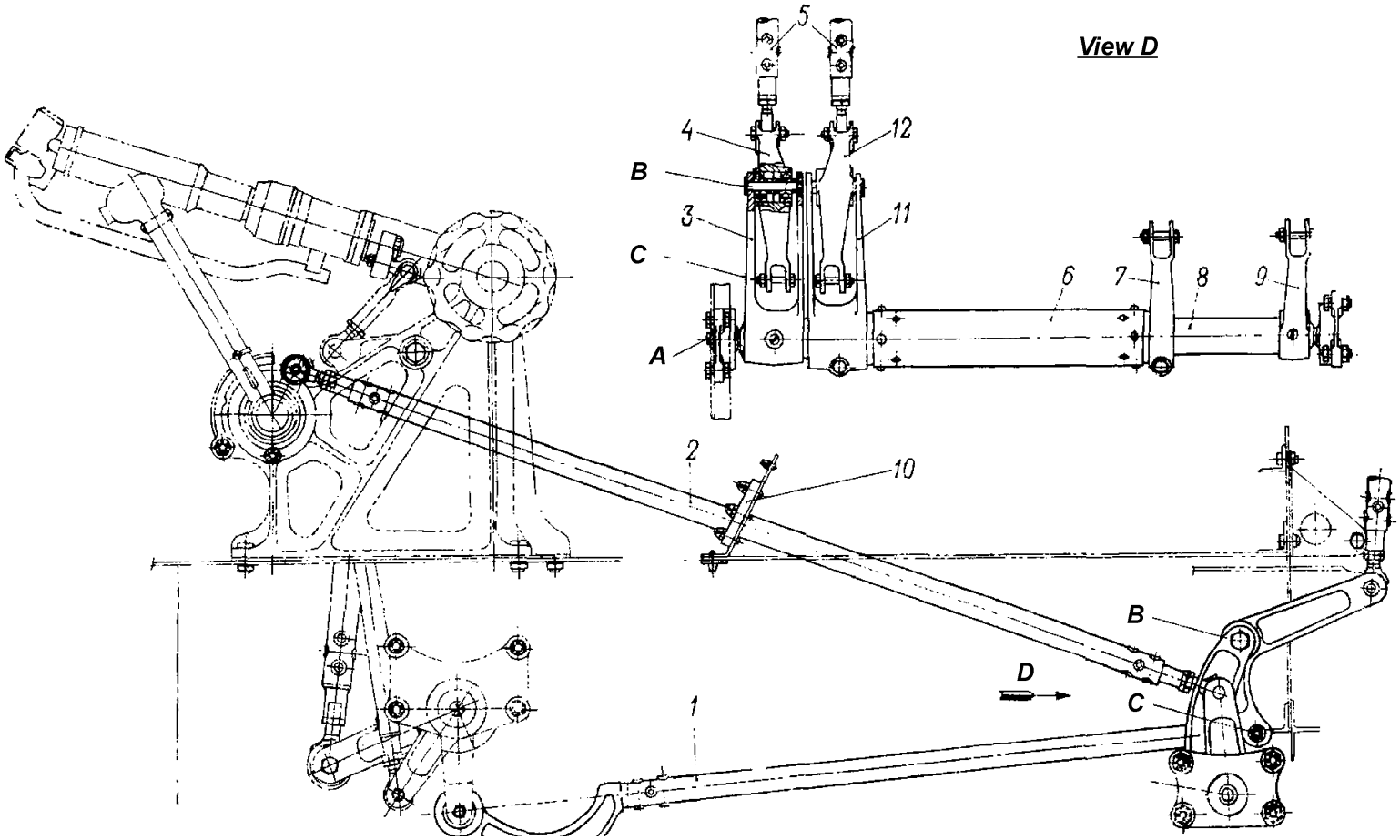


Fig. 19.60. Differential Assembly. Installation Diagram

1 – engine control rods, connected to locking shaft; 2 – engine control rods connected to decoupled control levers; 3. 11– levers; 4, 12 – arms; 5 – engine fuel pump actuating rods; 6 – outer shafts; 7, 9 – levers ;8 –inner shafts; 10 – seal

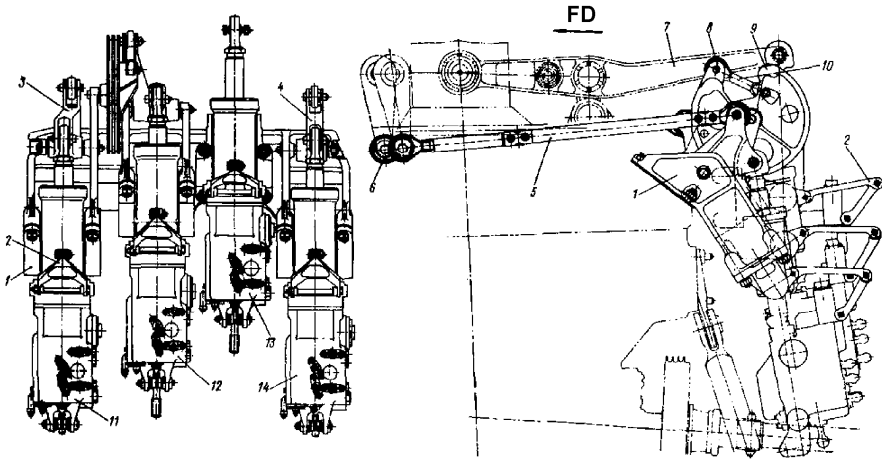


Fig. 19.61. Hydraulic Boost System. Installation Diagram

1 – arm; 2 – torsion link; 3, 4 – arms; 5 – longitudinal control rod; 6 – lateral control rod; 7 – MR collective pitch control swash plate lever; 8 – arm; 9 – quadrant; 10 – link; 11 – hydraulic booster KAU-3OB of the longitudinal control; 12 – hydraulic booster NA-60B of the directional control; 13 – hydraulic booster KAU-3OB of the collective pitch control; 14 – hydraulic booster KAУ-ZOB of the lateral control.

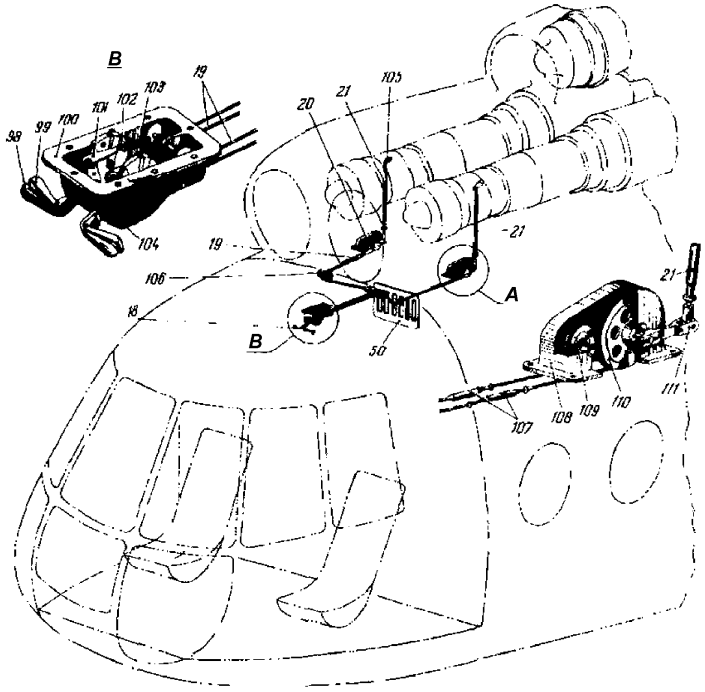


Fig. 19.62. Engine Shutdown. Installation Diagram

15 – engine shutdown lever ; 19 – engine shutdown control cable ; 20 – bracket with rollers ; 21 – engine shutdown actuating rod; 50 – upper angular arms carrier; 98 – handle; 59 – fixator with spring; 100 – arm; 101 – ear; 102 – roller; 103 – spring; 104 – restraint ; 105 – fuel pump regulator handle NR-40VR; 106 – roller; 107 – turnbuckle ; 108 – arms 109, Z – rollers; 111 – driver

Structural parameters of the helicopter assemblies and parts, obtained as a result of the computerized design, are added to data base and are a base for the helicopter designing and computer simulation of modifications. Further you can find models of the helicopter systems and assemblies (Fig. 19.63 – 19.85), those are essential to create helicopter geometrical arrangements and space allocation models and analytic samples of helicopter parts.

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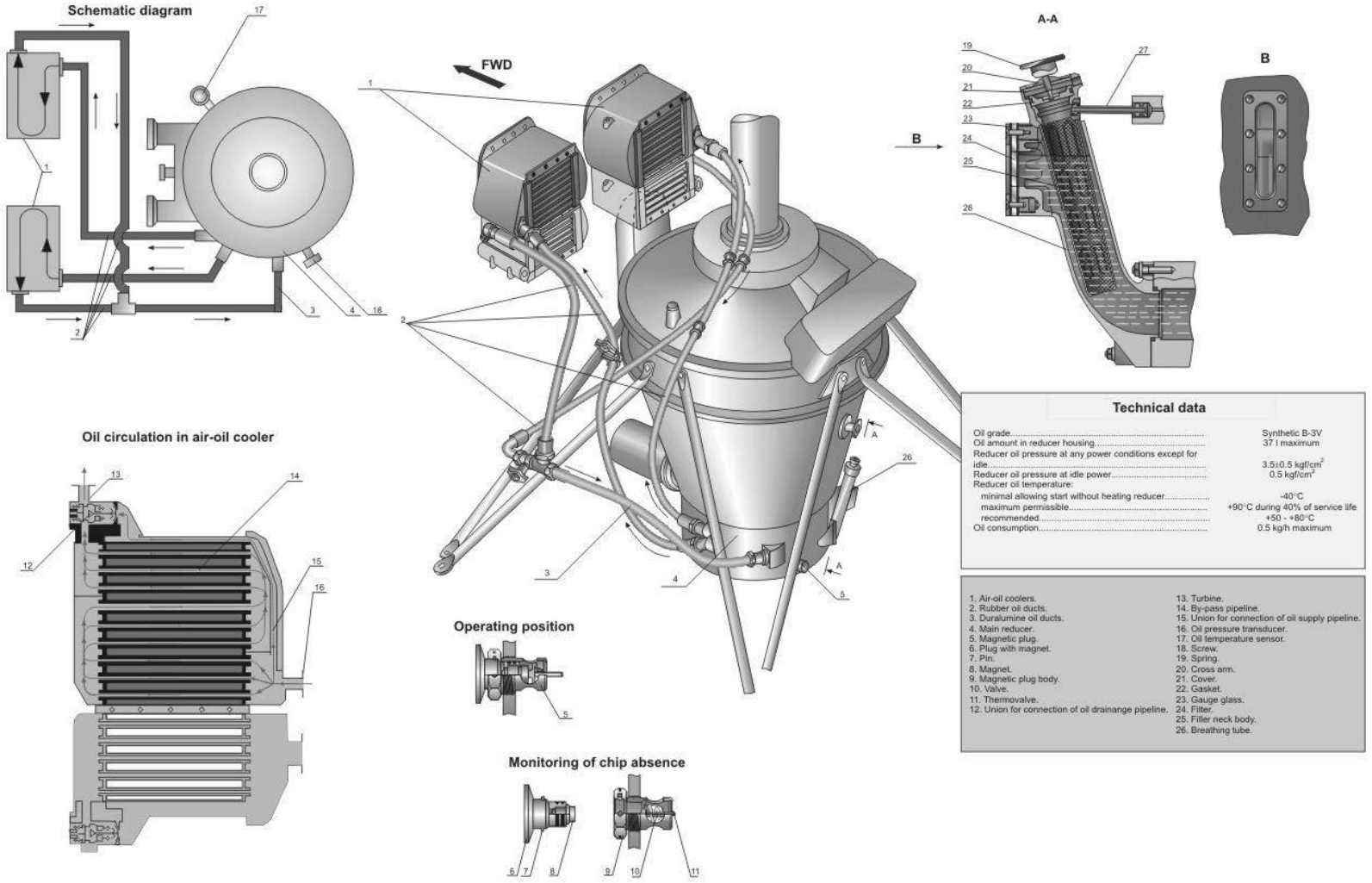


Fig. 19.63. Main Gearbox Oil System

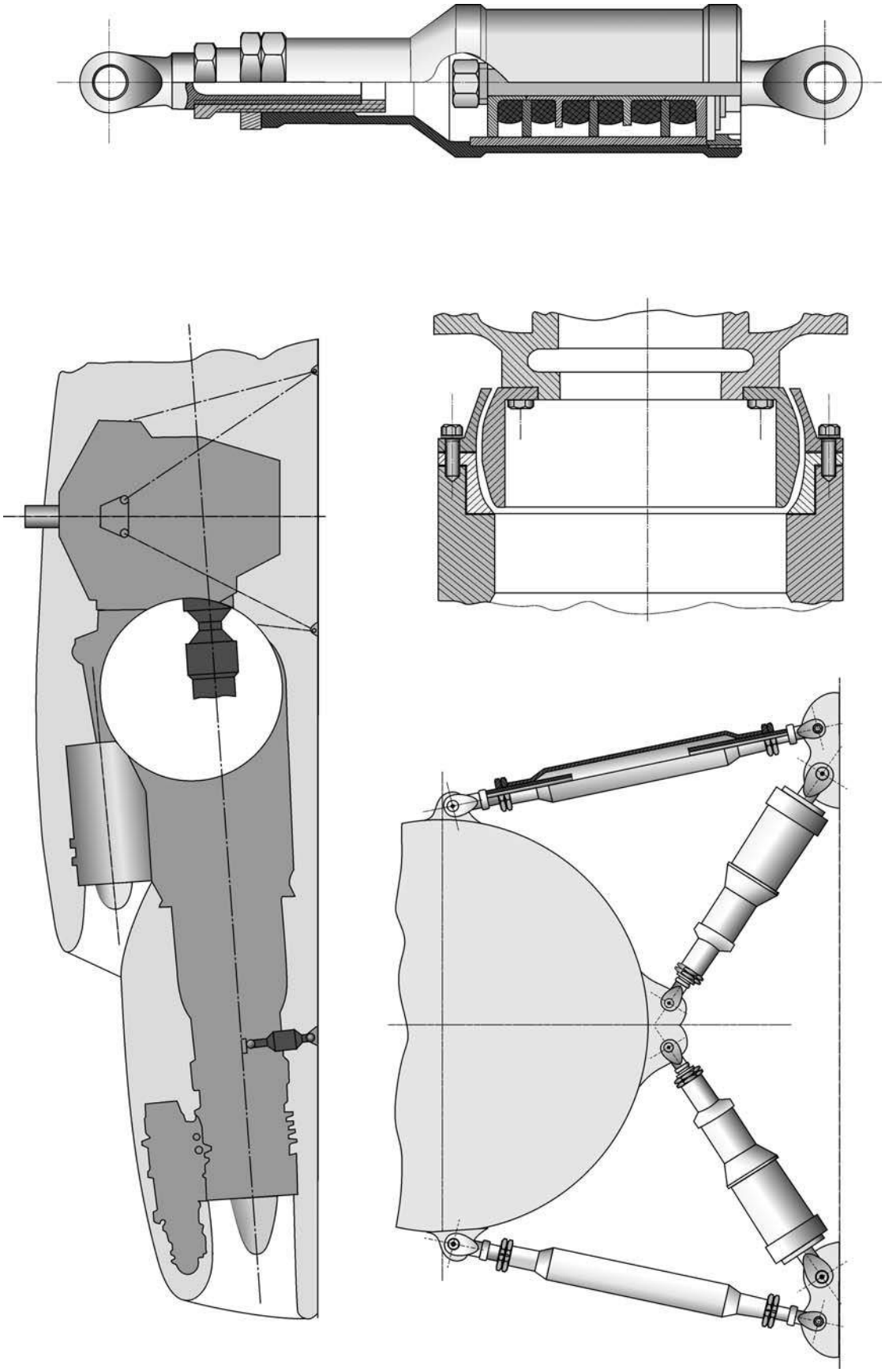


Fig. 19.64. Mi-8 Helicopter Engine Mount

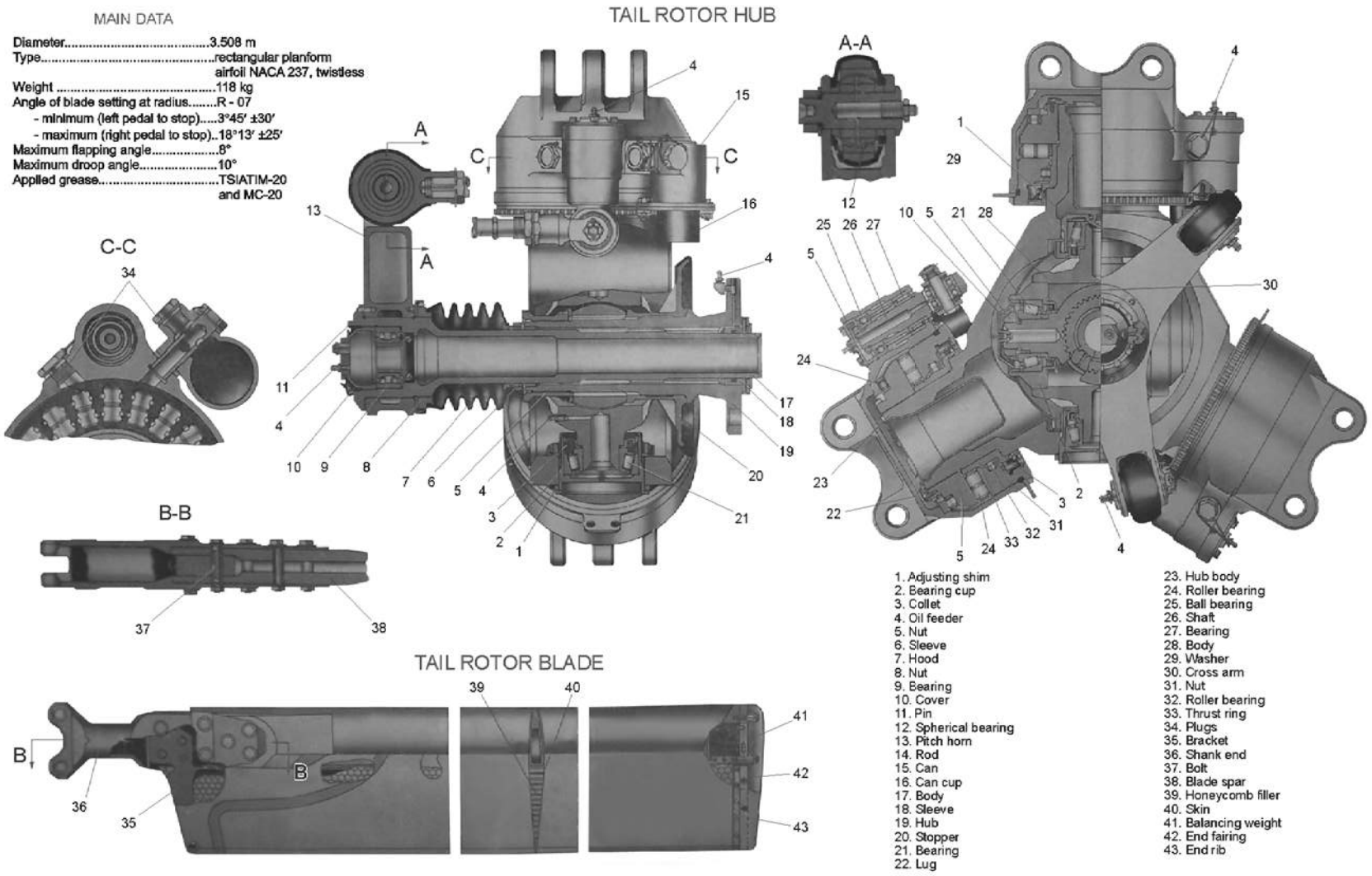


Fig. 19.65. Tail Rotor

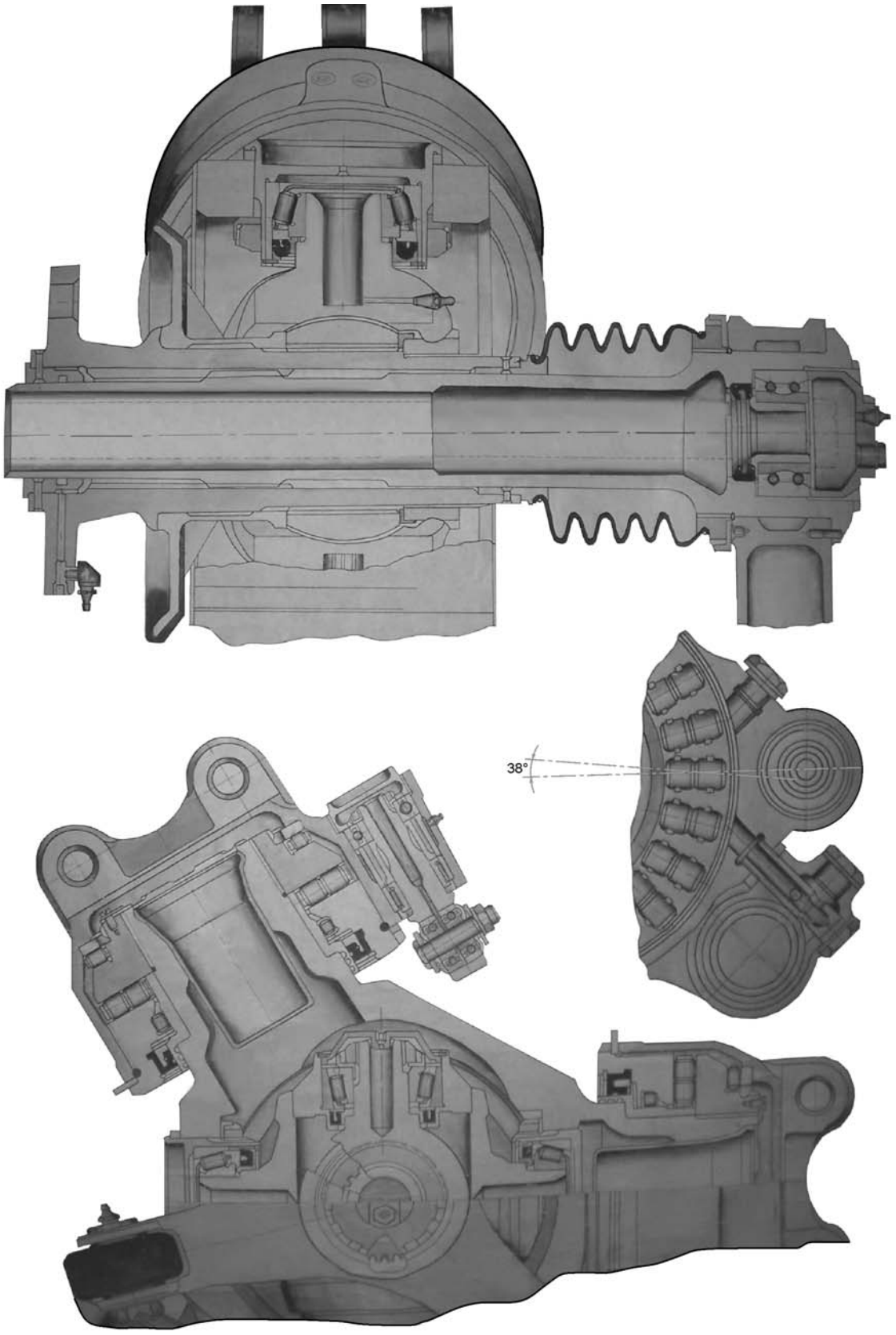


Fig. 19.66. Tail Rotor Hub

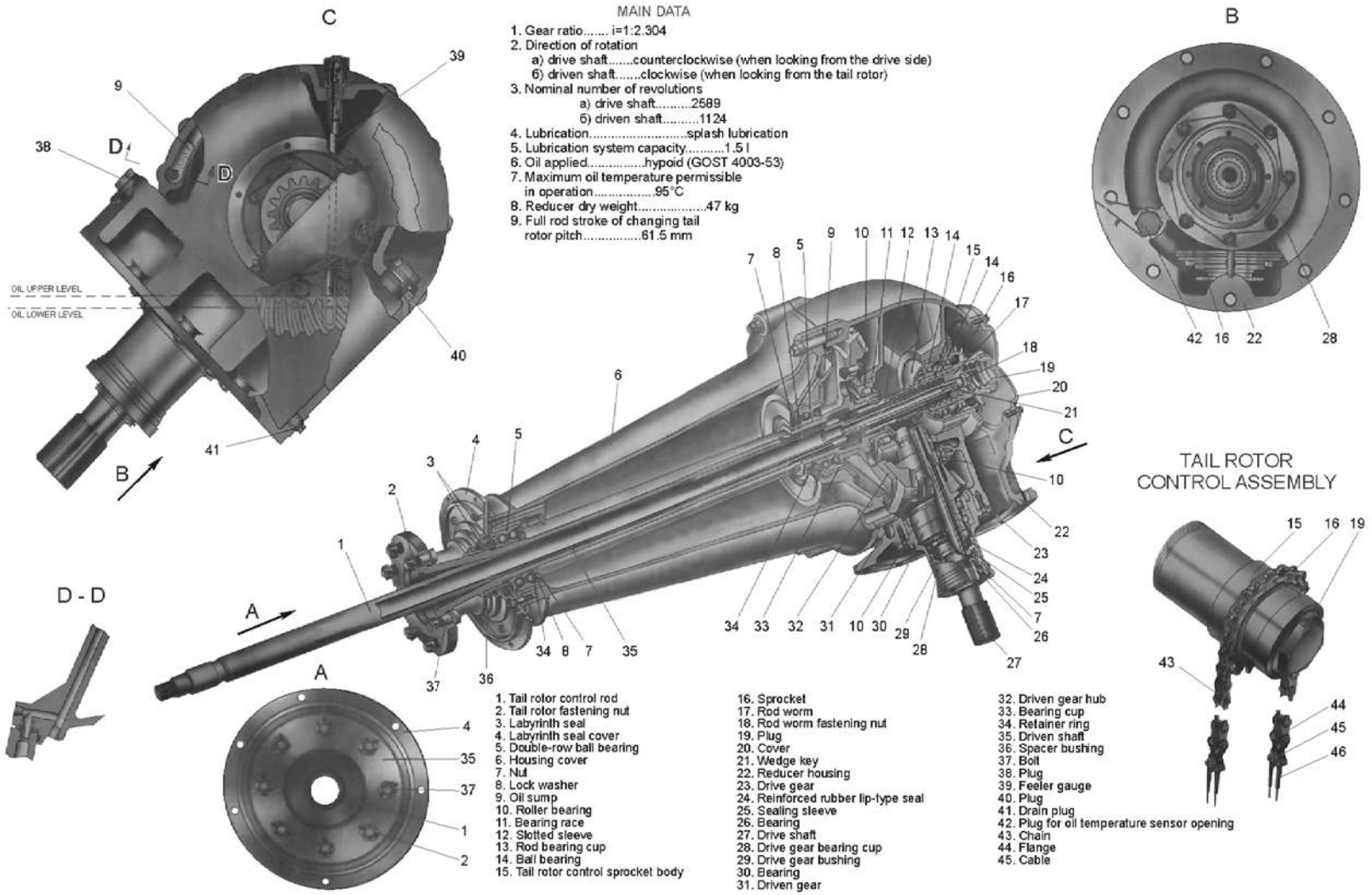


Fig. 19.67. Tail Gearbox

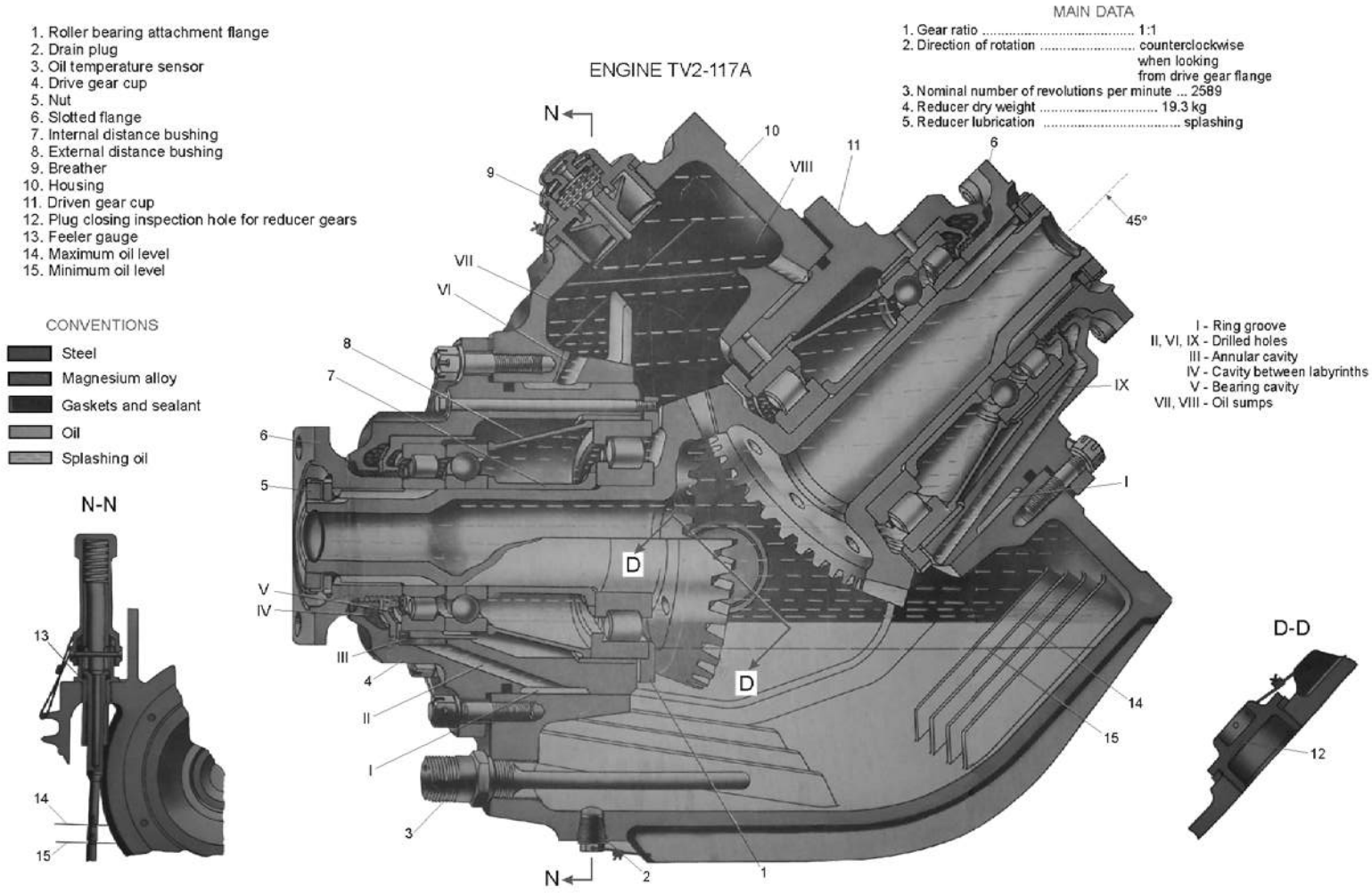


Fig. 19.68. Intermediate Gearbox

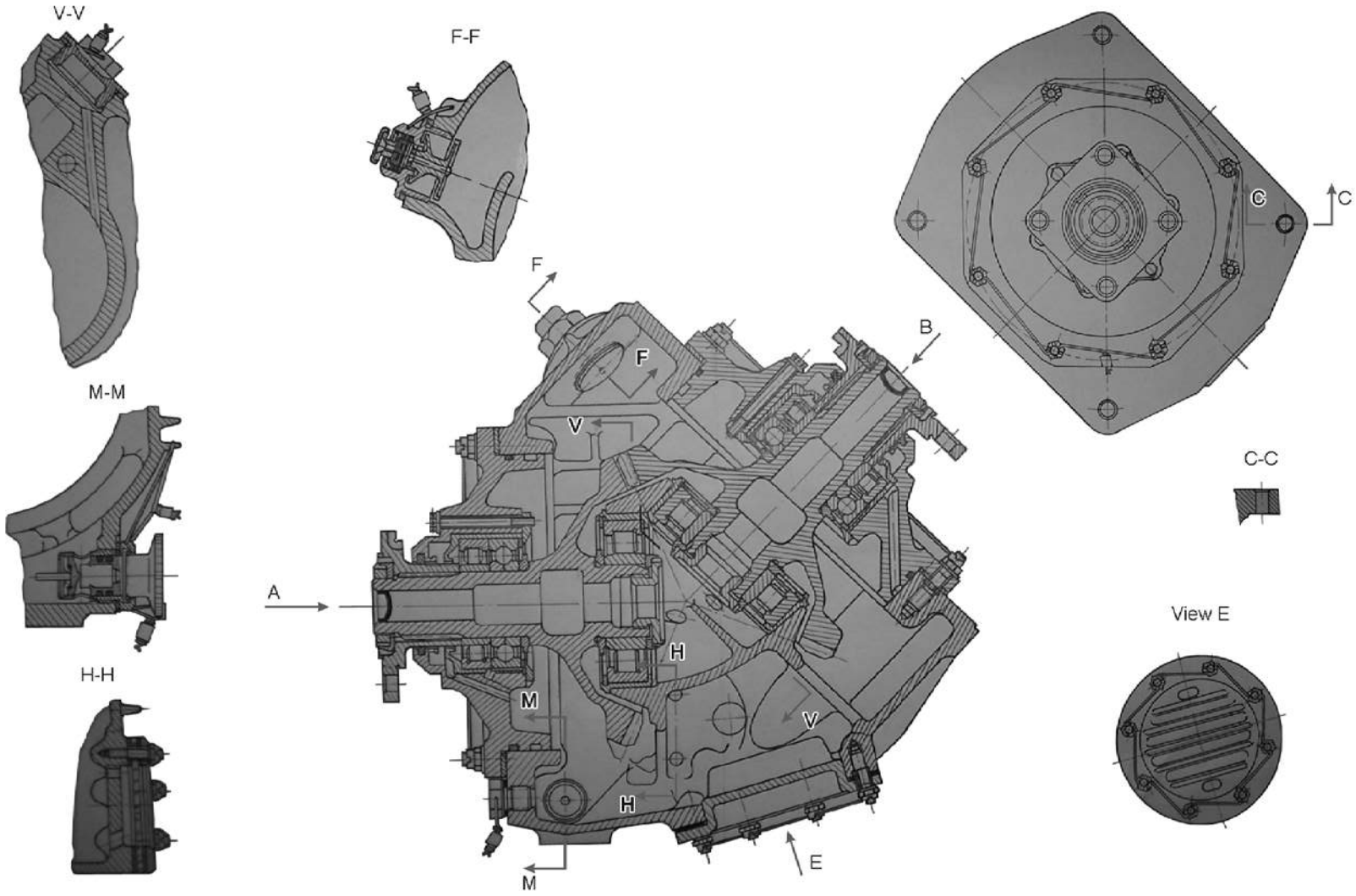


Fig. 19.69. Intermediate Gearbox

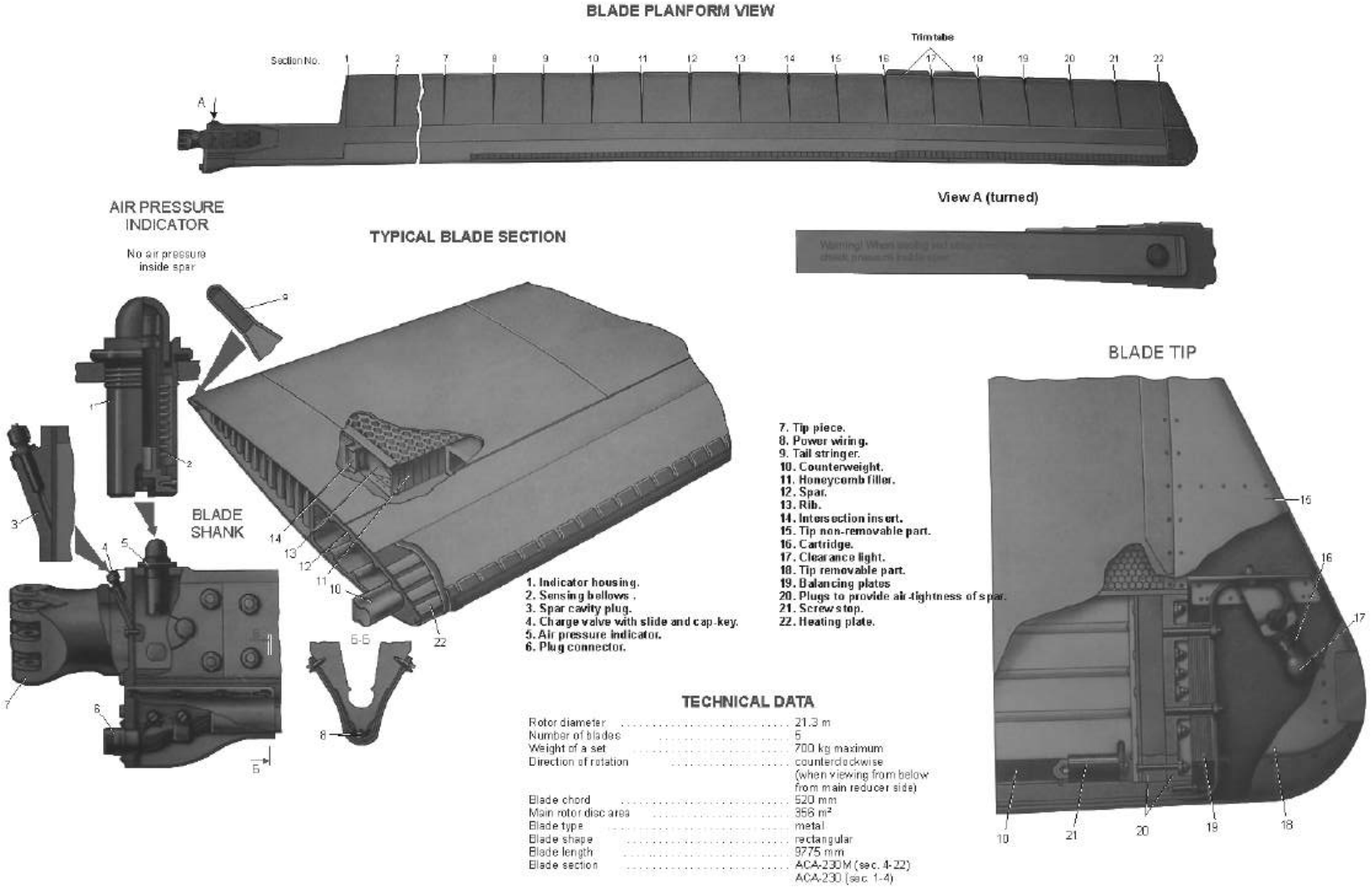


Fig. 19.70. Main Rotor Blade

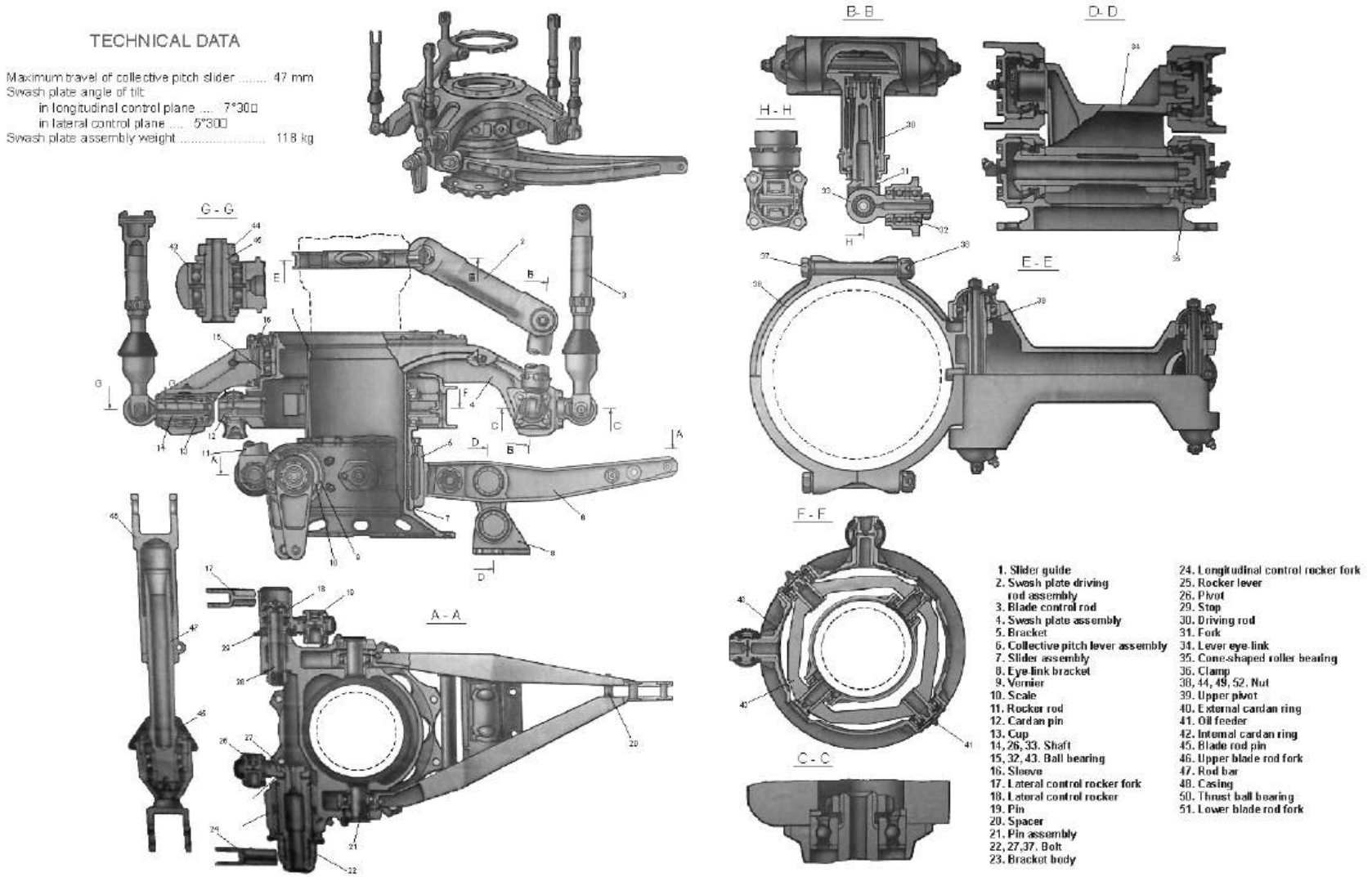


Fig. 19.71. Swash Plate Assembly

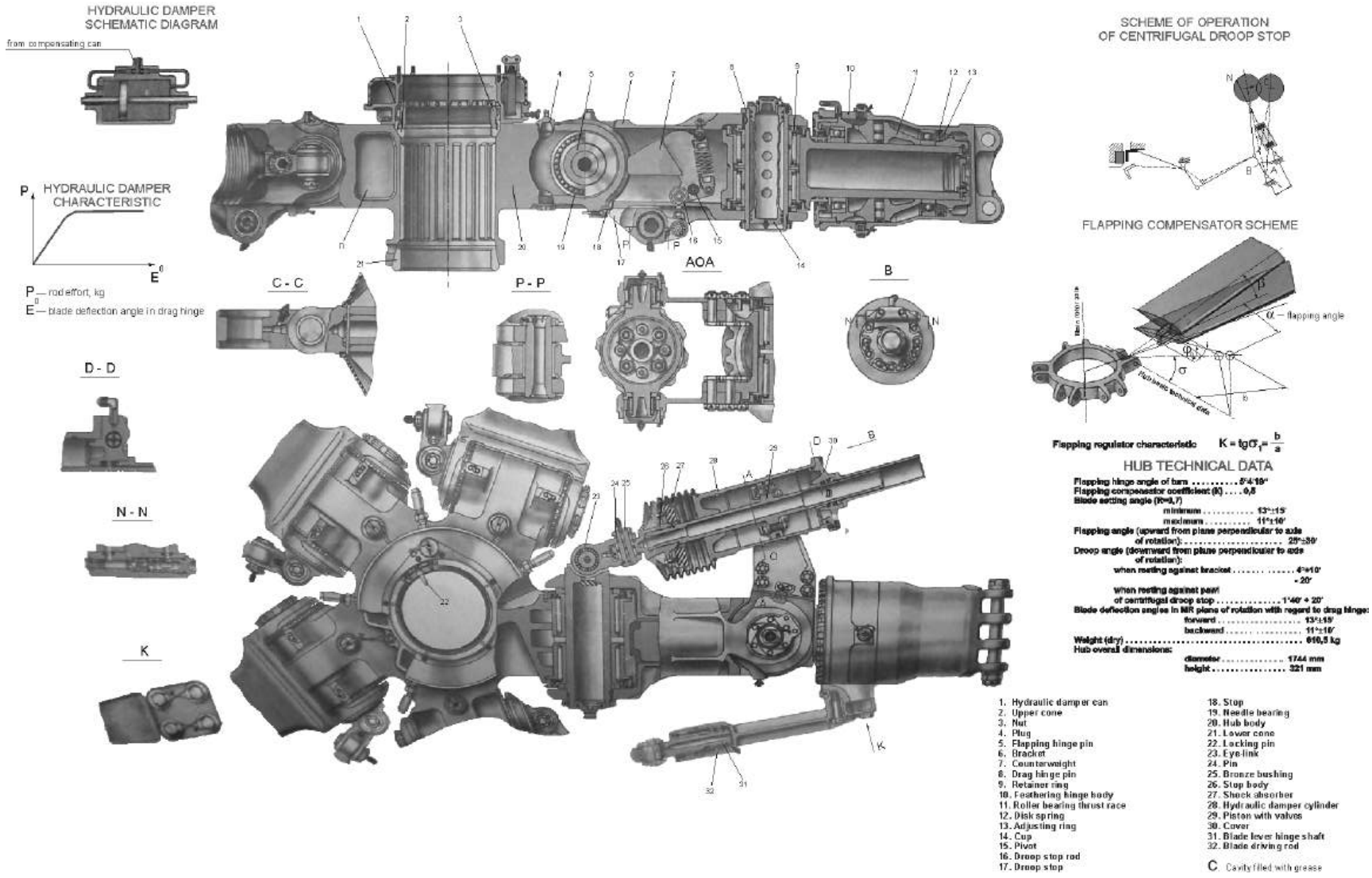


Fig. 19.72. Main Rotor Hub

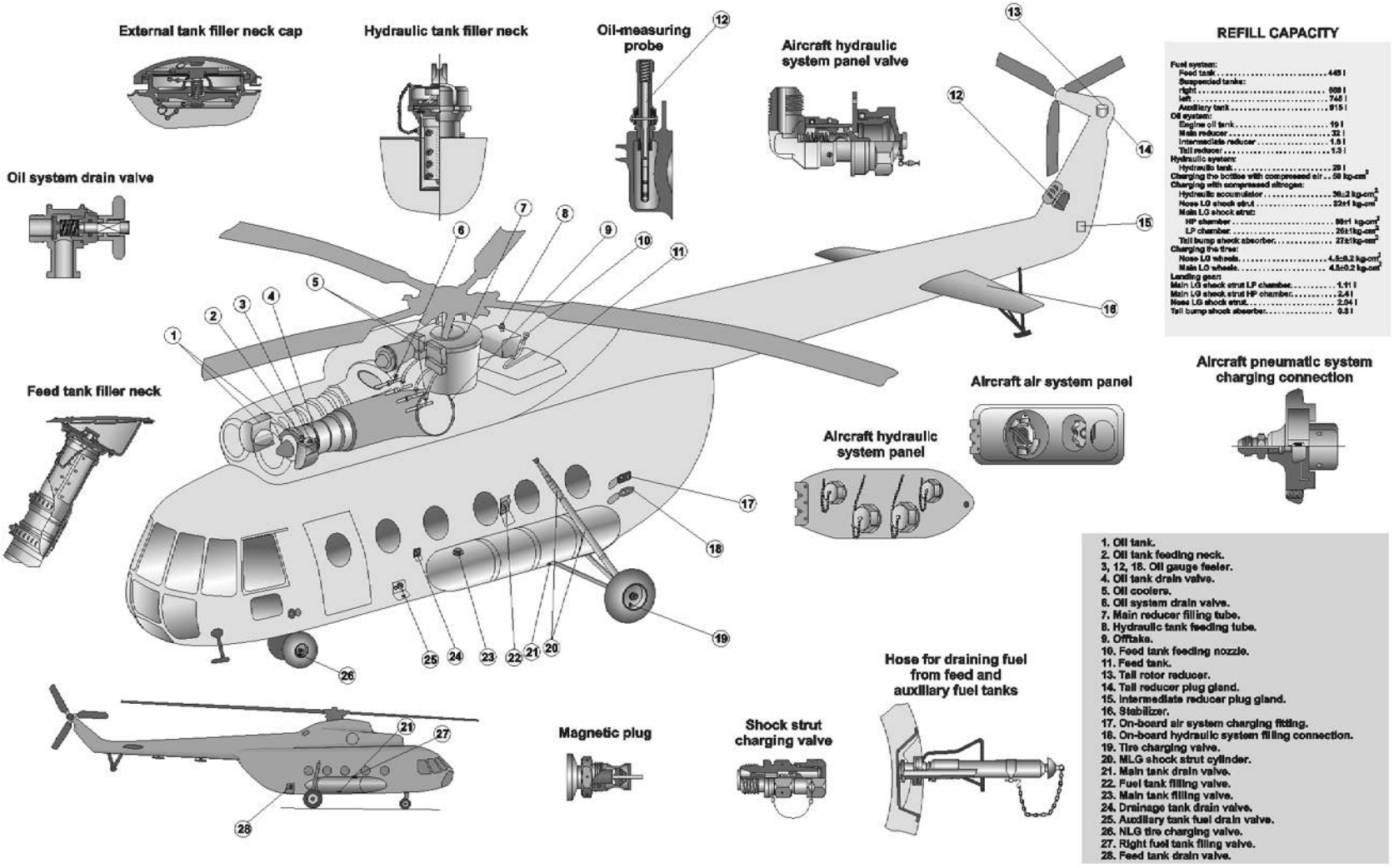


Fig. 19.73. Charge and Drain Units

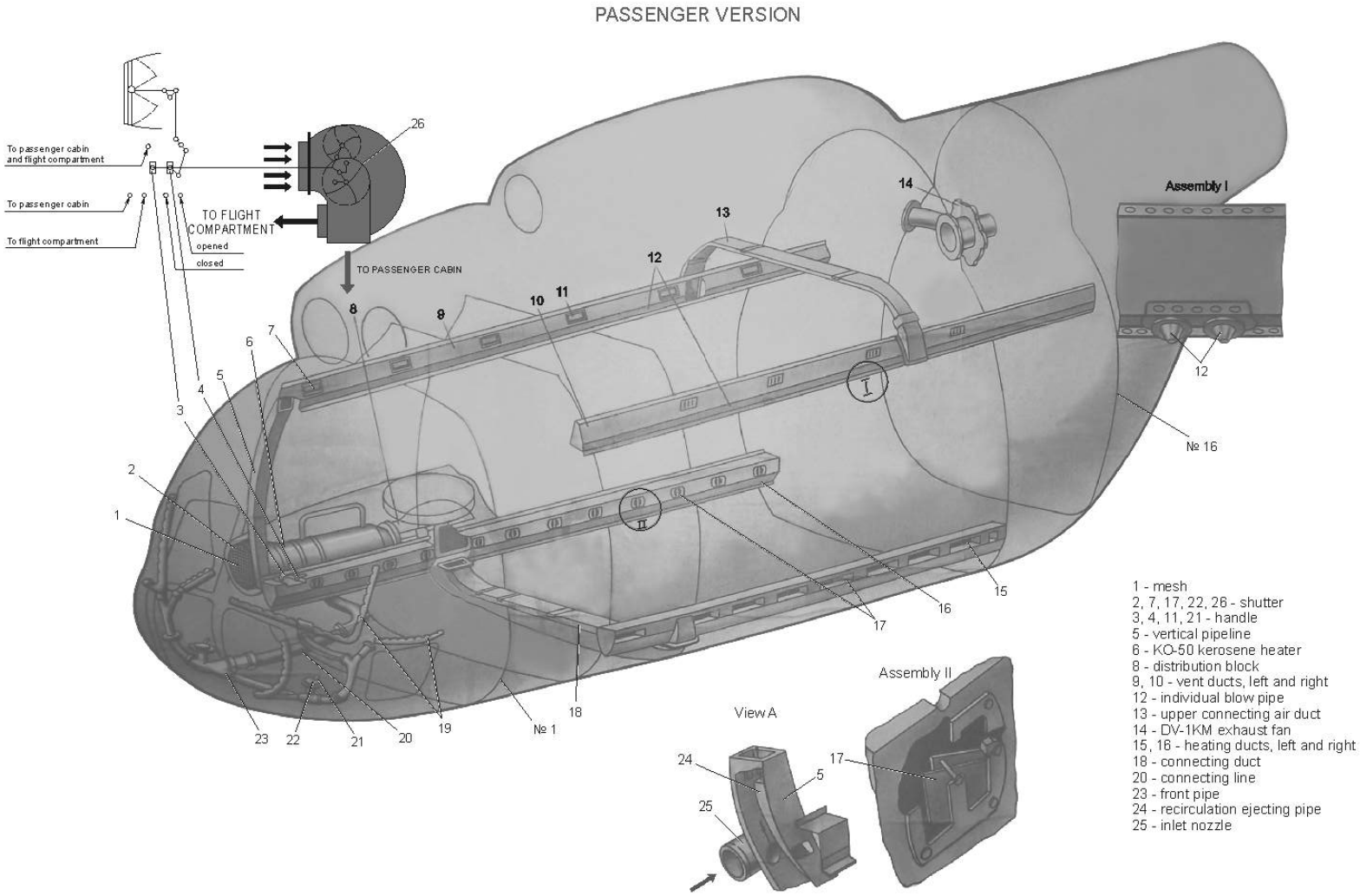
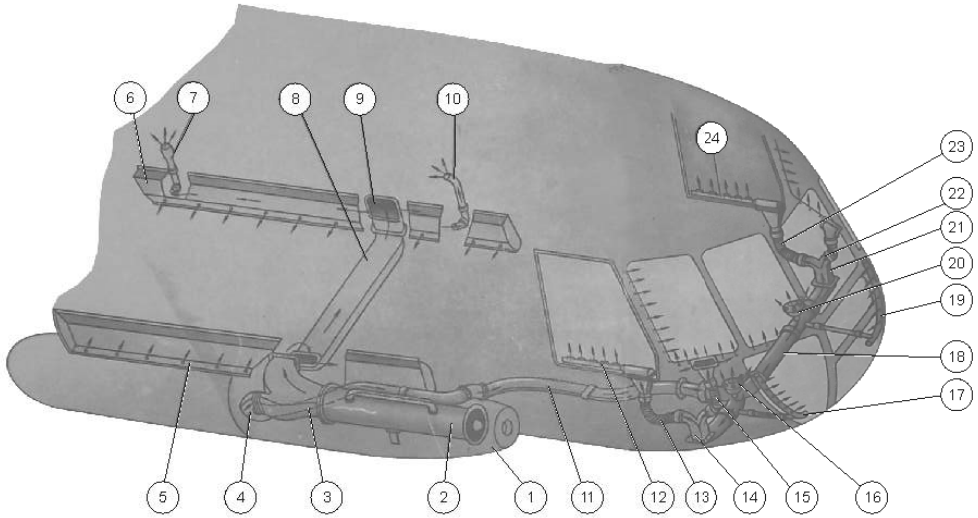
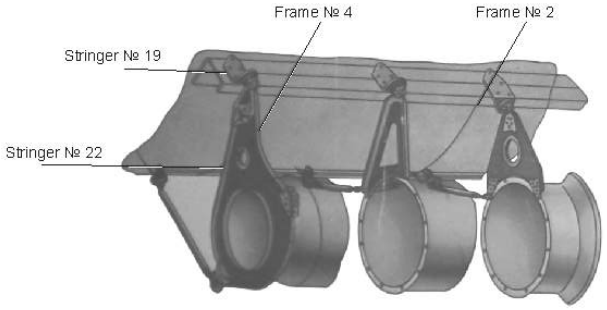


Fig. 19.74. Heating and Venting System

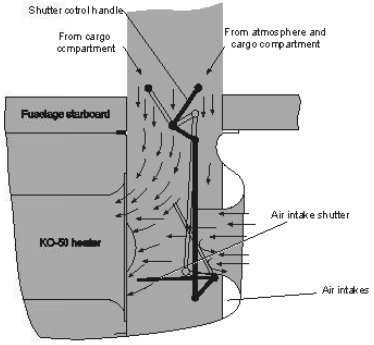


1. Air intakes.
2. KO-50 heater.
3. Distribution block.
4. Air duct.
5. Duct.
6. Duct.
7. Air system gravity filter heating pipe.
8. Air duct.
9. Pipe.
10. Fuel system drainage tank drain valve heating pipe.
11. Air duct.
12. Right movable blister window heating pipe.
13. Hose.
14. Pipe.
15. Hose.
16. Right pilot individual heating pipe.
17. Low right window heating pipe.
18. Air duct.
19. Low right window heating pipe.
20. Left pilot individual heating pipe.
21. Pipe.
22, 23. Hoses.
24. Left movable blister window heating pipe.

KO-50 HEATER ATTACHMENT SYSTEM



AIR INTAKE SHUTTER CONTROL SYSTEM



TECHNICAL DATA

Heater type	KO-50
Adjustment	automatic and manual
Heat productivity under automatic adjustment:	
minimum	5000 kCal·hr
average	35000 kCal·hr
maximum	50000 kCal·hr

Fig. 19.75. Heating and Venting System

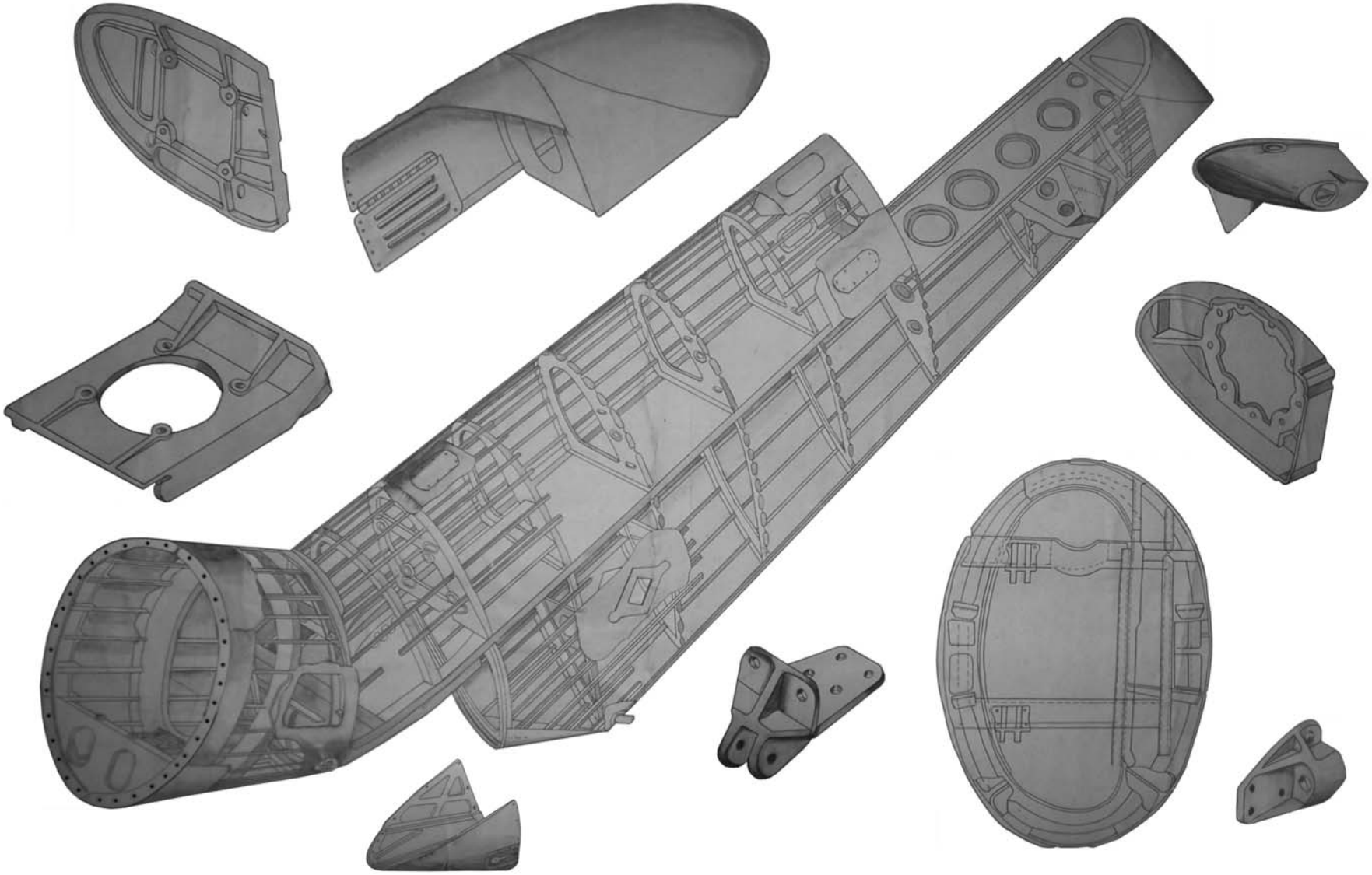


Fig. 19.76. Tail Pylon

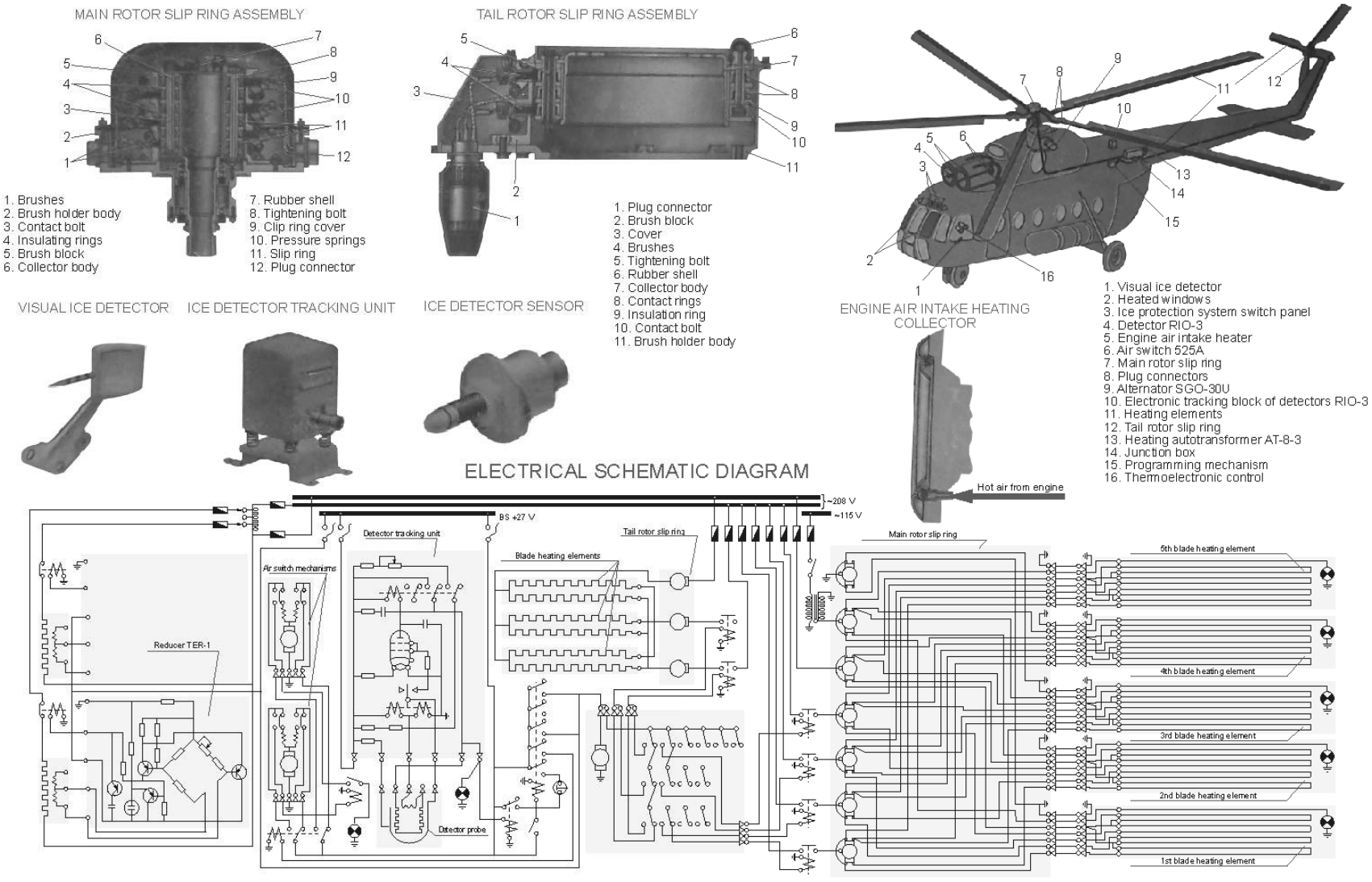


Fig. 19.77. Ice Protection System

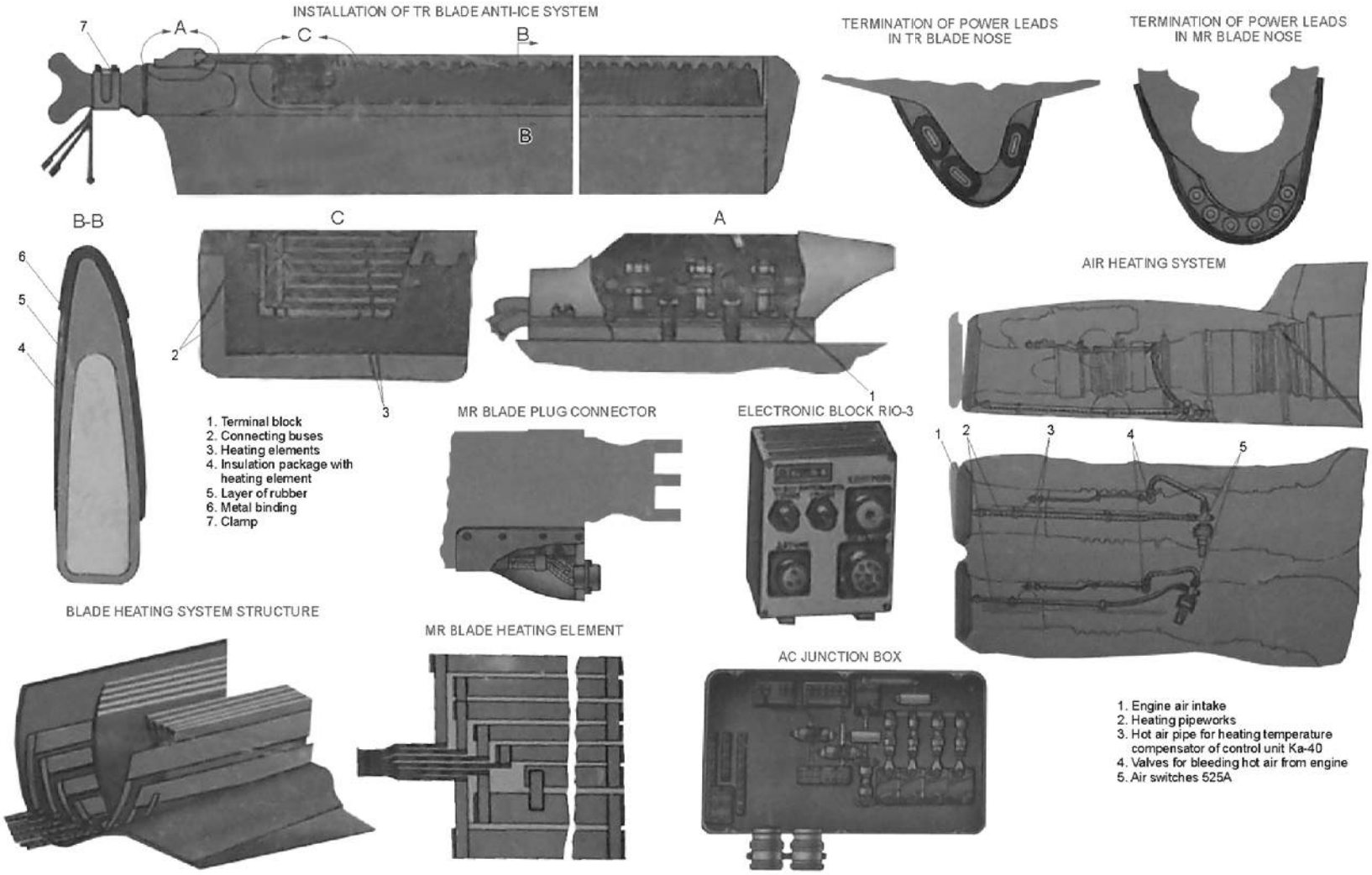


Fig. 19.78. Ice Protection System

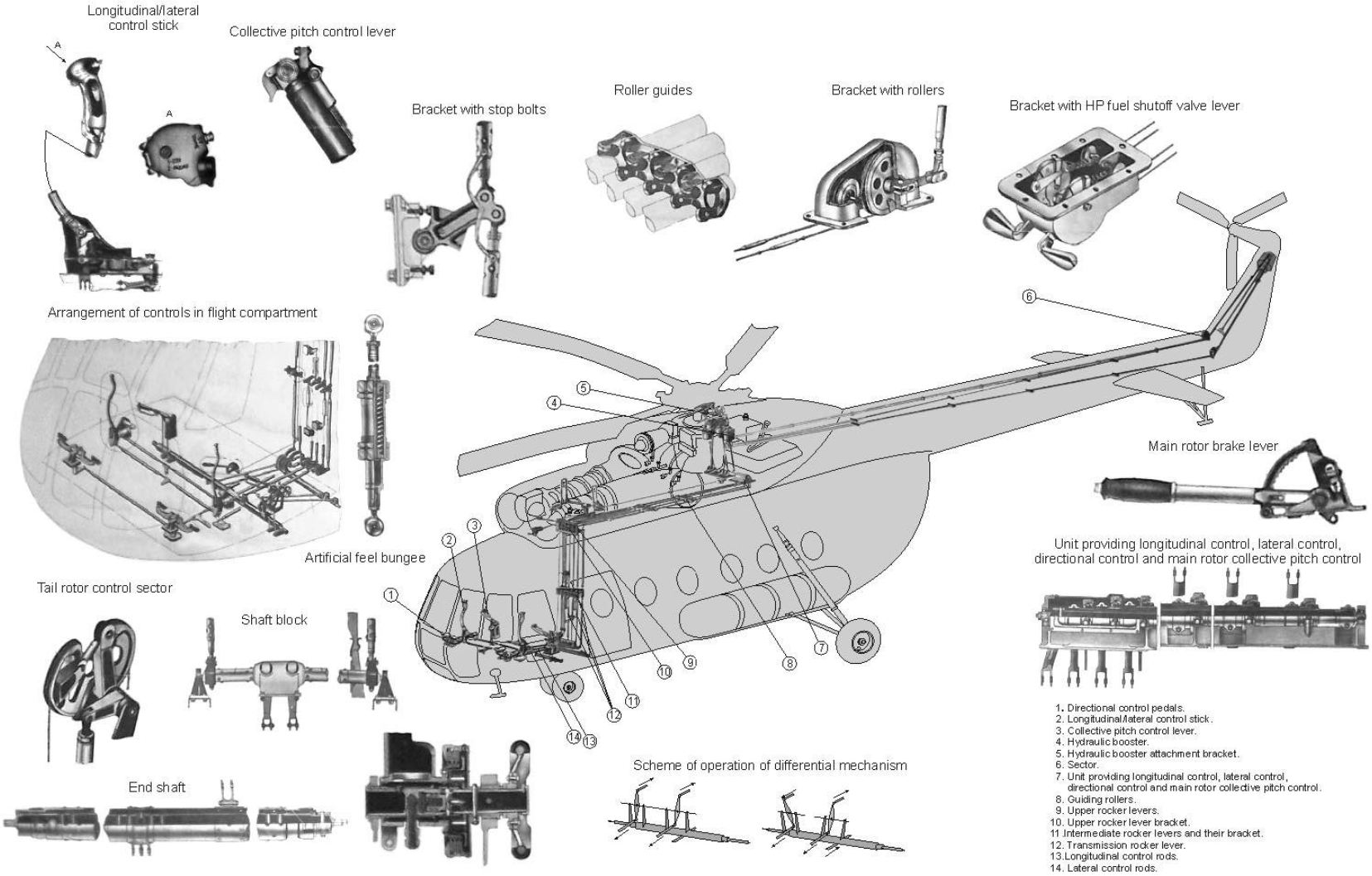


Fig. 19.79. Helicopter Control System

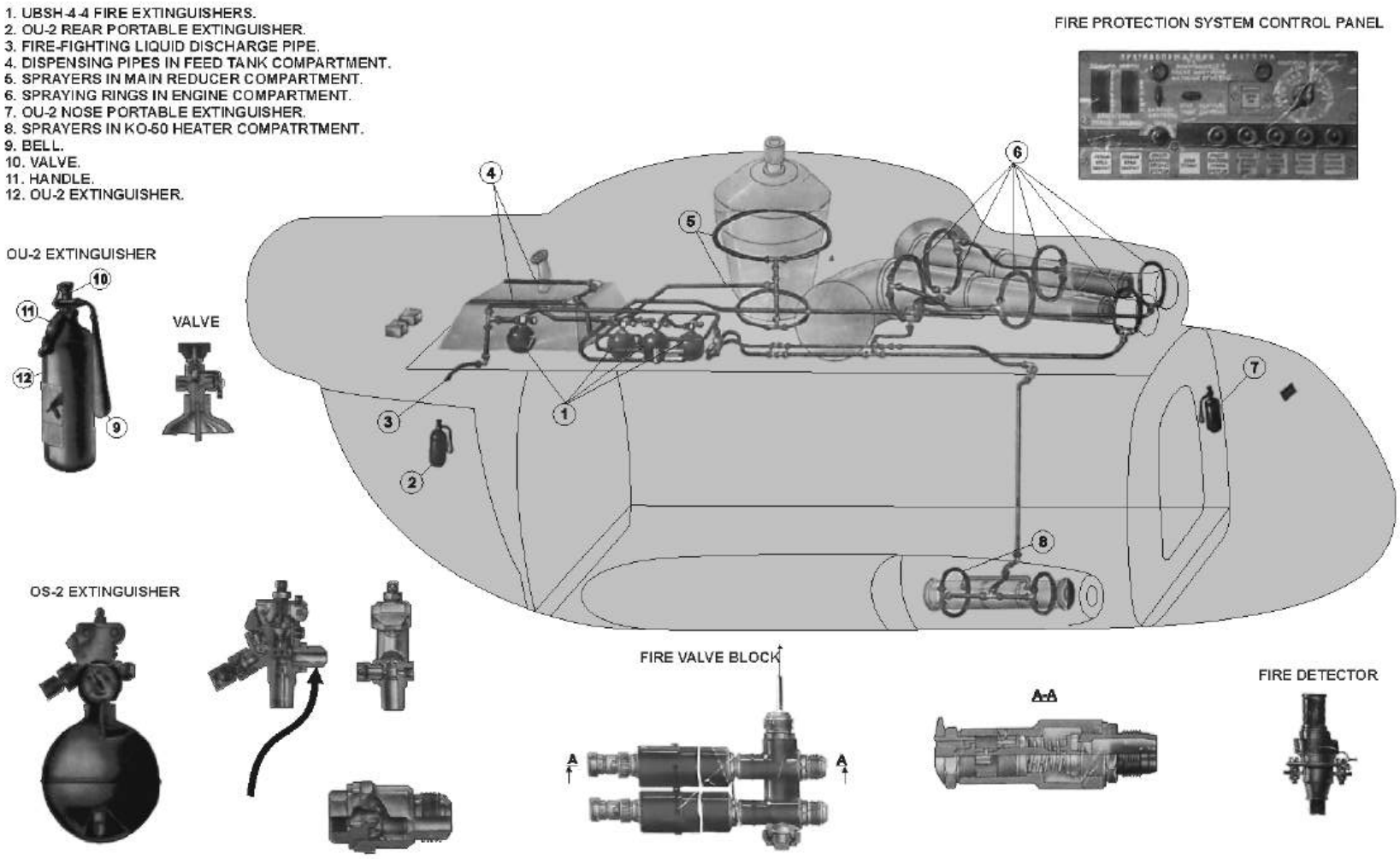


Fig. 19.80. Fire Protection System

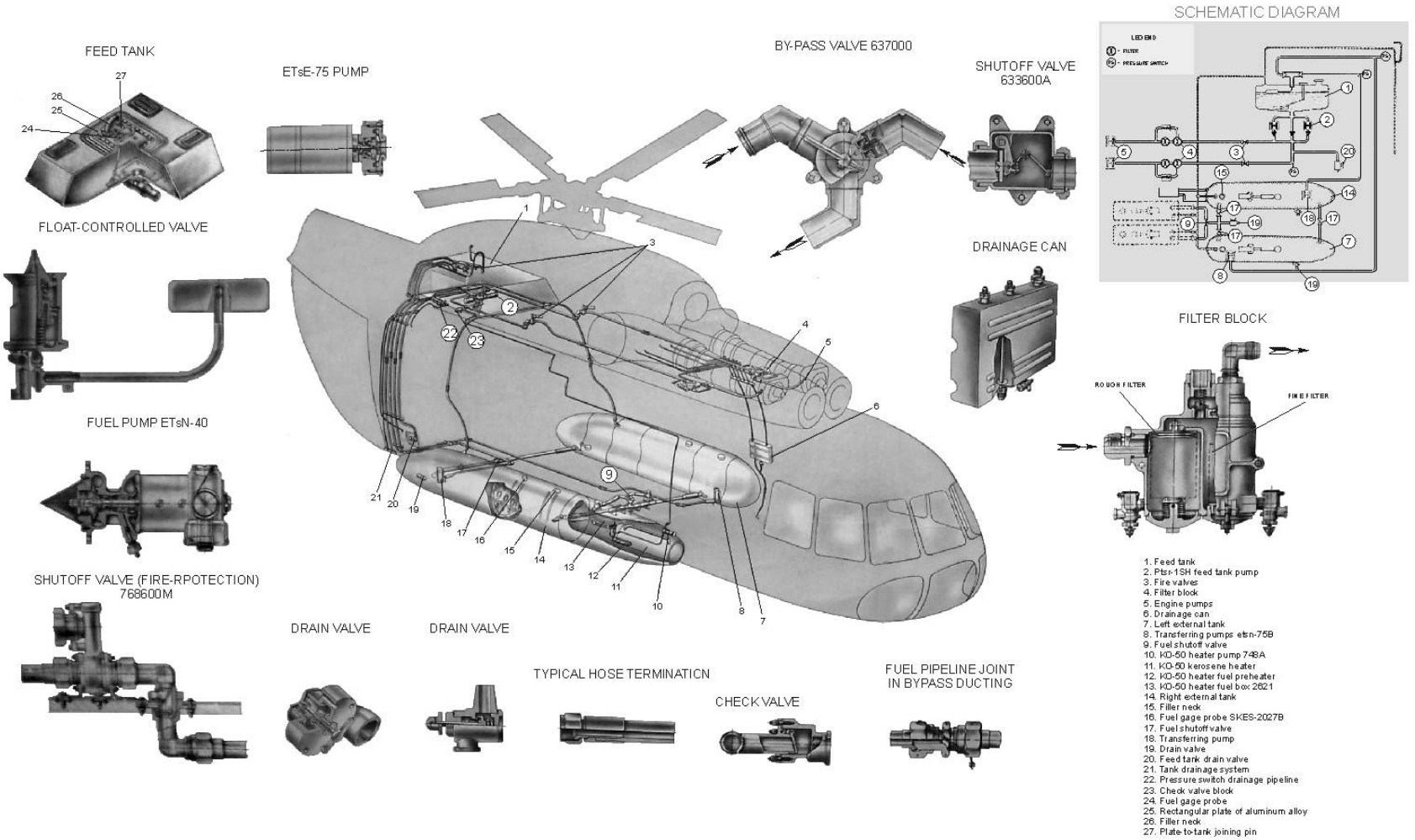
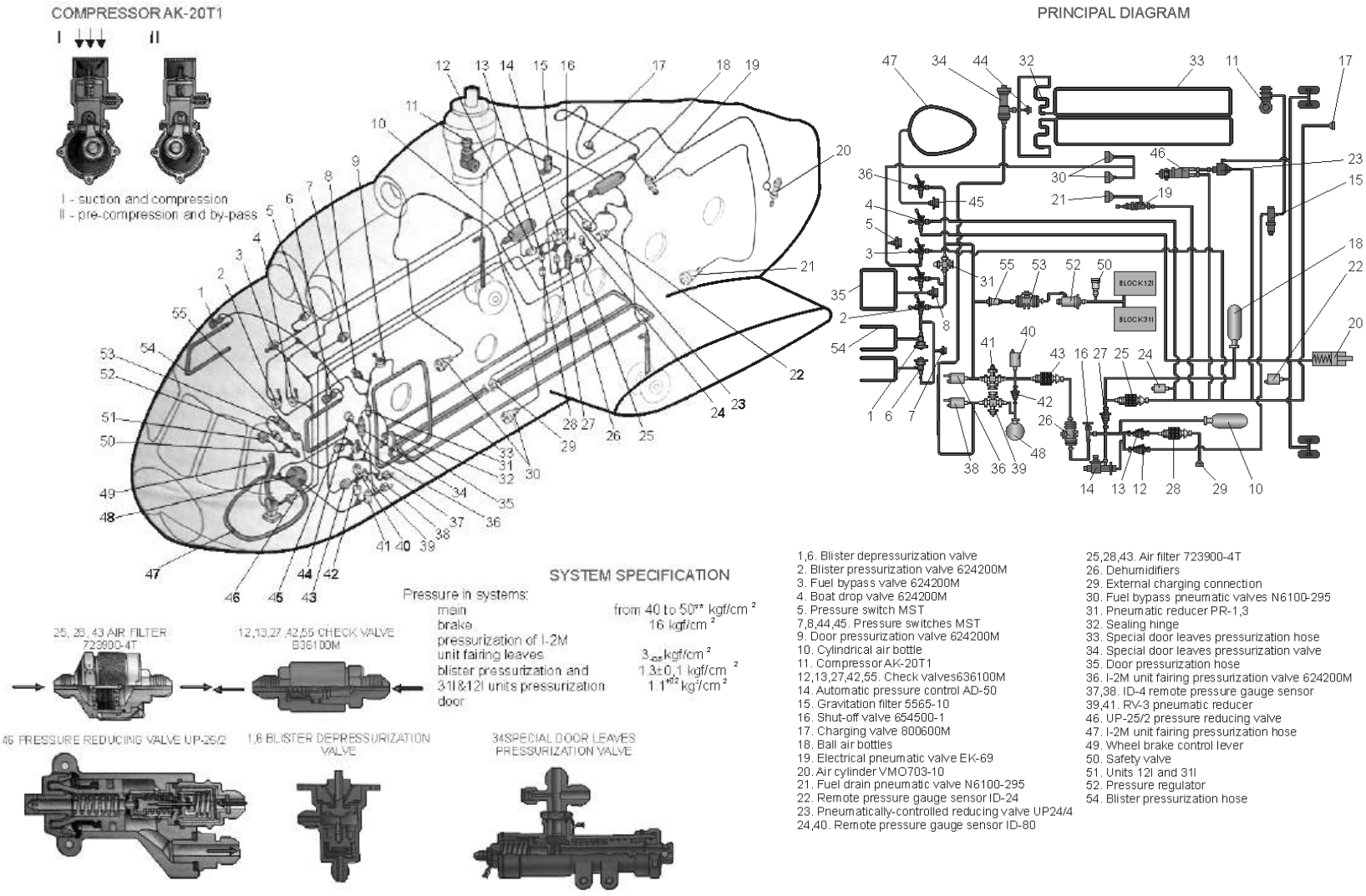


Fig. 19.81. Fuel System



1,6. Blister depressurization valve
2. Blister pressurization valve 624200M
3. Fuel bypass valve 624200M
4. Boat drop valve 624200M
5. Pressure switch MST
7,8,44,45. Pressure switches MST
9. Door pressurization valve 624200M
10. Cylindrical air bottle
11. Compressor AK-20T1
12,13,27,42,55. Check valves636100M
14. Automatic pressure control AD-50
15. Gravitation filter 5565-10
16. Shut-off valve 654500-1
17. Charging valve 800600M
18. Ball air bottles
19. Electrical pneumatic valve EK-69
20. Air cylinder VMO703-10
21. Fuel drain pneumatic valve N6100-295
22. Remote pressure gauge sensor ID-24
23. Pneumatically-controlled reducing valve UP24/4
24,40. Remote pressure gauge sensor ID-80
25,28,43. Air filter 723900-4T
26. Dehumidifiers
29. External charging connection
30. Fuel bypass pneumatic valves N6100-295
31. Pneumatic reducer PR-1,3
32. Sealing hinge
33. Special door leaves pressurization hose
34. Special door leaves pressurization valve
35. Door pressurization hose
36. I-2M unit fairing pressurization valve 624200M
37,38. ID-4 remote pressure gauge sensor
39,41. RV-3 pneumatic reducer
46. UP-25/2 pressure reducing valve
47. I-2M unit fairing pressurization hose
49. Wheel brake control lever
50. Safety valve
51. Units 12I and 31I
52. Pressure regulator
54. Blister pressurization hose

Fig. 19.82. Pneumatic System

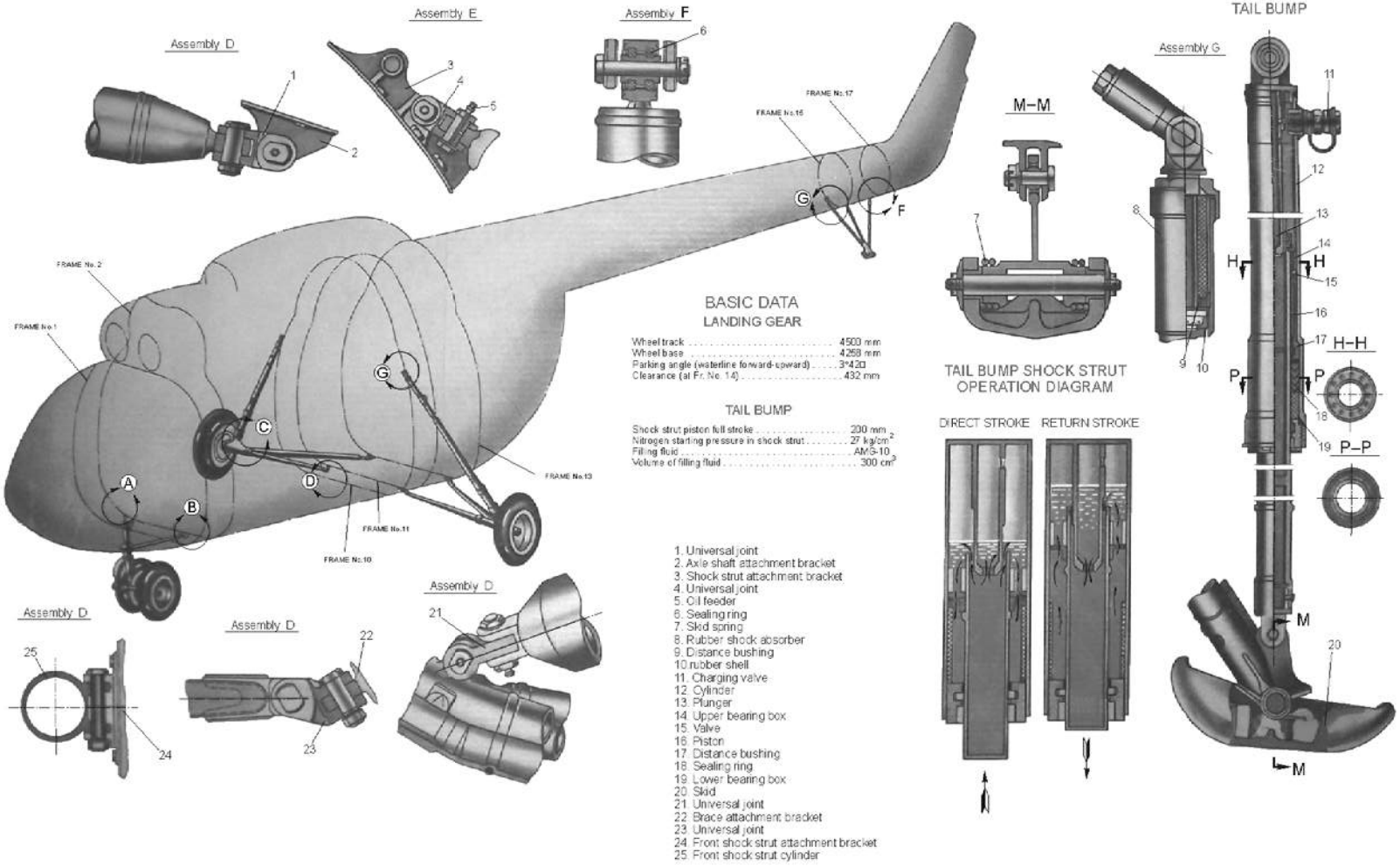


Fig. 19.83. Landing Gear

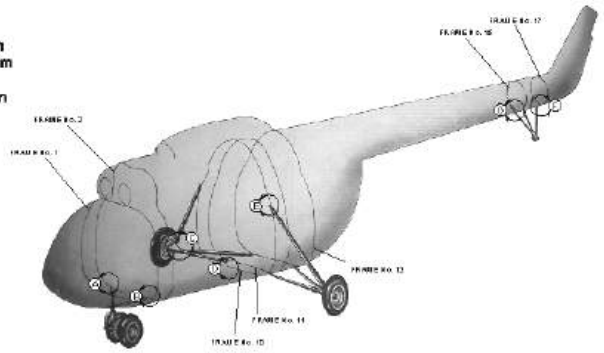
MAIN DATA

LANDING GEAR

Wheel track ..... 4500 mm
Wheel base ..... 4256 mm
Parking angle (waterline forward-upward) ..... 3°42'
Clearance (at Fr. No. 14) ..... 432 mm

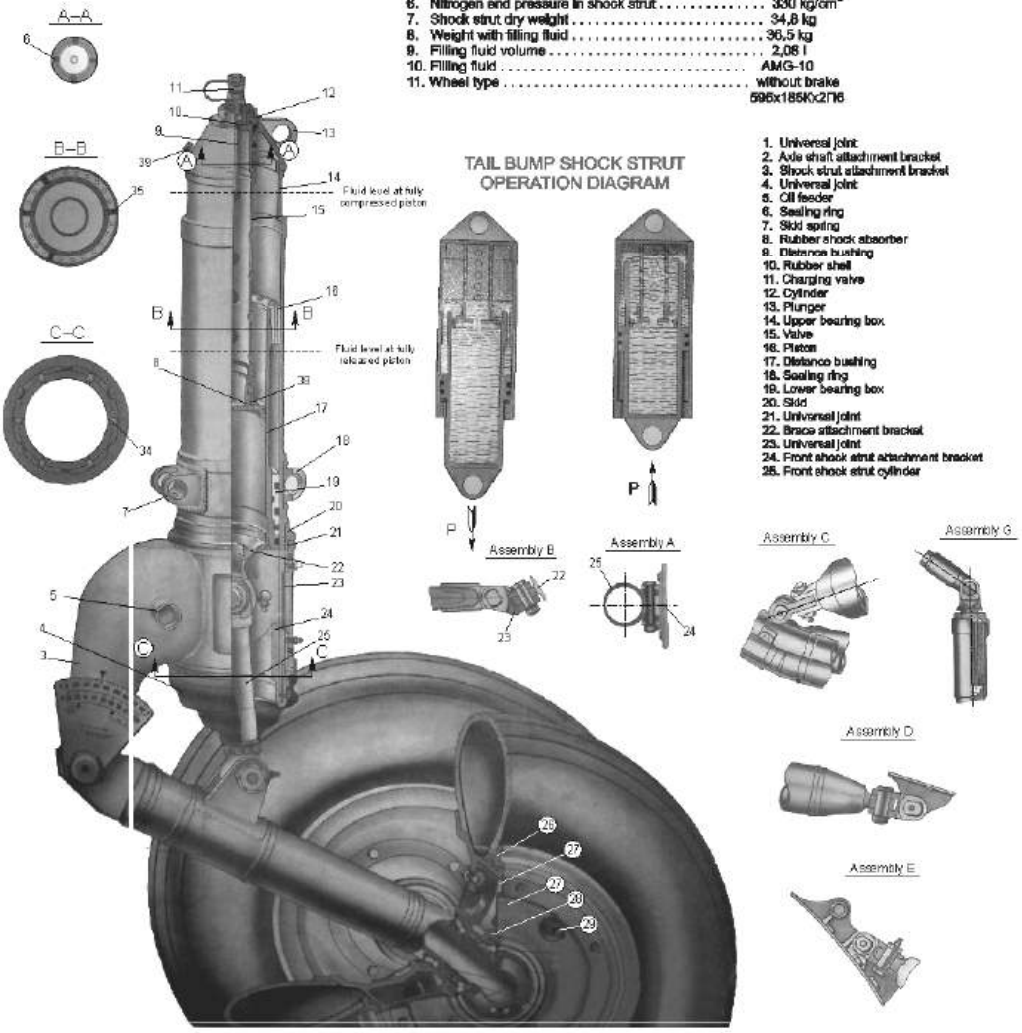
TAIL BUMP

Shock strut piston full stroke ..... 200 mm
Nitrogen starting pressure in shock strut .. 27 kg/cm²
Filling fluid AMG-10
Volume of filling fluid ..... 300 cm³



MAIN DATA

1. Shock strut piston working stroke (design) ..... 160 mm
2. Shock strut piston full stroke (to stop) ..... 165 mm
3. Shock strut piston parking stroke (takeoff weight) ... 130 mm
4. Nitrogen starting pressure in shock strut ..... 32 kg/cm²
5. Nitrogen parking pressure in shock strut ..... 108 kg/cm²
6. Nitrogen end pressure in shock strut ..... 330 kg/cm²
7. Shock strut dry weight ..... 34,8 kg
8. Weight with filling fluid ..... 36,5 kg
9. Filling fluid volume ..... 2,08 l
10. Filling fluid AMG-10
11. Wheel type ..... without brake 595x185Kx2P6



1. Universal joint
2. Axle shaft attachment bracket
3. Shock strut attachment bracket
4. Universal joint
5. Oil feeder
6. Sealing ring
7. Skid spring
8. Rubber shock absorber
9. Distance bushing
10. Rubber shell
11. Charging valve
12. Cylinder
13. Plunger
14. Upper bearing box
15. Valve
16. Piston
17. Distance bushing
18. Sealing ring
19. Lower bearing box
20. Skid
21. Universal joint
22. Brace attachment bracket
23. Universal joint
24. Front shock strut attachment bracket
25. Front shock strut cylinder

Fig. 19.84 Landing Gear

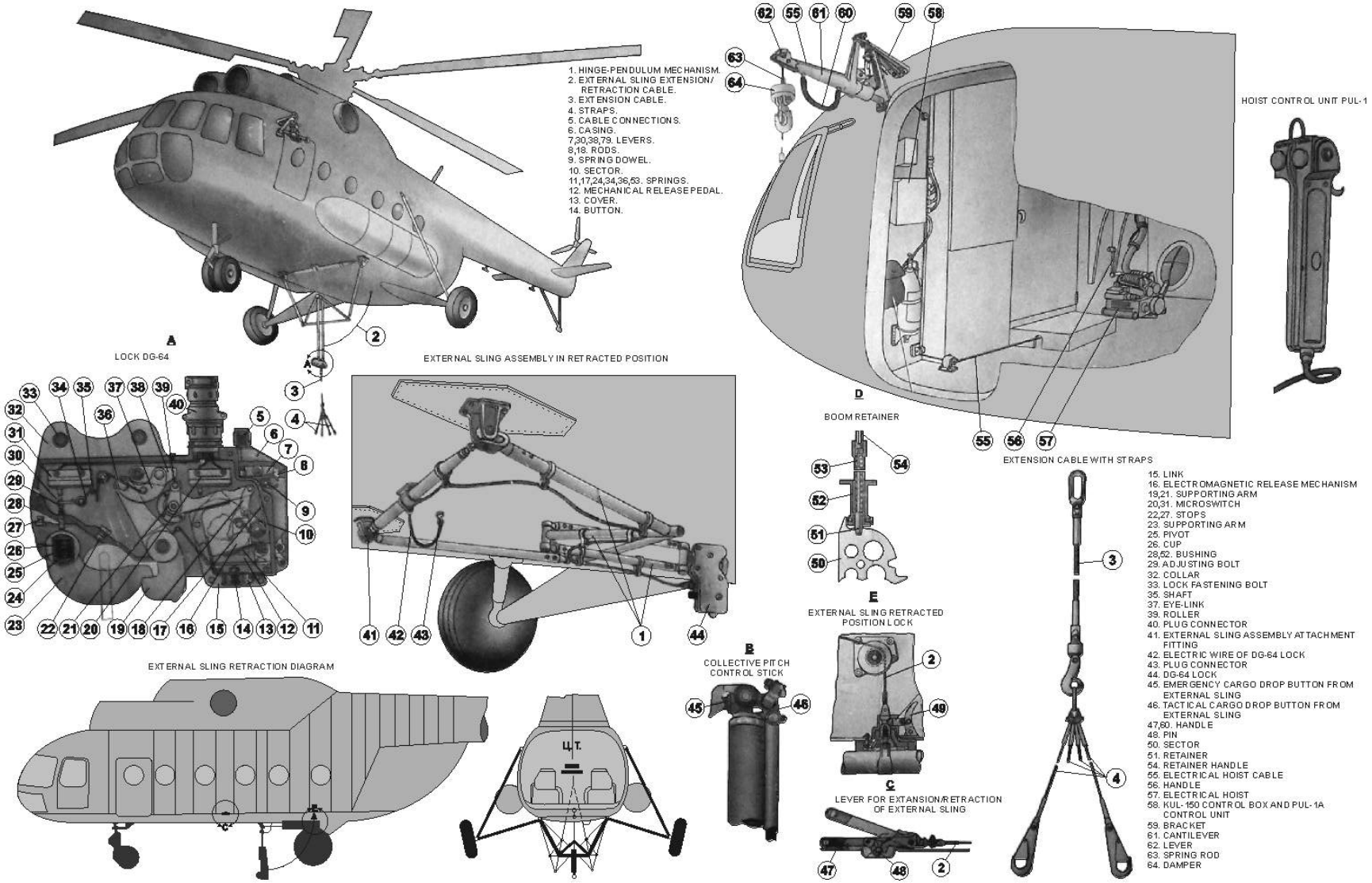


Fig. 19.85. External Sling Assembly and Jig

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