1 CLASSIFICATION, DESIGNATION, PROPERTIES, GENERAL STRUCTURE AND REQUIREMENTS FOR THE DESIGN OF VEHICLES

1.1 Classification and designation of cars

Vehicle rolling stock (vehicles) are divided by purpose into cargo, passenger and special. Freight rolling stock includes trucks, tractors, trailers and semitrailers.

Freight rolling stock, depending on the nature of its use, is divided into general purpose rolling stock and specialized rolling stock. Cars, trailers and semi-trailers of general purpose have a non-overturning body and are used for transportation of all types of cargo, except for liquids, without containers. Specialized vehicles include cars, trailers and semi-trailers designed for the transportation of certain types of cargo or equipped with special loading and unloading devices: dump trucks, self-loaders, vans and refrigerators, tanks, pipe carriers, metal carriers, container carriers , weight trucks, timber trucks, for transportation of construction structures, agricultural products and others Specialization of rolling stock is achieved mainly by installing specialized bodies and additional equipment on the chassis of basic cars, trailers and semi-trailers. At the same time, the type of body and its design depend mainly on the type of transported cargo, its properties and weight and dimensions parameters. Improvement of the design can be achieved by maximum use of the load carrying capacity of the chassis, avoidance of external influence on the quality of goods in the process of transportation, and improvement of the convenience and efficiency of loading and unloading operations.

Vehicles intended for permanent work with trailers or semi-trailers are called tractor vehicles. They are divided into truck tractors for work with semi-trailers and tractor cars for work with trailers. A truck tractor coupled with a trailer (semi-trailer) is called an articulated truck.

Passenger rolling stock includes buses, cars, passenger trailers and semi-trailers.

Special rolling stock (special vehicles) includes cars, trailers and semi-trailers and is intended for the performance of technological operations rather than transport work. Therefore, depending on the purpose, special equipment is installed on it - fire trucks, mobile cranes, mobile repair shops, etc.

Trucks (normal OH 025270) are divided into classes (table 1.1) according to gross weight and operational purpose (on-board, tractor, dump truck, etc.).

Table 1.1 – Indices of trucks

Full weight, i.e	Operational purpose of the car					
	On board	Tractors	Itself-resets	Tanks	Vans	Special
Up to 1.2	13	14	15	16	17	19
1.2 to 2.0	23	24	25	26	27	29
2.0 to 8.0	33	34	35	36	37	39
8.0 to 14.0	43	44	45	46	47	49
14.0 to 20.0	53	54	55	56	57	59
20.0 to 40.0	63	64	65	66	67	69
more than 40	73	74	75	76	77	79

Classes 18, 28, ..., 78 are reserve and are not included in the indexation.

The marking scheme for trucks, trailers and semi-trailers is shown in fig. 1.1.

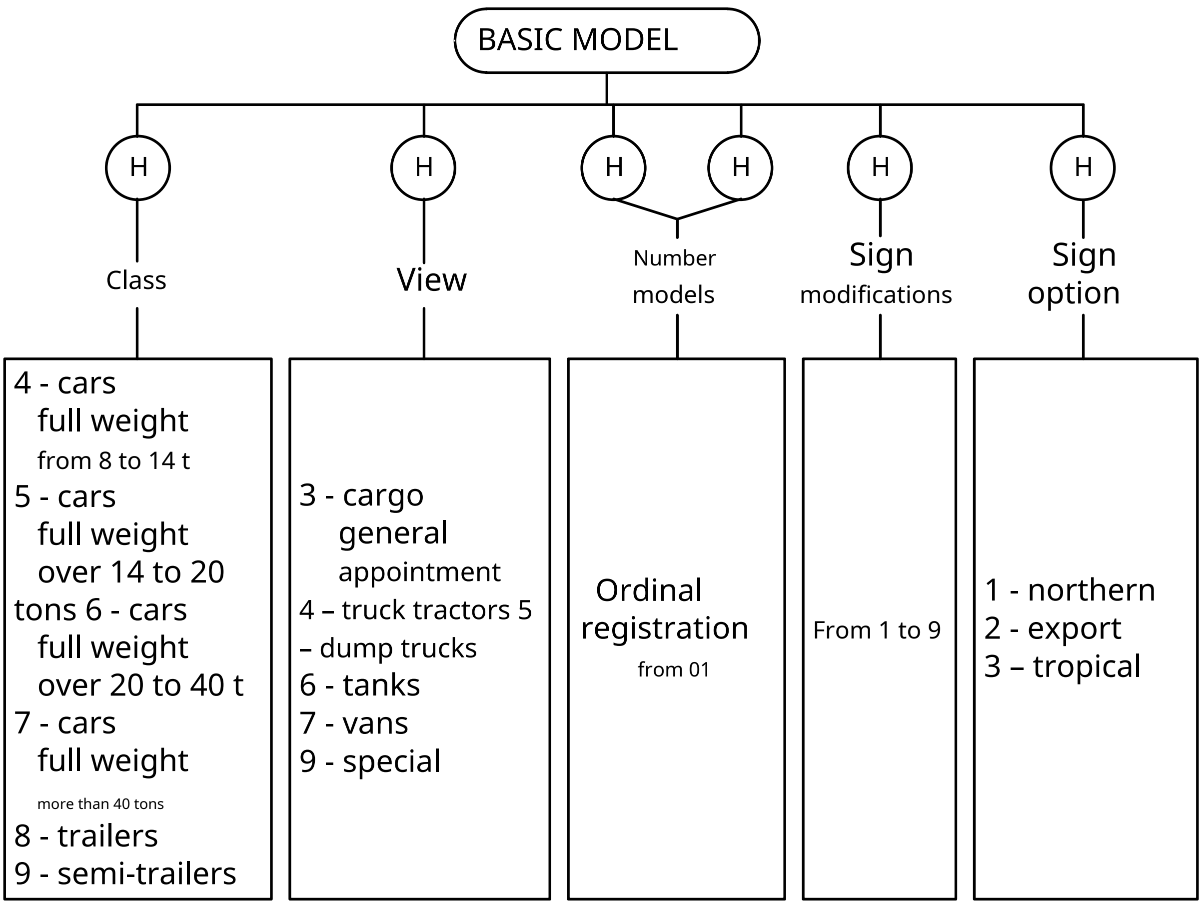


Figure 1.1 – Scheme of classifications and designations

For example, the designation ZIL-431413 is deciphered as follows: 4 - gross weight 8...14 t; 3 – general purpose trucks, 14 – model number, 1 – modification sign, 3 – tropical.

Passenger cars are classified according to the working volume of the engine in liters into 4 classes (from 11 to 41): 11 – especially small (up to 1.2), 21 – small (from 1.2 to 1.8), 31 – medium (from 1.8 to 3.5), 41 – large (more than 3.5 l).

Buses are classified by overall length in meters into five classes (from 22 to 62): 22 – especially small (up to 5 m), 32 – small (from 6.0 to 7.5 m), 42 – medium (from 8 up to 9.5 m), 52 – large (from 10.5 to 12 m) and 62 – especially large (16.5 m and more).

Indexes of trailers and semi-trailers are given in the table. 1.2.

Table 1.2 – Indices of trailers and semi-trailers

Types of trailers	Trailers	Semi-trailers
Lightweight	81	91
Bus	82	92
Cargo (board)	83	93
Dump trucks	85	95
Tanks	86	96
Vans	87	97

When considering the technical documentation of domestic and foreign motor vehicles, it is more convenient to use the classification adopted in the UNECE Rules (Table 1.3).

Table 1.3 – European classification of vehicles

Category ATZ	Type of motor vehicle	Full mass, i.e	Notes
M1	ATC with an engine, intended for the transportation of passengers, have no more than 8 seats (except for the driver's seat)	Do not rule-mented	Passenger cars
M2	The same having more than 8 seats (except the driver's seat)	Up to 5.0	Buses
M3	The same	Above 5.0	Buses, including articulated buses
N1	ATZ with an engine, intended for the transportation of goods	Up to 3.5	trucks, special cars
N2	The same	Over 3.5 to 12.0	Trucks- tractors, special vehicles cars
N3	The same	Over 12.0	The same
O1	ATZ without engine	Up to 0.75	Trailers and semi-trailers pegs
O2	The same	More than 0.75 up to 3.5	The same
O3	The same	Over 3.5 to 10.0	The same
O4	The same	Over 10.0	The same

1.2 Properties of cars

A car has a number of properties that characterize it not only as a vehicle intended for the transportation of goods, passengers and special equipment for performing non-transport works, but also as an object of safe movement for individual use (passenger car).

Usually, the following properties of cars are considered: operational, consumer and safety properties.

Operational properties characterize the car's performance of transport and special works. They determine the adaptability of the car to operating conditions, as well as the efficiency and convenience of using the car.

Operational properties of the car are divided into two main groups: operational properties related to the movement of the car and not related to its movement.

Traction, speed and braking properties, fuel economy, controllability, turning, maneuverability, stability, passability, smoothness of movement, environmental friendliness and traffic safety ensure the movement of the car and determine its regularities.

Capacity, strength, durability, adaptability to technical maintenance and repair, to loading and unloading operations, to boarding and disembarking passengers largely determine the efficiency and convenience of using a car.

The operational properties that ensure the car's movement depend significantly on the design and technical condition of the car's systems and mechanisms. The more perfect the design of the car and the better its technical condition, the higher the operational properties of the car. The systems and mechanisms of the car are designed in such a way as to acquire operational properties necessary for the given operating conditions and to ensure the effective use of the car in these conditions.

In fig. 1.2 shows the relationship between the operational properties that ensure the movement of the car and the systems and mechanisms of the car, the design and technical condition of which have the greatest influence on these properties.

Consumer properties are especially important for a passenger car, they characterize the car's ability to satisfy the demands of its owners. So, the consumer properties of a passenger car are: the convenience of boarding and disembarking the driver and passengers, the presence of effective systems of heating, ventilation, air conditioning, electric windows, audio system, built-in telephone, TV, as well as the quality of the materials for the interior of the body, the attractiveness of the exterior appearance of the car, its prestige and conformity to fashion.

Safety properties characterize the active, passive and environmental safety of the car.

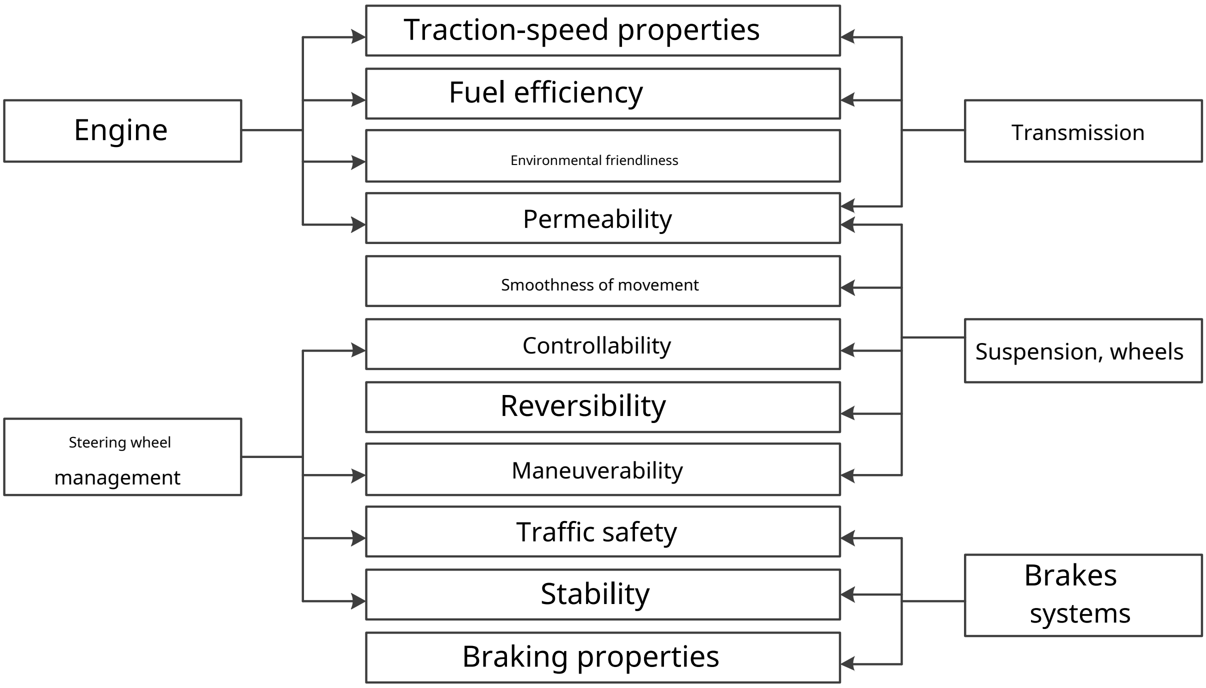


Figure 1.2 – Relationship of operational properties with systems and car mechanisms

Active safety is the ability of a car to prevent traffic accidents (traffic accidents). The active safety of the car is provided by its high traction-speed and braking properties, good stability, controllability, maneuverability, insufficient turning power, high smoothness of movement, good visibility and comfort, which sharply reduce driver fatigue and create conditions for long-term accident-free work.

Passive safety (internal and external) is the ability of a car to reduce the severity of the consequences of an accident, i.e. injuries to the driver, passengers and pedestrians, to ensure the safety of cargo and to prevent the possibility of a fire. The passive safety of the car is ensured by the high strength of the passenger compartment, which practically makes it impossible to deform it in the event of accidents, seat belts, inflatable airbags, trauma-safe steering, head restraints, special glass, reliable internal body equipment that reduces injuries to the driver and passengers , as well as the external shape of the body, which reduces injuries to pedestrians. The fire safety of the car is ensured by the design of the engine power system and the location of the fuel tank on the car.

Environmental safety is the ability of a car to reduce the damage caused during operation to passengers, the driver, people and the environment. The environmental safety of the car is ensured by the design of separate systems, mechanisms and their elements that reduce the noise created by the car and the toxicity of exhaust gases. In addition, environmental safety is achieved through the use of environmentally friendly materials and adaptive capacity of the car for disposal, i.e. reprocessing after failure of the car, its systems and mechanisms.

Safety parameters are regulated by state standards and UNECE Rules valid in Ukraine.

1.3 General structure and requirements for the design of cars

Key terms:

Mechanism (device, gear, mechanism) is a device designed to transform movement and speed.

Aggregate (unit) – connection of several devices into one whole. A system is a set of separate parts connected by a common function (for example, a power supply system, cooling system, lubrication system, etc.).

A car is a self-propelled machine that is set in motion with the help of an engine installed on it. Mechanisms, units and systems of a car contain thousands of parts. However, in most of the rolling stock, in which the car is the defining link, the principles of organization and operation of the main mechanisms are the same.

It consists of three main parts: engine, body and chassis. The engine converts thermal energy released during fuel combustion into mechanical energy. As a result of such a transformation, the drive wheels of the car are rotated through the transmission mechanisms. Most cars use piston engines - gasoline or diesel.

The body (body) serves to place the transported cargo. Passengers and a driver are placed in the body of a bus and a passenger car. The truck body consists of a platform (body floor, loading platform) for cargo and a driver's cabin.

Chassis (chassis, running gear*)*is a set of all mechanisms, intended for the transmission of torque from the engine to the driving wheels, vehicle movement and control. The chassis consists of a transmission, a supporting system, axles, suspension, wheels and control mechanisms.

Transmission (transmission; power drive) transmits the torque from the engine crankshaft to the driving wheels of the car and changes the magnitude and direction of this torque.

In fig. 1.3 shows a simplified diagram of the car (top view). From the engine *1*power is supplied to the driving co- forest*8.*The transmission consists of a clutch*2,*gearboxes*3,*gimbal transfers*4,*main gear5, the differential placed in it, and the axles *6.*

*clutch*intended for temporary disconnection of the box gears from the engine at the time of moving off and switching gears (with subsequent smooth connection to the engine).*Gearbox (gearbox)* serves to increase the torque obtained from the engine by engaging gears with different gear ratios. This ensures a change in the speed of the car. In addition, the gearbox is also used for reversing.

From the gearbox torque with the help *cardan transmission (cardan drive; driveline)* transferred to *main gear 5*, located in the rear axle, which can move relative to *frames* 9 during the deformation of the elastic element (spring) between the bridge and the frame. Cardan transmission transmits the torque from the gearbox to the rear axle at a variable angle and distance between them.

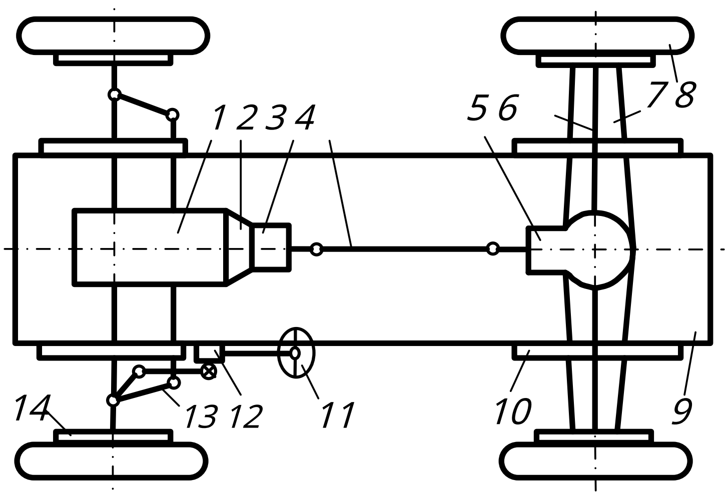


Figure 1.3 – Simplified scheme of the classic layout of the car

IN *main gear (final drive)* occurs further after the box gears for increasing the torque and transmitting motion at a right angle from the cardan shaft to *half axis 6* wheel

It is located together with the main transmission *differential (differential mechanism)*, which allows you to obtain different speeds of the wheels (on turns and when overcoming irregularities of different heights). *Half axis* are in the crankcase *7* leading (most often rear) bridge, and their outer ends are connected to driving wheels *8.*

frame *9,* which performs the functions of the supporting system in the considered scheme we, the wheels, are mounted on the driven and driven *7* bridges and suspensions *10*, which connect the bridges with the frame, form the car's undercarriage. The suspension and tires make it possible to soften shocks and impacts perceived by the wheels from road irregularities.

Controls include brake and steering systems. *Halmova system (braking system)* consists of brakes *14,* mounted on wheels, and reason for them. It serves to reduce speed, stop and hold the car in place.

Steering gear contains a steering wheel *11*, steering gear *12* and levers *13*. In this system, turning the steering wheel changes the position of the front wheels and thereby ensures the turning of the car.

The composition (assembling, arrangement) of the car can be very diverse, because it depends on:

- mutual location of the engine, cabin and body;

– type of supporting system (frame);

– the number of bridges and their location along the length of the car;

- type of transmission.

There are 4 car layout schemes according to the location of the engine: in front of the cab, under the cab or inside the cab, between the cab and the body, in the rear part of the chassis.

The first 2 engine installation schemes are typical for public transport and multi-purpose vehicles.

The arrangement of the engine under the cab (inside the cab) is more rational compared to installing it in front of the cab, because it provides large dimensions of the body platform or a shorter overall length of the car; better visibility for the driver, which contributes to increasing traffic safety.

Layout of tractor vehicles according to the "cabin over the engine" scheme is also appropriate in cases of arranging them with sleeping places located behind the seat.

The location of the engine behind the cabin is used for powerful tractors and long-wheelbase chassis of multi-axle vehicles.

It is advisable to keep the position of the cab in front of the engine in cases where there is a load-carrying body between the cab and the support-coupling device. Such tractors of the "camel" type are manufactured in the USA on long-wheelbase standard or special chassis of three- or four-axle cars. Recently, such tractor-trailers have been made four-axle, which makes it possible to increase the load capacity and reduce the turning radius (both front axles are a double steered cart) with normal tire sizes.

The advantages of "camel" type tractors are an increase in their towing weight, high stability and more complete use of the length of the road train.

The location of the engine in the rear part is typical for buses, because it is possible to increase the dimensions of the interior, as well as increase their environmental friendliness, because fuel vapors and thermal radiation of the engine do not affect passengers.

To increase the passenger body in some models of passenger cars (for example, VAZ-2114), the engine is placed in front across the longitudinal axis of the car, and the drive is carried out on the front wheels.

The supporting system of the car can be made in the form of a frame, body or combined structure. The frame construction can be spar or tubular in the form of a large-diameter central frame pipe. Its application is typical for trucks. Body or combined designs are typical for passenger cars and buses.

A number of requirements are put forward to the design of the car. They include production, operational, consumer and safety requirements. Let's define these requirements.

Production requirements – compliance of the car design with the technological capabilities of the manufacturing plant, as well as minimal consumption of materials, labor intensity and cost of production.

Operational requirements - high traction, speed and braking properties, fuel efficiency, good controllability, maneuverability, stability, smoothness of movement, passability, insufficient turning ability, reliability, technological maintenance and repair, minimum cost of transport works. Operating requirements largely depend on operating conditions, i.e., on which roads, in which climatic zones the car will operate, as well as which cargoes and which passengers are expected to be transported.

Consumer requirements are the low cost of the car and its operation, reliability and maintainability, ease of control, safety and comfort.

Safety requirements are put forward for active, passive (internal, external) and environmental safety of the car.

When designing a car, general and special requirements are additionally put forward to its systems, units and mechanisms.

The general requirements for all systems, aggregates and mechanisms are as follows: minimum dimensions and weight, simplicity of construction and maintenance, technology, maintainability and low noise level.

Special requirements are additionally put forward for each system, each unit and mechanism, taking into account their purpose, design features and work processes.

Questions for self-control

1. How are cars classified and marked?
2. What properties characterize the car?
3. What nodes, mechanisms, aggregates and systems are included in the design of av- a car?
4. What are the requirements for the design of the car?
5. What are the requirements for the design of systems, units and mechanisms had a car

2 LOADING MODES AND CALCULATION METHODS

2.1 Work process

For a mathematical description of the main features of the work process of aggregates, mechanisms and chassis systems, in many cases, when using the system equilibrium condition, it is enough to take into account only external forces (force) P (for rotary motion - moments M):

$$\sum M_n = 0, \tag{2.1}$$

and for transient processes (acceleration, deceleration) the mass is also involved m (for rotational motion - moments of inertia I):

$$\sum M_n + \sum I_n \ddot{\phi}_n = 0. \tag{2.2}$$

In some cases, for example, for a suspension, it is necessary to analyze oscillatory processes. At the same time, using the principle of independence, consider the behavior of the system under the influence of forces (or moments) caused by linear $e = x/P$ (or angular $e = \phi/M$) elasticity of system links, and forces caused by uneven mass movement:

$$\sum I_n \ddot{\phi}_n + \sum \frac{\phi_n + \phi_{n+1}}{e_{n,n+1}} = M_\phi, \tag{2.3}$$

where $M_\phi = M_a \text{son}(\phi \cdot t + \varepsilon)$ is the moment corresponding to the external disturbance influence When determining the self-oscillations of the system, take $M_\phi = 0$. Consideration of resistance forces (or moments) equivalent to viscous friction $C_b \dot{x}$ (or $C_b \dot{\phi}$) and dry rubbing $C_c \mu$, allows you to determine the degree of attenuation of co-pouring in time:

$$\sum I \ddot{\phi}_n + \sum \frac{\phi_n + \phi_{n+1}}{e_{n,n+1}} - C_b \dot{\phi} - C_c \mu = M_\phi. \tag{2.4}$$

In some cases, it is also necessary to take into account the presence of backlash in individual links of the system.

A significant simplification of the calculation and analysis of a multi-mass multi-link system is provided by replacing it with a simpler equivalent system, for example, a single-mass or two-mass system. At the same time, for systems with serially connected links, the following summation formulas are used:

$$I_e = \sum \frac{I_n}{i_n^2} + \sum \frac{m_n \cdot r_n^2}{i_n^2}, \tag{2.5}$$

$$e_e = \sum \frac{e_{n,n+1}}{i_n^2}, \tag{2.6}$$

where $i_n = \omega_e / \omega_n$ – transmission number from the link of reduction to the given link.

For systems that have open and closed (ring) branches, when defining I_e and e_e use special methods: the chain method fractions, remainder method [3].

When analyzing the work process of systems and subsystems with automatic regulation, a study of stability, speed and accuracy of system operation is additionally conducted.

When solving equations of type (2.1)-(2.4), we obtain initial characteristics, which for mechanical systems, of course, have the form of dependences of force parameters M or P from kinematics ω or v. On weekends, the character- Ristics also includes functions that allow you to evaluate the efficiency of the unit or system, for example, specific fuel consumption for heat engines, KD (coefficient of efficiency) for gears, etc. d. Oscillatory processes are, of course, evaluated using amplitude-frequency characteristics. Other types of characteristics are also used. It is also necessary to have information about the boundary conditions for the initial characteristics and the degree of influence on them by operating conditions: temperature, humidity, duration of work, etc.

Thus, the calculation of the initial characteristics should include:

– definition of nominal static (and dynamic) characteristics;

- calculation (if necessary) of the characteristics that allow the evaluation oscillating processes and operation of automatic subsystems;

– determination of the degree of influence of possible operating conditions on parameters three initial characteristics.

2.2 Strength calculation methods

During the operation of the car, its parts may be destroyed for various reasons. Failures, i.e. malfunctions of car parts, can be classified into two groups.

1. Failures that occur suddenly when the stress exceeds the limit strength of this part.
2. Failures arising as a result of gradual irreversible accumulation of damage in the form of destruction from fatigue (cracks) or in the form of wear and tear.

The calculation of limit states in the first and second cases is performed using fundamentally different calculation methods and conditions of loading parts.

2.2.1 Calculation of static (dynamic) strength

This calculation must be made based on the maximum loads that correspond to the driving modes of the car that are particularly difficult for this part, for example, driving over a significant unevenness, starting from a place when the clutch is suddenly engaged, etc. p. The resulting maximum short-temporary stresses should not exceed the strength limits of the material of this part. Particularly difficult calculation regimes will be different, different for car parts included in the transmission, suspension, control systems, frame or body, but, of course, their definition does not cause difficulties.

The criteria (criterion) of static strength (strength) can be coefficients of margin of strength, determined by limit or by allowable stresses.

When calculating according to ultimate stresses

$$\sigma_{do} \le \frac{\sigma_c}{p_c} \text{ or } \tau_{do} \le \frac{\tau_c}{p_c}. \quad (2.7)$$

Limit stresses are taken for plastic materials σ_c abroad this fluidity $\sigma_c = \sigma_t$, for fragile - beyond the limit of strength $\sigma_c = \sigma_{in}$. Di- the tension σ_{do} are taken according to the maximum stress determined according to the maximum load for the most dangerous section of the part. In the case of a complex stress state, the equivalent stress is determined $\sigma_{there\ are} = \sigma_{do}$. Of course, for plastic materials, the theory of the largest tangential stresses:

$$\sigma_e = \sqrt{\sigma^2 + 4\tau^2}, \quad (2.8)$$

$$\sigma_e = \frac{1-a}{2}\sigma + \frac{1+a}{2}\sqrt{\sigma^2 + 4\tau^2}, \quad (2.9)$$

where $a = \sigma_{vr} / \sigma_{all}$ – the ratio of tensile and compressive strength limits.

The value of the safety factor p_m is determined by the product of the coefficient cents K_1, K_2, K_3 and d, there are various factors. Of course, limit- are multiplied by two coefficients and accepted

$$p_m \approx n_0 = K_1 \cdot K_2, \quad (2.10)$$

where K_1 – coefficient of stability of material properties,

K_2 - the coefficient of responsibility of the part.

Accept for plastic materials $n_0 = 1, 2 \div 2.5$ (smaller values correspond to smaller ratios σ_t / σ_{in}, for brittle materials $n_0 = 2 \div 6$ (smaller values correspond to higher values of impact viscosity a_k).

Calculations based on allowable stresses are more often used in the automotive industry $[\sigma] = \sigma_c / p_m$ (or $[\tau] = \tau_c / p_m$), the values of which are have on the basis of the experience available in the industry, moreover

$$\sigma_{do} \le [\sigma] \text{ or } \tau_{do} \le [\tau]. \quad (2.11)$$

Thus, when calculating according to the ultimate stress σ_{do} need- but compare with σ_d / p_m – equation (2.7), and when calculating according to the permissible ones stresses - with a given value $[\sigma]$.

2.2.2 Calculation of fatigue strength

This calculation must be done according to the loading modes that correspond to the operating conditions characteristic of this vehicle, taking into account their duration. The calculated equivalent stresses, which characterize the fatigue of the material of this part with the known nature of the load change, are used to determine durability. To calculate the fatigue strength, taking into account vibrations, it is necessary to determine the conditions corresponding to resonant oscillations. For calculations on fatigue strength, it is necessary to have statistical data, the collection of which is quite time-consuming and requires a lot of time.

The fatigue strength criteria, as well as the static strength criteria, can be the strength margin coefficients determined by the limit or by the allowable stresses - equations (2.7) and (2.11).

Limit stresses are taken as the limit of endurance in a symmetrical stress cycle σ_{-1} (or τ_{-1}) or by stresses σ_{do} (or τ_{do}), which take into account the asymmetry of the cycle. Acting stresses σ_{do} (or τ_{do}) take according to the equivalent stress, which takes into account the specified nature of the stress change over time, $\sigma_{do} = \sigma_e$, the coefficient of safety in the calculation they take fatigue strength $p_m \approx n_0 = 1.1 \div 1.5$, and sometimes $n_0 = 1$.

If the operating conditions are known, then the following calculation procedure can be used [2].

1. Establish the classification of operating conditions for this car (road type, payload, speed mode, etc.).
2. Record the loading mode of this part in road conditions tests and build correlation tables for each of the accepted operating conditions.
3. Build part load distribution curves for each of accepted operating conditions.
4. Construct fatigue curves using data from bench tests experiences
5. Calculate the correlation equations of longevity.
6. Calculate the coefficient of the reserve of fatigue strength and the ultimate term parts service (in km mileage) for each of the accepted operating conditions.
7. Calculate the maximum service life of the part (in km mileage) for mixed operating conditions.

2.2.3 Calculation of wear resistance (contact strength)

It is expedient to make this calculation using the same loading regimes as when calculating the fatigue strength. A less reliable, but simpler calculation can be performed using conditional average loads and correction coefficients (coefficient, factor) that take into account the degree of non-stationarity of the regime.

2.3 Loading and calculation modes

The loading mode (working conditions) characterizes the real loads that the parts and units of the car withstand during operation. The real or conditional loading mode adopted when calculating the strength of car parts is called calculated. The calculation mode is established based on the analysis of loading modes. Different calculation modes are used for different units and systems of the car.

2.3.1 Calculation modes for transmission

1. According to the maximum torque of the engine. In this case, the calculation th moment (without taking into account the efficiency)

$$M_p = M_{emax} \cdot i_n \cdot a_n \tag{2.12}$$

where i_n – gear ratio from the motor shaft to this shaft;

a_n is the coefficient that takes into account the maximum possible part of the moment, transmitted by this shaft, if the other part of the moment is transmitted by other shafts (half axles, drive to the front and rear driving wheels, etc.).

This calculation mode is usually used for comparative verification calculations.

2. For maximum traction of the driving wheels with the road. In this you-padku

$$M_p = G'_{of} \cdot \phi_{max} \cdot \frac{r_k}{i'_n \cdot \eta'_n}, \tag{2.13}$$

where G'_{of} - the largest adhesion weight (adhesion weight) taking into account the possibility of during the acceleration of redistribution;

i'_n, η'_n – gear ratio and efficiency from the drive wheel to the given shaft;

r_k – rolling radius of the driving wheel;

ϕ_{max}= 0.7÷0.9 is the maximum (with margin) coupling coefficient (adhesion factor).

This calculation mode is usually used for cardan shafts and axles of multi-axle cars.

3. According to the maximum dynamic load, usually, for on-"throwing" clutch release. At the same time, it can be considered (without taking into account the efficiency)

$$M_p = M_{e\max} \cdot i_n \cdot a_n \cdot k_d, \qquad (2.14)$$

where $k_d = \beta \cdot \left(and_{tr} + 8 \right) / and_{tr}$ – dynamism coefficient;

and_{tr} – gear ratio of the transmission;

β – clutch reserve factor.

4. According to real operating loads (calculation on fatigue strength, in some cases taking into account bending and torsional vibrations).

In addition, individual parts of the transmission additionally count on heating (clutches, friction elements of gearboxes, and sometimes individual units), critical angular speed (cardan shafts), stiffness (gearbox shafts), wear and tear, etc. d. For the calculation of non-automatic drives (clutch pedal (pedal; foot bar), levers) the maximum calculated effort is taken R_p = 400 N. About calculation modes for semi-resolved of loaded semi-axles, see in p. 2.3.6.

2.3.2 Calculation modes for brakes

The calculation of brake mechanism parts is usually carried out based on the maximum traction of the wheels with the road:

$$M_p = G'_{of} \cdot \phi_{\max} \cdot r_k, \qquad (2.15)$$

where G'_{of} – the largest towed weight, taking into account the possible braking redistribution baths;

$\phi_{\max}$ = 0.8÷1.0 is the maximum coupling coefficient.

The calculation of the drive parts is carried out based on the maximum calculated force of 1500 N on the pedal (800 N on the lever) or on the maximum calculated pressure (with a pneumatic drive).

In addition, brake mechanisms (brake gear) count on heating.

2.3.3 Calculation modes for steering

1. By the maximum torque on the steering wheel

$$M_p = R_{in\max} \cdot r_{rk}, \qquad (2.16)$$

where r_{rk} – steering wheel radius,

$R_{in\max}$ - the maximum force applied by the driver to the steering wheel, 500 N (for cars you can take 250 N) regardless of whether there is an amplifier or not.

2. According to the maximum braking force applied to one steering wheel foot wheel (the braking force is balanced by the force on the steering wheel at φ_{max} = 0.8÷1.0, for other wheels they accept φ = 0)

$$R_p = G'_{of} \cdot \varphi_{max}. \tag{2.17}$$

3. According to the force that occurs when the driven wheel hits the highest gear damage while driving the car.

About the calculation modes for the parts of the rotary trunnion, see in p. 2.3.6.

2.3.4 Calculation modes for suspensions

1. According to static loads from the weight of the car, taking into account dynamism coefficient

$$R_p = G_{of} \cdot k_d. \tag{2.18}$$

Value k_d increases with increasing suspension stiffness, speed-car bones and road bumps. GAZ 3309 For example, for the root letter rear suspension: k_d = 1.3 at V_a=60 km/h. for the road from the river it covering; 1.6 at V_a = 50 km/h. for paving stones; 3.0 at V_a=20 km/h for off-road

2. According to real operating loads (calculation on fatigue strength).

2.3.5 Calculation modes for frames and bodies

1. Driving at high speed on a bumpy road. Calculation on bend at

$$R_r = P_{hundred} \cdot k_{yes}, \tag{2.19}$$

where $P_{hundred}$ – static load;

k_{yes} is the dynamism coefficient for *and* of the calculated section.

2. Overcoming large bumps with some wheels hanging out. Dis-account for torsion at

$$M_p = \frac{h}{B} \cdot \frac{c_n \cdot c_p}{c_n + c_p}, \tag{2.20}$$

where h – height of inequality;

B – track (wheel span);

c_n – angular stiffness of the suspension;

c_p – angular stiffness of the frame or body.

In addition, the calculation of deformations or damage to the body from an impact in the event of an accident (from the front, rear, side or top – overturning) is carried out - calculation for asymmetric loading according to the calculation scheme.

2.3.6 Design modes for bridge beams

The following calculation modes are used for bridge beams, as well as for steering knuckles and spindles and semi-unloaded half-axles.

1. For the maximum traction of the wheels with the road during a sharp brake-bath - equation (2.15).
2. According to the maximum loads that occur when the car skids of the mobile at the turn (they are neglected by traction forces).
3. According to the force that occurs when the wheel hits the highest obstacle under driving time of the car.

In addition, for semi-unloaded semi-axles, beams of driving bridges and their pivot pins, the calculation mode will be the mode with the implementation of the maximum traction force - equation (2.13).

Questions for self-control

1. What are the features of the mathematical description of the working processes of aggregates, Do you know the mechanisms and systems of the car chassis?
2. What are the initial characteristics of the work process?
3. How is static strength calculated?
4. What is the sequence of calculation for fatigue strength?
5. How to calculate the wear resistance?
6. What are the calculation modes for the transmission?
7. What are the calculation modes for brakes?
8. What are the calculation modes for steering?
9. What are the calculation modes for suspensions?
10. What are the calculation modes for frames and bodies?
11. What are the calculation modes for bridge beams?

3 CLUTCH

The clutch is designed for short-term disconnection of the engine shaft from the transmission and their subsequent smooth connection, which is usually necessary when the car starts from a standstill and after shifting gears while driving.

3.1 Requirements for clutches

1. Smooth switching on. This reduces dynamic loads in the transmission and improves smoothness of movement.
2. Complete (clean) shutdown. Allows you to avoid "driving" the car and reduces the danger of stalling the engine when the car is stationary, and also reduces the load on the gearbox synchronizers.
3. Full power on. Allows you to avoid clutch slippage when transmitting the maximum torque of the engine.
4. The minimum moment of inertia of the driven parts. It reduces work friction in gearbox synchronizers.
5. Effective heat removal. This eliminates the violation of normal ro- clutch boots due to overheating.
6. Wear resistance of friction surfaces and stability of the coefficient of friction at significant increase in temperature and wear of friction surfaces. This ensures increased reliability and durability of friction clutches.
7. Convenience and ease of management.

3.2 Classification of clutches

1. According to the nature of the connection between the leading and the driven parts:
   - mechanical (friction) clutches, dry or oil-operated;
   - hydraulic clutches (hydraulic couplings);
   - electromagnetic clutches with dry or liquid filler.
2. According to the management method:
   - non-automatic (usually with the driver's influence on the pedal) with or without an amplifier;
   - semi-automatic (usually with a signal to turn off or on from the movement of the fuel pedal or the gear shift lever);
   - automatic (usually controlled by the angular velocity of the motor shaft).

In addition, friction clutches are divided into: a) according to the shape of parts with friction surfaces:
- disk (single-disk, two-disk and multi-disk);

- conical;
- block;

b) according to the method of creating the clutch engagement force:

- semi-centrifugal (with springs and centrifugal sinkers);
- centrifugal;
- with springs (with peripheral springs or with a central twisted or diaphragm spring);
- with electromagnet;

c) according to the type of clutch disengagement drive:

- with mechanical;
- with hydraulic;
- with electric (electromagnetic);
- with a combined drive.

On most cars, permanently closed clutches are installed, that is, those that are constantly on and off by the driver when moving, shifting gears and braking. Permanently open clutches, which are turned off at low angular speed of the engine crankshaft and are automatically turned on when it increases, are used relatively rarely, mainly in automatic control.

Single-disc dry clutches are installed on cars and trucks of small and medium capacity. Two-disc clutches are used for heavy-duty trucks (KamAZ, KrAZ, MAZ), but sometimes, for the purpose of simplifying the design, a single-disc clutch is also used for them ("Magirus 290"). Multi-disc clutches are used extremely rarely and only on cars with a large carrying capacity.

Hydraulic couplings - hydraulic couplings (fluid coupling) - were used on Russian ZIM (GAZ-12) and MAZ-525 cars. Nowadays, hydraulic couplings are not used as a separate unit. In some hydromechanical transmissions, under certain conditions, the torque converter switches to the hydraulic clutch mode.

Electromagnetic powder clutches and clutches with electromagnetic pressure force creation in the 40s and 50s of the 20th century. received a certain application due to good adaptability to control automation. However, they did not become widespread, as well as automatic clutches of other types, which is mainly due to their complexity. In our country, electromagnetic powder clutches were installed on ZAZ cars for the disabled.

The clutch drive of passenger cars is mainly hydraulic, often with a servo spring, which facilitates control.

3.3 Working process of friction disk clutch

Features of the working process of the friction clutch are as follows:

– when switching on: in the smooth connection of the leading and driven parts due to the possibility of long-term mutual sliding (slipping) of friction surfaces;

– in the on state: in the transmission of the torque at the expense friction forces between the friction surfaces of the leading and driven parts, pressed against each other;

– when switching off: in the possibility of quick and unhindered transmission of torque due to disconnection of driving and driven parts;

– in the off state: in the absence of torque transmission when the drive is under load, the clutch is disengaged.

To describe the working process of the engaged state of the clutch, it is enough to use addiction $\sum M_n=0$. Only two external moments are applied to the clutch $M_1=M_2=M_{of}$. Driven disk sandwiched between flywheels com and pressure disk with effort R peripheral (or central) springs.

Parts of the clutch disengagement drive are unloaded.

In the disengaged state, the clutch does not transmit torque, but the drive parts and drive parts of the clutch are loaded with forces. In case of non-automatic control, the driver creates the shutdown force by applying force to the pedal R_p.

To describe the work process with this coupling state, it is enough to use the dependence $\sum R_n=0$.

The clutch is engaged by releasing the pedal. The effort on the pedal decreases to zero, and the moment transmitted by the clutch increases is from 0 to $M_{of\max}$. At the same time, skidding occurs for some time clutch because $\omega_1 \neq \omega_2$ at the beginning of the activation.

When switching gears, the driver quickly presses the clutch pedal and releases the fuel pedal (point Mr in fig. 3.1). Angular velocity ω_1 the engine shaft is reduced under the action of the engine braking torque. The driver uses the lever to shift gears and then relatively quickly releases the clutch pedal, lightly pressing the fuel pedal (point d in fig. 3.1). On point *same* slippage of the clutch ends. Full shift time - t_4, the time of the car coasting – t_3, time spent clutch slippage - T_b. Clutch slippage operation when switching gear ratio is much smaller than when starting from a standstill.

The clutch is disengaged by pressing the pedal. To avoid skidding, it is advisable to quickly disengage the clutch. The change in effort on the pedal depending on its movement is shown in Fig. 3.2. From fig. 3.2 it follows that the maximum loads on the parts of the clutch release drive when using coiled springs (curve *1*) from indicate the disengaged state of the clutch (point *b*), and 1 with a plate spring (curve 2) - the beginning of switching off (point *and*).

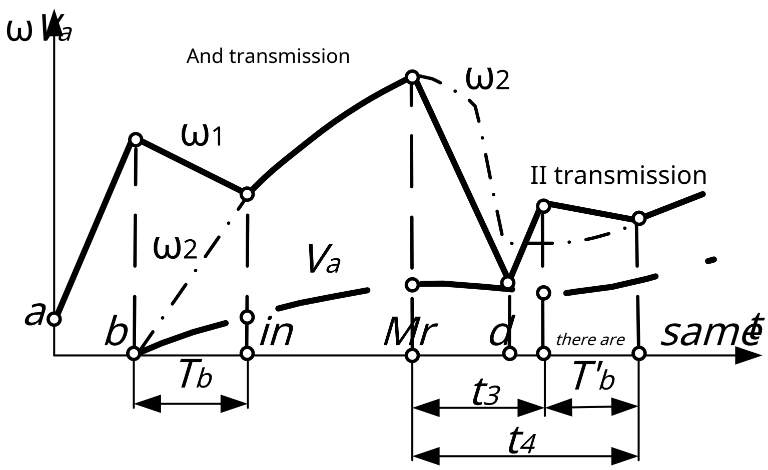


Figure 3.1 – Speed change diagram when switching gears

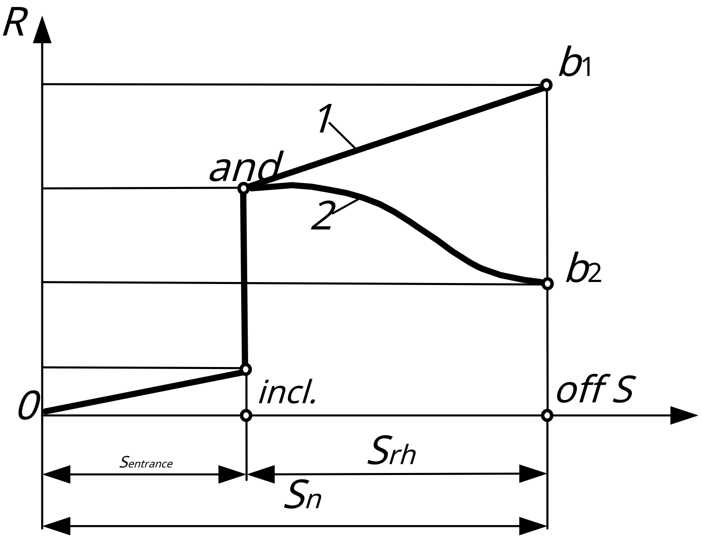


Figure 3.2 – Graph of the change of forces in the clutch drive

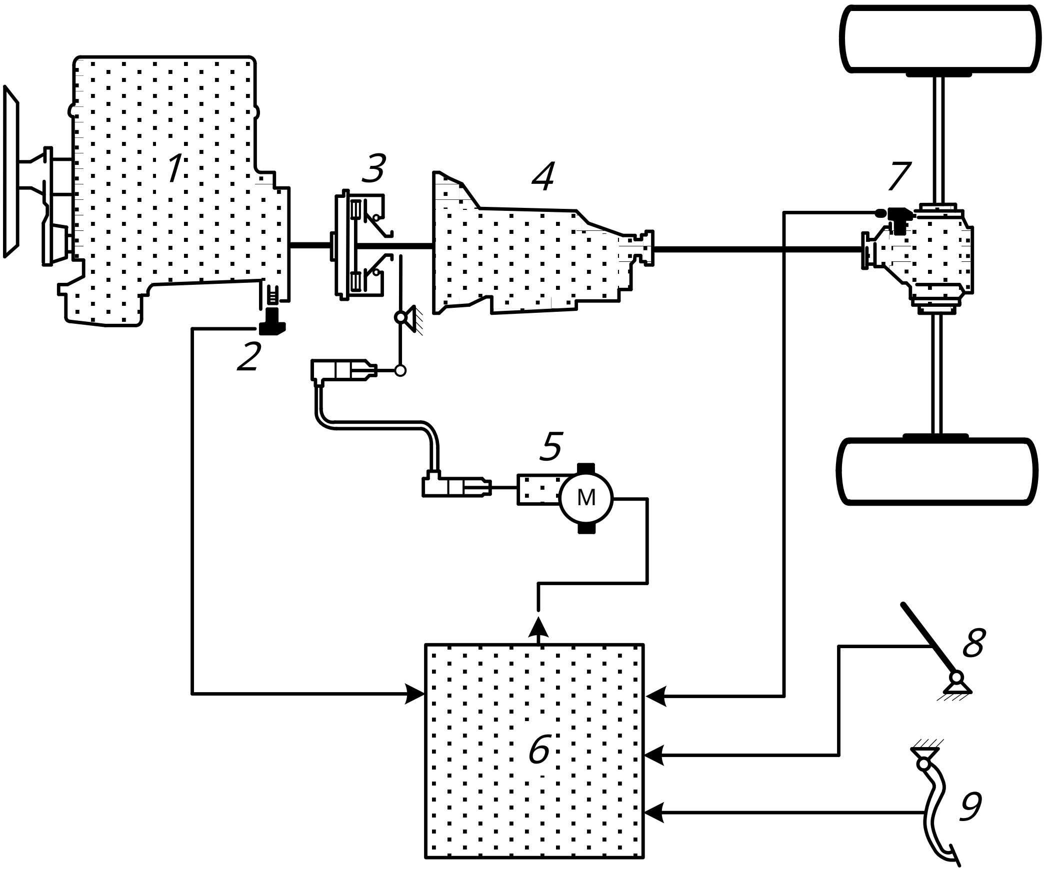
3.4 Clutch control automation

The use of semi-automatic (usually, with a signal to turn off or on from the gear shift lever) or automatic (usually, centrifugal) friction clutches allows you to significantly simplify driving by eliminating clutch pedals and by approximately two times [4] reducing the work of slipping when moving the car from place.

Semi-automatic clutches include, for example, clutches [3] with the switching force provided by an electromagnet. When applying current through the brushes to the ring winding of the electromagnet located in the ma-clutch, the driving disc with the pressure plate is attracted to the latter, pressing the pressure plate (clutch pressure plate) to the driven one. If the circuit of the electromagnet is open, the springs will move the pressure plate away from the driven plate (clutch driven plate). The smoothness of switching on is ensured by the gradual increase of the current in the electromagnet.

Automatic clutches include centrifugal clutches [4], for example, clutches. When the angular speed of the engine shaft increases, the pads attached to the flywheel are pressed against the inner cylindrical surface of the driven drum under the action of centrifugal forces. When the angular speed of the motor shaft decreases, the return springs move the pads away from the drum.

The automatic clutch (Fig. 3.3) allows smooth movement of the car from a standstill, and can also be used together with the servo-mechanism of inclusion in order to ensure fully automatic gear shifting. Other functions of the automatic clutch include controlling the compressive force during acceleration of the vehicle and interrupting the flow of power during braking.



1 – engine; 2 – engine crankshaft speed sensor; 3 – clutch;
4 – gearbox; 5 – servomotor; 6 – control unit (electronic control unit); 7 – speed sensor (sensing element); 8 – fuel supply pedal; 9 – clutch pedal

Figure 3.3 - Automatic clutch

3.5 Design and calculation of clutches

3.5.1 Selection of the type and design of the clutch

When choosing and substantiating clutch designs for a designed car, special attention should be paid to ensuring such requirements as smooth engagement, complete disengagement, durability, convenience and ease of control. For this, it is necessary to proceed from a critical assessment of the existing designs of domestic and foreign clutches and take into account the conditions of clutch operation.

For cars whose operating conditions require frequent use of the clutch (urban conditions, work in quarries, short distances, etc.), hydraulic or electrodynamic types of clutches can be used.

3.5.2 Determination of the dimensions of friction surfaces

The process involves the calculation of the outer and inner diameters of the friction pads of the driven clutch disc.

The maximum static moment transmitted by the clutch due to frictional forces, which prevents slipping of the working parts of the clutch, is determined from the dependence

$$M_C = \beta \cdot M_{e\max} = P_n \cdot \mu \cdot R_{cp} \cdot i, \tag{3.1}$$

where P_n is the total force of pressing the elastic element on the pressure one disk, H;

R_{cp} – average radius of friction, m;

i – the number of friction surfaces in the clutch mechanism;

μ – coefficient of friction. For different types of overlays, it varies within the limits from 0.2 to 0.5;

β – clutch reserve factor. Its value is selected depending on properties depending on the type and purpose of the car (Table 3.1).

Table 3.1 – Calculation parameters of clutches

Car type	Lightweight	Cargo	Bus, tractor vehicle
β	1.3...1.75	1.6...2.0	2.0...3.0
A_D	0.46	0.525	0.725

The dimensions of the friction pad of the driven clutch disc are determined empirically

$$D_{WITH} = A_D \sqrt{M_{e\max}}, \tag{3.2}$$

where A_D is the coefficient of the operating mode of the clutch, is accepted according to the table 3.1;

$M_{e\max}$ – maximum engine torque, N·cm;

D_{WITH} – the outer diameter of the friction pad of the driven disc, see

In design practice, the outer diameter of the driven clutch disc for single-disc clutches is selected within:

– for cars – D_{WITH}=170...225 mm;

– for trucks – D_{WITH}=250...400 mm. The inner diameter of the friction pad is assumed to be equal

$$d_B = (0.55...0.75) D_{WITH}. \qquad (3.3)$$

The average value of the friction radius is determined by the formula

$$R_{cp} = \frac{D_{WITH} + d_B}{4}.$$

3.5.3 Determination of total pressing force The calculation can be performed according to the dependence (3.1)

$$P_n = \frac{M_C}{\mu \cdot R_{cp} \cdot i}. \qquad (3.4)$$

The number of friction surfaces is equal to twice the number of driven clutch discs (2 for single discs, 4 for double discs).

To establish the correct selection of the main dimensions of the clutch disc, it is checked by the allowable specific pressures, which can be determined by the formula

$$P_0 = \frac{4P_n}{D_{WITH}^2 + d_B^2}.$$

Permissible specific pressure values for asbestos-based fractional materials should be within 150...300 kPa, and for metal-ceramic overlays 1000...1500 kPa. It is also necessary to keep in mind that for friction discs, in which D_{WITH} > 300 mm, you need to choose smaller values - not P_0 in order to reduce the speed of skidding on the periphery.

3.5.4 Calculation of compression springs

The process consists in determining the diameter of the spring D, the diameter of the wire d, with of which it is made, stresses τ and its maximum deformation λ_{max}.

The diameter of the cylindrical spring D varies within small limits (27...32 mm). The diameter of the spring wire d it is recommended to take it as an equal 3 ... 5 mm.

When the pressure springs are placed peripherally, their number must be taken as a multiple of the number of switch-off levers. The minimum number of springs is 3.

The number of springs (Table 3.2) is related to the dimensions of the clutch (its outer diameter D_{WITH}).

The force on each spring at peripheral location is determined by the formula

$$P = \frac{P_n}{n}, \tag{3.6}$$

where n – the number of springs of the clutch mechanism.

Table 3.2 – Influence of clutch dimensions on the number of used springs

The diameter of the driven disk, D_{WITH}, mm	Up to 200	200–280	280–380	more than 380
Number of springs	3–6	6–12	12–18	to 30

This effort should not exceed 600...700 N for medium-sized vehicles and 1000 N for heavy-duty vehicles. The maximum number of working springs in a single-row arrangement does not exceed 18, and in a double-row arrangement - 28...30.

The maximum stresses in the cylindrical springs when the clutch is disengaged exceed the working stresses by 15...25%, therefore the calculation formula has the following form:

$$\tau = \xi \frac{10 D \cdot P}{\pi \cdot d^3}, \tag{3.7}$$

where ξ is a correction coefficient that takes into account the influence of the curvature of the turns springs and depends on the ratio . $\frac{D}{d}$

Table 3.3 – Selection of correction factor

D/d	7	6	5	4	3
ξ	1,2	1.25	1.3	1.4	1.6

The calculated stresses in the springs should not exceed the allowable ones, which are equal to 700...750 MPa.

The maximum deformation of the spring is determined by the formula

$$\lambda_{max} = \frac{10 D^3 Mon}{G d^4}, \tag{3.8}$$

where G is the shear modulus [80000 MPa].

To ensure the normal operation of the clutch, it is necessary that, when the clutch is completely turned off, the gap between the coils of the spring remains at least f=1 mm. The total number of turns should be two turns more workers, because the extreme turns are bent and polished.

The length of the spiral cylindrical spring in the free state (without load) is determined by the formula

$$l = d(n+2) + f(n-1) + \lambda_{max}. \tag{3.9}$$

In clutch mechanisms, conical central springs can be used as a pressure element, in addition to twisted cylindrical springs of rectangular section and plate springs, the calculation of which is given in the literature [2, 3].

3.5.5 Indicators of durability or wear resistance

These indicators for the clutch mechanism are estimated by the specific work of skidding and the heating temperature when starting from a standstill.

The work of towing, which does not depend on the smoothness of switching on, is equal to

$$L = \frac{n_0^2}{180} \cdot \frac{I_a}{1 + \frac{I_a}{I_e}\left(1 - \frac{1}{\beta}\right)}, \tag{3.10}$$

where n_0 – the number of revolutions of the engine crankshaft per minute when switched on slow clutch (recommended to take 800 rpm);

I_a – moment of inertia of the car, reduced to the clutch shaft;

I_e – the moment of inertia of the rotating masses of the engine;

β – clutch reserve factor.

The moment of inertia of the translationally moving and rotating masses of the car, reduced to the crankshaft of the engine, is determined by the formula

$$I_a = \frac{G_a}{g} \cdot \frac{r_K^2}{i_0^2 \cdot i_{KI}^2}, \tag{3.11}$$

where G_a – total weight of the car, N;

r_K – kinematic radius of the wheel, m;

i_0 – gear ratio of the main gear;

i_{KI} – gear ratio of the first stage of the gearbox.

The specific work of clutch slippage is determined by the formula

$$C = \frac{L}{F_\Sigma}, \tag{3.12}$$

where F_Σ is the total friction surface of the clutch linings.

The heating of clutch parts during a single switch-on (neglecting radiation) is determined by the following formula:

$$\tau = \frac{\gamma \cdot L}{C \cdot m_D} = \frac{\gamma \cdot L}{427 \cdot 0.115 \cdot G_D}, \tag{3.13}$$

where m_D (G_D) is the mass (weight) of this part;

γ is a coefficient that takes into account how much of the work of friction is perceived clutch disc. For pressure disc and flywheel with single-disc clutch $\gamma = 0.5$; for a two-disc clutch $\gamma = 0.25$; for pressure disk and for the master medium disk $\gamma = 0.5$.

Numerical values of specific towing work and heating temperature when starting from a standstill in lower gears should not exceed the values given in table 3.4 (for one start-up).

Table 3.4 – Critical values of specific towing work and heating temperature

	C, $\frac{\text{MJ}}{\text{m}^2}$	C, $\frac{\text{kg}\cdot\text{see}}{\text{see}^2}$	τ, °C
For single cars	1	10	10
For cars with a trailer	1.5	15	20

3.5.6 Clutch control drive

The mechanism is calculated after substantiating and developing its design scheme.

When designing the clutch drive, it is necessary to ensure the correct selection of the main dimensions of the levers and parts, which affects the convenience and ease of control of the clutch clutch.

The selection of the gear ratio of the drive must be carried out taking into account the following requirements:

– the full stroke of the clutch pedal should not exceed 150 mm for passenger cars and 180 mm for trucks;

- the free travel of the pedal should be - 20...35 mm;

– the gap between the pressure clutch and the pressure levers should align 2...4 mm, the gap in each pair of friction surfaces 0.75...1.0 mm;

- maximum pressing force (P_{ped}) on the pedal when turning off the computer traction should not exceed 150 N for cars and 200 N for trucks.

Gear ratio (power) of the clutch drive

$$i_{Ave} = \frac{P_n}{P_{ped} \cdot \eta_{Ave}}, \tag{3.14}$$

or

$$i_{Ave} = \frac{\beta \cdot M_{e\max}}{\mu \cdot i \cdot R_{cp} \cdot P_{ped} \cdot \eta_{Ave}}.$$

For mechanical, hydraulic drives η_{Ave}=0.6...0.85.

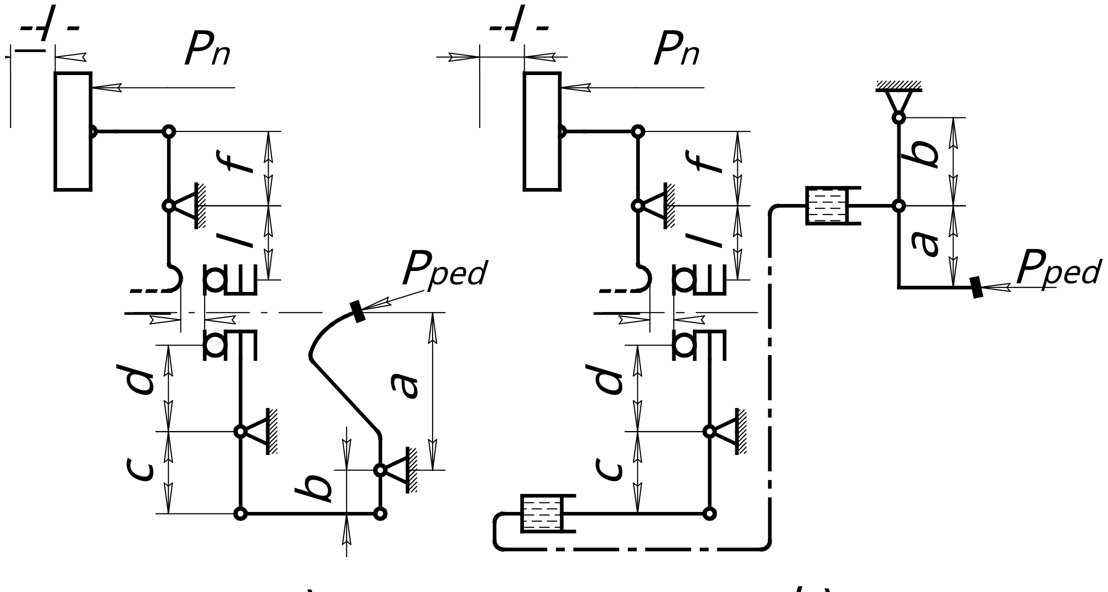
The gear ratios of the clutch drives of modern cars are in the range of 30...45.

Gear ratio of the mechanical drive (Fig. 3.4,*and*) is determined by dependence

$$i_{Ave} = \frac{a \cdot c \cdot e}{b \cdot d \cdot f}. \tag{3.15}$$

For a hydraulic drive, the gear ratio (Fig. 3.4,*b*) gift-bleats

$$i_{Ave} = \frac{a \cdot c \cdot d_2^2}{b \cdot d \cdot d_1^2}. \tag{3.16}$$



a) b)

Figure 3.4 – Kinematic schemes of mechanical and hydraulic clutch drives

The full stroke of the pedal (pedal movement) of the mechanical clutch drive is equal to

$$S_{pedM} = \Delta l_{ipr} + \delta \frac{a.c}{bd} \tag{3.17}$$

For a hydraulic drive, the full stroke of the pedal is equal to

$$S_{PedG} = \Delta l_{ipr} + \delta_2, \frac{d_2 a}{d_1 b} \tag{3.18}$$

where Δl_{Ave} – clearance (clearance) between the pressure and driven disk;

δ – the gap between the pressure clutch and the pressure levers. The use of the given dependencies makes it possible to solve the issue of choosing the structural dimensions of individual parts and the overall kinematics of the clutch drive. When determining the cross-sections and configuration of drive parts, special attention should be paid to the stiffness of levers, rods, rollers and other structural elements that affect both the amount of pedal travel and the frequency of engagement and disengagement of the clutch.

3.5.7 Calculation of vibration damper The calculation consists in determining the torsional stress of the damper spring

$$\tau = \frac{8 P D_{cp} K}{\pi \cdot d^3}, \tag{3.19}$$

where $K = \frac{4c-1}{4(c-1)} + \frac{0.615}{C}$; $C = \frac{D_{cp}}{d} = 4.5 \div 9.5$;

P – force acting on one spring, N;

d is the diameter of the spring wire, d=3...4 mm;

D_{cp} – the average diameter of the springs, D_{cp}=11...15 mm.

The full number of turns of the spring is accepted n=6. Moment of preliminary tightening of the damper springs

$$M_{software} = (15 \div 20\%) \cdot M_{e\max}. \qquad (3.20)$$

Permissible torsional stress in springs $[\tau]$ are taken as equals 650...800 MPa.

3.5.8 Checking the strength of the elements of the driven clutch disc and attachment water

The check is performed in accordance with the main provisions of the theory of strength

hundred

Torsional stresses along the inner diameter of the splined shaft (primary shaft of the gearbox) are equal

$$\tau = \frac{\beta \cdot M_{e\max},}{0.2 d_{in}^3} \qquad (3.21)$$

where d_{in} - the diameter of the shaft in the dangerous section,

see The stress of crumpling the slits is equal to

$$\sigma = \frac{2P}{zl(d_{WITH} - d_{in})a}, \qquad (3.22)$$

where d_{WITH} and d_{in} – external and internal diameters of the splined shaft;

l – length of splined connection;

Z – number of slots;

a – the coefficient of accuracy of the fit of the slots;

P is the force acting on the slots. The

slot shear stress is equal to

$$P = \frac{4\beta \cdot M_{e\max},}{d_{WITH} + d_{in}} \qquad (3.23)$$

$$\tau_{cp} = \frac{P}{zlbd}, \qquad (3.24)$$

where b is the width of the slot.

The stresses of the completed structures made of steels 40X, 18XHT, 30XHT, 12XHNZA are

for torsion - $[\tau_{Cr}]$=100...120 MPa;

for reference - $[\sigma_{zm}]$=60 MPa;

per cut - $[\tau_{cp}]$=30 MPa.

Slots are selected according to GOST 6033 - involute and GOST 1139 - spur.

When calculating a two-disc clutch, checking the strength of the splines is carried out taking into account the full size of the working surface of the hubs of both driven discs.

The driven disk is connected to the hub with rivets, less often with bolts. Rivets are designed for shear and crumpling, and bolts, in addition, for tension. The crumpling stress is determined

$$P = \frac{M_{e\max}}{Z_3 R_3 l d_3} \quad (3.25)$$

and cut

$$\tau_{cp} = \frac{4M_{e\max}}{\pi z_3 d_3^2 R_3} \quad (3.26)$$

where Z_3 and d_3 – the number of rivets and their diameter;

R_3 – the distance from the center of the shaft to the rivet axes;

l is the thickness of the driven disk.

The rivets that fasten the friction pads to the driven shaft are calculated similarly. Bending stresses are allowed up to 10 MPa, and shear stresses up to 6 MPa.

The parts of the clutch drive are calculated on the action of the maximum effort of pressing the pedal, taken as equal to 400 N, and the parts located after the limiter - on the force of the compression springs when the clutch is disengaged.

Questions for self-control

1. What is the clutch for?
2. What are the general clutch requirements?
3. By what features are clutches classified?
4. What types of clutches are installed on different cars?
5. Describe the working process of a friction disc clutch?
6. How is the type and design of the coupling selected?
7. How are the dimensions of friction surfaces determined?
8. How is the total clamping force determined?
9. What is the sequence of calculation of compression springs?
10. What are the indicators of durability or wear resistance of clutches?
11. What are the features of calculating the clutch control drive?
12. What is included in the calculation of the torsional vibration damper?
13. How to check the strength of clutch elements (disk, drive)?

4 GEAR BOX

The gearbox is designed to change the gear ratio of the transmission in order to obtain traction forces on the driving wheels and vehicle speeds within wider limits than can be achieved by changing the engine operating modes. In addition, the gearbox allows you to move the car in reverse and disconnect the engine shaft from the drive wheels for a long time, which is necessary when the engine is running in the parking lot or when coasting.

The gear ratio (kinematic) is defined as the ratio of the angular speed of the driving shaft ω_1 to the angular velocity of the driven driven shaft ω_1. Range of transmission numbers $D_{kp} = and_{n} / and_{in}$, where and_n, $\overline{and_{in}}$ gear ratios of lower and higher gears.

4.1 Requirements for gearboxes

1. Provision of high traction and dynamic properties and fuel car efficiency.
2. High efficiency (efficiency) in the operating range of reference numbers.
3. Minimal vibrations (vibration) and noise.
4. Provision of minimum dimensions and mass, simplicity of structure and serviceability, manufacturability, maintainability.
5. Ease and convenience of management.
6. Provision of power selection for driving additional equipment-transmissions (gearboxes of special and cargo specialized vehicles).

4.2 Classification of gearboxes

1. By the method of changing the gear ratio: stepped (with a gap or without interruption of the power flow), stepless, combined.
2. According to the nature of the connection between the driving and driven shafts: mechanical, hydraulic wooden, electric, combined.
3. According to the control method: non-automatic (usually with the influence of the driver on lever), semi-automatic, automatic, with combined control.

Mechanical step gearboxes (they usually consist of gear mechanisms), in addition, are divided into:

a) by type of gear mechanisms: with fixed gear axes, with moving axes of some gears (planetary), combined;

b) by the number of forward gears: two-, three-, four-, etc. d. degree-frequent;

c) by the number and location of the shafts: two-shaft, continuous coaxial or non-concentric, multi-shaft with the lower location of the driven shaft, etc. d.;

d) according to the method of switching gears: sliding gears, toothed clutches with or without synchronizers, clutches.

Stepless transmissions, like stepless transmissions, are divided according to the nature of the connection between the driving and driven shafts: mechanical (frictional, impulse), hydraulic (hydrovolumetric, hydrodynamic), electric. In addition, continuously variable transmissions are divided into two groups: dynamic and static. The first (frictional, hydrovolumetric) power parameters*M*do not depend on kinematicsω, in others (impulse, hydrodynamic) - depend.

Combined transmissions (stepless with stepless), hydromechanical, electromechanical are divided according to the method of their connection: with serial (one-flow) and with parallel (two- and three-flow) connection. The latter are divided into transmissions with internal - fig. 4.1,*in*,*Mr*(in stepless transmission -*G*) and external - fig. 4.1,*and*,*b*(in step transmission -*M*) branching of the power flow. Hydromechanical transmissions, consisting of continuously connected stepless hydrodynamic transmission (hydrotransformer) and stepless mechanical transmission, were most widely used in cars.

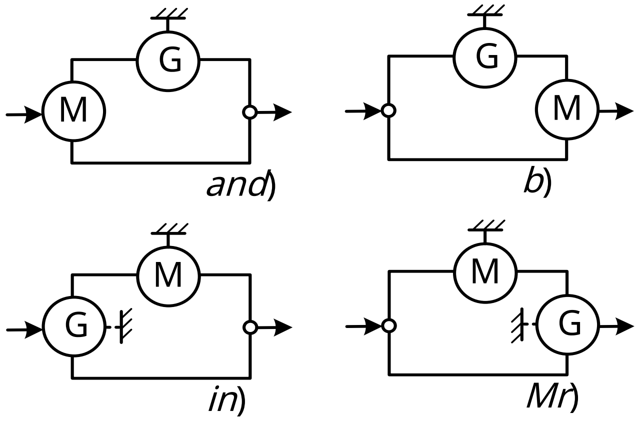


Figure 4.1 - Power flow branching diagrams of combined gears

Hydrotransformers (fluid converter) are classified according to the following characteristics-
we:

1. According to the presence of an external regulation system: unregulated and regulated
strolled;

2. According to the direction of rotation of the driven shaft: forward, reverse
movement, reversible;

3. According to the sequence of location and number of blade wheels;

4. According to the type of turbine: with centripetal, with axial, with centrifugal turbine
she

Unregulated direct-drive torque converters with a centrifugal turbine are usually used on cars, and they have a reactor installed on the freewheel mechanism. This allows them to work not only in torque transformation modes, but also in hydraulic coupling modes. Such torque converters are called complex.

In addition to the main gearbox, some cars use additional gearboxes: demultiplier, divider, etc.

4.3 Working process of a mechanical step gearbox

Four states can be distinguished for the gearbox, which works together with the clutch: 1) switching on the gear; 2) on state; 3) turning off the transmission; 4) off state (neutral).

4.3.1 Enabled state

Dependency is used to describe the workflow $\sum M_n = 0$.

There are three external points: M_1 – on the drive shaft, M_2 – on the driven shaft and M_c is the reactive moment perceived by the crankcase, the existence of which is necessary but for receiving $M_2 \neq M_1$.

The noise of the gearbox mainly depends on:

– accuracy of gear manufacturing;

- stiffness (stiffness) of shafts and crankcase (case, casing, housing), as well as crankcase material (a cast-iron crankcase is less noisy than an aluminum one);

– gear balancing values;

– the size of the lateral gap between the teeth in engagement (extrusion oils at high speed);

– structural elements of gears (noise decreases when changes in the module and the diameter of the gears, with helical gears, with the extension of the hub, etc. d.).

Gear ratios with fixed gear axes and external engagement can be determined by the expression

$$i_m = (-1)^n \frac{Z_2}{Z_1} \cdot \frac{Z_4}{Z_3} \cdots \frac{Z_{2n}}{Z_{2n-1}}, \tag{4.1}$$

where n - the number of engagements through which power is transmitted on this transmission womanhood;

Z – numbers of gear teeth.

Efficiency coefficient of mechanical gearbox (mechanical gearbox)

$$\eta = \eta_{with} \cdot \eta_p \cdot \eta_{zm}, \tag{4.2}$$

where η_{zm} – the coefficient that takes into account the consumption of power for lubrication (by spraying or under pressure);

η_p – coefficient that takes into account losses in bearings;

η_{with} – Efficiency of the gear mechanism itself.

For automotive gearboxes, usually η = 0.98÷0.99 - on a straight line, 0.97÷0.98 – on downgrades and 0.95÷0.96 in first gear.

If we take into account that when the ratio of the number of teeth of the gears, which are in engagement, is more than 3, the efficiency begins to noticeably deteriorate, then with a higher direct transmission, we will have a limitation in the range of gear ratios of the gearbox: a) two-shaft D_{kp}<4; b) long coaxes

D_{kp}<10.

The temperature of the oil under intense operating conditions can reach 120-140°C. The oil level is usually 35-45 mm above the axis of the intermediate shaft, and when using a pump - to the axis.

4.3.2 Enabling gears

Engaging with sliding gears leads to shock loads on the gear teeth if their angular velocities do not correspond to the gear being engaged. To describe the working process of switching gears, the dependence between the force impulse and the amount of movement during an inelastic impact is used [1]. Switching on with the help of synchronizers eliminates shock loads on the teeth.

Activation with the help of frictions is usually used in hydromechanical transmissions. When shifting gears, the clutch of the off gear is disengaged and the clutch of the on gear is engaged. If automatic shifting is used, then the shifting process occurs without reducing the fuel supply, which leads to an increase in the slippage work of the engaged clutch.

A decrease in the rate of activation over time, a decrease in the difference between static and dynamic coefficients of friction, etc. contribute to the reduction of dynamic loads when the friction is engaged. d.

4.4 Features of the working process of the planetary gearbox [5]

The main feature of the working process of planetary mechanisms is that the axes of the planetary gears (satellites) can rotate around some common axis. In automobile gearboxes, of course, coaxial planetary mechanisms with cylindrical gears are used, for which the common axis coincides with the axis of rotation of non-planetary gears (Fig. 4.2).

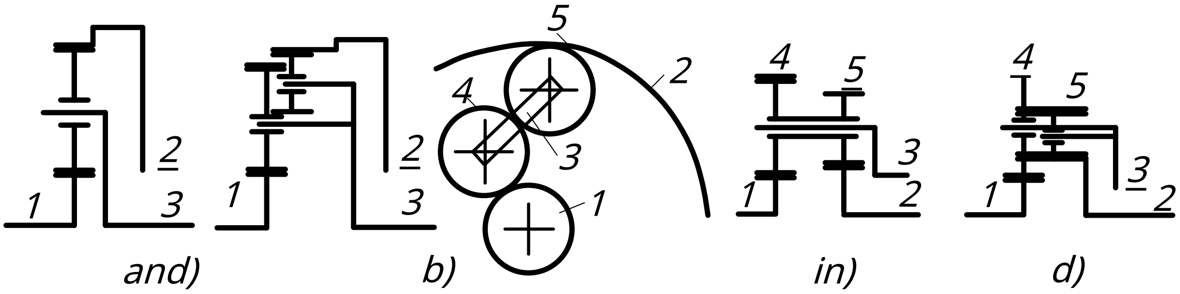


Figure 4.2 – Schematics of automotive planetary gearboxes

Peculiarities of planetary coaxial mechanisms in comparison with non-planetary ones:

– the possibility of the formation of several parallel power flows, why hydraulic or electric transmissions can be installed in parallel lines;

– the possibility of obtaining a higher efficiency, because part of the strength is transmitted by portable motion without loss;

– smaller dimensions and significantly greater load transfer at the some gear ratios;

– absence of radial load on the shafts (with 3 or more satellites years), except for satellite axes;

– output shafts can only be coaxial;

– more significant limitations in ensuring the given transmission numbers, because their value is determined not only by the parameters of the gears, but also by the connection scheme of the components of the planetary mechanisms between themselves and with the output shafts;

– higher production cost;

- a higher noise level, because with three satellites there will be six tooth contact points instead of one for a pair of gears.

We note two more features.

1. Frictions of non-planetary gears are loaded with torque, transmitted shaft, on which they are located

$$M_{fn} = M_{entrance} \cdot i_p \tag{4.3}$$

where n – friction index;

i_p – gear ratio to the location of this clutch.

Brake and locking clutches are mainly used in planetary gears. The first ones perceive the reactive moment and are loaded by the difference (the sum for reverse gears) of the torques on the driven and driving shafts, and the second ones are loaded only with a part of the transmitted torque, because the other part is transmitted in parallel through locked gears

$$M_{fgn} = M_{entrance}\left(i_t - 1\right), \tag{4.4}$$

$$M_{fbn} = M_{entrance} \cdot and \quad , \tag{4.5}$$

where i_t – transmission ratio;

and is the share of the torque transmitted by the locking clutch.

2. The number of teeth of the gears, which are components of the elementary planetary mechanism, must be in certain ratios that satisfy the conditions: concentricity, assembly, neighborhood (location of satellites), absence of trimming, i.e. $Z_{min} > 13$.

The condition of concentricity ensures the coincidence of the axes of the driving and driven shafts: for fig. 4.2, *and*

$$Z_2 - Z_1 = 2 \cdot Z_{Sat}, \tag{4.6}$$

for rice 4.2, *in*

$$t_1(Z_1+Z_{Sat1})=t_2(Z_2+Z_{Sat2}),$$

where t_1 and t_2 – coupling modules.

The condition of assembly ensures the possibility of assembly of gears:

for fig. 4.2, *and*

$$Z_1+Z_2=AND\cdot p, \qquad (4.7)$$

for rice 4.2, *in*

$$Z_{Sat1}\cdot Z_2-Z_{Sat2}\cdot Z_1=AND\cdot p,$$

where p – the number of satellites; AND – coefficient (whole number).

The neighborhood condition ensures sufficient gaps between the teeth of the satellites:

for fig. 4.2, *and*, *in*

$$\text{son}\frac{\pi}{p}>\frac{Z_{cam}+2f_{Goal}}{Z_1+Z_{cam}}, \qquad (4.8)$$

where $f_{Goal}=h_{Goal}/m$ is the tooth head height coefficient, the equation is taken by 1 for uncorrected gear teeth.

A planetary transmission with numbers of gears that do not meet the conditions of concentricity or assembly can sometimes be formed with the help of adjusting the teeth of the gears (GAZ-13) or uneven placement of satellites (Mercedes 600).

The equation of the connection between the power parameters – torques – on each of the three links of the planetary mechanism can be obtained by writing them down through the circular forces and the radii of the pitching circles of the gears:

$$M_1=P\cdot R_1, M_2=P\cdot R_2, M_3=-2P\frac{R_1+R_2}{2}.$$

For M_3 a force equal to −2 is taken from the condition of satellite equilibrium P and applied to the axis of the satellite. Marking $\frac{R_2}{R_1}=a$ and dividing M_2 and M_3 on M_1, get mo

$$M_1:M_2:M_3=1:a:-(1+a), \qquad (4.9)$$

The equation of the connection between the kinematic parameters – the angular velocities of the links of the planetary mechanism – can be obtained from the energy conservation equation

$$N_1+N_2+N_3=0,$$

by writing it in the form

$$M_1\cdot\omega_1+M_2\cdot\omega_2+M_3\cdot\omega_3=0,$$

and applying equation (4.9)

$$\omega_1+a\cdot\omega_2-(1+a)\cdot\omega_3=0. \qquad (4.10)$$

From equation (4.10), it is possible to determine the angular velocity of any of the three links, if the angular velocities of the other two are known.

The angular velocity of the satellite relative to its axis can be determined from the expression

$$\omega_{cam} = (\omega_1 - \omega_3)\frac{Z_1}{Z_{cam}}. \quad (4.11)$$

In equations (4.9) and (4.10) α –parameter of the planetary mechanism. It corresponds to the gear ratio (taken with the opposite sign) of a planetary mechanism turned into a non-planetary one, that is, when the water is stopped

Lee Therefore, $a = \frac{Z_2}{Z_1}$ - for Fig. 4.2, *and*, *Mr*, $a = -\frac{Z_2}{Z_1}$ - for Fig. 4.2, *b*;

$a = -\frac{Z_2 \cdot Z_{cam1}}{Z_1 \cdot Z_{cam2}}$ - for Fig. 4.2, *in*.

The gear ratios of the planetary gearbox are determined in the following sequence. Write down as many coupling equations (4.10) as there are planetary series involved in torque transmission. Then conditions are substituted that take into account the connections between links (link, member) on this transfer. By jointly solving equations (4.10) is determined by the equation for the gear ratio in the form $and_i = \frac{\omega_{acc}}{\omega_{exit}}$, vira-driven by the parameters of the planetary series.

As an example, let's determine the gear ratio for the first gear (the clutch is on F_2 rice. 4.3) GMP of the ZIL-114 car, moreover $a_1 = a_2 = 55 / 23 = 2.39$. Having written down

$$\begin{cases} \omega_{with1} + a_1 \cdot \omega_{to1} - (1 + a_1) \cdot \omega_{in1} = 0 \\ \omega_{with2} + a_2 \cdot \omega_{to2} - (1 + a_2) \cdot \omega_{in2} = 0 \end{cases}$$

and considering that in 1st gear

$$\omega_{entrance} = \omega_{to1} = \omega_{with2}, \omega_{exit} = \omega_{to2}, \quad \omega_{in1} = \omega_{in2}, \quad \omega_{with1} = 0,$$

we will get

$$and_1 = \frac{\omega_{with2}}{\omega_{to2}} = \frac{a_1 \cdot a_2 + a_2}{a_1 \cdot a_2 - 1} = 1.72.$$

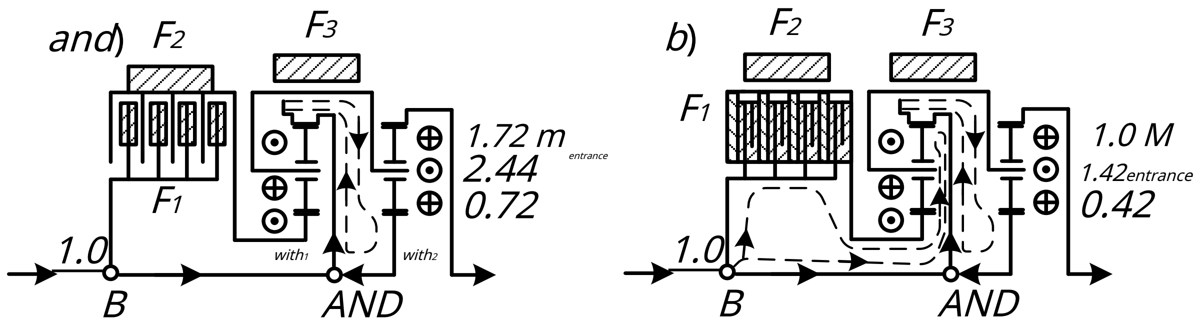


Figure 4.3 – Load distribution for the first (*and*) and the second (*b*) gears GMP ZIL-114

For convenience in the solution, indices were adopted, indicating: 1 – the first and 2 – the second planetary series, *with* – solar (with external teeth) and *to* – the crown (with internal teeth) of the gear, *in* - drove.

Determination of the efficiency of planetary mechanisms can be carried out according to the method of M. AND. Kreines, which allows you to determine the efficiency without the HMP scheme and without the analysis of power circulation, using only the expression for the transmission number, by dividing the power transmission number $\bar{and}_t = \frac{\omega_{exit}}{\omega_{entrance}}$ to the cinematic gear ratio $and_t = \frac{\omega_{entrance}}{\omega_{exit}}$.

The efficiency is determined in the following sequence. It is set as it changes and_t as each parameter increases a. All a_p divided into two groups: the first consists of a_p, the increase of which leads to an increase and_t, and the second - from a_p, the increase of which leads to a decrease and_t. In general in this form, it can be determined by the sign of the expression

$$\operatorname{sign}\frac{a_n}{i_m}\cdot\frac{\partial i_m}{\partial a_n}. \qquad (4.12)$$

If the sign is plus, then a_p belong to the first group, and if minus, then - to the second Then it is folded $\bar{and}_t$ by multiplying by η_0 everyone a_p the first groups and division into η_0 everyone a_p of the second group in the expression for and_t. Finally recognized is waiting $\eta_3 = \bar{i}_m / i_m$.

For elementary planetary mechanisms, when turning them into non-planetary mechanisms by decelerating the carrier, we obtain $\eta_0 = \eta_{n3} = \eta_{in3} = 0.96$ – for rice. 4.2, *a*; $\eta_0 = \eta^2_{n3} \cdot \eta_{in3} = 0.936$ - for rice. 4.2, *b*; $\eta_0 = \eta^2_{n3} = 0.95$ - for rural areas and 4.2, *in*; $\eta_0 = \eta^3_{n3} = 0{,}926$ - for fig. 4.2, *Mr*. Accepted $\eta_{n3} = 0.975$ and $\eta_{in3} = 0.985$.

As an example, let's determine the efficiency for the first transmission of the HMP of the ZIL-114 car (Fig. 4.3). Because a_1 and a_2 belong to in this case, to the second groups, then, dividing them into $\eta_0 = 0.96$, we will get at $a_1 = a_2 = 2.39$

$$\eta_3 = \frac{a_1 \cdot a_2 + a_2 \cdot \eta_0}{a_1 \cdot a_2 - \eta_0} \cdot \frac{a_1 \cdot a_2 - 1}{a_1 \cdot a_2 + a_2} = \frac{1.67}{1.72} = 0.97.$$

Let's determine the value of the moments transmitted by the ZIL-114 HMP frictions. According to equation (4.4)

$$M_{f2} = M_{entrance}\left(\frac{-a_1 \cdot a_2 + a_2}{-a_1 \cdot a_2 - 1} - 1\right) = 0.72 M_{entrance},$$

$$M_{f3} = M_{entrance}(-a_2 - 1) = -3.39 M_{entrance}.$$

Locking clutch F_1 transmits only that part of the torque that passes through it to the sun gear $with_1$ of the first planetary row, i.e., according to equation (4.9), $1/(1+a_1)$ from the moment on led M_{in1}. From the condition of equilibrium of the link consisting of two carriers, $M_{in2}=M_{in1}$, but the carrier of the second planetary row must be loaded is equal to the moment that is part of $(1+a_2)/a_2$ in and $M_{to2}=M_{exit}=M_{entrance}$, as direct transmission is considered. Therefore,

$$M_{f2}=\frac{1+a_2}{a_2}\cdot\frac{1}{1+a_1}=0,42M_{entrance}.$$

Thus, you can find the load as a percentage of $M_{entrance}$ for every transmission on each link of the planetary mechanism. In fig. 4.3 presents the distribution of loads (not including losses) for the first (*and*) and the second (*b*) transmission of the ZIL-114 HMP. Arrows show the direction of power flows. The icons in the circles indicate the direction of action of the circular forces: the dot is towards us, the cross is away from us.

In the ZIL-114 HMP, closed circuits are formed on the forward gears (branching points of the power flow -*AND* and *B* rice. 4.3), and in a circuit with a branching point *AND*, as the analysis shows, there is circulating power. Its value can be determined by the parameters of the link that transmits only circulating power. Such a link for the ZIL-114 GMP is a link $with_2$:

for the first transmission $N_c=\frac{M_{with2}\,\omega_{with2}}{M_{entrance}\,\omega_{entrance}}N_{entrance}=0.72N_{entrance}$,

for the second transmission $N_c=0,42N_{entrance}$.

Circulating power can be when there is a closed circuit, which is possible in planetary mechanisms with two or more planetary rows, as well as when using locking frictions. However, the existence of a closed circuit does not yet determine the presence of circulating power. It is possible to transfer power by parallel flows without circulating power.

Circulating power, consisting on some links (see Fig. 4.3) with transmission power, increases the load and losses on these links, and therefore leads to an increase in dimensions, a decrease in durability and efficiency. However, these disadvantages manifest themselves insignificantly with high efficiency of overloaded circuits and with small circulating capacities. In blocked links, losses from circulating power do not increase, since there is no mutual movement of links.

Efficiency was obtained for the first transmission of the ZIL-114 HMP $\eta_{in}=0.97$. Phrad - the exact number 1.72 can be obtained with the help of one planetary series (Fig. 4.2,*and*) at $a=1,4$, if link 1 is stopped, and link 2 is leading. In this in this case, the efficiency will be $\eta_3=\frac{\eta_0+a}{1+a}=0.984$ (about 1.5% higher).

4.5 Working process of the torque converter

Features of the working process of the torque converter (GDT):

1. Power and kinematic connections between blade wheels are carried out by-flows only through the working fluid, which is a single whole annular link, which is in force interaction simultaneously with the blades of all blade wheels;

2. There are quite high speeds with which the liquid flow flows around the blades the reason for the existence of significant hydraulic energy losses (mainly due to friction and impact), which leads to limitations in the initial characteristics, mainly in terms of efficiency;

3. Force interaction of fluid and blades in the absence of rigid kinematics matrix connection between the driving and driven shafts is the reason for the interdependence of the power forces M and kinematic ω parameters, which determines the ability of the GDT to load the engine with torque, which is uniquely determined by the operating mode *and* and angular velocity ω_1 drive shaft

4.5.1 Stable modes of operation [3, 16]

The initial characteristics of the GDT are used in dimensionless form:

$$K = f(i), \eta = f(i), \lambda_1 = f(i). \tag{4.13}$$

where $i = \omega_1/\omega_2$ – gear ratio;

$K = M_2/M_1$ – transformation coefficient;

$\eta = N_2/N_1 = K \cdot i$ – efficiency;

$\lambda_1 = \dfrac{M_1}{\rho \cdot \omega_1^2 \cdot D^5}$ – torque coefficient of the drive shaft;

ρ – density of the working fluid, for HDT oils $\rho \approx 88$ kg·s^2/ m^4;

D – the active (largest) diameter of the working cavity of the DHT in m. The following parameters are used to evaluate the initial characteristics in traction modes of operation (Fig. 4.4):

- K_0 is the transformation coefficient at $i = 0$; and
- η_{max}, $\eta_{max\,m}$ – maximum efficiency in moment transformation modes and hydraulic couplings;
- $D_{and} = i_{max}/i_{min}$ is the kinematic working range, which is determined by modes $\eta = 0.8(D_{0.8})$ or $0.75(D_{0.75})$;
- P – transparency coefficient.

$$P = \frac{\lambda_{1\,max}}{\lambda_{1m}} = \frac{M_{1\lambda}}{M_{1m}} \cdot \left(\frac{\omega_{1m}}{\omega_{1\lambda}}\right)^2 \tag{4.14}$$

Transparency is the property of mechanical energy to change the mode of operation of the engine when the moment or angular speed of the driven shaft changes. Cut-are not transparent (P≈1), with a line (P>1) and the inverse (P<1) about by stardom

Of course, in automobile GDT K_0= 3÷2, η_{max}= 0.85÷0.92, $\eta_{max m}$= 0.93÷0.97, $D_{0.8}$= 2, 4÷2, P= 1, 2÷2.5 (higher values correspond to smaller K_0 and $D_{0.8}$).

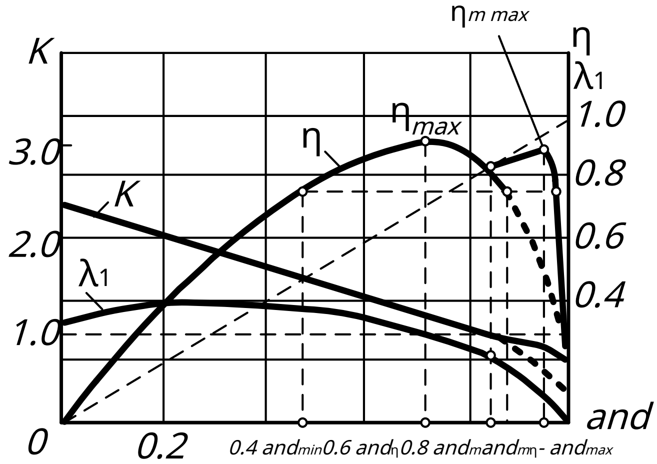


Figure 4.4 – Output characteristics of the GDT in traction modes of operation

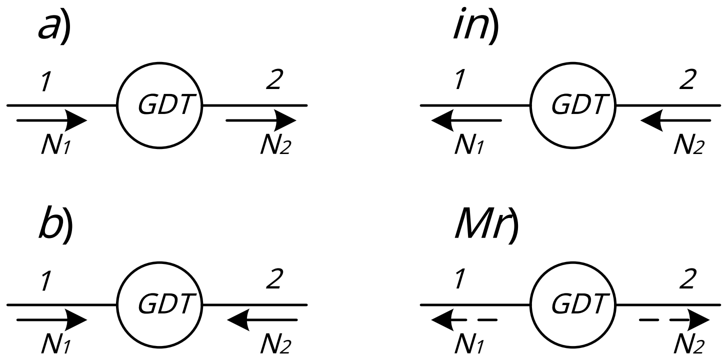


Figure 4.5 – GDT operation modes

HDT operation modes are divided into four main groups according to the direction of power transmission (Fig. 4.5) with ω_1> 0.

1. Traction: N_1> 0 , N_2< 0 – power is transmitted from the engine to the driving wheels (Fig. 4.5, *and*).

2. Brakes: N_1≥0, N_2≥0 – the power is only supplied to the GDT. If the transition from traction to braking modes occurs when the sign of y changes ω_2 are anti-rotation modes, if y M_2 are overtaking modes.

3. Inverted: N_1< 0 , N_2> 0 – power is transmitted from the drive wheels to the engine (Fig. 4.5, *in*).

4. Engines: $N_1 < 0$, $N_2 < 0$ – power is only diverted from the HDT. These modes are possible only as short-term unstable ones, because transmission cannot be a source of energy (Fig. 4.5, $M r$).

Calculation of the initial characteristics according to the given geometric parameters (blade angles β_n and radii R_n location of input and output edge) can be performed according to the jet theory, that is, considering the averaged parameters of the flow of the working fluid: flow rate Q, m^3/s and pressure H_n, m, attributed to the average stream.

Using the well-known principle of mechanics, according to which the impulse of a force is equal to the amount of motion $P \cdot t = m \cdot V$, and taking the district component P_{u1n} strength, since it is she who gives the torque when multiplied by R_{1n}, we will get an expression for the force of fluid interaction with the blades at the entrance to n-e blade wheel.

$$P_{u1n} = \frac{m \cdot V_{u1n}}{t} = Q \cdot \rho \cdot V_{u1n}.$$

Then the torque will be equal to

$$P_{u1n} \cdot R_{1n} = Q \cdot \rho \cdot V_{u1n} \cdot R_{1n}.$$

Adding the moments of the amount of movement from the entrance to the exit from the vane wheel and taking into account the fact that the entrance conditions are completely determined by the exit conditions from the previous vane wheel, we obtain

$$M_n = Q \cdot \rho \cdot \left(V_{u2n} \cdot R_{2n} - V_{u2(n-1)} \cdot R_{2(n-1)} \right). \tag{4.15}$$

The parameters included in equation (4.15) can be expressed through their proportional values ω_1 and characteristic linear dimension D – active diameter GDT $Q \approx \omega_1 \cdot D^2$, $V_u \approx \omega_1 \cdot D$, $R \approx D$, todi

$$M_n = \lambda_n \cdot \rho \cdot \omega_1^2 \cdot D^5. \tag{4.16}$$

Impeller pressure

$$H_n = \frac{\omega}{g} \cdot \left(V_{u2n} \cdot R_{2n} - V_{u2(n-1)} \cdot R_{2(n-1)} \right). \tag{4.17}$$

Frictional losses

$$H_{tr\,n} = \zeta_n \cdot \frac{W_{cp\,n}^2}{g} = \zeta_n \cdot \frac{V_m^2}{g \cdot \mathrm{son}^2 \beta_{cp\,n}}. \tag{4.18}$$

where $\zeta_n \approx 0.16$ is the coefficient of friction resistance, $W_{cp\,n}$ – relative (along the blade) speed. Losses per hit

$$H_{udn} = \phi_p \cdot \frac{W_{support}^2}{2g} = \frac{\phi_p}{2g} \cdot \left(V_{u2(n-1)} \cdot \frac{R_{2(n-1)}}{R_{1n}} - V_{u1n} \right)^2, \tag{4.19}$$

where $\phi_n = 1$ – loss ratio per hit.

During the design of the GDT according to the specified parameters of the initial characteristics, the geometric parameters are determined by the average stream, and then profile the blades from the inlet to the outlet (along the flow) and from the inner torus to the outer (across the flow).

4.5.2 Unstable modes of operation

Acceleration of the car. Neglecting the elasticity of the links and skidding in the mecha- gears, let's present a car with GDT in the form of a two-mass system for which the equations are valid

$$I_1\dot{\omega}_1 = M_e - M_n, \tag{4.20}$$

$$I_2\dot{\omega}_2 = M_n K - M_c . \tag{4.21}$$

These equations cannot be solved analytically because M_n proportion- he ω_1 and λ, a λ_1 is a non-linear function $and = \omega_2/\omega_1$. Besides, K is a non-linear function and.

Solving equations (4.20), (4.21) by the Runge-Kutta method under given M_c= const, K and λ_1 polynomials of the first degree of the form $a+bx$, M_e - friend th degree of appearance $a+bx+cx^2$ shows that the entire process of acceleration when starting from a standstill with a previously engaged gear and an instant increase in fuel supply to the maximum can be divided into three stages.

1. Acceleration of the engine shaft before the start of rotation of the turbine shaft. On this stage ω_2 = 0.

2. Acceleration to the zone with $\omega_1 and \approx \omega_2$. At this stage (the stage of disordered hijacking) acceleration ω_1 and ω_2 development differ significantly.

3. Acceleration at $\omega_1 and \approx \omega_2$ – stage of orderly acceleration. For him with equal babysitter (4.20), (4.21)

$$\frac{M_e - M_n}{M_n K - M_c} = \frac{I_1}{I_2 i}, \tag{4.22}$$

$$M_n = M_e \frac{i + \dfrac{M_c}{M_e}\dfrac{I_1}{I_2}}{i + K_1\dfrac{I}{I_2}}. \tag{4.23}$$

These features make it possible to apply a simplified method of calculating the initial acceleration phase of the car.

Engine braking. For analysis, you can use equations (4.20), (4.21). The entire process of engine braking through the GDT with a sudden release of the fuel pedal can be divided into three stages.

1. Entering braking mode. Value ω_1 big $\omega_2 \approx 0$, i.e $\omega_2 \approx \omega_{2n}$ = const and can be accepted take a linear change M_n from t: $M_n = A - B \cdot t$.

2. Irregular braking. Value ω_1 and ω_2 substantially distinguish- sia, but the stage time is short.

3. Orderly braking. At this stage $\omega_1 and \approx \omega_2$ a fair relations learning (4.22), (4.23).

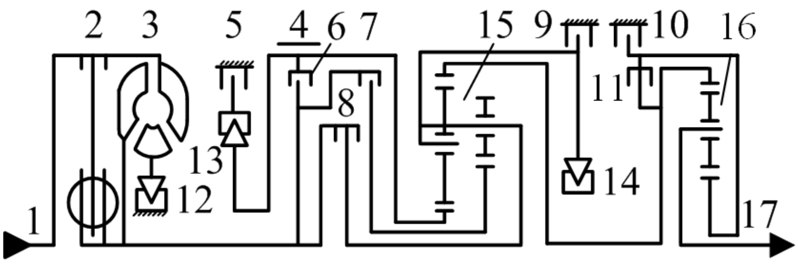
These features make it possible to apply a simplified calculation method, eliminating the second stage.

According to experimental data [1], it was found that the shortest time to fully press or release the fuel pedal is about 0.1 s. However, the engine torque changes much more slowly. Tempo $\frac{dM_e}{date} = \frac{dM_T}{date}$ almost three times smaller when switching to engine braking by HDT, than without HDT, and the amplitude of moment oscillations is 5 times smaller.

4.6 Features of operation of automatic gearboxes

Automatic transmissions perform gear shifting operations without the driver's participation. Power losses in an automatic transmission are significantly greater than in a mechanical one. However, this is compensated by the advantages associated with the possibility of maintaining the engine operation in the most economical mode. The automatic gearbox contains (Fig. 4.6):

– torque converter (always used in gearboxes dacha of passenger cars; on trucks, of course, a Trilok-type design is used - with a centrifugal turbine): designed for starting from a standstill, increasing torque and absorbing torsional vibrations;



1 – drive shaft; 2 – locking clutch; 3 – torque converter; 4 – band brake; 5-11 – multi-disc clutches and brakes; 12-14 – free movement mechanisms; 15 and 16 – planetary mechanisms; 17 – driven shaft

Figure 4.6 – Diagram of a five-speed automatic gearbox (ZF S HP 18)

– in the gearboxes of passenger cars (as a rule) and trucks in these cars (always) the torque converter is supplemented with a locking clutch (joint box, clutch);

– several planetary mechanisms;

– multi-disc clutches with a hydraulic drive, disc or band brakes (intended for switching without interrupting the power flow);

– free-wheeling mechanisms together with switching elements for op-
automatic gear shifting;

– a control system for selecting and smoothly shifting gears according to the program set by the car driver (Table 4.1);

– engine-driven hydraulic pump: provides the pressure required for the operation of switching elements, supplies fluid to the torque converter, provides lubrication and cooling of the gearbox.

4.6.1 Design options

Automatic transmissions installed on passenger cars have 3, 4 (mainly) or 5 forward gears.

Table 4.1 – Gear shift diagram

Re- cottage	Transmission element (Fig. 4.6)									Transmission numeric
	2	4	5	6	7	8	9	10	11	
1	○				●		○	●		3.67
2	○	●	●		●			●		2.00
3	○	●	●		●				●	1.41
4	○		●		●	●			●	1.00
5	○	●	●			●			●	0.74
Posterior course	○			●			●	●		4.10

●– enabled;○ – can be enabled

The range of mechanical conversion is between 3.0 (three-speed gearbox) and 5.0 (five-speed unit).

Automatic transmissions for trucks can have from 3 to 6 forward gears. The mechanical conversion range varies from 2 to 8. These gearboxes often have built-in hydrodynamic retarders, as well as a hydraulic pump, a large sump and a fluid cooler.

4.6.2 Electronic gearbox control system Control systems of automatic transmissions, in which only hydraulics are used, are beginning to be replaced by systems in which elements of electronics and hydraulics are combined (hydraulic drive is kept only in clutches). The advantages of using electronics include:

– the ability to install several different programs for switching editor;

– great smoothness of switching on the transmission;

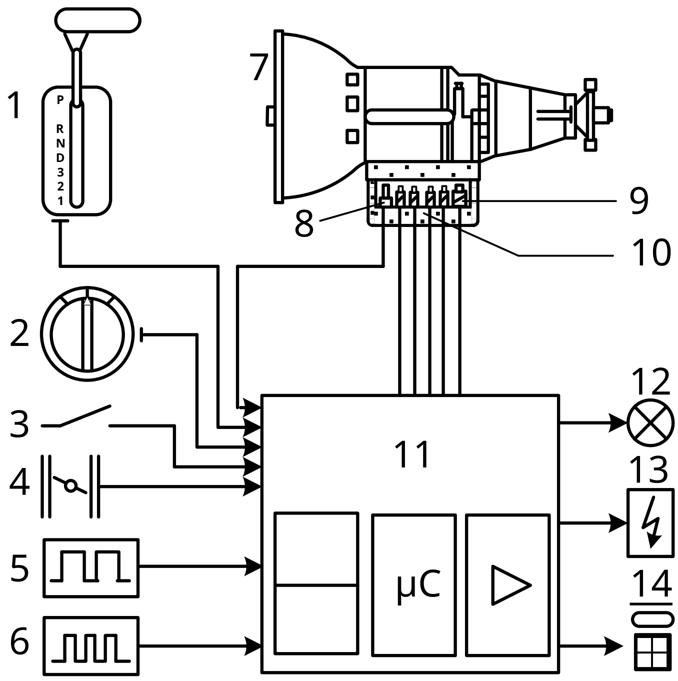
– flexibility and adaptability to different types of cars;

– application of simplified hydraulic control circuits and mechanical no freewheeling.

The measuring transducers of the system determine the load, the position of the gear shift lever, the position of the program switch and the "kick-down" mode, as well as the frequency of rotation of the engine shaft and the driven shaft of the gearbox. The control unit processes this data in accordance with the set program and produces transmission control signals.

Electrodynamic transducers form the link between the electronic and hydraulic circuits, while the solenoid valves actuate the clutches. At the same time, analog or digital pressure regulators are used.

Gear shifting control. When selecting the required gear, the system asks for data on the speed of the driven shaft of the gearbox and the engine before the corresponding solenoid valve is activated. The driver can select the necessary gear shifting program, for example, to ensure maximum fuel economy or maximum speed mode. You can also intervene in the gear shifting process at any time using the manual gear shift lever.



1 – gear shift lever with positional shift; 2 – program switch; 3 – forced activation of a reduced gear ("kick-down"); 4 – turning angle sensor throttle valve; 5 – engine torque (signal *t*);6 – rotation frequency engine crankshaft (ignition signal); 7 – gearbox; 8 – driven shaft speed sensor (min); 9 – pressure regulator; 10 – solenoid valves; 11 – electronic control unit (ECU); 12 – failure indicator; 13 - decrease engine torque by adjusting the ignition; 14 – diagnostic block

Figure 4.7 – Scheme of electronic control of the gearbox

Intelligent gear shifting programs optimize car driving by supplementing standard gearbox control data with such auxiliary parameters as longitudinal and lateral acceleration, speed of movement of brake pedals and fuel supply. A sophisticated control program allows you to choose the appropriate gear both for the current driving conditions of the car and for the driving style. For example: Porsche Tiptronic (Fig. 4.8) is provided by the ZF 4 HP 22 gearbox, which works according to the intelligent gear shifting program. This system combines automatic and active individual driving modes.

In addition to the standard shift positions, the gearshift lever can switch to a second (parallel) logic circuit, in which a simple, light movement of the lever with a jerk is enough to immediately change gears (provided that the engine shaft speed is not exceeded).

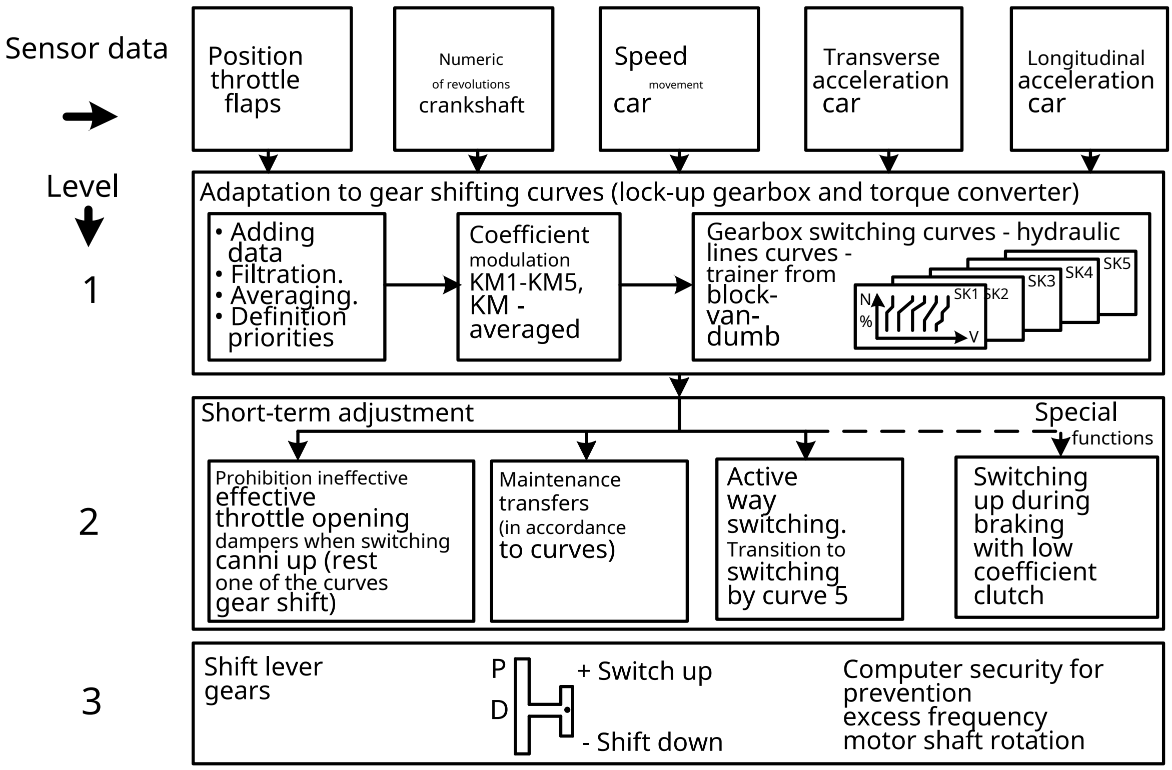


Figure 4.8 – Diagram of the Tiptronic gear shifting process

Blocking of the torque converter. A mechanical locking clutch can be used to increase the efficiency of the gearbox by eliminating slippage in the torque converter. The variables used to determine the torque converter lockout conditions are the engine load and the speed of the gearbox driven shaft.

Switching quality control. The accuracy with which the pressure in the friction elements is regulated depending on the magnitude of the transmitted torque affects the quality of switching; this pressure is established with the help of a special regulator. The smoothness of gear shifting can be increased due to a short-term reduction in the output power of the engine during the gear shifting period.

Protective circles. They are designed to avoid damage to the gearbox due to driver error, while the system is activated by returning to the backup mode for faulty functions in the electrical circuit.

End controls. Such elements of electrohydraulic conversion as solenoid valves and pressure regulators provide communication between electronic circuits and hydraulic circuits.

4.7 Design and calculation of stepped gearboxes

Stepped gearboxes with gear wheels are most widely used in modern cars due to their simple construction and high efficiency.

The methodical sequence of designing and calculating stepped gearboxes is as follows.

1. Selection of the type and structural scheme of the gearbox.
2. Determination of wheelbase.
3. Selection of the number of gears, modules and numbers of gear teeth of different stages

not

4. Checking the static strength of gear teeth.
5. Checking the contact strength of gear teeth.
6. Calculation of gearbox shafts for strength and hardness.
7. Calculation of slotted joints.
8. Calculation of synchronizers (synchronizer).
9. Selection of gearbox bearings.

4.7.1 Selection of the type and structural scheme of the gearbox

The choice is made taking into account the type and purpose of the designed car, as well as based on the analysis of modern and promising design solutions of similar mechanisms.

In the modern automobile industry, four-speed gearboxes are usually installed on especially small and small-volume passenger cars, four- or five-speed on medium-sized cars, and hydromechanical transmissions are used on high-class cars.

Trucks, especially those of small and medium capacity, in most cases have four-, five- and six-speed gearboxes. Trucks, especially those with a large carrying capacity, are usually equipped with a hydromechanical transmission.

4.7.2 Determination of center distance

Determination of interaxial distance A is carried out according to the empirical form-tallow

$$A = a \sqrt[3]{M_{e\max}}, \tag{4.24}$$

where $M_{e\max}$ – maximum torque of the engine, N·m;

a – experimental coefficient depending on the type of car.

For trucks and buses at their base a=17...19.5; for passenger cars a=14.5...16; for cars with diesel engines a=20.5...21.5; for additional and distribution boxes a=17...21.5.

4.7.3 Selection of the number of gears and determination of gear ratios

The selection of the number of gears and determination of gear ratios is carried out on the basis of traction calculation [8].

If the gear ratio is 7, then a box scheme with 4 stages is chosen, at i_{KI} = 6.6...10 – with 5 degrees; for heavy machines in which i_{KI}= 8.5...10, - with 6th, 8th and a large number of degrees.

Gear ratio of gears of permanent engagement $i_{software}$ is roughly equal to the gear ratio of the first gear i'_{I}

$$i_{software} \cong i'_{I} \cong \sqrt{i_{KI}}, \tag{4.25}$$

where i_{KI} – gear ratio of the first gear.

The engagement module is determined by an empirical formula

$$m = \frac{2A\cos\beta}{Z_1(i_{software}+1)}, \tag{4.26}$$

where Z_1 - the number of teeth of the driving gear (provided the teeth are cut $Z_1 \geq$ 13);

β – spiral angle of helical gears.

The normal coupling modulus calculated according to the dependence must be specified in accordance with GOST 9563.

The number of teeth of the gear of the first gear of the intermediate shaft is determined from the condition of operation of the teeth without undercutting.

$$Z_{m-1} \geq Z_{\min} = \frac{2f}{\text{son}_2 a}\cos^2\beta, \tag{4.27}$$

where $f \leq 0.8$ – tooth height factor;

a=20° – engagement angle;

β=20...30° – for trucks;

β=30...45° – for cars.

With spur gears of the first gear β= 0°. The number of hex teeth the pitch of the first gear of the driven shaft is determined

$$Z_m = Z_{m-1} \cdot i'_{i}. \tag{4.28}$$

The direction of the helical line of teeth in all helical gears of the intermediate shaft must be the same.

After determining the number of gear teeth of the gearbox, it is necessary to specify the interaxial distance by selecting the angle of inclination of the teeth β.

$$A = \frac{m_n(Z_1 + Z_2)}{2\cos\beta_{1-2}} = \frac{m_n(Z_3 + Z_4)}{2\cos\beta_{3-4}} = \ldots, \tag{4.29}$$

where m_n – normal engagement module, mm.

The width of the teeth of spur gears is approximately equal $b = (4{,}4 \ldots 7)m_n$ mm, for helical gears $b = (7.0 \ldots 8.6)m_s$.

4.7.4 Checking the static strength of gear teeth

Checking the static strength clarifies the correctness of the preliminary selection of the modulus value. The bending stress at the base of the teeth is calculated according to the following formulas:

for spur gears

$$\sigma = 0.36\frac{P}{bmy}, \tag{4.30}$$

for helical gears

$$\sigma = 0.24\frac{P}{bm_n y}, \tag{4.31}$$

where P is the circumferential force during contact in the engagement pole, which bleats

$$P = \frac{M}{r_D}, \tag{4.32}$$

where M is the torque on the gear shaft, which is calculated, and is waiting for $M_{e\max}$;

r_D – the radius of the dividing circle;

y is the coefficient of the tooth shape, which is chosen according to the tables or calculated for helical gears according to the given number of teeth.

$$Z_{PR} = \frac{Z}{\cos^3\beta}. \tag{4.33}$$

The width of the gear is determined empirically

$$b = (0.17 \ldots 0.25)A.$$

Permissible stress for cemented and cyanidated steels, for spur gears of the first and second gear, reverse 350...850 MPa (8500 kgf/cm^2), smaller values for passenger cars,

for gears III, IV, V gears 150...400 MPa (1500...4000 kgf/cm^2).

The tooth shape factor for uncorrected spur and helical gears is determined according to the table. 4.2.

If it is assumed to perform angular correction for the gears of the designed gearbox, then the value y, taken from table 4.2, it is worth remembering live on the correction factor K_a, the value of which is taken depending on bearing from the engagement angle:

$a°$	15°	17°30'	22°30'	25°
K_a	1.14	1.07	0.935	0.875

Table 4.2 – The value of the tooth shape factor

The number of teeth Z or Z_{PR}	Form factor tooth y	The number of teeth Z or Z_{PR}	Form factor tooth y
16	0.101	28	0.117
17	0.102	30	0.120
18	0.104	32	0.123
19	0.105	35	0.128
20	0.106	37	0.131
21	0.108	40	0.136
22	0.110	45	0.142
24	0.112	50	0.145
26	0.114	60	0.150

When applying a high correction, the tabular values of the coefficient y multiply by 1.14.

4.7.5 Checking the contact strength of gear teeth

The check of contact strength serves for an approximate assessment of the durability of gear teeth and is based on the use of the Belyaev–Hertz dependence. For a pair of gears made of the same material, the contact compressive stress is determined by dependence

$$\sigma_K = 0.418 \cos\beta \sqrt{\frac{PE}{b \sin\alpha \cdot \cos\alpha}\left(\frac{1}{r_1} + \frac{1}{r_2}\right)}, \qquad (4.34)$$

where r_1 and r_2 – radii of the initial circles of the leading and driven gears;

E – modulus of elasticity of the first kind, kgf/cm^2;

P is the circumferential force determined by the formula

$$P = \frac{i_{\text{software}} M_P}{r_1}, \qquad (4.35)$$

where $M_P = 0.5 M_{e\max}$ - calculation moment.

Permissible contact stress $[\sigma]_K$ (Table 4.3) should be within

Table 4.3 – Values of permissible contact stresses

Transfer \ Types of heat treatment	Cemented gears	Cyanized gears
First gear and reverse gear	1900...2000	950...1000
Constant engagement and higher gears	1300...1400	650...700

After performing the calculation, it is recommended to summarize the main gear parameters (module, number of teeth, diameter of the initial circle, height of the head and leg of the tooth, engagement angle, correction method, angle of inclination of the teeth to the initial cylinder, calculated bending stresses and contact stresses) in a separate table.

Gearbox gears are made of alloy steels 25KhGT, 25KhGM, 18KhGT, 30KhGT, 35Kh, 40Kh, 12KhN3A.

4.7.6 Calculation of gearbox shafts

The strength and stiffness calculation of shafts is used to check the strength and stiffness of predetermined shaft diameters and lengths.

The diameters of the shafts are chosen from the conditions of the greatest hardness. For intermediate and driven shafts

$$d \cong 0.45A, \tag{4.36}$$

where A – interaxial distance, mm.

For the drive shaft

$$d = 4.9\sqrt[3]{M_{e\max}}, \tag{4.37}$$

where $M_{e\max}$ – maximum torque, N mm. The diameter of the shafts can be determined from the ratios:

$\frac{d}{l} = 0.16 \ldots 0.18$ – for driving and intermediate shafts,

$\frac{d}{l} = 0.18 \ldots 0.21$ – for the driven shaft.

The oil level should be 35...45 mm above the axis of the intermediate shaft. The volume of the oil bath for the gearbox:

trucks

$$Q = (2.6 \div 3.3) \cdot 10^{-3} N_e m, \text{ l}, \tag{4.38}$$

passenger cars

$$Q = (0.92 \div 1.4) \cdot 10^{-3} N_e m, \text{ l}, \tag{4.39}$$

where m – the number of all gears in the gearbox;

N_e – engine power, kW.

The strength of the gearbox shafts is checked under the joint action of torsion and bending. The material for the shafts is steel 25ХГМ, 35Х, 40Х.

With known force loads on the shafts, the reactions of the shaft supports are determined starting from the secondary shaft. The calculation is performed for all stages of the gearbox. The reactions of the supports are determined using equilibrium equations for the spatial system of arbitrarily located forces [4].

The strength of the gearbox shafts is determined based on the torsional and bending deformation of the shafts. In fig. 4.9, 4.10 and 4.11 show the calculation schemes of the secondary, intermediate and primary shafts with helical gears.

The forces acting in helical gearing are determined by the dependence we:

district force

$$P_X = \frac{M_X,}{r_X} \tag{4.40}$$

radial force

$$R_X = \frac{tg\alpha_X,}{\cos\beta_X} \tag{4.41}$$

axial force

$$Q_X = P_X tg\beta_X, \tag{4.42}$$

where M_X – moment on the shaft H-th gear, due to the maximum torque of the engine.

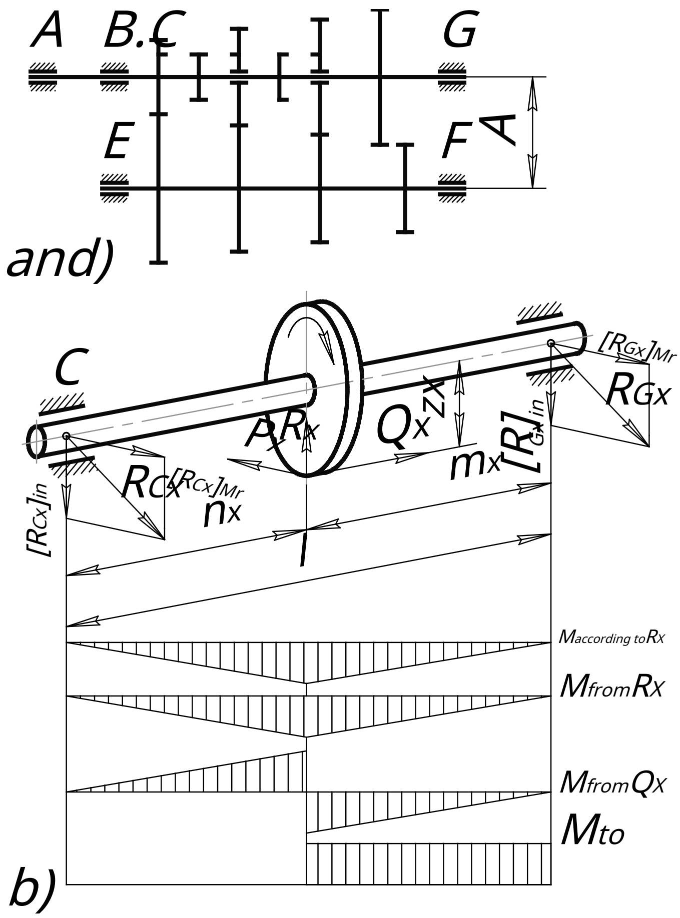


Figure 4.9 – Calculation schemes of the secondary shaft with helical gears

For all three shafts, it is recommended to submit two calculation schemes in the vertical and horizontal planes. According to the found value of the moments that bend the shaft in the vertical M_{call} and in horizontal $M_{according\ to}$ planes, the full bending moment is determined

$$M_n = \sqrt{M_{call}^2 + M_{according\ to}^2}. \qquad (4.43)$$

According to the found value of bending M_{with} and cool M_{to} moments, the equivalent moment is determined

$$M_E = \sqrt{M_{with}^2 + M_{to}^2}. \qquad (4.44)$$

The equivalent bending stresses are determined by dependence

$$\sigma_E = \frac{M_E}{W_I},$$

where W_I – axial moment of resistance,

see[3]. For a solid shaft

$$W_I = 0.1\,d^3. \qquad (4.45)$$

If there is a slotted cut on the shafts

$$W_I = \frac{\pi d_{in}^4 + B_{Sh}\,(d_H + d_{in})^2\,(d_H - d_{in})\,Z_{Sh}\,d^3}{32\,d_H}, \qquad (4.46)$$

where d_H and d_{in} – external and internal diameter of slots;

Z_{Sh} and B_{Sh} – the number of slots and their width.

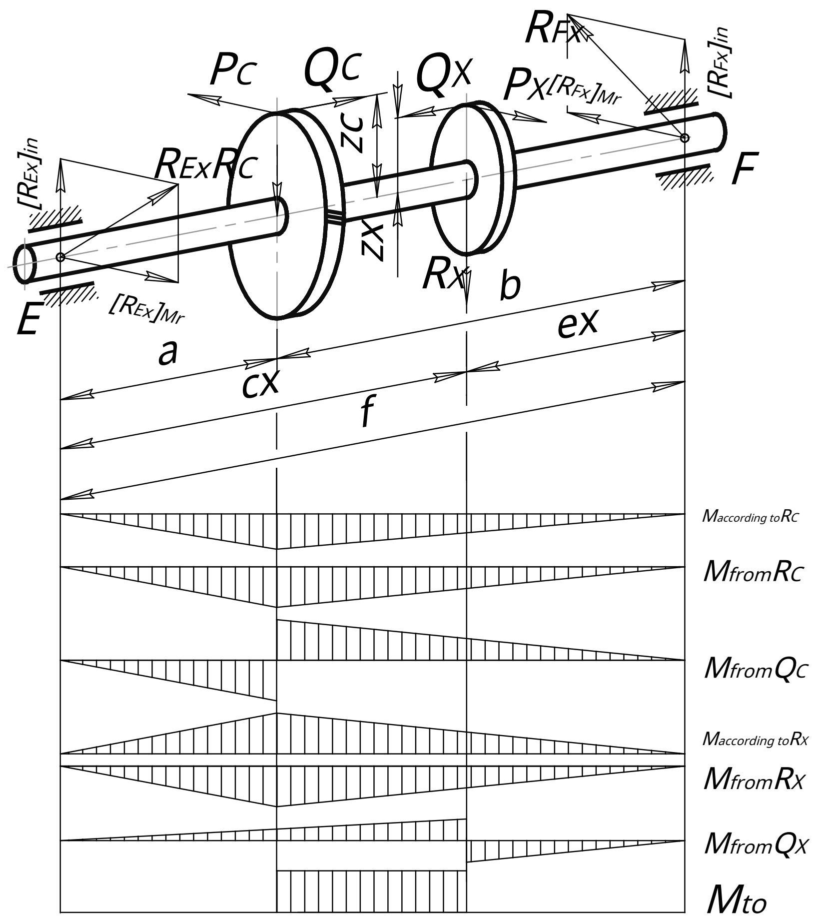


Figure 4.10 - Calculation scheme of the intermediate shaft with helical gears

The equivalent stresses obtained according to dependence (4.44) should not exceed $[\sigma]_{r}$=60...70 MPa (600...700 kgf/cm2).

When designing a gearbox with an intermediate shaft in the form of a block of gears, only its axis, which works on bending deformation, is calculated.

Deflections of shafts and angles of rotation of cross-sections are determined separately for acting forces P_X, R_X and in the locations of the gears.

Total deflection f and the angle of rotation of the sections Q_Σ should not exceed to touch $[f]$=0.13...0.15 mm at higher stages and $[f]$=0.15...0.25 mm below whose degrees The maximum angles of rotation of the shaft sections should not exceed 0.2 mm.

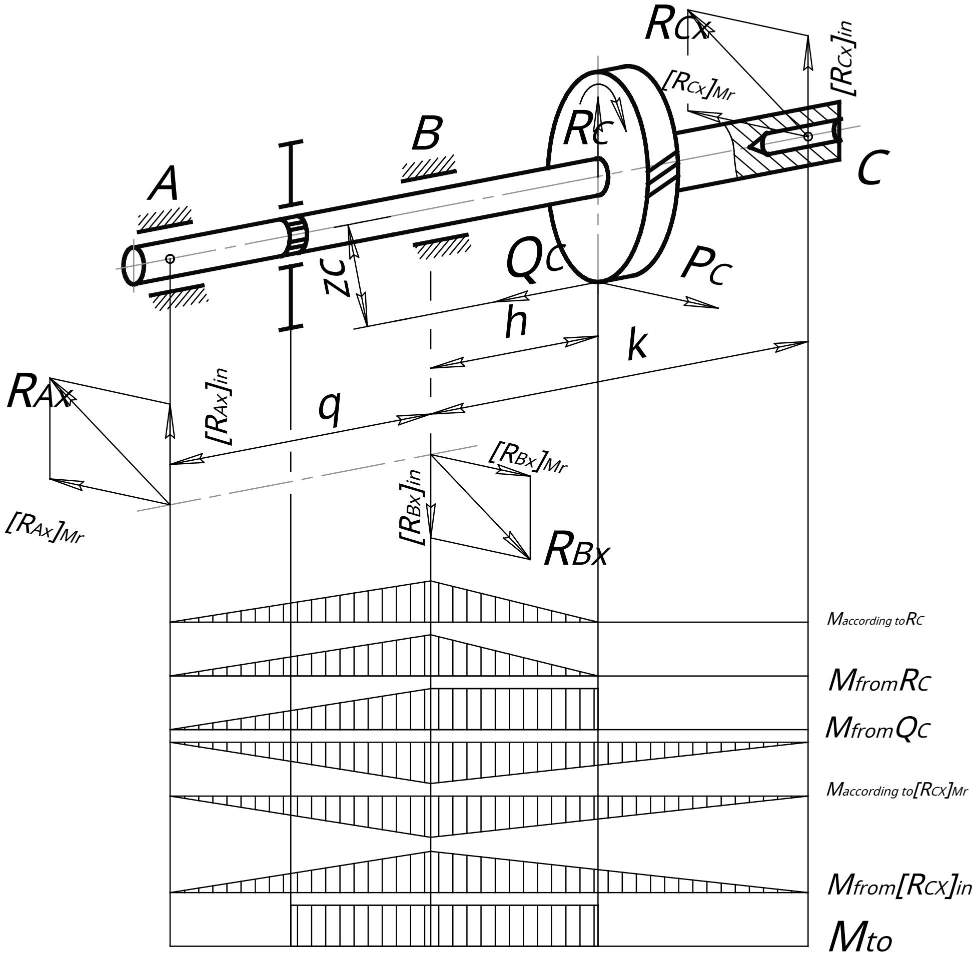


Figure 4.11 - Calculation scheme of the primary shaft with helical gears

4.7.7 Calculation of spline joints of gears

The calculation of slotted joints is performed only for crumpling, because the cut of the slots is not observed in practice. The crumpling stress is determined by the formula:

$$\sigma_{Sh} = \frac{8 M_{K\,\max}}{0.75 Z_{Sh} l_{Sh} \left(d_H^2 - d_{in}^2\right)}, \tag{4.47}$$

where $M_{K\max}$ – maximum torque on the calculated shaft; Z_{sh} and l_{sh} – the number of slots and their working length; d_H and d_{in} – external and internal diameters of the slots.

The crumpling stresses calculated by formula (4.47) should not exceed 40 MPa (400 kgf/cm^2).

4.7.8 Calculation of synchronizers

The calculation consists in the correctness of the preliminary selection of individual structural elements of the synchronizers. If the cones are not jammed, their angle should be greater than the friction angle $tg\alpha \geq \mu$ and is accepted within a = 7...12°. The moment of friction on the cones is equal

$$M = T\mu r = \frac{P}{\mathrm{son}\,a}\mu r, \qquad (4.48)$$

where μ is the coefficient of friction adopted in the calculations μ=0.1...0.12;

P – the force acting on the synchronizer cones from the driver;

T – normal pressure on the cones;

r is the average radius of the cone.

The axial force required to equalize the rotation speeds of the shaft and gear is determined by the formula:

$$Q \geq \frac{M\,\mathrm{son}\,a}{r\mu}. \qquad (4.49)$$

To prevent premature switching off of the transmission, the following ratio is necessary:

$$P_1 \geq Q\,tg\beta, \qquad (4.50)$$

where P_1 – circumferential force holding the locking pin in the deep lazy, equal to the ratio; $\frac{M}{r_1}$

β - half of the angle at the top of the locking pin (angle of course is within 35...40°);

r_1 is the average radius of the blocking surfaces.

If the basic geometric parameters of the synchronizer are correctly selected, the condition must be fulfilled

$$tg\beta \leq \frac{M \cdot r}{\mathrm{son}\,r_1}. \qquad (4.51)$$

4.7.9 Selection of gearbox bearings

The choice is based on ensuring the longevity of the bearings and is determined by the efficiency factor based on the average operating mode.

The coefficient of serviceability of the bearing is calculated according to the formula

$$C = Q_E K_K \cdot K_E \cdot K_\tau (n_E h)^{0.3}, \qquad (4.52)$$

where K_K, K_E, K_τ – coefficients characterizing the operating modes of bearing and take into account, respectively, the rotating ring, the influence of dynamic loads and temperature conditions. When calculating for stepped gearboxes should be accepted $K_K = K_E = K_\tau = 1.0$;

n_E – equivalent number of revolutions of the bearing;

h – the estimated durability of the bearing. The equivalent load is determined by the formula

$$Q_E = \sqrt{\frac{a_1}{100}\beta_1 Q_1^{3.33} + \frac{a_2}{100}\beta_2 Q_2^{3.33} + \ldots + \frac{a_n}{100}\beta_n Q_n^{3.33}}, \qquad (4.53)$$

where $a_1, a_2, \ldots, a_n$ - the average operational duration of the use of different number of stages of the gearbox, adopted according to table 4.4;

$\beta_1, \beta_2, \ldots, \beta_n$ is the coefficient of the number of revolutions of the corresponding shaft on each editors:

$$\beta_1 = \frac{1}{i_{K\mathrm{I}}},\ \beta_2 = \frac{1}{i_{K\mathrm{II}}}, \ldots, \beta_n = \frac{1}{i_{Kn}},$$

$Q_1, Q_2, \ldots, Q_n$ – combined reactions in the bearing at each gear, determined calculated by the formula:

$$Q_1 = R_1 + mA_1,\ Q_2 = R_2 + mA_2,\ \ldots,\ Q_n = R_n + mA_n,$$

where $R_1, R_2, \ldots, R_n$ – radial reactions acting on the bearing;

m – coefficient of reduction of axial load to radial load (m=1.5);

$A_1, A_2, \ldots, A_n$ – axial loads acting on the bearing.

Table 4.4 – The average operational duration of the use of different stages of the gearbox (%)

Types cars	Values for different degrees gearboxes										
	I	II	III	IV	V	VI	VII	VIII	THEM H		Z.h.
Lightweight	1 1	4 3	20 6	75 15	 75						0.3
Cargo and buses	1 0.6 0.5 0.4 0.3	3 1.8 1.5 0.8 0.5	14 7.6 5 1,2 1	82 20 10 2.6 1.8	 79 23 6 4.4	 60 14 8	 25 12	 50 18	 54		0.5
The same, with the addition forge KP	0.2	0.4	0.8	1,2	2.4	5	8	12	20	50	0.5
Cars-dump trucks	4 3	11 6	18 11	26 16	41 23	 41					2.5

The radial and axial load on the bearing is determined by the calculated moment on the primary shaft of the gearbox, taken equal to half of the maximum torque of the engine.

The value of the equivalent number of revolutions of the bearing n_E determined by based on average operating speeds (V_E= 30... 35 km/h for trucks, V_E= 35...40 km/h for passenger cars
cars and long-distance buses).

Estimated durability of the bearing h is determined by dependence

$$h = \frac{L_{KP}}{V_E} \tag{4.54}$$

where L_{KP} – mileage of the car before overhaul, km.

The minimum service life of gearbox bearings should be 6000...7000 hours.

Questions for self-control

1. What is a gearbox and what is its purpose?
2. List the main requirements for a gearbox.
3. How and by what features are gearboxes classified?
4. What are the types of step gearboxes? Rename them weights and disadvantages.
5. On what types of cars and why are multi-shaft co- gear making?
6. Describe the working process of a mechanical gearbox country house
7. What do you know about the features of the working process of the planetary box gears?
8. List and describe the features of the hydrotrans- shaper
9. What are the features of automatic transmissions?
10. Name the sequence of construction and calculation of stepped gearboxes.
11. How are gearbox bearings selected?
12. What is the sequence of calculation of gearbox shafts?
13. What are the features of the calculation of spline joints of gears?
14. What is a synchronizer and how is it calculated?

5 CARDAN TRANSMISSION

Cardan transmission is designed to transfer torque from one unit to another if the axes of their shafts change their relative position or do not lie on the same straight line.

5.1 Requirements for cardan transmissions

1. Transmission of torque at all possible operating conditions angular velocities ω and angles γ between the axes of the shafts.
2. High efficiency even at significant angles γ.
3. Minimal vibrations and noise.
4. Absence of significant axial forces and activations in compensating junction assembling.

5.2 Classification of cardan gears

1. According to the design of the cardan transmission:

a) open or closed (inside the crankcase, pipe, etc.); b) single-link or multi-link (with intermediate supports). 2. According to the design of cardan joints (hinge, joint, pivot):

a) unequal angular velocities (elastic γ< 3-5° and tough γ<15-20°);

b) equal angular velocities (balls with dividing grooves γ<32° , ball with a dividing lever γ<38° , cams γ<40°).

5.3 Working process of cardan gears

5.3.1 Cardan joints

When transmitting rotary motion at an angle with the help of hinges, difficulties arise in ensuring uniform rotation of the shaft located after the hinge. From fig. 5.1, *and* it follows that $V_A=\omega_1 r_1=\omega_2 r_2$, and know-what is the condition $\omega_2=\omega_1$ feasible if $r_2=r_1$. When turning shaft 1, for example, by 180°, r_1 and r_2 will decrease, but their equality must be preserved. So, the point of contact *AND* shafts 1 and 2 should move along the bisector plane OO $(\phi_1=\phi_2)$ when rotating the shafts.

This condition is not fulfilled in a rigid joint with unequal angular velocities (Fig. 5.1, *b*), because spikes AND_1 and IN_1 crosshairs 3 move in the plane AT_1AT_1, perpendicular to the shaft axis 1, a spikes AND_2 and IN_2- in the plane AT_2AT_2, perpendicular to the shaft axis 2.

If shaft 1 turns to an angle a_1, shaft 2 will return to the corner a_2, why?

$$tg\alpha_1 = tg\alpha_2 \cos\gamma. \qquad (5.1)$$

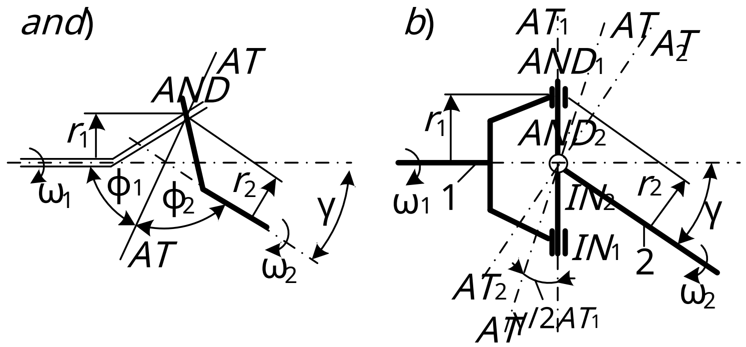


Figure 5.1 - Diagrams of cardanic gear joints

Differentiating (5.1), we get

$$\frac{\omega_1 date}{\cos a_1} = \frac{\omega_2 date}{\cos a_2}\cos\gamma$$

and after transformations

$$i_K = \frac{\omega_1}{\omega_2} = \cos\gamma\left(1 + \text{son}^2 a_1 tg^2\gamma\right). \qquad (5.2)$$

It follows from (5.2) that $\omega_2 \neq \omega_1$, and the unevenness coefficient (Fig. 5.2, *and*)

$$K_n = \frac{\omega_{2\,max} - \omega_{2\,min}}{\omega_1} = \frac{1}{\cos\gamma} - \cos\gamma = \frac{1-\cos^2\gamma}{\cos\gamma} \qquad (5.3)$$

at $\gamma < 10°$ is small. Uneven rotation of the shaft can be converted into uniform, if you put the second hinge with a broken angle $\gamma_2 = \gamma_1$ the first hinge, and the forks of the hinges of the unevenly rotating shaft must lie in the same plane. This rule is valid for any even number of hinges. With three hinges (Fig. 5.2, *b*)

$$\cos\gamma_1 \cos\gamma_2 = \cos\gamma_3, \qquad (5.4)$$

moreover, the cosines of the angles of those hinges whose forks are located in the same way are multiplied (1 and 2 in Fig. 5.2, *b*).

In fig. 5.3 shows the diagram of the forces acting on the forks and the hinge cross, without taking into account the efficiency η_K:

$$P_1 = \frac{M_1}{2R}, \quad P_2 = \frac{M_1 i_K}{2R}, \qquad (5.5)$$

$$T_1 = \frac{M_1 \text{son}\, a_1\, tg\gamma}{2R}, \quad T_2 = \frac{M_1 \cos a_1 \cdot \text{son}\,\gamma}{2R}\sqrt{1 + \text{son}^2 a_1 tg^2\gamma}, \qquad (5.6)$$

$$Q_1 = \sqrt{P_1^2 + T_1^2} = \frac{M_1}{2R}\sqrt{1 + \text{son}^2 a_1 tg^2\gamma}, \quad Q_2 = \sqrt{P_2^2 + T_2^2}. \qquad (5.7)$$

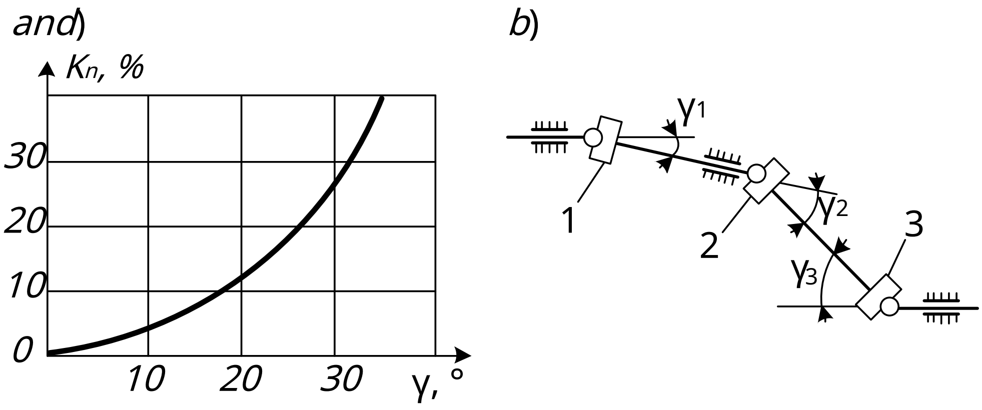


Figure 5.2 - Features of rotation of cardan shafts

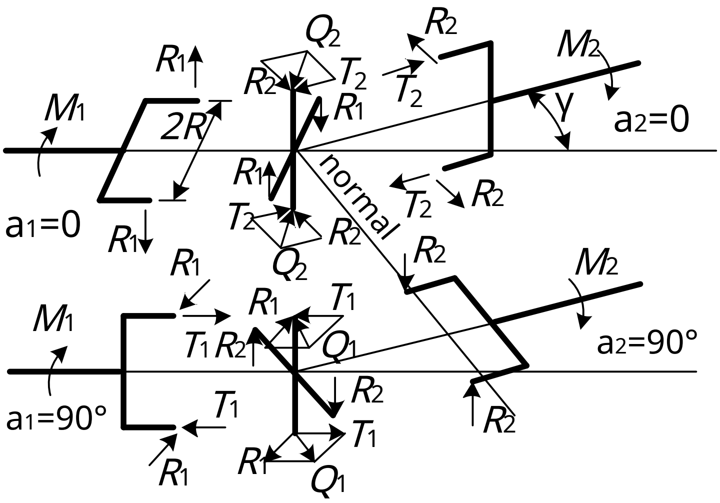


Figure 5.3 – Diagram of the forces acting on the forks and the hinge cross

Maximum values $Q_{1\,max}$, $T_{1\,max}$, $P_{2\,max}$ will be at a_1 = 90° , a $Q_{2\,max}(<Q_{1\,max})$ and $T_{2\,max}(<T_{1\,max})$ - at a_1 = 0°.

Moments: at the crossroads $M=2Q_1R$ – steep, on the first fork $M_1 = 2P_1R$ - cool and $M_{1u}=2T_1R$ – flexible, on the second fork $M_2 = 2P_2R$ - cool and $M_{2u}=2T_2R$ - bending Reactive moment on drive shaft $M_3=M_2=M_1=M_1(i_K-1)$ when changing i_K during rotation according to equation (5.2). The intermediate shaft of a two-hinged cardan transmission does not have a reactive moment, if $\gamma_2=\gamma_1$ and the forks of its hinges are harvest in one plane.

In addition, a pulsating torque occurs on shaft 2 (Fig. 5.3).

due to the fact that $\frac{d\omega_2}{date}\neq 0$.

$$M_I=AND_{sq2}\frac{d\omega_2}{date}, \qquad (5.8)$$

where AND_{sq} is the moment of inertia of the shaft2.

Efficiency of cardan joints η_{ksh} depends on the angle γ and is equal to 1÷0.99 for a rigid hinge of unequal angular velocities at γ=0...15°; 0.98 for hinges of equal angular velocities at γ= 25°.

5.3.2 Vibrations of cardan gears

Consider the condition of balance of forces $\sum P_n=0$ rotating cardan shaft (driveshaft), considering that its ends, which are installed in supports, do not have transverse movements. Centrifugal force

$$F=m(y+a)\omega^2$$

causes additional deflection in shear *and* center of gravity due to imbalance (m –mass of the shaft). The force is balanced by the elastic force of the shaft

$$P=cy\frac{EI_p}{l^3},$$

where l –shaft length, cm; I_p– polar moment of inertia of the section, see4; *with*– a coefficient that depends on the nature of the load and the type of supports.

From the condition $F=R$ we receive

$$y=\frac{ma\omega^2}{\frac{cEI_p}{l^3}-m\omega^2}. \tag{5.9}$$

From (5.9) $y=\infty$, if the denominator is zero. So, the critical angle speed

$$\omega_{KP}=\sqrt{\frac{cEI_p}{Jr^3}}. \tag{5.10}$$

For a solid shaft with a diam D: $I_p=\frac{\pi D^4}{64}$, $m=\frac{\pi D^2}{4}l\frac{\gamma}{g}$. For shaft that rests freely in the supports and is uniformly loaded along its length, $c\approx100$, for a shaft with engaged ends $c\approx20$. Let's take c=100 and from (5.10) we obtain

$$n_{KP}=12\cdot10^4\frac{D}{l^2}, \tag{5.11}$$

where D and l in m and γ = 7800 kg/m³ for steel.

For tubular shaft $I_p=\frac{\pi}{64}\left(D^4-d^4\right)$, $m=\frac{\pi\left(D^2-d^2\right)}{4}l\frac{\gamma}{g}$ and

$$n_{KP}=12\cdot10^4\frac{\sqrt{D^2+d^2}}{l^2}. \tag{5.12}$$

It is recommended n_{max}, which corresponds to $V_{a\max}$ and there are 20÷35% lower than n_{KP}. However, damage to the cardan shaft and extension was observed gearbox in this case too. It turned out that it takes in fluctuations participation is not only the cardan shaft, but also the entire transmission and engine. At the same time, several resonant forms of oscillations are possible. In fig. 5.4 shows the dependence

ness of movements $\bar{f} = \frac{f}{P}$, attributed to the disturbing force, from the frequency of pouring *n* in Hz.

points *B* and *WITH* correspond to resonant oscillations of the power unit and cardan shaft, point *AND* – resonant oscillation of only the cardan shaft with zero movement along the front and rear joints. mode *AND* corresponds to the equation (5.12), but mode *IN* occurs at lower frequencies.

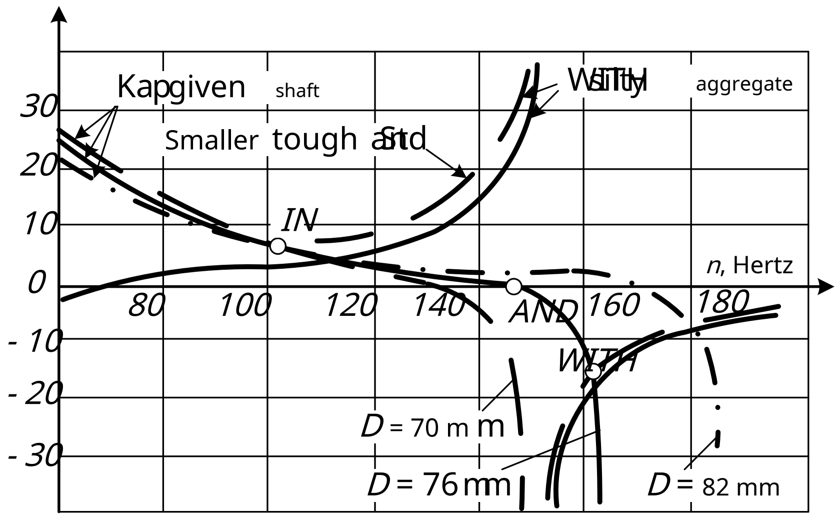


Figure 5.4 – Dependence of displacements attributed to the disturbing force, from the frequency of oscillations

From fig. 5.4, it can be seen that increasing the diameter of the cardan shaft really significantly shifts the point *AND*, but does not practically affect the position of the point *B*. Shift point *B* is provided by increasing the rigidity of the power unit. Reco- is required to provide below the point *B* approximately on 600 rpm (10 Hz).

Nowadays, the calculation of vibration of the transmission and the power unit is usually carried out on the basis of the Michaelestad method with a number of clarified provisions and assumptions and with the use of a computer.

The folded cardan shaft is subject to dynamic balancing with the capacity (15...25)·10 N·m, for cars with a carrying capacity below 5 t and 100·10^4- with a load capacity of 5 tons and above, (10...20)10 N·m – for cars.

5.4 Design and calculation of cardan gears

When designing a cardan transmission, it is necessary to consistently solve such issues.

1. Selection of the type and structural scheme of the cardan transmission.

2. Selection of the cross-section sizes of cardan shafts and verification of their strength those.

3. Checking the selected sizes and cross-sections of shafts during critical operations tach

4. Determination of shaft twisting angles.

5. Checking the strength of the forks and crosspieces of cardan joints, as well as splined joints.

6. Selection of needle bearings of cardan joints.

5.4.1 Selection of the type and structural scheme of the cardan transmission

The choice is made based on the type and purpose of the car's layout scheme, as well as on the basis of the analysis of the designs of similar mechanisms installed on the cars of the selected prototypes [2].

5.4.2 Selection of cross-section sizes of cardan shafts and their verification strength

Length l cardan shaft depends on the layout scheme of the car mobile and the static angle of inclination of the cardan shaft. External D_n and internal her D_{in} the diameters of the cardan shaft pipe are selected from the structural not, and the thickness of the pipe wall is usually 1.8...3.5 mm.

The torsional stress in the dangerous section of the shaft during the transmission of the maximum torque is calculated by the formula:

$$\tau = \frac{16 \cdot K_\sigma \cdot M_{e\max} \cdot D_n \cdot i_{kI} \cdot i_o \cdot i_p}{\pi \cdot (D_n^4 - D_{in}^4)}, \quad (5.13)$$

where K_σ is a coefficient that takes into account the influence of normal compressive stresses and bend in cardan transmission. In the calculations, it is accepted for passenger cars mobiles $K_\sigma = 2{,}0...3{,}0$; for cargo $K_\sigma = 1{,}5...2{,}0$.

The splined shank of the cardan shaft is also subject to torsional calculation:

$$\tau = \frac{16 \cdot K_\sigma \cdot M_{e\max} \cdot i_{kI} \cdot i_o \cdot i_p}{\pi \cdot d^3}, \quad (5.14)$$

where d – the diameter of the shaft of the cardan shaft along the internal slots. Permissible cardan shaft stresses are 200...250 MPa (2000...2500 kgf/cm^2).

5.4.3 Checking the selected sizes and cross-sections of the shafts in case of critical ones revolutions

In the designs of modern cars, cardan transmissions of the open type have become predominant. The critical number of revolutions of such a cardan shaft is determined by the formula:

$$n_{cr} = 12 \cdot 10^6 \cdot \frac{\sqrt{D_n^2 + D_{in}^2}}{L}, \quad (5.15)$$

where L - the working length of the cardan shaft, cm.

The margin at the critical number of revolutions for cardan shafts lying freely in the supports is equal to:

$$\frac{n_{cr}}{n_{max}}=1.2...2.0;$$

where n_{max} is the maximum number of revolutions of the corresponding cardan shaft the maximum speed of the car.

If there is a suspension bearing in the cardan transmission, only the main shaft can be calculated for the critical number of revolutions, because its length is longer.

5.4.4 Determination of twisting angles

Determination of twisting angles is performed only for long shafts according to the dependence:

$$\theta=\frac{180}{\pi}\cdot\frac{32\cdot M_{e\max}\cdot i_{tr}\cdot a}{G\cdot(D_n^4-D_{in}^4)},\qquad(5.16)$$

where i_{tr} – gear ratio of the car transmission.

The allowable turning angle for modern cars is 3...9 in a lower gear° per one meter of shaft length. Cardan shafts are pressed from steels 15 and 20.

5.4.5 Checking the strength of the forks and crosspieces of cardan shaft hinges the editor

The check is performed at maximum torque, with the first gear of the gearbox engaged. The calculation scheme of the cardan fork is shown in Fig. 5.5. The cardan shaft fork is designed for bending and torsional deformation in the dangerous section N–N.

Bending moment in dangerous section

$$M_{and}=\frac{M_{there\ are\max}\cdot i_{tp}\cdot l}{2\cdot r},\qquad(5.17)$$

torque in the dangerous section

$$M_{to}=\frac{M_{there\ are\max}\cdot i_{tp}\cdot with}{2\cdot r}.\qquad(5.18)$$

Moment of resistance of the fork section: axial

$$W_u=\frac{b\cdot h^2}{6},\qquad(5.19)$$

polar

$$W_p=a\cdot b^2\cdot h,\qquad(5.20)$$

where a – coefficient depending on the ratio of the sides of the cross section of the fork, taken according to the table. 5.1.

If the cross-section of the cardan fork has an oval shape, then the moments of resistance are determined by dependencies: axial W_u=0.1·b·h^2, polar W_b=0.2·b^2·h.

Table 5.1 - Value of the coefficient a

h/b	1.0	1.5	1.75	2.0	2.5	3.0	4	10
a	0.208	0.231	0.239	0.246	0.258	0.267	0.282	0.312

The bending and torsional stress in the cross section of the fork of the cardan joint is determined according to known dependencies

$$\sigma_u=\frac{M_u}{W_u};\quad \tau_{Cr}=\frac{M_{to}}{W_p}. \tag{5.21}$$

If the simultaneous action of bending and torque moments is taken into account, then the equivalent stress equal to

$$\sigma_\theta=\sqrt{\sigma^2_u+4\cdot\tau^2_{Cr}}. \tag{5.22}$$

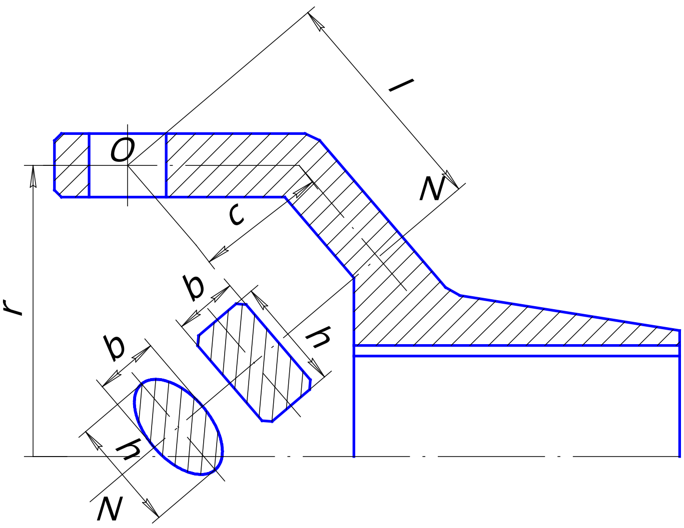


Figure 5.5 – Calculation diagram of the cardan fork

Working stresses should not exceed 150...200 MPa (1500...2000 kgf/cm^2). Forks are made of steels 35, 40, 45 and alloyed steels 30X, 35X.

The splines of the cardan shaft fork and the splined shank are calculated for cut and crumple according to a method similar to the calculation of the splines of the clutch shaft, however, the value is taken as the calculated moment $M_{emax}\cdot i_{p\ t}$. The permissible shear stress should be taken as equal30 MPa (300 kgf/cm^2), and crumpling - 65 MPa (650 kgf/cm^2).

The spike of the gimbal crosshead is designed for shear deformation, bending and crumpling. The circumferential force acting on the spike of the cross (Fig. 5.6),

$$P_{sh}=\frac{M_{there\ are\max}\cdot i_{tr}}{2\cdot r\cdot\cos\gamma_0}, \qquad (5.23)$$

where γ_0 – the angle of inclination of the cardan shaft on a stationary car with a full payload in the body; if $\gamma_0 < 10°$, then the value of $\cos\gamma_0$ is close to unity.

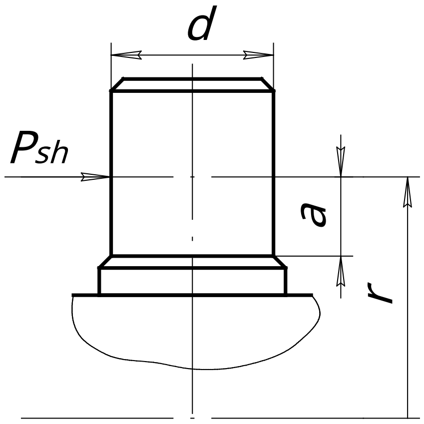


Figure 5.6 – Calculation scheme of the gimbal cross

Bending, shearing and crumpling stresses caused by force P_{sh}, determines are based on dependencies

$$\sigma_u=\frac{P_{sh}\cdot and}{0.1\cdot d^3}, \qquad (5.24)$$

$$\tau=\frac{4\cdot P_{sh}}{\pi\cdot d^2}, \qquad (5.25)$$

$$\sigma_{em}=\frac{P_{sh}}{2\cdot and\cdot d}. \qquad (5.26)$$

5.4.6 Selection of needle bearings of cardan joints

The choice consists in determining the permissible radial load on the bearing, the size of which determines the parameters of the bearings from the catalog.

$$P_{add}=790\,\frac{Z\cdot l\cdot\delta}{\sqrt[3]{n_n}}, \qquad (5.27)$$

where Z – the number of rollers (needles) of the bearing; l – working length of the roller, cm; δ – roller diameter, cm;

n_n – conditional number of revolutions of the bearing in minutes, determined by the maximum by the small value of the angular speed of rotation of the crosshead depending on the dependence

$$n_n=\frac{n_m}{i_{tr}}\cdot tg\,\gamma_0, \qquad (5.28)$$

where n_m – number of revolutions of the crankshaft of the engine at minimum torque

The durability of the bearing significantly depends on the crumpling stresses, which are determined by the average operational loading modes of the transmission

$$\sigma_{see} = \frac{M_{eEK},}{2 \cdot r \cdot d \cdot l} \tag{5.29}$$

where M_{eEK} is the average operating torque transmitted transmission and adopted depending on the value of the ratio $G_a / M_{e\max}$. If $G_a / M_{e\max} >$ 3.3; then $M_{CD} = M_{there\ are\max}$; at $G_a / M_{e\max} < 3.3$ magnitude M_{CD} is taken as a percentage of $M_{e\max}$ according to the schedule (Fig. 5.7).

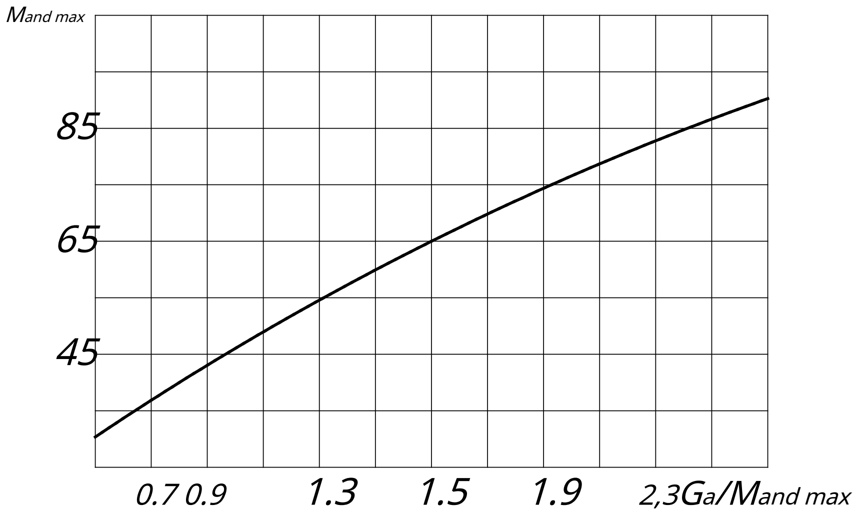


Figure 5.7 – Graph for determining the average operating profit of the car's transmission mode

The working stresses of crumpling, determined by dependence (5.29), should not exceed the permissible values, the values of which are accepted for trucks and buses of 8 MPa (80 kgf/cm^2) and cars 6 MPa (60 kgf/cm^2).

Questions for self-control

1. Purpose of cardan gears and cardan joints.
2. According to what features are cardan transmissions classified?
3. Requirements for cardan gears.
4. What is the critical frequency of rotation of the cardan shaft?
5. What parts and what loads are calculated in the cardan transmission what?

6 MAIN GEAR

The main gear is designed to increase the torque and reduce the speed of rotation to the values required for the driving wheels, as well as to transmit rotational movement at an angle.

In passenger cars, the main transmission in most cases consists of the leading and driven gears, hepoid (with the engine longitudinally located) or cylindrical (with the engine transversely located). The main gear ratio is usually in the range between 2.2:1 and 5.0:1 for the makes of cars produced.

On trucks, it is rarely possible to achieve the required values of transmission ratios with the help of a single unit consisting of a gear wheel and a gear; for this purpose, more complex main gears are used: single (up to i_0= 7); double central with leading six-
a thorn located in front or above; two-speed (i_0= 9); double
spaced (wheel gearbox with cylindrical gears); double spaced with a planetary wheel gearbox.

The use of spaced main gears with wheel and on-board gearboxes (including planetary ones) makes it possible to reduce the dimensions of the main gear and half-axle shafts, and, accordingly, to increase the road clearance (road clearance), which can be quite large even in the case of using high-power layout schemes.

6.1 Requirements for main transmissions

1. Ensuring high traction and dynamic qualities and fuel eco- nomics
2. High efficiency.
3. Low level of vibrations and noise.
4. Minimum dimensions in relation to the height from the center line: down - to ensure burning road clearance, upwards - to lower the floor level.
5. Placement of the through shaft in the main gear of the middle bridge for driving the rear axle. This allows you to avoid the use of a transfer box and simplify the cardan transmission in cars with wheel formula 6x 4.

6.2 Classification of main gears

1. According to the layout: a) separately from the gearbox, namely: in the driving bridge or in the form of on-board gears and wheel reducers; b) in one unit with a gearbox or power unit.

2. By type of transmission: chain, worm, gear (cylindrical, co-night, hepoid), combined.

3. By the number of pairs of engagements: a) single (cylindrical, conical, hepoid, worms); b) double central (flat or angular) or spaced.

6.3 Working process of main gears

The working process of the main gear is similar to the working process of the gearbox when the gear is engaged. However, unlike cylindrical gears, the axial shift of bevel gears breaks the engagement. This circumstance, as well as significantly greater loads with small dimensions, require the use of, in addition to increasing the rigidity of the crankcase, special measures that increase the rigidity of the structure:

a) conical radial thrust bearings with preload $f_0 = 0.02 + 0.04$ mm, which reduces the axial deformation and the total load sion on bearings;

b) a stop, fixed on the crankcase, for the driven bevel gear, which reduces its deformations under heavy loads at low gears;

c) a third bearing is placed at the apex of the cone of the driving gear, in addition to the cantilever installation of the gear.

Damage to the teeth from fatigue usually has the appearance of "smallpox-like wear" (pitting) on the lateral surfaces. The reduction of wear is ensured by increasing the rigidity of the structure (deflections of no more than 0.075 mm and only 0.25 mm push-off of the driven gear are allowed), the accuracy of the tooth profile, and the quality of the tooth surface. The same measures also reduce engagement noise. However, the errors that are the cause of the increased noise are usually tiny compared to the errors that were the cause of the increased wear.

The load on the leading bevel gear (Fig. 6.1), expressed in terms of the total force R, decompose into three mutually perpendicular forces P_1, N_1, S_1:

district force $P_1 = \dfrac{M_1}{r_{cp1}}$;

axial force $Q_1 = N_1 \mathrm{son}\,\delta_1 - S_1 \cos\delta_1 = P_1 \cdot tg\beta_1 \cdot \cos\delta_1 \left(\dfrac{tg\alpha \cdot tg\delta_1}{\mathrm{son}\,\beta_1} - 1 \right)$;

radial force $R_1 = N_1 \cos\delta_1 \pm S_1 \mathrm{son}\,\delta_1 = P_1 \cdot tg\beta_1 \cdot \cos\delta_1 \left(\dfrac{tg\alpha}{\cos\beta_1} \pm tg\delta_1 \right)$.

The sign in parentheses corresponds to the direction of the force from the apex of the cone to the base.

Axial force Q_1, directed to the base of the cone, makes it impossible to invoke gears whirring. Therefore, of course, the drive gears are on the left (view from the top of the cone) spiral, which allows you to avoid jamming when moving forward.

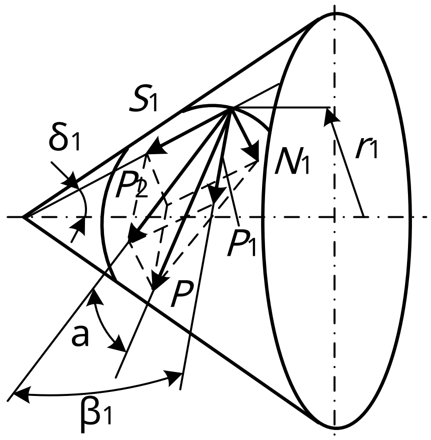


Figure 6.1 – Load on the leading bevel gear

Load on the driven bevel gear $(a_1 = a_2, \beta_1 = \beta_2)$: district power $P_2 = -P_1$; axial force $Q_2 = -R_1$; radial force $R_2 = -Q_1$.

Load on the driven hepoid gear $(a_1 = a_2, \beta_1 \neq \beta_2)$:

district force $P_2 = -P_1 \dfrac{\cos\beta_2}{\cos\beta_1}$;

axial force $Q_2 = -P_2 tg\beta_2 \cdot \cos\delta_2 \left(\dfrac{tg\alpha_2 \, tg\delta_2}{son\beta_2} - 1 \right)$;

radial force $R_2 = -P_2 tg\beta_2 \cos\delta_2 \left(\dfrac{tg\alpha_2}{\cos\beta_2} \pm tg\delta_2 \right)$.

Let's consider the features of single main gears (tables 6.1, 6.2).

Table 6.1 – Design features

Transfers	Worms	Hepoid	Conical oblique teeth
1	2	3	4
1	Small dimensions and weight	Moderate dimensions	Increased dimensions due to the driven gear
2	Almost any value $i_0 = k\dfrac{Z}{Z_r}$	$i_0 = \dfrac{D_2 n_0 \cdot \cos\beta_2}{D_1 n_0 \cdot \cos\beta_1}$	$i_0 = \dfrac{D_2 n_0}{D_1 n_0} = \dfrac{Z_2}{Z_1}$ <8, that it is not always possible due to the size and rigidity of the driven gear

Continuation of the table. 6.1

3	Possible upper or bottom location worm and passage drive shaft	Possible increase to of the vaginal lumen or lowering the floor level, as well as placement of the through shaft	The axes of the gears intersect, which does not allow placing the through shaft

Note. Z_k – the number of teeth of the worm wheel, Z_r is the number of steps taken by the worm, D_{n0} is the diameter of the initial circle, β – spiral angle ($\beta_1 \approx 45^0$, $\beta_2 = 25^0$ for hepoid six-thorn).

Table 6.2 - Features of the work process

Transfers	Worms	Hepoid	Conical oblique teeth
1	High smoothness for traction, low noise ness	Good grip smoothness quietness, little noise	Good engagement smoothness, moderate noise
2	Significant longitudinal teeth slippage, efficiency 0.90-0.94	Efficiency 0.96-0.98	High efficiency of 0.98
3	Low pressure in the contact of the teeth	Moderate pressures and axial loads	Large axial loads
4	Reduced requirements for manufacturing accuracy	Moderate requirements for accuracy and surface cleanliness	Increased requirements for the accuracy of manufacturing gears and crankcases

Double main gears are used: a) if specified i_0 can't implement in one pair of gears; b) if the driven gear is excessively large; c) to increase the road clearance or lower the floor level; d) to reduce loads on the differential and half-axle (spaced double main gear).

6.4 Design and calculation of the main transmission

When designing the main gear, it is recommended to specify the gear ratio of the main gear i_0, defined earlier in traction-dynamic calculation of the car. The clarification comes down to the selection of the numbers of teeth of the gears of the main gear. The choice of teeth must be made based on the existing designs of the main gears, which are similar in type and purpose to cars. The smallest number of gear teeth (smaller gear wheel of a pair) is selected depending on the gear ratio of the main gear (Table 6.3).

Table 6.3 – Dependence of the smallest number of gear teeth on the gear ratio of the main gear

i_0	2.5	3	4	5	6...8
$Z_{sh\min}$	15	12	9	7	6

The number of teeth of the wheel $Z_{to}=i_0 \cdot Z_{sh}$. Choosing Z_{sh} and Z_{to} for a conical pair, it is impossible to allow their multiple connections to avoid increased wear of gear teeth.

The spiral direction of the gear teeth for bevel gears is left; for hepoid gears – left at lower displacement and right – at upper displacement. For a conical pair, the end module m_s in the basis of division-th cone can be approximately determined by dependence

$$m_s = \frac{2 \cdot Z}{\sqrt{Z^2_{sh} + Z^2_{to}}}, \text{ mm} \tag{6.1}$$

where Z is the length of the generating cone of the dividing cone, roughly determined from

$$Z = A \cdot \sqrt[3]{M_{e\max} \cdot i_{kI}}, \tag{6.2}$$

where A – empirical coefficient. For truck bevel gears mobiles A=30, for hepoid transmissions of passenger cars A=25;

$M_{e\max}$ – maximum engine torque, kg·m;

i_{kI} – gear ratio of the first gear of the gearbox.

The obtained values of the module should be compared with the data for similar main gears of cars and, if necessary, clarified.

Width of toothed crown $b \approx 0.3 \cdot Z$.

End module in the middle section of the tooth

$$m_{with\ cp} = m_s \left(1 - \frac{b}{2 \cdot Z}\right), \text{ mm} \tag{6.3}$$

Average radius:

gears

$$R_{USSR} = \frac{m_{with\ cp} \cdot Z_{sh}}{2}, \tag{6.4}$$

wheels

$$R_{Kss} = \frac{m_{s\ c} \cdot Z_{to}}{2}.$$

Normal modulus and normal pitch in the middle section

$$m_{nss} = m_{with\ sp} \cdot \cos \beta_g,$$

$$t_{nss} = \pi \cdot m_{nss}, \tag{6.5}$$

where β_g is the angle of the spiral in the middle section $\beta_g \approx$ 35...45°. Spiral angle β_g is determined taking into account the overlap of the teeth:

$$\mathrm{tg}\beta_g = \frac{\varepsilon \cdot t_{nsr}}{k \cdot b},$$

where ε – tooth overlap ratio, ε= 1.25...1.5;

k is a ratio-dependent coefficient $\frac{b}{Z}$ [5]. For $\frac{b}{Z}$ = 0.3 k=1.18.

The value of the normal modulus pre-selected according to dependence (6.5), rounded to the nearest value provided by GOST 9563, is refined taking into account the bending stresses acting at the base of the tooth.

Checking the static strength of gear teeth is performed based on the maximum torque transmitted by the main gear.

The bending stress of gear teeth is determined by the following dependence thu:

$$\sigma = \frac{3 \cdot M}{y \cdot R_0 \cdot t_n \cdot Z \cdot (1 - \lambda^3)}, \text{ N/m}^2 \tag{6.6}$$

where M – calculated moment, N·m (for gears $M_{sh} = and_{kr} \cdot M_{there\ are\ max}$, for the wheel $M_{to} = and_{kr} \cdot i_0 \cdot M_{e\max}$);

R_0 – the radius of the base of the dividing cone, m: $R_0 = \dfrac{m_s \cdot Z}{2}$;

λ – tooth length factor: for a conical pair

$$\lambda_{sh} = \lambda_{to} = AND - \frac{b}{Z}$$

for a hepoid pair

$$\lambda_{sh} = AND - \frac{b_{sh}}{Z_{sh}}, \lambda_{to} = AND - \frac{b_{to}}{Z_{to}};$$

t_n – normal step at the base of the dividing cone, m:

$$t_n = \pi \cdot m_s \cdot \cos\beta_g; \tag{6.7}$$

y – tooth shape factor (value y is taken from the tables for given number of teeth Z_{Ave})

$$Z_{Ave} = \frac{Z}{\cos\phi \cdot \cos^3\beta_g}, \tag{6.8}$$

where ϕ is the angle of the generating dividing cone.

Table 6.4 gives the reference values of the allowable bending stresses of the teeth of some gears.

Table 6.4 – Allowable stress values for heat-treated (cemented) alloy steel gears

Transmission type	Conical		Hepoid	
The location of the presenter gears	console	between supports	console	between supports
Permissible stress, MPa	450	550	520	620

Contact stresses in the engagement of gear teeth and wheel

$$\sigma_{to} = 0.418 \cdot \sqrt{\frac{2 \cdot R \cdot THERE\ ARE}{b \cdot \text{son}\ 2a} \left(\frac{1}{\rho_{yes}} + \frac{1}{\rho_{ec}} \right)}, \tag{6.9}$$

where ρ_e is the equivalent radius of curvature of the gear tooth profile, wheel in the engagement strip, see:

$$\rho_{yes} = \frac{R_{sh.Wed}}{sos^2 \cdot \beta_{gsh} \cdot \cos\phi_{sh}}, \quad \rho_{ec} = \frac{R_{to.Wed}}{sos^2 \cdot \beta_{gk} \cdot \cos\phi_{to}},$$

where a – engagement angle;

R – circumferential force ($R = \frac{M_p}{R_{sh.Wed}}$);

M_p – calculated moment ($M_p = and_{kr} \cdot M_{there\ are\ \max}$).

Permissible contact stress in engagement $[\sigma]_{to}$=700...900 MPa (7000...9000 kg/cm^2).

When designing hepoid main gears, the angles of inclination of the helical line of the drive teeth β_{gsh} and conducted β_{gk} gears at the same angle of clutches are different in size; for automotive hepoid transmissions $\frac{\cos\beta_{gsh}}{\cos\beta_{gk}}$=1.3...1.5. The minimum number of teeth of the driving gear of the hepoid the main gear for trucks is 5, and for cars - 9.

Spiral angle β_{gsh} gears are selected depending on the number of teeth Z_{sh} (tab.6.5).

Table 6.5 – Dependence of the gear spiral angle on the number of its teeth

Z_{sh}	6-13	14-15	16 and over
β_{gsh}	50	45	40

Bias *THERE ARE* the axis of the gear must be within the range, mm:

$$E \leq A_E D_{approx}, \tag{6.11}$$

where $AND_{THERE\ ARE}$ – coefficient of relative sizes;

D_{approx} – diameter of the base of the dividing cone of the wheel, mm:

$$D_{approx} = AND_e \cdot \sqrt[3]{M_p}, \tag{6.12}$$

where AND_e is an empirical coefficient corresponding to the calculated moment M_p, calculated according to the driving wheel coupling conditions:

$$M_p = m_2 \cdot G_2 \cdot \phi, \text{ kg} \cdot \text{see}$$

Coefficients $AND_{THERE\ ARE}$, AND_e, and ϕ determined according to the table 6.6.

Table 6.6 – Values of coefficients $AND_{THERE\ ARE}$, AND_e, and ϕ

Odds / Cars	Lightweight	Cargo
$AND_{THERE\ ARE}$	0.2	0.125
AND_e	6.65	5.8...6.6
ϕ	0.65	0.85

Wheel calculation

Width of toothed crown b_{to} along the derivative of the dividing cone

$$b_{to} = (0.125...0.167) \cdot D_{approx}. \tag{6.13}$$

End module m_{sk} at the base of the dividing cone

$$m_{sk} = \frac{D_{approx}}{Z_{to}}. \tag{6.14}$$

Average radius $R_{to\ Wed}$ wheels

$$R_{to\ Wed} = \frac{D_{approx} - b_{to} \cdot \text{son}\,\phi_{to}}{2}, \tag{6.15}$$

where ϕ_{to} is the angle of the generating dividing cone of the hepoid gear wheel, equiv. valence according to the gear ratio of the conical pair $\text{tg}\,\phi_{to} = i_0$.

Angle ϕ_{to}' of the tangent in the engagement strip R

$$\text{tg}\,\phi_{to}' = \frac{\text{tg}\,\phi_{to}}{\cos a_0}, \tag{6.16}$$

where a_0 – the difference in the spiral angles of the gears

$$\text{son}\,a_0 = \frac{E}{R_{to\ Wed}}. \tag{6.17}$$

The spiral angle of the teeth of the wheels

$$\beta_{gk} = \beta_{gsh} - a_0.$$

Average normal modulus $m_{n\ av}$ and a normal step t_{nsr}

$$m_{n\ av} = \frac{2 \cdot R_{to\ Wed} \cdot \cos\beta_{gk}}{Z_{to}}, \qquad t_{n\ av} = m_{n\ av} \cdot \pi. \tag{6.18}$$

Gear calculation

The angle of the generating pitch cone of the gear $\phi_{sh} = 90 - \phi_{sh}'$, at the same time $\phi_{sh} + \phi_{to} < 90°$.

Average gear radius

$$R_{USSR} = \frac{m_{n\ av} \cdot Z_{sh}}{2 \cdot \cos\beta_{gsh}}. \tag{6.19}$$

Width b_{sh} toothed crown on the working of the dividing wheel

$$b_{sh} = \frac{\sqrt{R_{ok}^2 - E^2} - \sqrt{R_{k\min} - E^2}}{\cos\phi_{sh}}, \tag{6.20}$$

where $R_{ok} = \frac{D_{approx}}{2}$, $R_{k\min} = 2 \cdot R_{cr} - R_{approx}$.

End module at the base of the dividing cone

$$m_{ssh} = \frac{m_{nss}}{\cos\beta_{gsh}} + \frac{b_{sh}}{Z_{sh}} \cdot \text{son}\,\phi_{sh}. \tag{6.21}$$

Radius of the base of the dividing cone of the gear

$$R_{osh} = \frac{m_{ssh} \cdot Z_{sh}.}{2}$$

The length of the creative Z_{sh} dividing cone

$$Z_{sh} = \frac{R_{ok}.}{\mathrm{son}\,\phi_{sh}} \qquad (6.22)$$

At checks performed calculations necessary that $b_{sh} \approx (0.2...0.3) \cdot Z_{sh}$.

The calculation of hepoid gears for strength and durability is similar to the calculation of bevel gears.

For a double main gear, the total gear ratio

$$i_0 = \frac{Z_2 \cdot Z_4,}{Z_1 \cdot Z_3}$$

where Z_1 and Z_2 – the number of teeth of the leading and driven bevel gears;

Z_3 and Z_4 – the number of teeth of the leading and driven cylindrical gears.

The selection of the number of teeth should be carried out based on the existing designs of the main gears, which are similar in type and purpose of cars.

In order to relieve from the axial forces of the bearings of the intermediate shaft of the double main gear, the direction of the helical line of the teeth of the driving cylindrical gear is selected so that the direction of the axial force from this gear is opposite to the axial force from the driven bevel gear.

The selection of the module and the check of the static strength of the cylindrical gears of the double main transmission, as well as the assessment of their durability, are carried out in accordance with the main provisions of the methodology outlined for the gears of stepped gearboxes.

The calculation of the shafts of the main gear consists in checking their static strength under a complex load transmitted by torque and the action of bending moments in the vertical and horizontal planes from the forces that arise in gear engagements.

For helical-bevel gears, the circumferential, radial, and axial forces acting on the teeth and in engagement are determined by the following formulas:

$$P = P_1 = P_2 = \frac{M_{e\max} \cdot i_{tr}}{R_{USSR}}, \qquad (6.23)$$

$$R_1 = Q_2 = P \left(\frac{tg\alpha \cdot \cos\phi}{\cos\beta_g} - tg\beta_g \cdot \mathrm{son}\,\phi \right), \qquad (6.24)$$

$$Q_1 = R_2 = P \left(\frac{tg\alpha \cdot \cos\phi}{\cos\beta_g} + tg\beta_g \cdot \cos\phi \right), \qquad (6.25)$$

where P_1, R_1, Q_1 – circumferential, radial and axial forces acting, respectively on the tooth of the driving gear;

P_2, R_2, Q_2 - the same forces that act on the tooth of the driven gear.

Dependencies (6.23), (6.24) and (6.25) are valid for the case when the direction of rotation of the driving gear is opposite to the direction of the angles of inclination of the helical line of its teeth. If the direction is opposite to the specified one, then in the dependencies (6.24) and (6.25) the signs before the second term should be changed to the opposite.

In a hepoid pair of gears, the radial and axial forces are determined by the dependencies:

$$R_1 = \frac{P}{\cos\beta_{gsh}}(\mathrm{tg}\,a\cos\phi_{sh} + \mathrm{son}\,\beta_{gsh}\,\mathrm{son}\,\phi_{sh}), \tag{6.26}$$

$$R_2 = \frac{P}{\cos\beta_{gk}}(\mathrm{tg}\,a\cos\phi_{to} - \mathrm{son}\,\beta_{gk}\,\mathrm{son}\,\phi)_{to}, \tag{6.27}$$

$$Q_1 = \frac{P}{\cos\beta_{gsh}}(\mathrm{tg}\,a\cos\phi_{sh} - \mathrm{son}\,\beta_{gsh}\cos\phi_{sh}), \tag{6.28}$$

$$Q_2 = \frac{P}{\cos\beta_{gk}}(\mathrm{tg}\,a\,\mathrm{son}\,\phi_{to} + \mathrm{son}\,\beta_{gk}\cos\phi)_{to} \tag{6.29}$$

Dependencies (6.26), (6.27), (6.28) and (6.29) are given for the case when the direction of the angle of inclination of the helical line of the teeth coincides with the direction of rotation of the gears. If the direction is reversed, before the second term in the dependencies given above, the sign should be changed to the opposite.

If the calculation is performed for a double main gear, the forces acting in the cylindrical pair are determined.

Calculation formulas for determining the forces in helical cylindrical gears are given in Chapter 4.

The driving and intermediate shaft of the double main gear are subject to strength calculations. For this purpose, the reactions of the shaft supports are determined, followed by the construction of curves of bending moments in the vertical and horizontal planes. When determining the forces acting on the shafts, focus is on the maximum loading mode of the transmission (maximum torque of the engine when the first gear of the gearbox is engaged).

According to the known value of the moments that bend the shaft in the vertical M_{call} and horizontal $M_{according\ to}$ planes, the full bending moment is determined

$$M_{with} = \sqrt{(M_{call})^2 + (M_{according\ to})^2}. \tag{6.30}$$

The equivalent moment acting on the shaft, taking into account the influence of the bending moment,

$$M_{there\ are} = \sqrt{(M_{with})^2 + (M_{Cr})^2}. \tag{6.31}$$

Working stresses in the dangerous cross-section of the shaft are determined by dependence

$$\sigma_{there\ are} = \frac{M_{there\ are}}{0.1 \cdot d^3}, \qquad (6.32)$$

where d – the diameter of the shaft in the dangerous section.

Working stresses in dangerous sections of shafts should not exceed the permissible level of 300 MPa.

To increase the rigidity of the drive bevel gear shaft, it is placed on three supports, and sometimes on four, on a number of car models. In this case, the two front bearings are installed in the crankcase and on the shaft with pretension. Since with such a design, the high stiffness and strength of the shaft are provided structurally, its calculation is not carried out, and the dimensions of the sections and the shape of the shaft are determined in accordance with the designs of similar main gears.

The splines of the main gear shafts are calculated according to a method similar to the calculation of the splines of the clutch and gearbox. When calculating the drive shaft of the main gear, the calculated moment should be taken as equal to the maximum torque of the engine when the first gear of the gearbox is engaged.

Recently, single hepoid main drives of driving bridges instead of double ones have become widespread. This is explained by the general tendency to reduce the gear ratio of the driving bridge and, therefore, the possibility of using a single main gear, the advantages of which compared to a double main gear are the simplicity of the design, higher efficiency, and a significant reduction in metal consumption.

Questions for self-control

1. What is the main gear and what is its purpose?
2. By what features are the main gears classified, on which the automobile where are they used?
3. What are the requirements for the main gears and what constructive measures are they performed?
4. What are the features of the work process of the main transmission?
5. What are the advantages of the hepoid main gear that provide it widely used on cars?
6. What adjustments and why are they performed in the main transmission?
7. What is the sequence of designing and calculating the main transmission.

DIFFERENTIAL

A differential is a transmission mechanism designed to distribute engine torque between the drive wheels and drive axles of the car. The differential serves to provide the drive wheels with different rotation speeds when the car is moving on turns or over bumps.

The different speed of rotation of the drive wheels, which travel a different path on turns and uneven roads, is necessary for their rolling without slipping and skidding. Otherwise, the vehicle's resistance to movement will increase, fuel consumption and tire wear will increase.

7.1 Requirements for differentials

1. Distribution of torque between the output shafts in the specified co-ratios (distribution proportional to hitch weight provides increased passability).
2. High efficiency and small dimensions.
3. Ensuring good stability (without skidding) when driving on turns and on uneven roads, as well as (for locked differentials) high traction properties when driving off-road.

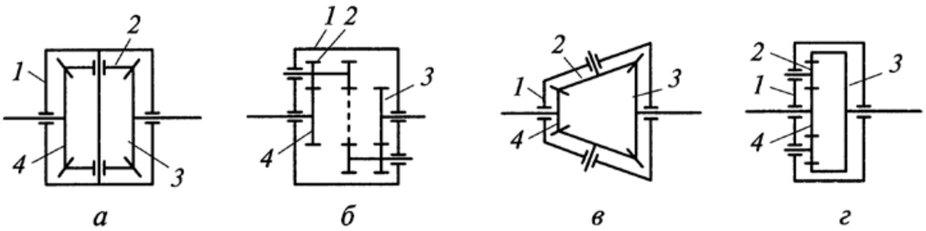
7.2 Classification of differentials

Depending on the type and purpose of cars, different types of differentials are used on them, which are divided as follows:

1. By destination (location): inter-wheel, inter-axle, inter-starfish, etc. d.;
2. According to the ratio of moments on the output shafts: symmetrical and bearing metric;
3. By type of transmission: gears with cylindrical or conical gears worms, worm, cam, etc. d.;
4. According to the possibility of blocking: unblocked and blocked with forced locking or self-locking (partially or completely).

The differential, which distributes the torque of the engine between the driving wheels of the car, is called interwheel. The differential, which distributes the engine torque between the driving axles of the car, is called interaxle. Conical differentials, symmetric and low-friction, are used on most cars.

A symmetrical differential distributes the torque equally. Its gear ratio is equal to one, i.e. the gears of the semi-axes*3*and*4*(rice.7.1,*and*, *b*) have the same diameter and the same number of teeth. Such differentials are usually used as inter-wheel differentials.

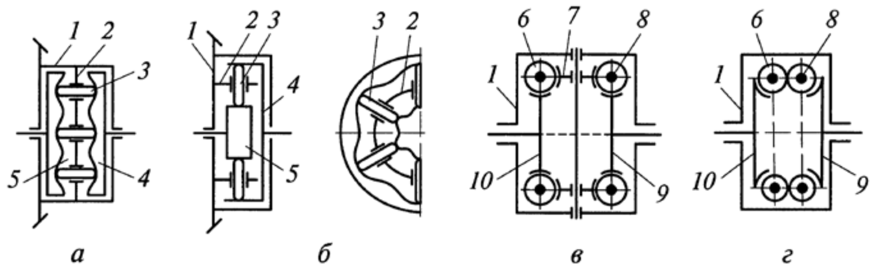


and,*b* -symmetrical;*in*,*d* -asymmetrical;*1* –corps;*2* –satellite;*3, 4* –gears

Figure 7.1 – Gear differentials

An asymmetrical differential distributes torque unevenly. Its gear ratio is not equal to one, but constant, that is, the gears of the semi-axes*3*and*4*(rice. 7.1,*in*,*Mr*) have different diameters and a different number of teeth. Asymmetric differentials are usually used as interaxle differentials, when it is necessary to distribute the torque in proportion to the loads on the driving axles.

Cam differentials can be with horizontal (Fig. 7.2,*and*) or radial (Fig. 7.2,*b*) by the arrangement of crackers. Crackers*3*placed in one or two rows in the holes of the clip *2*of the body*1*of the differential between the star cams of the semi-axes*4*and*5*, which are installed on the slots of the half-axes. Crackers in the differential play the role of satellites.



1 –corps;*2* –clip;*3* –rusk;*4, 5* –asterisks;*6, 8* –worms;
7 –satellites*; 9, 10* –gears

Figure 7.2 - Cam (*and*,*b*) and worm (*in*,*Mr*) differentials

When the car is moving in a straight line on a flat road, the nuts are stationary relative to the clamps and sprockets of the half-axles. All parts of the differential rotate as one, and both drive wheels of the car rotate at the same speed.

When the car is moving on a turn or on an uneven road, the breadcrumbs move in the holes of the clips and provide the drive wheels with different speeds of rotation without slipping and skidding.

Cam differentials are high-friction differentials that increase the total traction force on the driving wheels of the car by 10-15%, which helps to increase the traction properties and passability of the car. Cam differentials are relatively simple in design and have a small mass. They are widely used on cars with increased and high traffic.

Worm differentials can be with satellites or without satellites. In the wormhole with satellites (Fig. 7.2,*in*) torque from the body *1* dif- rential through worm satellites *7* and worms *6* and *8* transmitted by worms gears *9* and *10* semi-axes, which are installed in the slots of the semi-axes connected to the car tires.

When the car is moving in a straight line on a flat road, the body, satellites, worms and gears of the half-axles rotate as a single unit. When the car moves on a turn and on uneven roads, the different speed of rotation of the driving wheels is provided due to the relative rotation of the satellites, worms and gears of the half-axles.

In the worm differential without satellites (Fig. 7.2,*Mr*) worm gears *9* and *10* semi-axes are engaged with worms *6* and *8*, which are also interlocked with each other. The torque from the case 1 of the differential is transmitted to the gears *9* and *10* semi-axes through worms *6* and *8*.

Worm differentials have significant internal friction, which increases the total traction force on the drive wheels by 10-15%. This contributes to increasing the traction properties and passability of the car. However, worm differentials are the most complex in design. They are the most expensive of all differentials, since their satellites and half-axle gears are made of tin bronze. In this regard, currently worm differentials are used very rarely on cars.

7.3 Working process of the differential

The automobile differential is a three-link planetary mechanism for which the dependencies given in subsection 4.4 are valid. However, the degree of influence of the differential on the operational properties of the car depends significantly on the blocking coefficient

$$K_\delta = \frac{M_{in} - M_{with}}{M_{in} + M_{with}} = \frac{M_r}{M_0}, \qquad (7.1)$$

where M_r – moment of friction (friction losses),

M_0 – the torque applied to the differential;

M_{with}, M_{in} – moments on the leading edge (the wheel skids) and the you- lagging edge drive shafts, and for a symmetrical differential

$$M_{in} = \frac{M_0}{2} + \frac{M_r}{2} \quad \text{and} \quad M_{with} = \frac{M_0}{2} - \frac{M_r}{2} \qquad (7.2)$$

K_δ varies from 0 at $M_r = 0$ to 1 at $M_r = M_0$. Often used

use another expression: $K_\delta' = \frac{M_{in}}{M_{with}}$, then K_δ' changes from 1 to ∞. Value

K_δ for differentials: 0.05÷0.15 – conical unblocked; 0.4÷0.5 – fist; 0.4÷0.5 – wormy; 0.3÷0.6 - increased friction.

Fuel efficiency. When driving when turning a car 4x2

$$\eta_d = \frac{M_{in}\omega_{in} + M_{with}\omega_{with}}{M_0\,\omega_0} = 1 - \frac{\omega_{with} - \omega_{in}}{\omega_0} K_\delta = 1 - \frac{K_\delta}{\frac{R_n}{B} - \frac{1}{2}}, \qquad (7.3)$$

where R_n – turning radius (on the outer non-steerable wheel),

IN- track

If $\frac{R_n}{B}=10$ and $K_\delta=0.1$, then $\eta_d=0.99$, and without a differential due to the wheel alignment $\eta_{to} = 1 - \frac{\omega_{kz} - \omega_{in}}{\omega_{to0}} = 0.895$, that is, losses will be higher by 9.5%.

For cars with all drive axles, in straight-line driving on the highway, if the front axle is engaged through the differential, fuel savings will be 3-6% compared to turning off this axle and 5-8% compared to locked drive.

Traction properties and passability. If in equation (7.1) we replace the torques with the traction forces maximum in terms of traction (we assume for the lagging wheel ϕ_{max}, for a running-in wheel, $-\phi_{min}$), then

$$K_\delta = \frac{\phi_{max} - \phi_{min}}{\phi_{max} + \phi_{min}}. \qquad (7.4)$$

The use of locked differentials significantly increases the passability and traction properties of the car on off-road and with a significant difference in the coupling coefficients of the driving wheels.

Stability and controllability. If when braking with the wheel brakes with the engine not disconnected, there will be different braking forces on the left and right wheels, then the differential helps to maintain the lateral stability of the car, reducing the difference in braking forces on the left and right wheels. When braking sharply on a slippery road with the parking brake mechanism installed between the engine and the differential, it is possible to brake the differential housing until the car stops. At the same time ($a=1, \omega_3 = 0$) from the equation babysitter (4.10) $\omega_1 = -\omega_2$, that is, the drive wheels will rotate in different directions wounds, which will lead to a loss of transverse stability. The locked drive of the front and rear wheels worsens controllability, increasing the turning radius and leading the front wheels. Inter-wheel and inter-axle differentials improve handling.

7.4 Design and calculation of the differential

The design and calculation of the differential consists in choosing its type and structural scheme depending on the technical and operational qualities and purpose of the car.

On modern cars, simple conical differentials, the method of calculation of which is considered below, have become the most common.

The calculation of the conical differential should begin with the selection of its geometric parameters. The number of satellite teeth is recommended to be equal Z_c = 10, 11, 12; for a semi-axial gear Z_n = 14, 16, 18, 20, 22. The angle of grip a = 20°; 20°30´.

The working height of the tooth is h_0 = 1,6·m_n, full - h = 1,788·m_n.

The normal (end) modulus can be approximately determined from the dependence

$$m_{n(s)} = \sqrt[3]{M_{e\max} \cdot i_0}, \tag{7.5}$$

where $M_{e\max}$ – maximum torque, kg·m;

i_0 is the gear ratio of the main gear.

Width of toothed crown b = (0.25-0.3) Z, mm, where Z is the length of the generator dividing cone.

$$Z = \frac{m_n \cdot Z_n,}{2 \cdot \mathrm{son}\,\phi_n} \tag{7.6}$$

where ϕ_n – half of the angle at the apex of the semi-axial gear. His majesty on can be determined by dependence

$$\mathrm{tg}\,\phi_n = \frac{Z_n.}{Z_c} \tag{7.7}$$

Half of the angle at the top of the satellite gear ϕ_c = 90 - ϕ_n.

Differential gear teeth are checked for bending stress

$$\sigma = \frac{3(1 + k_\delta) \cdot M_{\text{there are}\max} \cdot i_{kI} \cdot i \cdot \eta_t,}{m_n^2 Z_n \cdot q \cdot Z \cdot (1 - \lambda_3) \cdot \pi \cdot y} \tag{7.8}$$

where q – the number of satellites;

k_δ is the blocking factor for a simple conical differential k_δ = 0.05...0.15;

η_t – transmission efficiency.

Allowable stresses are $[\sigma]$ = 500...700 MPa for gears from steels 18KHGT, 15KHGNTA, 40Kh.

The end surface of the satellites is checked for crumpling stresses

$$\sigma = \frac{4 \cdot Q_c}{\pi \cdot (d_1^2 - d^2)}, \tag{7.9}$$

where Q_c – axial force, $Q_c = P_c \cdot \mathrm{tg}\,a \cdot \mathrm{son}\,\phi_c$;

d - the diameter of the spike of the cross (Fig.7.3);

d_1 – external diameter of the end washer of the satellite;

P_c is the total circumferential force applied to one satellite.

$$P_c = \frac{M_{e\max} \cdot i_{kI} \cdot i_0 \cdot \eta_{AND},}{q \cdot r_{Wed}} \tag{7.10}$$

where r_{Wed} is the average radius of the satellite according to the dividing cone.

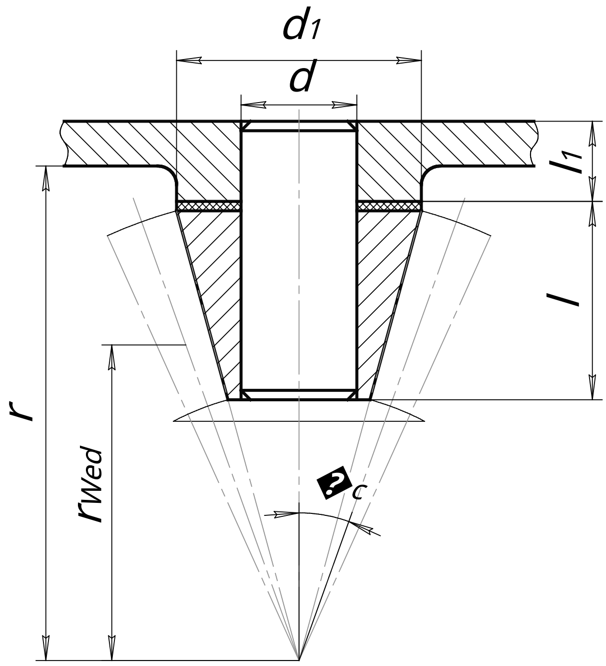


Figure 7.3 – Calculation scheme of differential satellites

The crosshead spike is also checked for crumpling stresses

$$\sigma_{Cr} = \frac{P_{C,}}{d \cdot l} \qquad (7.11)$$

where l is the length of the supporting surface of the satellite.

The spike of the cross is checked according to the direction of the cut

$$\tau_{Wed} = \frac{4 \cdot M_{e\max} \cdot i_{kI} \cdot i_0 \cdot \eta_{AND}.}{\pi \cdot q \cdot r \cdot \phi \cdot d^2} \qquad (7.12)$$

The crumpling working stresses obtained according to dependencies (7.9) and (7.11) should not exceed 70...80 MPa (700...800 kg/cm2), and according to dependence (7.12) – 100 MPa (1000 kg/cm2).

The splines of the semi-axial gears are checked for shearing and crumpling.

Questions for self-control

1. What is a differential and what is it for?
2. What are the requirements for differentials and what constructive solutions are they provided?
3. How are differentials classified and on which cars are they used? are they
4. Why is conical symmetry widespread on cars? differential?
5. What role does friction play in the differential?
6. How is a conical differential designed and calculated?

The driving wheels are driven through the half-axles of the driving axles and with the help of cardan gears.

Transmission shafts that connect the differential with the wheels of the car's drive axle are called half-axles. They serve to transmit engine torque from the differential to the drive wheels.

Metal beams with wheels are called car bridges. Bridges are used to install wheels and support the supporting system of the car (frame, body). Bridges perceive vertical, longitudinal and transverse forces acting on the wheels when the car is moving.

The driver is the bridge with the driving wheels, to which the torque from the engine is supplied. On cars, the driving axles can be only the front axle, only the rear axle, as well as the middle and rear axles or all axles at the same time. Rear drive axles on off-road vehicles with a 4×2 wheel formula, intended for use on paved roads and dry dirt roads, have become the most common.

A steerable bridge is a bridge with driven steerable wheels, to which the torque of the engine is not supplied. The front axles are steerable on most cars.

A bridge with driving and controlled wheels at the same time is called combined. Combined axles are used as front axles in front-wheel-drive passenger cars with limited cross-country ability, in all-wheel-drive cars with increased cross-country ability, and on high-cross-country cars intended for operation in difficult road conditions.

A bridge with driven non-steerable wheels is called a supporting one. Support bridges were most widely used on trailers and semi-trailers. They are also used on multi-axle trucks and as rear axles on front-wheel drive cars.

The drive of the driving wheels with the help of cardan gears is used to transfer the engine torque to the driving wheels when using combined bridges.

8.1 Requirements for drive wheels

In addition to the general requirements for the design of the car, the following special requirements are put forward for the half-axles and drive shafts:

1. Provide torque transmission to the drive wheels of the car mobile without pulsation when they rotate with different angular velocities;

2. Perform the functions of a fuse in case of excessive dynamic loads changes in the drive wheel drive mechanism system.

Other requirements for cardan drives are given in section 5. Special requirements for vehicle axles are as follows:

1. Have a minimum mass, the smallest overall dimensions, high rigidity bone;

2. To ensure stable installation angles of controlled wheels and axles their rotation (pivots).

8.2 Classification of drive wheels

1. Drive (actuator) with semi-axes (for non-separated and non-controlled bridges): unloaded, semi-unloaded, 3/4 unloaded, the latter are rarely used.

2. Cardan gear drive (with independent suspension or with steering them wheels - is considered in section 5. Cardan transmission).

8.3 Working process of drive wheels (half axles)

Unloaded axles transmit only torque. However, with significant deflections of the bridge beam and technological uncertainty, significant bending stresses are possible. Maximum loads on semi-unloaded half-axles (Fig. 8.1) can occur in the following cases:

1. At maximum traction force (during acceleration)

$$X_p = X_{e.g} = \frac{m_{2p} G_2 \phi}{2}, \quad Z_p = Z_{e.g} = \frac{m_{2p} G_2}{2} - G_{to}, \quad Y_p = Y_p = 0; \tag{8.1}$$

2. During sharp braking

$$X_{lg} = X_{pg} = \frac{m_{2Mr} G_2 \phi}{2}, \quad Z_{lg} = Z_{pg} = \frac{m_{2Mr} G_2}{2} - G_{to}, \quad Y_{lg} = Y_{Mrp} = 0; \tag{8.2}$$

3. Skidding

$$H_{lz} \approx H_{pz} \approx 0, \quad Z_{lz} \neq Z_{nz} = \frac{G_2}{2}\left(1 \pm \frac{2H\phi}{B}\right) - G_{to},$$
$$Y_{lz} \neq Y_{pz} = \frac{\phi G_2}{2}\left(1 \pm \frac{2H\phi}{B}\right). \tag{8.3}$$

Plus signs when drifting to the left – for the left wheel (Fig. 8.1), when drifting to the right – for the right wheel;

4. In the event of impact while moving over bumps

$$X_{ln} = X_{mon} \le X_p \text{ or } X, \quad Z_{ln} = Z_{mon} = k_{dpr}\frac{G_2}{2} - G_{to}, \quad Y_{ln} = Y_{mon} = 0, \tag{8.4}$$

where the dynamism coefficient k_{dpr}=1.7 for asphalt; 2.2 – for a dirt road; 4.2 - for off-road [1]. In equations (8.1)-(8.3) we take

ϕ=0.8÷1, m_{2p}=1,2÷1.4 for cars and 1.1÷1.2 for trucks, respectively m_{2Mr}=0.8 ÷0.85 and 0.9÷0.95.

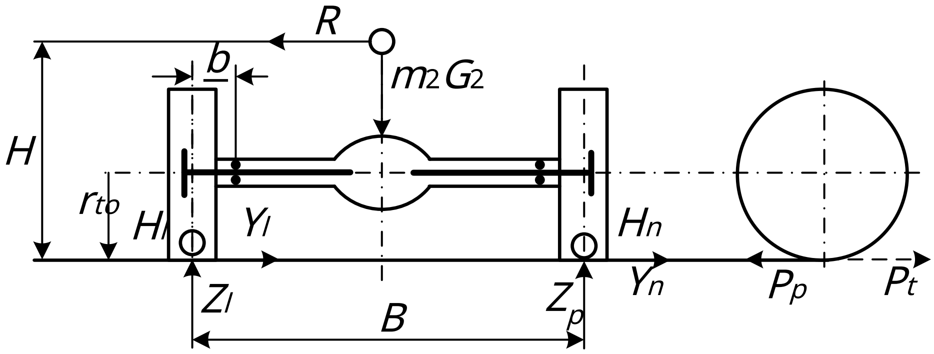


Figure 8.1 – Loads acting on semi-unloaded axles driving wheels

Moments from acting forces:

$$M_{Cr}=X\cdot r_{to}, M_X=X\cdot b, M_Z=Z\cdot b, M_Y=Y\cdot r_{to}. \quad (8.5)$$

Total bending moments:

$M_{with}=b\sqrt{X^2+Z^2}$ during acceleration and braking;

$M_{evil}=Y_l r_{to}-Z_l b$ and $M_{from}=Y_n r_{to}+Z_n b$ at drift to the left ($Y_l>Y_p, Z_l>Z_p$).

By M_{Cr} and M_{with} calculate the total stresses - equation (2.8) from with a safety margin of 2-2.5. However, the main calculation method for half-axes is the fatigue strength calculation (see clause 2.2.2).

The layout scheme with all-wheel drive improves the traction effort of cars, SUVs and trucks on wet and slippery road surfaces and uneven terrain.

In cars with permanent four-wheel drive and torque distribution equally between the driving axles, a conical differential or a planetary mechanism is used. The torque distribution is changed using automatic or controlled high-friction differentials.

All-wheel drive control (with rigid drive on the front and rear axles, viscous coupling or transfer box) includes locking the differential in the main gear and the transfer box (which has a reduction gear for driving on steep slopes, at low speeds and for transmitting high torques).

A viscous clutch (a sealed multi-disc mechanism with a highly viscous organosilicon fluid) is another means of activating the all-wheel drive. As soon as the limit traction force on the permanently connected bridge is exceeded, the coupling, reacting to the increase slippage, begins to transmit the torque to the second driving bridge.

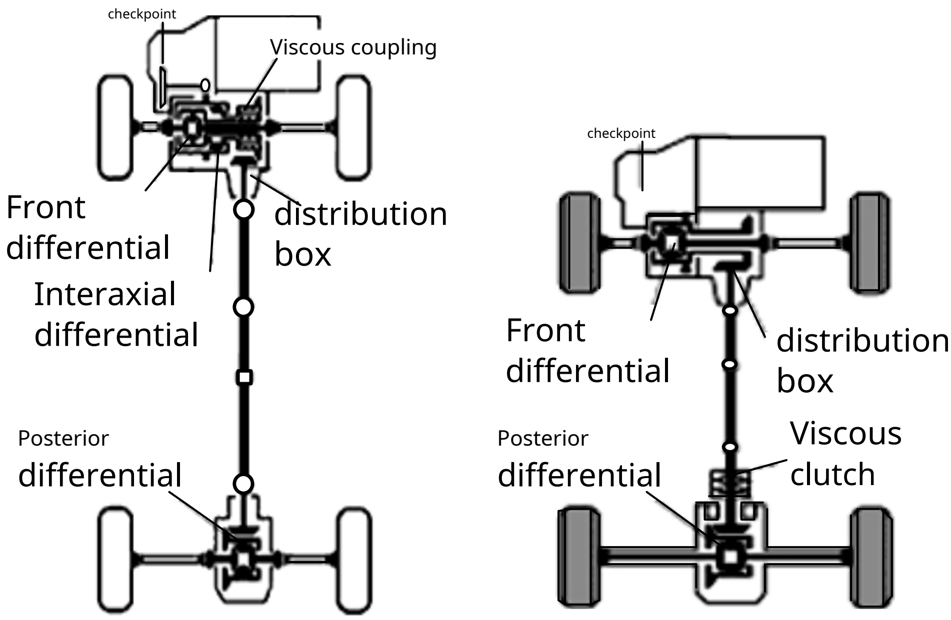


Figure 8.2 – Schemes of all-wheel drive with a viscous coupling

On more modern cars, they began to use additional locking of the differential in the transfer box, which is carried out in accordance with the intelligently controlled functioning of the brakes.

Self-locking differentials, in which a device that prevents the relative rotation of the driven links operates automatically, are gradually being replaced by electronic systems, for example, the traction control system (TCS). Such a system slows wheel spin by applying the brakes while power continues to be transmitted from the transmission to the braked wheel.

8.4 Strength calculations

Semi-unloaded and fully unloaded half-axles are most common on modern cars. As a rule, half-unloaded half-axles of the flange type are used in the designs of most passenger car models.

Fully unloaded axles are widely used in the driving axles of trucks. Semi-axles unloaded by 3/4 are not widely used in the automotive industry nowadays, so the method of their calculation is not considered.

The calculation schemes of semi-unloaded and fully unloaded semi-axles are shown in fig. 8.3.

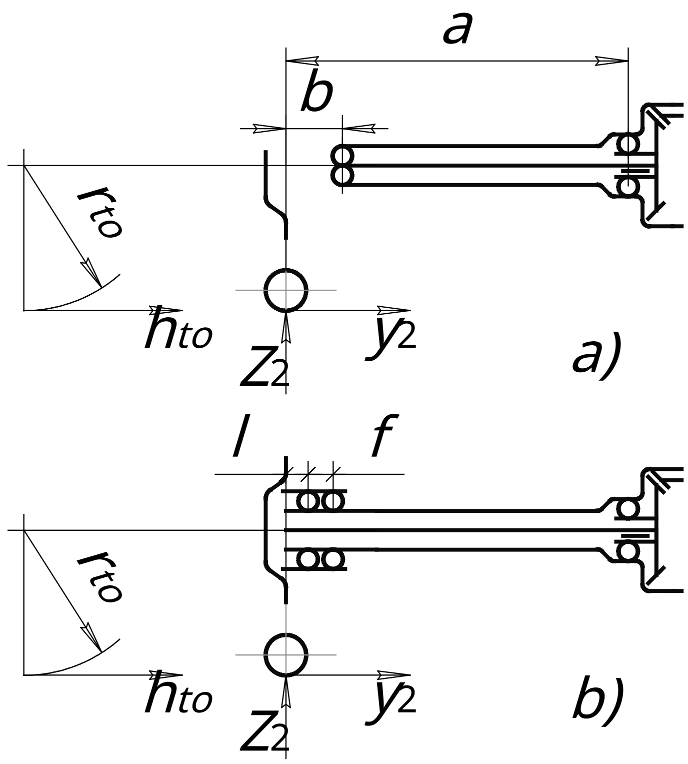


Figure 8.3 – Calculation schemes of semi-unloaded and fully unloaded drawn semi-axis

The semi-unloaded half-axle is calculated according to three main design load modes:

1) at the maximum value of the tangential force in the contact coforests with a road $X_2 = Z_2 \cdot \varphi$, at the same time $Y_2 = 0$; $Z_2 = \frac{G_2}{2}$;

2) at the limit value of the lateral force $Y_2 = Z_2 \cdot \varphi_1$, at the same time $X_2 = 0$; $Z_2 = \frac{G_2}{2}$; $\varphi_1 = 1.0$;

3) at the maximum value of the vertical dynamic load sion $Z_2 = K \cdot \frac{G_2}{2}$, at the same time $X = \varphi_{to}$ 2. Coefficient dynamism (K) is taken equal to 1.75 for cars and buses and 2.5 for trucks.

For the first calculation mode, the half-axis is checked for stresses from the joint action of bending and torsion.

The bending moment, which is the same for both semi-axes, is determined by dependence

$$M_{with} = b \cdot \sqrt{X_{2l}^2 + Z_{2l}^2} = b \cdot \sqrt{X_{2p}^2 + Z_{2p}^2}, \tag{8.6}$$

where $X_{2l(p)}$ – tangential forces on the left (right) semi-axes

$$X_{2l} = X_{2p} = \frac{m \cdot G_2}{2} \cdot \varphi, \tag{8.7}$$

$Z_{2l(p)}$– vertical reactions in contact between the wheels and the support surface for li- of the left and right hemispheres

$$Z_{2l}=Z_{2p}=\frac{m_2\cdot G_2-g_{to}}{2},\tag{8.8}$$

where m_2 is the redistribution coefficient of normal reactions, taken as by him 1.2;

g_{to}– the weight of the drive wheel in folded form, taken as an estimate by analogy with the prototype.

A torque acts on the semi-axis

$$M_{to}=x_{class}\cdot r_{to}=x_{kp}\cdot r_{to}.\tag{8.9}$$

The equivalent moment acting on each of the semi-axes is determined by dependence

$$M_e=\sqrt{b^2(x_{class}^2+Z_{2l}^2)+x_{class}^2\cdot r_{to}^2}=\sqrt{b^2(x_{kp}^2+Z_{2p}^2)+x_{kp}^2\cdot r_{to}^2}\tag{8.10}$$

Equivalent stress in the dangerous cross-section of a shaft (semi-axle) of round cross-section

$$\sigma_e=\frac{M_e}{0.1\cdot d^3},\tag{8.11}$$

where d is the diameter of the semi-axis in the calculated section.

The permissible equivalent stresses of the semi-axes are 400...500 MPa (4000...5000 kg/cm^2).

The strength of the half-axes according to the second calculation mode is checked by bending stresses. Since joint actions of vertical $Z_{2l(n)}$ and lateral $Y_{2l(n)}$ reactions cause unequal bending moments in the semi-axes, then mean a greater bending moment for one of the semi-axes (in the given case – the left one). The total bending moment from the action of vertical and tangential forces is equal to

$$\sigma_n=\frac{(Y_{2l}\cdot r_{to}+Z_{2l}\cdot b)}{0.1\cdot d^3},\tag{8.12}$$

$$Z_{2l}=\frac{G_2}{2}\cdot\left(1+\frac{2\cdot h\cdot g\cdot\phi_{to}}{B}\right),\tag{8.13}$$

$$im_{2l}=\frac{G_2}{2}\cdot\left(1+\frac{2\cdot h\cdot g\cdot\phi}{B}\right)\cdot\phi.\tag{8.14}$$

It is worth accepting in calculations ϕ= 1.0, and the height of the center of gravity of the car la h and track width B choose by prototype. Work stress is not must exceed 400...500 MPa (4000...5000 kg/cm^2).

In the calculations according to the third calculation mode, it is considered that the half-axis works on the bending deformation under the action of the bending moment

$$M_{with}=\frac{G_2\cdot K\cdot b}{2}.\tag{8.15}$$

The bending stress in the dangerous section is determined by dependence

$$\sigma_{with} = \frac{1}{0.2 \cdot d^3} \cdot G_2 \cdot K \cdot b. \tag{8.16}$$

The calculation of a fully unloaded semi-axis is based on the maximum torque, the value of which is equal to

$$M_{Cr} = Z_2 \cdot \phi \cdot r_{to} = 2 \cdot m_2 \cdot \frac{G_2}{2} \cdot \phi \cdot r_{to}. \tag{8.17}$$

The torsional tangential stresses during transmission of the maximum torque by the semi-axis are equal

$$\tau_{Cr} = \frac{G_2 \cdot m_2 \cdot \phi \cdot r_{to}.}{0.4 \cdot d^3} \tag{8.18}$$

The maximum tangential stresses in the semi-axes should not exceed 600 MPa (6000 kg/cm2).

In addition to the above, half-axes are subject to calculation for stiffness based on the twist angle when transmitting the maximum torque

$$\theta = \frac{180}{\pi} \cdot \frac{M_{e\max} \cdot i_{kI} \cdot i_0 \cdot l}{2 \cdot G \cdot I_p}, \tag{8.19}$$

where l – the working length of the semi-axis;

G – modulus of elasticity of the second kind G=80000 MPa (800000 kg/cm2);

I_p is the polar moment of inertia of the cross-section of the semi-axis.

The working values of the twisting angles of the semi-axes should be within 10-13° per one meter of length.

Questions for self-control

1. What are the methods of ensuring the driving of the drive wheels of automobiles?
la?
2. What are car half-axles, what are they?
3. What are car bridges, what are they for?
4. Name the main requirements for driving the driving wheels.
5. How are drive wheels classified?
6. How to prevent breakdowns of the main gear and the drive differential the bridge?
7. Explain the working process of drive wheels.
8. How is the calculation of the drive of the drive wheels of a car the strength of the mobile phone?

9 PENDANT

The suspension ensures an elastic connection of the frame or body with the wheels, softening the shocks and impacts that occur when the wheels hit bumps, transferring all the forces and moments between the wheels and the frame.

9.1 Requirements for suspensions

1. Ensuring the smoothness of the stroke (mainly determined by the value static deflection).
2. Ensuring the movement of the car on uneven roads without impact limiter (mainly determined by the amount of dynamic deflection).
3. Ensuring effective damping of body vibrations.
4. Prevention of tilting of the body during acceleration, braking, and turns.
5. Alignment with steering kinematics.
6. Transmission to the body or frame of forces and reactive moments from wheel

9.2 Classification of pendants

Usually, the classification of suspensions is considered depending on the type of direct, elastic and extinguishing devices that make up the suspension.

1. By type of guiding device:

a) independent (lever, telescopic, combined) suspension with wheel movement in the longitudinal plane (with transverse levers); in the transverse plane (with longitudinal levers); in both planes; along the straight line (candle-telescopic suspension); combined (lever-telescopic suspension);

b) dependent for the wheels of a given axle - with a rigid beam between the left and right wheels (bridge beam or additional transverse beam) without reactive rods or with them;

c) dependent for the wheels of different axles (blocked), and the wheels of each axle can have dependent or independent suspension. Leveling levers-balancers are usually used to connect the wheels of adjacent axles, such suspensions are called balancers.

2. According to the type of the main elastic element in the elastic device: a) metal (leaf spring, twisted spring, torsion);

b) non-metallic (rubber, pneumatic, hydraulic); c) combined homogeneous (spring-spring, hydraulic) or pneumatic heterogeneous;

d) without an elastic element (rigid attachment to the frame of the wheel axis or bridge beam, or balancer axis).

3. According to the type of extinguishing device with friction: a) only in the spring and hinges;

b) only in shock absorbers;

c) in shock absorbers and other elements, for example, in springs.

9.3 Structural diagrams and structural elements of suspensions

The ability to limit vertical movements of the body and reduce angular oscillations around the transverse and longitudinal axes depends on the geometry of the suspension and its stiffness (Table 9.1). An important role in the oscillation process is played by structural elements (Table 9.2), which make up the suspension.

9.4 Working process of the suspension

The suspension provides vertical transmission *Z*, longitudinal *H* and transverse *Y* forces acting on the wheel from the road and their moments. In the main this function is performed by the guiding device. It also determines the kinematics of body and wheel movements. The elastic device provides a reduction in dynamic loads caused mainly by the action of force *Z*. The presence of an elastic device causes vibrations of the body and wheels. Damping of oscillations is provided by a damping device.

The advantages of dependent suspension are simplicity of design and low cost. The advantages of an independent suspension are better than with a dependent suspension, adaptability to bumps, less weight of direct parts and greater static deflection - better smoothness of the ride.

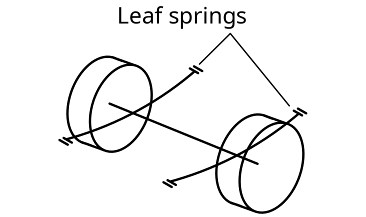
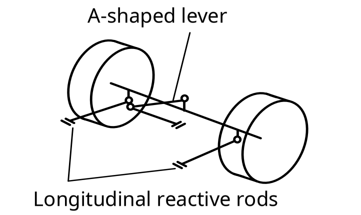
The advantages of the balancer suspension are half the displacement of the body when one wheel is moved relative to the other; for some schemes – the equality of vertical forces during acceleration and braking.

9.4.1 Types of oscillations

Such characteristics as depreciation and damping of the suspension are mainly related to the vertical oscillations of the car. Driving comfort (loads felt by passengers and cargo) and operational safety of the car (distribution of forces in relation to the road surface) are largely determined by the characteristics of the suspension. The comfort of the vehicle is mainly determined by the smoothness of body oscillation. Axis oscillations largely determine the safety of driving a car. Both types of oscillations are characterized by the ratio of frequencies and amplitudes.

In the table 9.3 shows the influence of various characteristics on the two-mass model (templet).

Table 9.1 – The main types of suspensions and their characteristics

<table>
<tr><td colspan="5">Dependent suspension</td></tr>
<tr>
<td>
</td>
<td>
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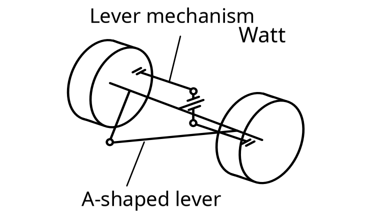
<td>
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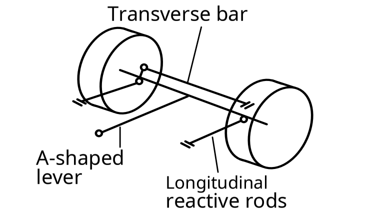
<td>
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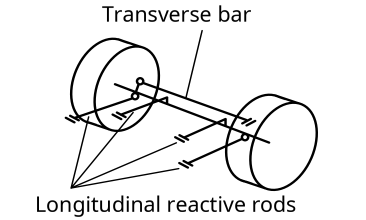
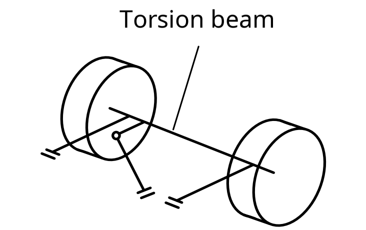
<td>
</td>
</tr>
<tr><td colspan="5">Used for the rear drive axle; for front and rear axles of heavy-duty vehicles and all-terrain vehicles. The track, toe-in and camber of the wheels remain constant in relation to the road surface even during body roll; reliable track retention</td></tr>
<tr>
<td rowspan="2">Low production costs; oscillation of the axis in the longitudinal plane; high unsprung mass; high rigidity under the action of transverse force and moment</td>
<td colspan="4">Absence of lateral movement of the body during suspension travel, absence of undesirable wheel positions as a result of lateral and longitudinal forces</td>
</tr>
<tr>
<td>Projected deviation from the transverse axis; high weight and costs</td>
<td></td>
<td>The transverse bar encourages the transverse movement of the body during suspension travel</td>
<td>The projected level of angular deviation from the transverse axis</td>
</tr>
<tr><td colspan="5">With a torsion beam</td></tr>
<tr>
<td>
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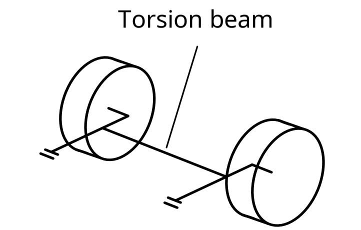
<td>
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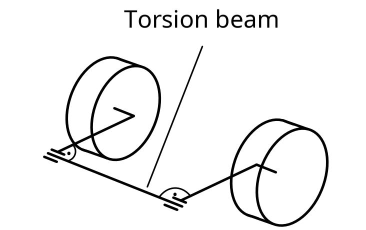
<td>
</td>
<td></td>
<td></td>
</tr>
<tr>
<td colspan="2">They are used for the rear axle with front-wheel drive</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr><td colspan="5">A large distance between supports minimizes structural stresses; acceptable transfer of forces to rigid longitudinal elements; simple manufacturing; two connection points; ease of assembly; extremely strong construction; limited kinematic capabilities</td></tr>
<tr>
<td>The axis of torsion is above the center of the wheel</td>
<td>The axis of torsion is below the center of the wheel</td>
<td>The axis of torsion coincides with the axes of rotation of the levers (all axles are united on the car body)</td>
<td></td>
<td></td>
</tr>
</table>

Continuation of the table. 9.1

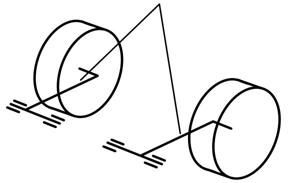
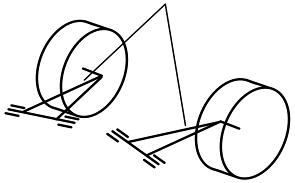
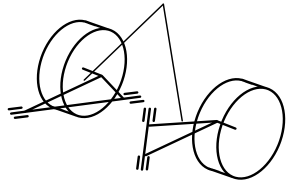
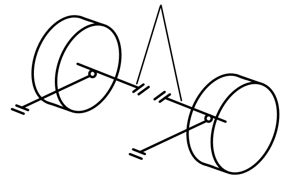
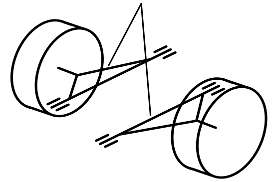
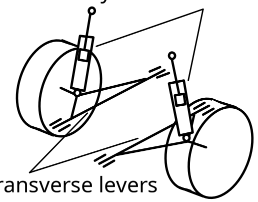
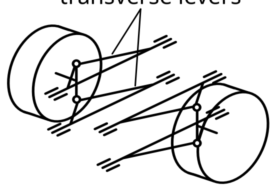
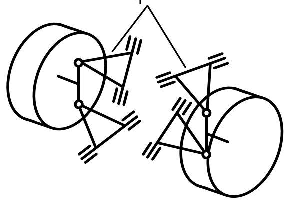
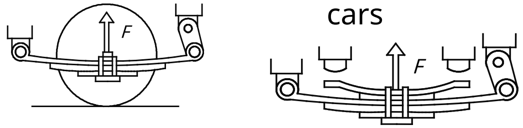
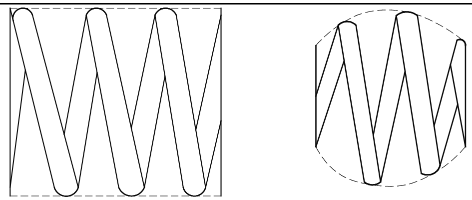
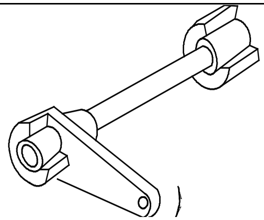
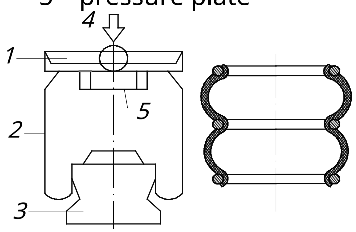
Independent suspension				
Longitudinal levers 	Oblique levers 	Oblique levers 	Transverse levers 	Transverse levers 
Rear axle designs for front or rear wheel drive vehicles				
Moderate requirements for accommodation space; low costs; limited changes in kinematics: change in camber of the wheels, significant change in the longitudinal inclination of the axis of rotation of the steered wheels, determined position of the axis of rotation; high tensions	Ease of manufacture; wide kinematic possibilities; insufficient elastokinematic properties; excessive controllability as a result of longitudinal and circular forces; high steering forces	Moderate expenses; limited kinematic capabilities; transverse forces forcing the body to move upwards on turns; supporting effect with positive wheel camber		
Stoyak-McPherson  Transverse levers		Upper and lower A-shaped transverse levers 	Oblique levers 	
They are used on bridges with front and rear wheel drive			On the front axle with front and rear wheel drive	
Moderate need for space (allows to increase the width of the car); a large distance between supports reduces structural stresses; few hinges; ease of installation; low mass; relative insensitivity to installation tolerances; limited kinematic possibilities in changing the camber of the wheels and the angle of ascent; the position of the center of oscillations in relation to the longitudinal and transverse axes; space required for springs; high installation height		Maximum kinematic capabilities; costs associated with a large number of joints; narrow structural tolerances (without subframe); due to the relatively small distances between the attachment points, rigid supports are required in order to prevent changes in the orientation of the wheels (reduced driving comfort)		
			The forces from the upper communication link are transmitted to the fire barrier	

Table 9.2 – Structural elements of suspensions

Amortize-elements	Scheme	The influence of the navan factor taking your own time totu vibrations of the body	Characteristics
1	2	3	4
Steel electric mentees Sheet reso-ra	Leaf springs with spring a truck driver cars  		Single or multilayer; certainly do not require a guiding device; in some cases, inter-sheet friction can be reduced by plastic inserts (noise reduction); usually without inserts for trucks; necessary maintenance and repair; reliable transfer of effort to the frame
Spiral spring		The natural frequency decreases increases with load; Of course characteristics are linear	The elasticity of the conical spring has a progressive characteristic; shock absorbers can be located in springs; absence of damping, possible self-guidance of spring oscillation; advantages: need for limited space, low weight, lack of maintenance; Disadvantages: a guiding device is required
Torsion			They are made of round bars or rolled flat sheet steel (smaller weight with round bars); it is possible to adjust the height of the car; no wear, maintenance and repair; a bundle of flat steel sheets can be used to absorb additional loads during bending
Pneumatic shock absorber sleeved toroidhe	Element with a constant volume of gas: 1 – frame automobile phone; 2 – cylindrical flexible element; 3 – pusher; 4 – air supply; 5 – pressure plate  	When the load changes the natural frequency remains constant; forms curves are determined by the properties of the gas, the shape of the pusher, the angle of the cord in the shock absorber	Similar spring elements are especially effective for trucks and buses; more and more widely used in passenger cars to adjust the suspension level of the rear axle or all wheels; increased driving comfort is achieved; the movement of the wheels must be determined by the suspension guide; low pressure (< 10 bar) implies large volumes; the geometry of the toroidal shock absorber does not contribute to the achievement of low vertical stiffness

Continuation of the table. 9.2

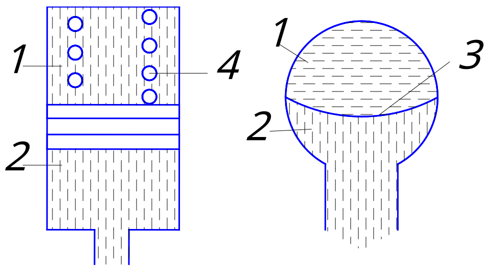
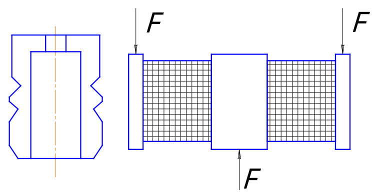
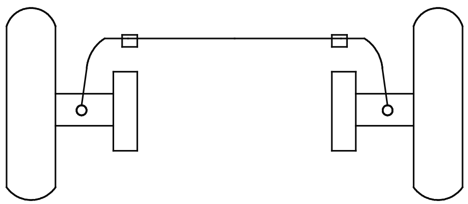
1	2	3	4
Hydropneumatic pendant Hydraulic diaphragm-my battery Piston battery new type	Damping element with constant mass of gas 1 – gas; 2 – liquid; 3 – diaphragm; 4 – steel spring 	When increasing the amount of tension increases own frequency; curves are a function initial pressure of the battery simulator	The characteristics are determined by the volume of gas in the accumulator (separated from the liquid by a piston); the liquid compresses the gas according to the load on the wheel; damping valves are combined with the shock absorber and are connected between the riser and the battery; the rubber diaphragm requires maintenance due to possible gas diffusion
Rubber elastic ele-mentees		The natural frequency is influenced by the load due to the nonlinearity of the elasticity characteristic	Resisting elements made of vulcanized rubber between metal parts are more often used with hydraulic shock absorbers; are used to install units (engine, transmission, gearbox) and as additional vibration isolation elements
Stabilizer		It does not affect the even deflection of the suspension both sides; half forces associated with stiffness, acts in the case of one-sided deflection of the suspension, and the total resistance to torsion corresponds to mutually opposite by moving the wheels	Reduces body roll and also affects handling characteristics (excessive and insufficient handling); usually made in the form of a U-shaped rod or tubular support, the ends are often made flat in order to adapt to bending loads; mounting points should be at extreme distances from each other to ensure the minimum diameter of the stabilizer; the relative position of the stabilizer and the bending radii of the shoulders should be chosen to ensure torsional stresses without a large influence of bending loads

Table 9.3 – Influence of structural characteristics on the vertical oscillations of the car

Constructive characteristics	Influence on your own frequency of oscillations body	Effect on frequency intermediate link	Effect on natural frequency axis oscillations
Body data			
Resilience	Significant on com- control fort by car	Average on com- control fort by car	Insignificant for the safety of driving a car
Less elasticity	Increase in frequency and amplitude, lower degree of comfort		Increased frequency, slight decrease in amplitude
Great elasticity	Reduced frequency and maximum amplitude, increased level of comfort		Slight increase in amplitude at low excitation frequencies
Decrement	Significant for the comfort of driving a car		Significant for changes in the dynamic load on the wheel
	<u>It must be correctly selected for the given application conditions</u>		
more (more shock absorber torus);	Reduction of acceleration	Increase in the roots	Increasing acceleration, reducing changes in the dynamic load on the wheel
less (less shock absorber torus)	An increase in acceleration	Reduction of the roots	A slight decrease in acceleration, increased changes in the dynamic load on the wheel
Mass	Insignificant on the amplification factor for changing the load on the wheel; the gain factor for acceleration decreases with increasing load (an empty car is less comfortable than a loaded car and not as safe)		
Tire and wheel data			
Elasticity (with increased degree of elasticity hundred tires)	The natural frequency and amplitude remain virtually constant		The reduction of the natural frequency and amplitude due to the acceleration of the body and changes in the load on the wheel are approximately proportional to the decrease in the vertical stiffness of the tire
Decrement	Frequency and amplitude are not affected by bus damping		Tighter damping is the result of a slight decrease in amplitude during body acceleration and changes in wheel load
	As a result of the heating, the damping of the tire must be kept to a minimum in order to provide a significant level of elasticity for a non-rigid tire		
Wheel mass	The reduced weight of the wheel actually does not affect driving comfort		The minimum weight of the wheel increases the level of traffic safety

Angular oscillation around the transverse axis is associated with the rotation around the transverse axis of the car during acceleration of the car from a state of who The kinematic characteristics of the suspension are selected in such a way as to minimize angular fluctuations during acceleration and braking.

Angular sway relative to the longitudinal axis is associated with the rotation of the car around the longitudinal axis, which usually passes through the lower front and upper rear parts of the car; oscillation about the longitudinal axis occurs in response to steering actuation. Transverse stability stabilizers on the front and rear axles reduce this effect.

9.4.2 Guide suspension device

The kinematic characteristic of the suspension, that is, the dependence of movements x_k, y_k and tilt angles a_k, β_k, y_k wheels (Fig. 9.1, b) from the vertical transdisplacement z_k are obtained by graphic constructions or analytical development by Hunks In fig. 9.1, the following characteristics are given for a single-lever independent suspension according to the scheme (Fig. 9.1, a) with ϕ = 60° i R=0.5r_k [3]. With this- of the example, you can get individual cases:

a) at ϕ = 0 wheel movements will be only in the transverse plane, moreover x_k=y_k=a_k=0;

b) the same with a lever parallelogram suspension will give x_k=y_k=a_k=β_k=0;

c) when replacing a parallelogram with a trapezoid x_k=y_k=a_k=0, β_k≈0, y_k≈0.

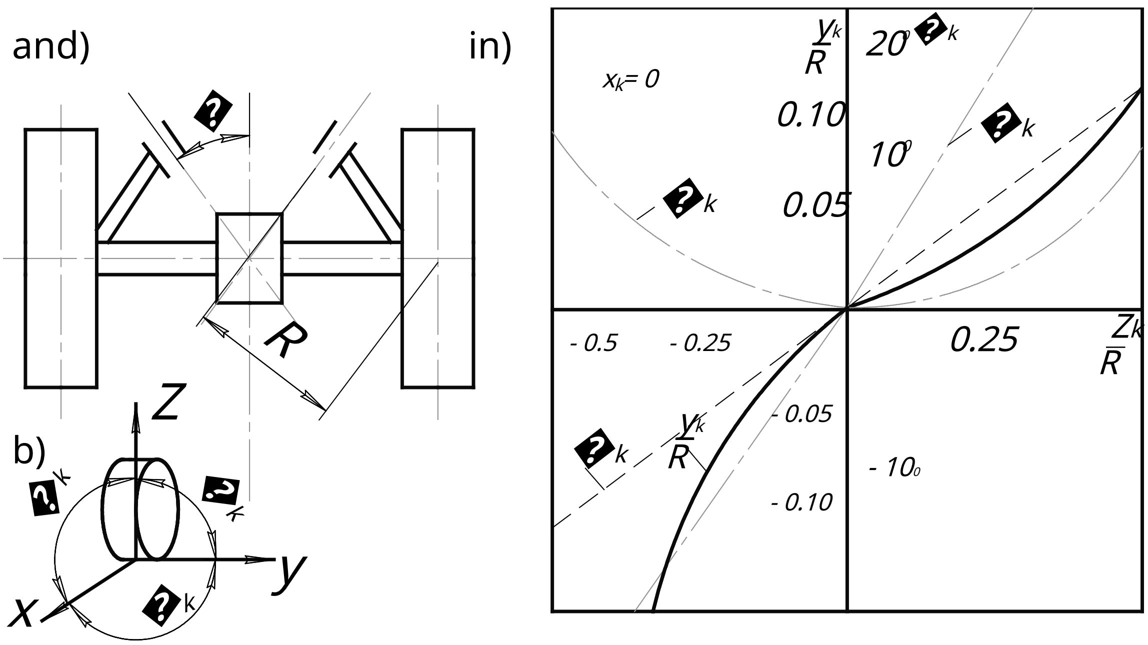


Figure 9.1 – Single-lever independent suspension
and its kinematic characteristics

It is desirable that when moving the wheels vertically $x_k, y_k, a_k, \beta_k, y_k$ were minimal, especially for the drive wheels. For example, $x_k \neq 0$ sometimes distorts the kinematics of the steering drive, $\beta_k \neq 0$ contributes to the appearance of noise and increased tire wear, $a_k \neq 0$ worsens wheel stabilization, $y_k \neq 0$ changes the trajectory of the car.

Longitudinal roll depends on the intensity of braking or acceleration, the height of the center of gravity and the base of the car, as well as on the elastic characteristics of the suspension and the type of guide device.

Transverse roll of the body affects the smoothness of the ride, stability, controllability, and tire wear. Roll angle

$$\beta = \frac{M_\beta}{C_{\beta 1} + C_{\beta 2}}, \qquad (9.1)$$

where $M_\beta = G h_{k.p} (\mu + \beta)$ is the moment causing body roll;

$h_{k.p}$ – roll shoulder – distance from the point of lateral force application (wind or centrifugal force) to the roll axis, and the roll axis is a straight line connecting the roll centers of the front and rear suspensions;

$\mu \approx \frac{y}{G}$ – specific lateral force;

G – weight of spring-loaded parts;

$C_{\beta 1}$ i $C_{\beta 2}$ – combined angular stiffnesses of the elastic elements of the front and rear suspension.

Height h the roll center of the suspension is determined by:

– for dependent suspension – a point slightly below the axes of the lugs spring;

- for suspensions of a candle or with the movement of the wheel in the longitudinal plane - no - a point lying on the road (h=0);

- for a single-lever suspension with transverse arms - a point that lies at the intersection of the lines drawn through the axis of rotation of the lever and the point of contact of the tire with the road;

- for two-lever suspensions - a point that lies at the intersection of the lines, conducted through the instantaneous center of rotation O_l or O_p (rice.9.2, a) and point contact of the tire with the road.

Angular stiffness of the suspension

$$C_\beta \approx 2 C_p d_p^2, \qquad (9.2)$$

where C_p – vertical stiffness of the suspension,

$2d_p$ – spring track (Fig.9.2, b).

From fig. 9.2, b: $M_\beta - \left(Z_{rl1} - Z_{rp1}\right) \cdot d_{p1} - M_{CM} = 0$. Neglecting the elasticity of the tire and taking into account the moment $M_{SM} = WITH_{\beta SM} \cdot \beta$ stabilizer tand $Z_{rl1} = Z_{rp1} = 0.5\, WITH_p d_{p1} \beta$, we will get $C_{\beta 1} = WITH_{p1} d_{p1}^2 + C_{SM}$ and $\beta = \frac{\mu G h_{Cr}}{WITH_{\beta 1} + WITH_{\beta 2} - G h_{Cr}}$.

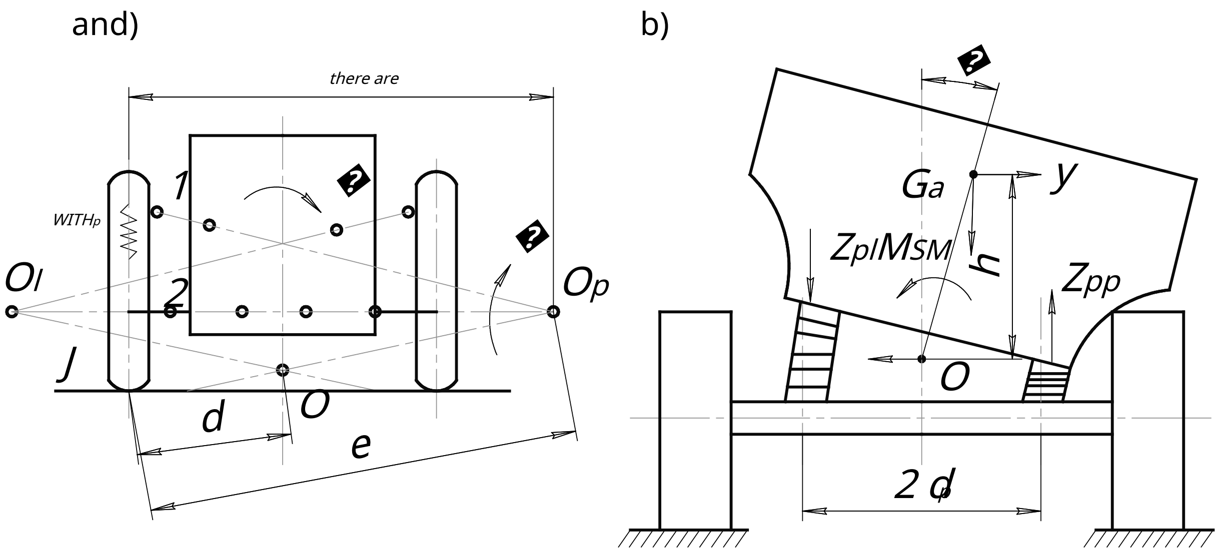


Figure 9.2 – Calculation scheme of the suspension

The transverse stability stabilizer reduces roll by 20-40%. It is considered permissible β= 6÷7° with μ=0.4.

Diagrams of forces acting on parts of suspensions are presented in fig. 9.3. For the dependent suspension according to fig. 9.3, and:

$$R_{Z1}=Z\frac{l_2}{l_1+l_2},\quad R_{Z2}=Z\frac{l_1}{l_1+l_2},\quad R_{X2}=R_{Z2}\,tg\alpha, R_{X1}=X-R_{X2}.$$

In addition, these forces, as in the following diagrams, create moments. For the independent suspension of the non-driving wheels according to fig. 9.3, b:

$$R_{zn}=Z, R_{xb}=R_{XH}=Z\frac{m}{a+b} \text{ and } P_{Ave}=Z\frac{t+s}{t}.$$

In addition, when braking $R_{TV}=R_t\frac{r_k-b}{a+b}$ and $P_{so\,on}=R_t\frac{r_k+b}{a+b}$. Reactions $R_{x1}=R_{xb}, R_{x2}=R_{xn}, R_{Z2}=Z\frac{s}{t}$.

For the balancing suspension according to fig. 9.3, in car 6x2 with symmetrical springs: $\frac{Z_3}{Z_2}=\frac{a}{b}\cdot\frac{l_p\pm2dy_3}{l_p-2dy_2}$, where y_2, y_3 – specific longitudinal power of the second and third axes; the upper sign is for braking, the lower one is for acceleration. With sharp braking Z_3 comes out in 2-3 times more than Z_2. That is why they use auto-locking of the balancer, which turns on when braking.

For the pendant according to fig. 9.3, g of the MAZ car, the reactive moments perceive the upper rods and the balancer, therefore $Z_2=Z_3$ and when braking and at dispersal

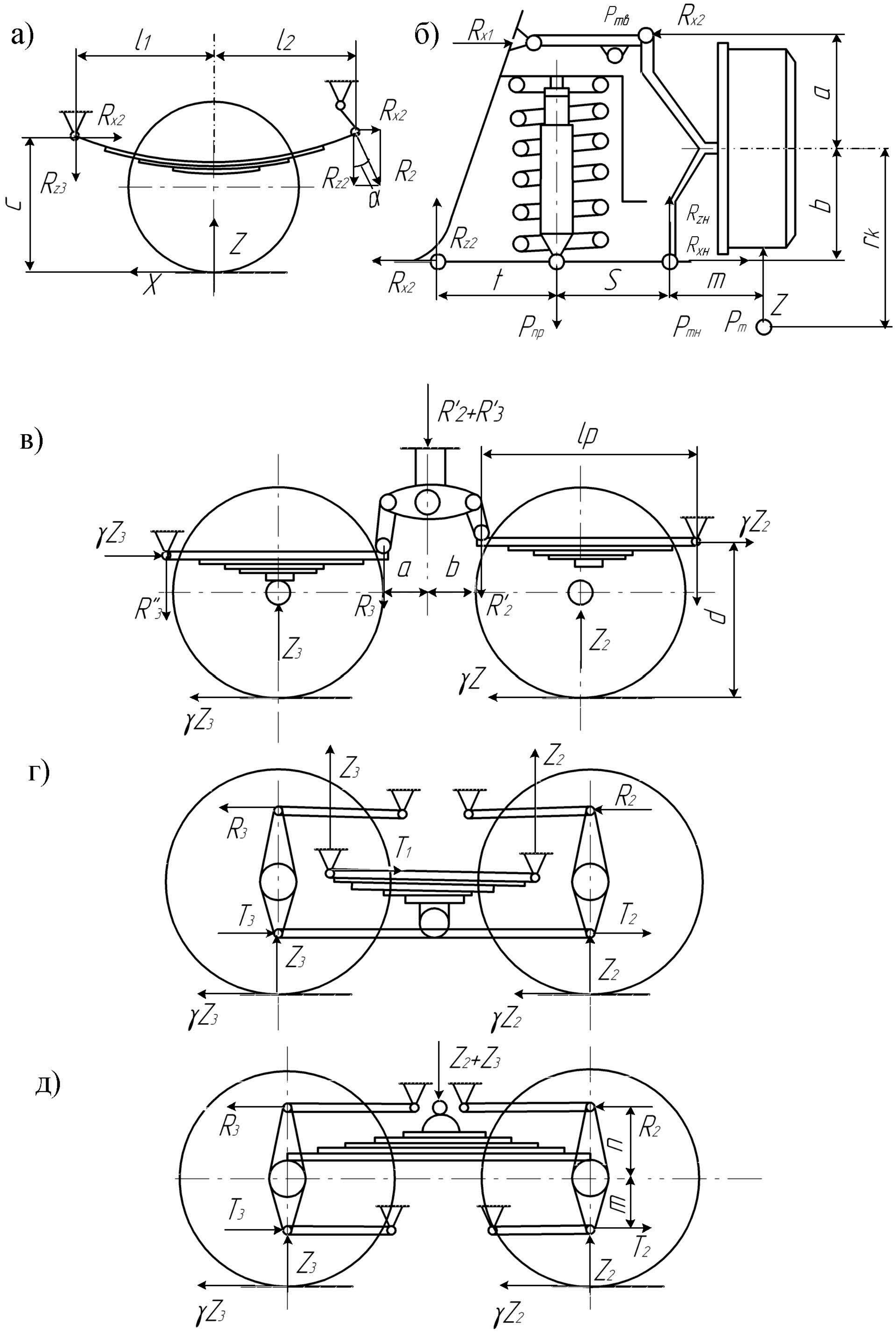


Figure 9.3 – Schemes of forces acting on parts of suspensions

For the pendant according to fig. 9.3, d – ZIL, KrAZ, Urals: $R_i = \frac{y_i Z_i (r_k - m)}{m+n}$,

$T_i = y_i Z_i - \frac{(r_k + n)(0.5c - e)}{m+n} \pm e$, where *with* - the distance between the lower bars, *there are* – c-- displacement of the upper rod from the longitudinal plane. Load on the frame $Z_2 + Z_3$ transmitted through one point.

9.4.3 Elastic suspension device

The elastic device is evaluated by the elastic characteristic of the suspension (Fig. 9.4, a). The estimated value is the conditional static deflection

$$f_p = \frac{M_0 g}{2C_p} = g\omega_c^2 \tag{9.3}$$

where $M_0 g$ – weight of spring-loaded parts,

$2C_p = tg\alpha$ – suspension stiffness,

ω_c is the natural frequency of oscillations of the spring-loaded parts.

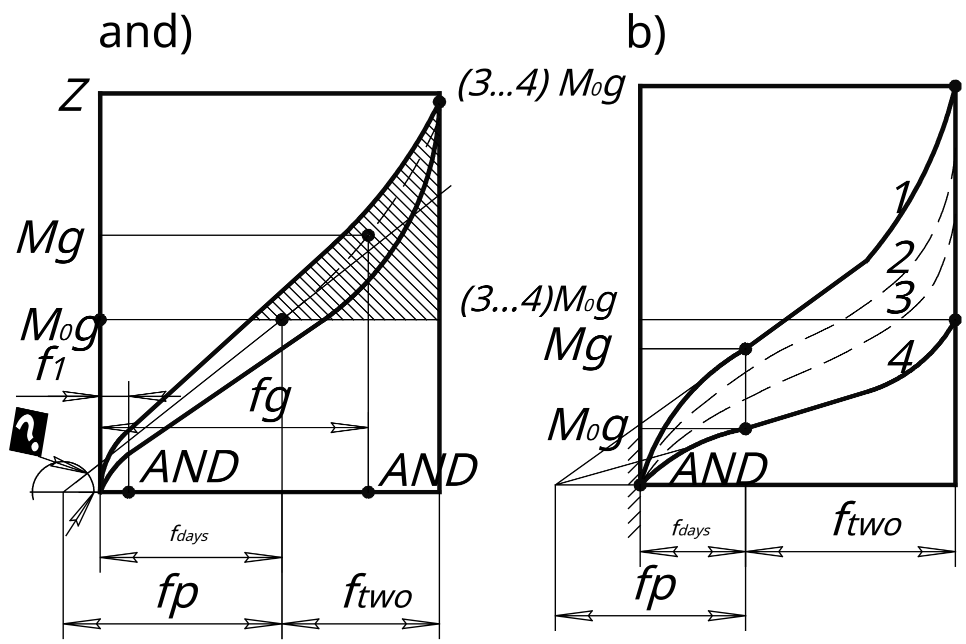


Figure 9.4 – Elastic characteristic of the suspension

The shaded area corresponds to the potential energy received by the suspension when hitting the surface. The larger it is, the lower the probability of hitting the limiter when driving on an uneven road. To maintain sufficient dynamic capacity at low ϕ_{two} the necessary nonlinear elastic characteristic Ristic with $Z_{max} = (3 \div 4) M_{unit}$. If the weight of the spring-loaded parts changes when moving with and without a load, then it is desirable that the elastic characteristic consists of a family of curves with the same f_p and f_{two}, (Fig.9.4, b). This is mo- easily with pneumatic and hydraulic elastic elements. Non-linearity can be obtained with metal main and additional elastic elements. Radial stiffness of tires *WITH* 6-12 times higher than the spring stiffness $_{sh}$ element (2-3 times for cars 4X4 and 6X6).

Static deflection (due to hardness) significantly affects the smoothness of the stroke. Reducing stiffness reduces the amplitude of body and wheel movements at low-frequency resonance, shifting it to lower frequencies, where the probability of resonance is lower.

Metal elastic elements. Leaf springs (bow springs) often perform the functions of all three suspension devices. Disadvantages: short durability, inter-sheet friction, which impairs smoothness of movement, large dimensions and weight. Potential energy per unit volume, $A_{question} = \frac{A}{V} = \frac{1}{6} \cdot \frac{G^2}{E}$ for a leaf spring in 4 times smaller at $\sigma = \tau$, than for spring and torsion, in which $A_{question} = \frac{1}{4} \cdot \frac{\tau^2}{G}$ (from-wear modules of elasticity $\frac{E}{G} \approx 2{,}6$).

Deformations of a leaf spring, twisted spring, torsion.

$$f_p = \delta P_e \frac{l^3 (1-\varepsilon^2)}{48 E I_0}, \quad f_n = P_p \frac{8 n D^3}{d_n^4 G}, \quad f_m = M_p \frac{32 l}{\pi d_m^4 G}, \tag{9.4}$$

where δ = 1.25÷1.5 – deformation coefficient;

$l_e = l - l_0$ – the effective length of the spring (full length minus the distance between the stirrups);

$\varepsilon = \frac{l_2 - l_1}{l}$ – coefficient of asymmetry ($l_1 + l_2 = l$);

$I_0 = \frac{b}{12} \sum h_T^3$ – the total moment of inertia of the spring in the middle section;

THERE ARE $= 2{,}2 \cdot 10^5$ MPa;

n and D – the number of turns and the diameter of the spring;

d_n, d_m – wire and torsion diameters;

$G = 8.5 \cdot 10^4$ MPa;

l is the length of torsion bar without slotted ends.

The force loading the main spring before the additional spring starts to act: $P_0 = c_1 f_0$, where c_1 – stiffness of the main spring. When both work spring $P_p = P_0 + (c_1 + c_2)(f - f_0)$, where f – full deflection of the main spring, c_2 – stiffness of the additional spring. So

$$f = \frac{P_p + c_2 f_0}{c_1 + c_2}. \tag{9.5}$$

Non-metallic elastic elements. In Ukraine, double rubber cylinders (Fig. 9.5, a) are used for pneumatic suspensions D = 0.3 and 0.25 m, on loading P_{max}= 1.5 and 2 t, pressure during static deflection

p_c=0.25÷0.5 MPa. For the balloon $P=pF_{ef}$, where $F_{ef}=\pi R_{ef}^2$ – effective planning yes p – excess pressure, which changes with a dynamic load change it goes like this:

$$p=(p_c+0.1)\left(\frac{V_c}{V}\right)^k-0.1, \qquad (9.6)$$

where V_c and V – the total volume of the cylinder and the additional tank in the static name and settlement provisions,

k=1.3÷1.35 is the polytropy index. The stiffness of the balloon

$$c=\frac{dP}{df}=k\frac{p+0.1}{V}F_{ef}^2+p\frac{dF_{ef}}{df}, \qquad (9.7)$$

where the first term characterizes the influence of the volume, the second – the shape of the ball on.

In hydropneumatic suspensions, the pressure (up to 20 MPa) of gas *1* (rice.9.5, b) transmitted from the liquid *3* through a separate piston *2*. The stiffness of such an element that does not have back pressure from below on the piston *4*, can be determined from equation (9.7), considering $\frac{dF_{ef}}{df}=0$ and $p+0.1\approx p$.

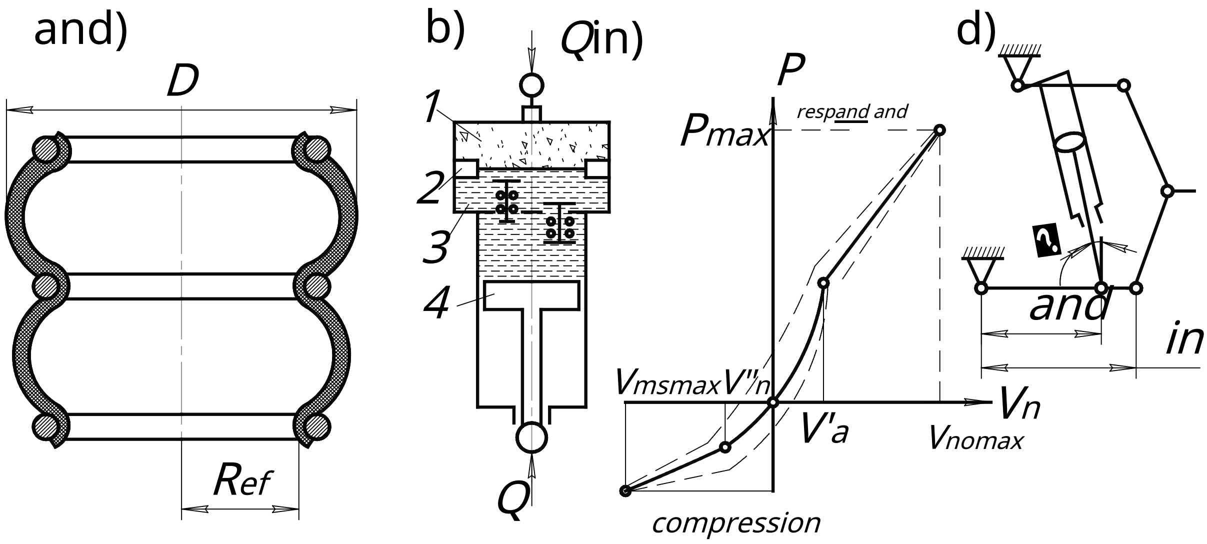


Figure 9.5 – Elements of hydropneumatic suspensions, characteristics and shock absorber installation diagram

9.4.4 Shock absorbers

Telescopic shock absorbers (shock absorbers) convert vibrations of the body and suspension into heat. They are attached to the body and axle with the help of elastic elements to reduce noise.

Single-tube shock absorbers

Advantages: easily adapted to different suspension designs, as the large diameter of the piston allows for low operating pressures. There is enough space for valves and channels. The heat is dissipated directly through the outer part of the cylinder. Can be installed in any position.

Disadvantages: long length. The outer side of the cylinder, which serves as a guide for the movement of the piston, is subjected to deformations from flying stones, etc. p.

The structural scheme of the suspension should provide sufficient space for moving the moving part of the shock absorber without mechanical obstacles. The piston rod seal is subject to damping pressure.

Two-pipe shock absorbers

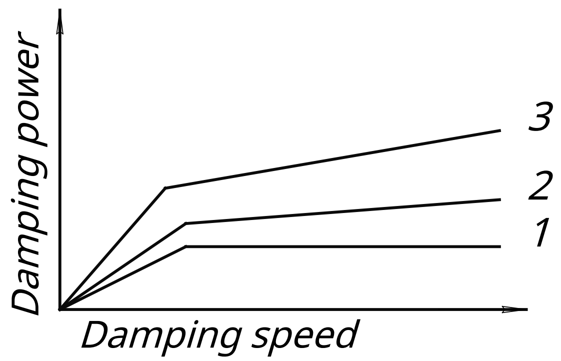
Atmospheric or low pressure damping devices. *Advantages*:

insensitive to external damage. Unlike single-tube shock absorbers, mechanical means can be used on the outer surface of the cylinder to ensure the laying of pipelines in limited areas of the body. They have a short length, because the compensation chamber is located around the working cylinder.

Disadvantages: sensitive to overload (damping failures). Only a certain position is possible when installing on a car.

Damping characteristics

The characteristics obtained as a result of damping in the hole and in the spring-loaded valve that closes it are shown in Fig. 9.5, c and 9.6. The spring reacts to the pressure by increasing the outlet opening. The diameter of the piston and the spring can be specially selected to approximate the linear dependence of the damping characteristics. An internal adjustment mechanism can be used to obtain several characteristics for one shock absorber. Compression force values are often only 30...50% of the reverse stroke values.



1- comfortable;*2*– standard;*3*- sporty

Figure 9.6 – Characteristics of shock absorbers (return modes)

Electronically controlled shock absorbers (active adjustment to operating conditions) are used to increase driving comfort and safe driving of the vehicle.

A semi-active type of control is often used, in which the shock absorber is adjusted in accordance with the speed of the car.

Characteristics of the shock absorber $P=f(V_n)$ is usually nonlinear, but its mo- should be approximated by two straight lines for the initial and valve sections (Fig. 9.5, c) with resistance forces $P_H=k_H V_{m\ n}$ i $P_K=P_H' + k_K\left(V_n-V_n'\right)_m$, where k– drag coefficient; m is an exponent, usually $1\le m\le 3$; V_n–fast- piston bone.

Area under the curve $P=f(V_n)$ gives the dissipated power. Having determined N_c for compression stroke and N_{in} break-off stroke, can be found for conditional amortization congestion with a linear characteristic of the average value $k_{cpc}=\dfrac{2N_c}{V^2_{n.c\max}}$ and $k_{cpo}=\dfrac{2N_o}{V^2_{no\max}}$, and is also equivalent tny drag coefficient $k_e=\dfrac{k_{cpc}+k_{cpo}}{2}$. Then drag coefficient reduced to the wheel:

$$k_P=k_e i^2_{am}\cos^2\gamma_{Am}, \tag{9.8}$$

where $i_{Am}=\dfrac{b}{a}$ and γ_{Am} – gear ratio and shock absorber setting angle (Fig. 9.5, d).

The main estimated parameters of the shock absorber:

a) coefficient of aperiodicity (affects fluctuations)

$$a=\frac{k_P,}{\sqrt{MK}} \tag{9.9}$$

where M– weight of spring-loaded parts, c – suspension stiffness; usually $a=0.2\div0.4$;

b) maximum force during compression and rebound, moreover $P_{\max c}\le P_{\max o}$;

c) critical piston speeds V_n' and V_n'' fig.9.5, c.

The thermal calculation of the shock absorber is usually carried out at V_n near 0.3 mc^{-1} (boundaries of initial plots). The heat absorbed by the amortization rum $Q_{Am}=\dfrac{\left(P_{PC}+P_{after}\right)V_n t}{2\cdot427}$ kcal Heat transfer equation

$$Q_{see}=a\cdot S\left(t^{o}_{Am\max}-t^{o}_{district}\right)\cdot t, \tag{9.10}$$

where $S=\pi D_1\left(\dfrac{D_1}{2}+l\right)$ – the outer surface of the shock absorber,

$t°_{Am}$max– permissible temperature of the outer surface (120-130°C) shock absorber during operation t=1 hour;

$t°_{district}$– air temperature;

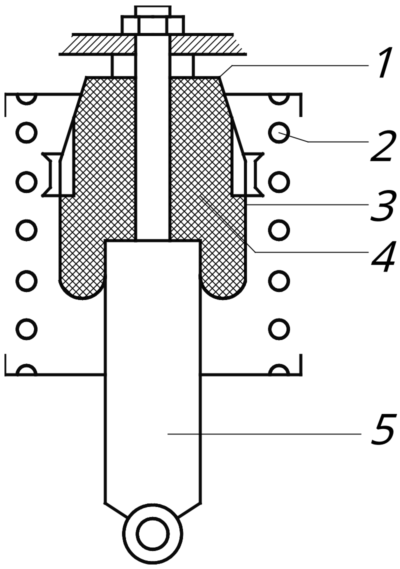
a=50÷70 kcal/m2· °C is the heat transfer coefficient.

9.4.5 Controlled suspension systems

Adjusting the suspension consists in changing its characteristics and parameters when the weight of the transported cargo or road conditions (load equalization system) changes. It is carried out in pneumatic and hydropneumatic suspensions, where automatic body position regulators and suspension stiffness regulators are used [1].

Partially loaded systems(rice.9.7, 9.8) The use of non-rigid springs leads to an increase in the compression of the car suspension under load. In order to maintain the height of the car body at an acceptable level, auxiliary pneumatic or hydropneumatic springs are used.

The system may also include electronic load balancing control units acting on solenoid valves.



1 – air fitting; 2 – steel spring;
3 – additional pneumatic spring; 4 – gas chamber; 5 - shock absorber

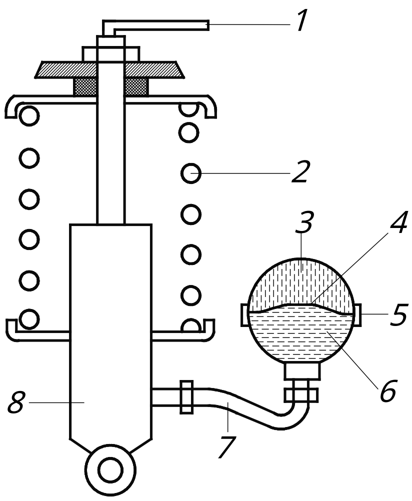
Figure 9.7 – Air suspension with load equalization

Advantages of the electronic system:

- reduced energy consumption due to the elimination of intermediate cycles during braking, acceleration and when driving on turns;

− the response of the system to an increase in the speed of the car changes changing the height of the suspension to save fuel;

- increasing the height of the suspension when driving on road surfaces of fourth and fifth categories; increased stability of movement in turns, which is achieved by transverse blocking of suspension elements on one axis.



1 – liquid supply; 2 – steel spring; 3 – battery; 4 – gas chamber; 5 – rubber diaphragm; 6 – liquid; 7 – hose; 8 – shock absorber

Figure 9.8 – Hydropneumatic load leveling system

Additional advantages for heavy-duty trucks:

changing the height of the suspension for replacing bodies and containers; the height of the vehicle can be adjusted, for example, to align the load-bearing surface with the loading platform; control of the lifting axle: the lifting axle is automatically lowered when the maximum load on the axle is exceeded; the lifting axle is raised for a short time (2...3 minutes) in order to increase the load on the drive axle (increase the compression force).

Fully loaded suspension systems

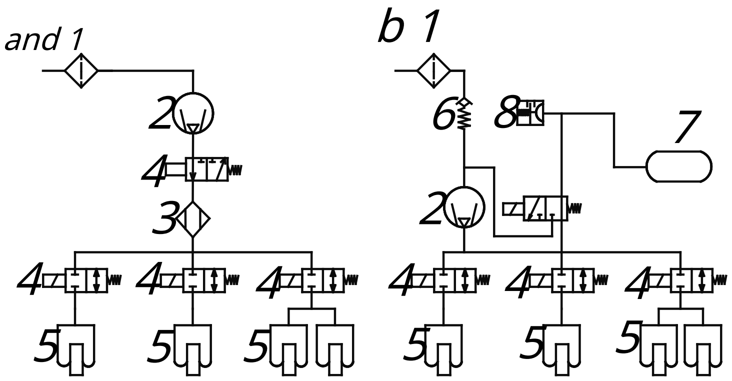
The elastic action is provided by a gas element of the suspension, which does not have spiral springs. One or both axles of the vehicle can be steered.

If it is necessary to control all axles, the system must have an electronic control unit with a special control program that takes into account factors such as changes in axle load to prevent the vehicle from tilting or tipping over, and at the same time recognize system errors.

Open system

Advantages: relatively simple design and management.

Disadvantages: high output power of the compressor, necessary for short periods of time of active control; the need to ensure air drying; noise during intake and exhaust periods.



and– open system; *b*– closed system; 1 – filter; 2 – compressor;
3 – dryer; 4 – solenoid valve; 5 – pneumatic shock absorber;
6 – non-return valve; 7 – pneumatic cylinder; 8 – pressure sensor in

Figure 9.9 - Load equalization system
(fully loaded system)

Closed system

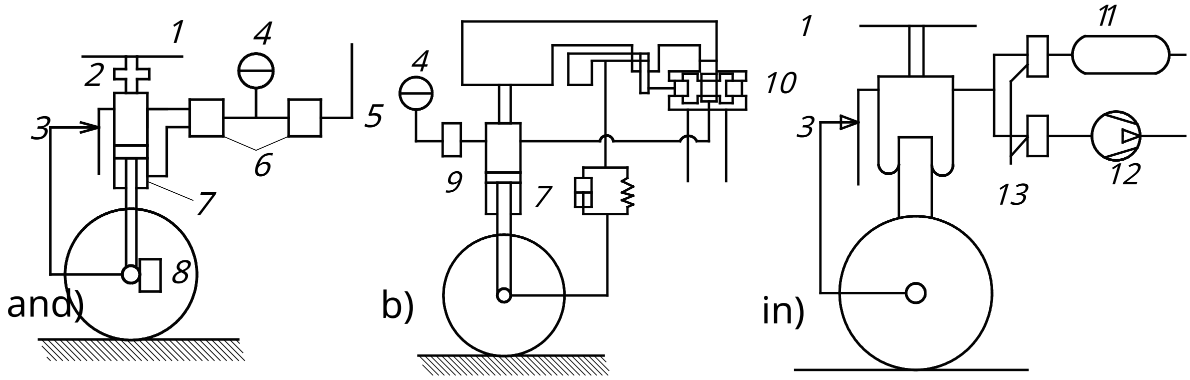
Advantages: low output power of the compressor (minimum pressure drop between the accumulator and the suspension element), no dryer.

Disadvantages: relatively complex construction.

Pneumatic shock absorbers are much smaller in weight than hydropneumoshock absorbers.

9.4.6 Active suspension

In the active suspension (Fig. 9.10), the parameters of both elasticity and damping are controlled.



a – hydraulic suspension; b – hydropneumatic suspension;
c – pneumatic suspension; 1 – car body; 2 – wheel load sensor;
3 – the sensor movement; 4 – battery; 5 – line from the pump; 6 – servo valve;
7 – hydraulic cylinder; 8 – acceleration sensor; 9 – damper;
10 – plate dividing valve; 11 – receiver; 12 – compressor; 13 – solenoid valve

Figure 9.10 – Active suspension

Structures containing a hydraulic cylinder

With the help of an external source, energy is generated for accelerated adjustments of the hydraulic cylinder, sensors provide communication between the cylinder and the car body. Sensors of wheel load, displacement and acceleration transmit signals to the electronic control unit (ECU) within a few milliseconds.

The control system allows you to achieve a constant load on the wheel while maintaining a constant average height of the car. Steel springs or hydropneumatic suspension elements are used to support the static load on the wheel.

Constructions of hydropneumatic suspensions

Structural oscillations are regulated using the distribution of hydraulic fluid flows in the hydropneumatic suspension circuit. In order to reduce energy requirements, the system's action is limited to smoothing irregular low-frequency oscillations; the gas accumulator is connected to the hydraulic cylinder and dampens oscillations of higher frequencies.

Designs of pneumatic suspensions

The movement of the body is controlled by adjusting the air supply to the pneumatic shock absorbers. Closed damper systems are limited to controlling low-frequency oscillations and oscillations from steering.

Since the system balances transverse forces, it allows the use of springs.

9.5 Strength calculations

Design loads. The calculation of the fatigue strength of a metal elastic element is carried out depending on the operating conditions, the calculation of the static strength is based on the towing weight, taking into account the coefficient of dynamismP_p=$G_{University}k_d$, moreover$k_d$is taken depending on the conditions operation [1].

Materials. Leaf springs, stabilizers and springs are made from spring steels 50XHA, 65G, etc., torsion bars - from 45X-NMFA, 60C2A, 70C2A, hardness HRC 45-50. The details of the guide device are made of cast steel 35L, 45L, forged steel 20, 30X, 40X, and stamped rough sheet steel 14G2, 30T.

Calculations. A leaf spring is designed for bending (the root leaf with traction and braking forces taken into account), a coiled spring - for compression, a torsion spring - for torsion, and for full bending - with a knocked-down rubber buffer. Spring fingers - for bending and crumpling, levers of the guide device - for bending with compression taken into account.

A load that acts on a pneumatic spring suspension device and causes a change in the effective area F_{ef} and effective radius R_{ef} pneumoba-womb

$$P_b = p_p F_{ef} = p_p \pi R_{ef}, \tag{9.9}$$

where p_p – air pressure in the pneumatic cylinder.

When changing the dynamic load of the air pressure in the pneumatic cylinder

$$p_p = (p_{Art} + 1)\left(\frac{V_0 + V_{res}}{V_b + V_{res}}\right)^k - 1 \tag{9.10}$$

where p_{Art} – air pressure in the cylinder under static load; V_0 – cylinder volume under static load; V_{res} – the volume of the additional air tank;

V_b – cylinder volume at any load;

k=1.3 is the polytropy index at speeds corresponding to the natural frequency of oscillations of the sprung mass of the car.

Questions for self-control

1. What is the purpose of a car suspension?
2. Name the main requirements for pendants.
3. By what features are pendants classified?
4. What are the advantages and disadvantages of different types of suspensions?
5. Explain the working process of suspension.
6. What are the types of suspension vibrations?
7. Purpose and principle of operation of the suspension guide device.
8. What is an elastic suspension device?
9. What types of shock absorbers do you know and their advantages?
10. The principle of operation of controlled suspension systems.
11. Explain the principle of active suspension and its design features
12. How is suspension strength calculated?