10 WHEELS AND TIRES

10.1 Wheels

Modern car wheels usually contain a wheel rim and a hub or wheel disk. The size of the wheel is mainly determined by the diameter of the brake drum and the load capacity of the tire. The most important design parameters are: rim width, diameter of the rim (diameter at rim seat), diameter of the central hole, diameter of the circle on which the holes for the wheel mounting studs are located, the number of such holes, the depth of the studs, as well as the offset of the rim.

10.1.1 Variants of construction of rims

The rims differ from each other (depending on the type of tire used) in the number of individual structural elements and the shape of the cross section of the rim. The most important parts of the rim are the bead, the surface of the tire bead on the rim, and the base of the rim. Rims have one of the following cross-sectional shapes:

- with a recess in the center (deep rim);
– flat base;
– the landing surface of the side of the tire with a taper of 5°;
– the landing surface of the side of the tire with a taper of 15°.

When installing the wheel on the hub, two requirements must be met: the centering of the wheel to ensure its correct rotation and the transfer of loads from the wheel to the hub.

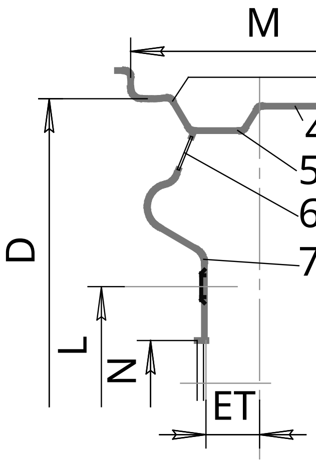
10.1.2 Wheels of passenger cars

Mass-produced wheels are made of sheet steel, and special-type wheels are made of stamped metal, cast aluminum, or magnesium alloy.

Wheels made of sheet aluminum and split wheels, created on the basis of a classic stamped wheel, are the lightest, but, at the same time, expensive to manufacture. Also used are cast aluminum wheels with rims made by the rolling method, and prefabricated wheels specially created for transmitting large forces.

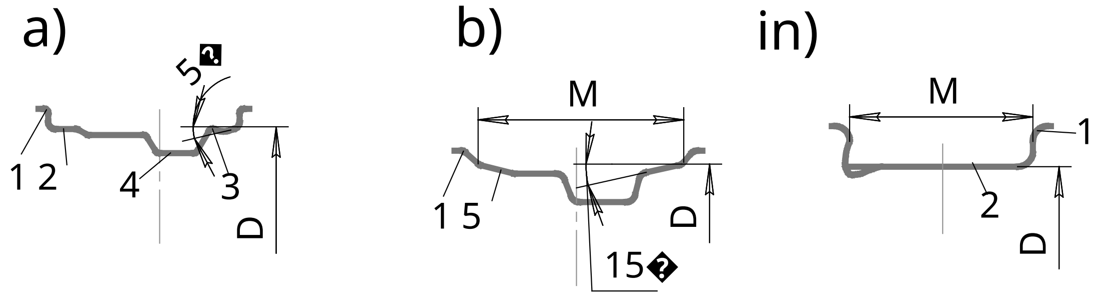
The disc and rim of the wheel made of sheet steel are connected to each other by welding; when manufacturing wheels by the method of hot three-dimensional stamping or casting wheels from light alloys, these two elements of the wheel are usually manufactured as one unit. Multi-element wheel designs, as well as wheels made of different materials, are used only in special cases and for racing cars. Rims for car wheels

(Fig. 10.1) almost always have a central recess with a double hump H2 (very rarely with a flat part FH), a conical landing surface (with a slope of conical shelves of 5°) and a J-shaped side. Less high sides with a B profile are often used on the wheels of small cars; sides with a large JK and K profile are rarely used and only on heavy trucks (Fig. 10.2).



1 – rim edge (for example, J-shaped); 2 – the landing surface of the side of the tire, the slope of the of the shelves (conicity) 5°; 3 – hump (or hump), for example, double hump H2; 4 – dinner; 5 – central recess (stream); 6 – ventilation hole; 7 – disc; D is the diameter of the rim (for example, 14"); L is the diameter of the circle on which the holes for pins; M is the width of the rim (for example, 6"); N is the diameter of the central hole; ET is the distance from the middle plane of the rim to the contact plane of the disc

Figure 10.1 - Wheel (for example, size 6Jx14H2)



a – hump on the rim of a passenger car wheel; b – a rim with a conical landing surface (15)° for trucks (for tubeless tires); c – lunch with cone dish with a flat surface (5)° for trucks; 1 – board; 2 - conical landing surface; 3 – hump; 4 – stream; 5 – conical landing surface (15°); *M*– rim width;*D*– rim diameter

Figure 10.2 – Constructions of the wheel rim

Recent wheel developments that have gone into mass production include the TR (metric) rim wheels created by MICHELIN for use with the TRX series tires, which provide more space for brake mechanisms, and also coscrees with DUNLOP rims and "Denloc" chutes, which also require the installation of special tires on them, designed to provide increased safety in the event of a tire puncture. The TD (TRX-Denloc) system requires wheel and tire compatibility for its application. Contrary to generally accepted practice, in all three of the above-mentioned wheels, other rim or tire designs cannot be used, or they can be used to a very limited extent.

A completely new development is a wheel with a tire clamp outside the rim (CONTINENTAL). This design makes it possible to significantly increase the diameter of the brake mechanism, although at the same time, when the tire interacts with the rim, some changes in the characteristics of these elements are noted. This design allows the car to move at a reduced speed for several hundred kilometers with flat tires and eliminates the need to use a spare wheel. This version also did not become popular in the tire market and has limited use.

In view of the efforts made to reduce the space in the car for the placement of the spare tire, as well as to reduce the total weight of the car, special spare wheels with tires that do not have a long life are increasingly being used.

Design criteria for passenger car wheels: high strength of individual wheel elements, effective brake cooling, reliable installation of the wheel on the car, high degree of concentricity, small space, low weight, low cost, ease of mounting the tire on the wheel, reliable fastening balancing loads and aesthetic appearance (especially in the case of using light alloy wheels). Recently, great attention has been paid to wheels that allow to reduce the coefficient of aerodynamic resistance c_w.

The wheels are usually fixed on the car with the help of 3-5 wheel nuts of a special shape corresponding to this wheel. Correct installation of the wheel is achieved by centering the wheel on the hub with the help of a centering collar. The so-called central wheel mounting is used only on racing cars.

Wheel caps are fixed on the wheels with the help of clamps or (which is not so common) with the help of threaded connections due to aesthetic requirements, the need to reduce aerodynamic resistance c_w and cooling wheels and brakes.

Traditional materials for the manufacture of wheel caps are steel and aluminum. Recently, plastics have become increasingly used, which allow to reduce the weight and cost of the wheel.

10.1.3 Truck wheels

The main requirements for truck wheels are:

– high fatigue strength and long service life;

– the maximum possible reduction of wheel mass;
- ensuring a large load capacity of the wheel;
– increased quality when processing the wheel disc;
- reduction of wheel runout and its imbalance (unbalance);
– easy assembly of a tire with a rim.

For tubeless truck tires, a modern design of a non-separable rim is used, which has a landing surface for the side of the tire with an inclination of conical shelves of 15°.

Advantages:
– integral design of the wheel allows to reduce its weight;
– increased rim diameter;
- provision of sufficient free space;
– a unified valve (valve), located in a precisely selected one place and at a sufficient distance from the brake drum or disc brake bracket;
– unified form of balancing loads;
- the possibility of central centering of the wheel;
- the possibility of using automatic or semi-automatic tire mounting equipment.

Radial wheel runout is one of the main causes of vehicle vibrations during movement. The permissible radial runout when using a rim with a conical landing surface (15)° today reaches a value of 1.25 mm. Transverse wheel knocking has a smaller negative effect on the movement of the car. Wheel imbalance is also less of a problem than tire imbalance, which is characterized by a larger magnitude.

Wheel centering using lugs and spherical washers or only spherical nuts has been replaced by central centering in order to reduce excessive radial runout of the wheel.

Any unevenness in the contact zone of the tire with the road (waviness, slopes, presence of vegetation, etc.) affects the brake drum when the wheel nuts are tightened, causing fluctuations in the braking force when the wheel rotates. These fluctuations, in turn, lead to vibrations in the steering system. The tests showed that the maximum allowable waviness of the contact surface should be 0.15 mm, and the slope from the center of the wheel to the outside should be 0.2 mm.

Discless wheels

The discless wheel is made of steel and includes a rim. The removable rim of the wheel is connected to the hub with the help of clamps and screws. In order to simplify the installation and disassembly of tube tires, the wheel rim consists of three parts, separated from each other in the transverse direction (Fig. 10.3). The same hub can be used in combination with a rim intended for mounting tubeless tires on it.

A new development can be called an integral cast rim with a conical landing surface (15°), which weighs as much as a steel rim with an optimized mass.

One of the reasons for using light alloy rims is to reduce the weight of the entire wheel with a rim that has a conical landing surface (15°).

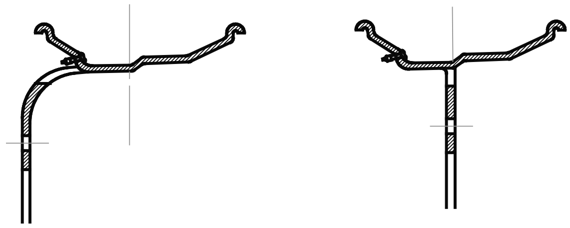
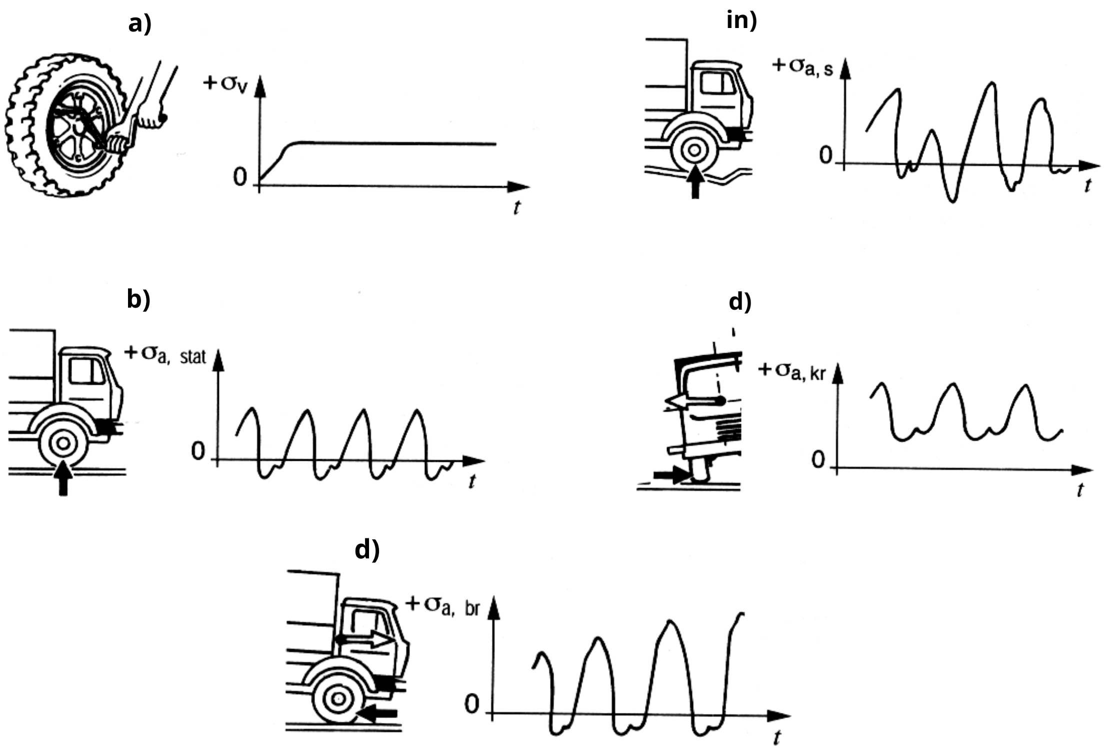


Figure 10.3 – Discless wheel with an offset rim (left)
and rim without shift (right)

Loads on truck wheels Pre-tensioning of the wheel (Fig. 10.4) takes place under the combined effect of stresses obtained during assembly and forces arising when the tire is inflated with air.



andis the preliminary stress σ_v; b – static stress $\sigma_{a.stat}$; c – tension during movement uneven surface σ_{as}; d – stress during movement on the turn $\sigma_{a.kr}$;
e – stress during braking $\sigma_{a.br}$

Figure 10.4 – Types of loads on the wheel and stress distribution depending on time*t*

When the wheel rolls at a low speed on an absolutely flat road surface under conditions of static nominal load on the wheel, periodic stresses arise in it.

Additional loads are caused by dynamic forces created when the car is moving on an uneven road, during car maneuvers, when the steering wheel of a stationary car is turned, during braking and acceleration.

Weaknesses of the wheel design

Such elements of the wheel as the side of the rim, flange of the disc, ventilation holes, the place where the rim is welded to the disc and the radial part of the central depression of the rim are affected by heavy loads and damage. The flange connection is partially affected by stresses.

Radial cracks are usually formed in the area of the central hole, and tangential cracks are formed between the holes arranged in a circle.

10.2 Tires

10.2.1 Tire classification

The main characteristics – dimensions, load values, established value of internal pressure and recommended speed of movement – are standardized in order to ensure the interchangeability of tires, which are divided into 7 groups or categories (Table 10.1). In addition to pneumatic tires, solid tires designed for driving at speeds not exceeding 25 km/h or 16 km/h for tires mounted on non-suspension wheels are also used.

Table 10.1 - Categories of tires

Category	Tire application
1	Two-wheeled motor vehicles (motorcycles, mopeds, scooters with an engine with a working volume of less than 50 cm^3)
2	Cars, including cargo and passenger cars
3	Light commercial vehicles, including delivery vehicles
4	Commercial vehicles, including multi-purpose vehicles
5	Earthmoving machines (transport machines, loaders, graders)
6	Industrial trucks, including h cars equipped with solid rubber tires
7	Agricultural machines and equipment (tractors, mechanisms, aggregates, trailers)

Tires of categories 2-4 are divided, depending on road conditions, into: standard, road tires; special, increased traffic (M + S and off-road).

The main requirements for all tires are presented in the table. 10.2, although it should be noted that for tires intended for heavier vehicles, the main attention should be paid to the last three criteria (especially #6).

Table 10.2 - Operating characteristics

Basic criteria	Additional criteria
1. Comfort when moving	Soft suspension, low noise level, smooth rolling (small deviations from the rounded shape of the tire)
2. Manageability	Perception of efforts that occur during turns, ensuring accuracy in driving a car1)
3. Stability of movement	Ability to maintain straightness of movement1), cornering stability1)
4. Traffic safety	Reliable landing of the tire on the rim, good adhesion of the tire to the road1)
5. Durability	Structural strength, possibility of movement at high speeds, tightness, high resistance to punctures
6. Economy	Expected service life (mileage), nature of wear, wear of the sidewall, rolling resistance, possibility of retreading

1)The main criteria in the conditions of winter traffic on the road

10.2.2 Tire design

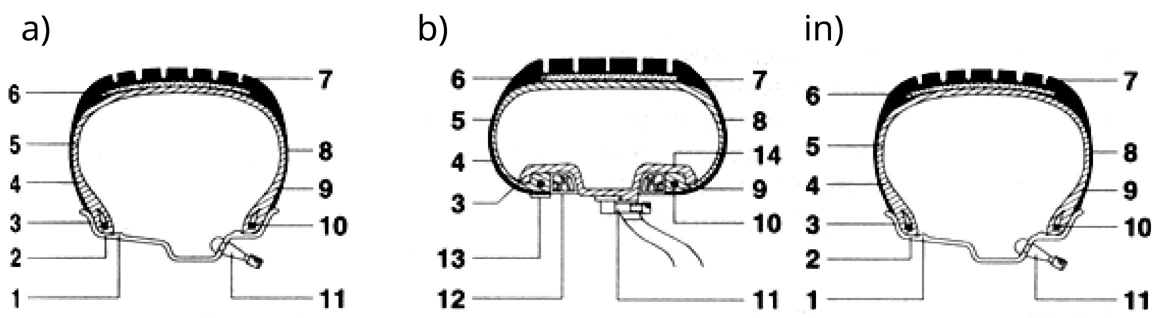
Nowadays, only radial tires are installed on passenger cars. Today, diagonal tires are used only on motorcycles, mopeds, earthmoving machines and vehicles serving industry and agriculture. Diagonal tires are used less and less on trucks.

Diagonal tires got their name because the cord threads in the tire tread are located diagonally (at a certain angle to each other) and cross each other. In radial tires, the cord threads are placed at the shortest distance between the sides. The breaker belt limits the relatively thin and elastic frame (carcass of tire) to ensure sufficient stability of the tire.

Tires with diagonally arranged breaker belt layers have become widely used in the USA, they contain an additional belt outside the diagonal frame. The characteristics of these tires are not as good as those of radial tires.

ST type tires are characterized by a flat tread tread. In the deflated state, such tires, resting on the seat of the wheel rim, ensure the possibility of driving the car for a distance of several hundred kilometers. In the TRX/Denloc system, a special protrusion in the side of the tire fits into a corresponding depression in the wheel rim. Such a device allows you to fix the side of the tire.

When using tubeless tires (Fig. 10.5), the tube is replaced by a vulcanized inner layer characterized by high air permeability. The side of such a tire should sit on the rim of the wheel more hermetically in order to ensure the necessary sealing. Sometimes special additional sealing elements in the form of rings made of elastic material are used. The absence of a camera in the tire allows you to reduce the weight of the tire and simplify the operations of its installation on the wheel rim.



a) – radial tire for passenger cars; frame: two layers of radial cord viscose threads; breaker: two layers of intersecting steel threads and two layers of nylon threads current located in a circle; b) – tire type ST with a flat running track; c) – radial tire for heavy trucks; frame: one layer of radials thread (monolayer); breaker: four layers of intersecting steel threads.
1 – hump on the wheel rim; 2 – landing surface for the side of the tire on the wheel rim; 3 – edge of the rim; 4 – tire frame; 5 – airtight inner layer; 6 – breaker belt; 7 – protector; 8 – side panel; 9 – board; 10 – board core; 11 – valve; 12 – locking ring; 13 – balancing load; 14 – the edge of the rim

Figure 10.5 – Examples of tubeless radial tires

The diameter and construction of the wheel rim

For tires in categories 3 and 4, a single-element deep rim with a 15° taper (for tubeless tires) is more preferable than a multi-element rim with a 5° taper. This type of rim is identified by diameter code designations (the code unit corresponds to 25.4 mm) ending in 0.5; for example, 17.5; 19.5 and 22.5. Numbers 16 and 20 are codes for deep rims with a taper of 5°. Standard deep rims with diameter code designations expressed as a whole number (10, 12, 13, etc.) and special designs of rims, the diameter of which is indicated in mm, are used to install passenger car tires on them.

The ratio of the height of the tire profile to its width

It is determined by the formula: (H/ W) · 100, where N– cross-sectional height of the tire profile; W– cross-sectional width of the tire profile.

This ratio for modern standard car tires is in the range of 80 to 50; a ratio equal to 35 corresponds to tires for sports cars, from 65 to 100 - for truck tires.

With different values of this ratio, one tire mounting diameter is used to facilitate the interchangeability of tires. Tires for passenger cars with a small ratio of the height of the tire profile to its width provide high stability when driving on turns. Also, with a small ratio of the tire's height to its width, the width of the tread footprint can be preserved while simultaneously increasing the diameter of the wheel rim, which allows for more space to accommodate brake mechanisms.

10.2.3 Tire marking

The tire marking is stamped on its sidewall (Table 10.3) and reflects the standard requirements considered in UNECE Regulations No. 30 for passenger car tires, No. 54 for heavy-duty vehicle tires (for speeds of 80 km/h and higher) and No. 75 for tires of mechanical two-wheel vehicles (vehicles with engines, the working volume of which exceeds 50 cm^3, and move at a set speed of more than 40 km/h). The exception is tires of types V, VB, VR, ZB and ZR, designed for driving at speeds above 210 and 240 km/h.

Tires tested for compliance with UNECE Regulations are identified by a code printed on the sidewall of the tire near its edge. This code designation consists of the capital letter "E" and the code number located in a circle, followed by the number corresponding to the number registration of this tire design.

Example: E4 020 427

The capital letter "E", applied on the sidewall of the tire in accordance with European recommendations 92/93, has the same significance in the code marking.

The width of the tire, its design (R – radial; "–" – with intersecting threads; B – diagonal) and the rim diameter are the minimum information required for tire marking. Industrial truck tires are also, of course, marked with the tire diameter. On the tires of two-wheeled vehicles, cars and trucks, this information is often supplemented with the value of the ratio of the height of the tire to its width (in %), this ratio is plotted directly after the information about the width of the tire (separated by a blank). According to EEC regulations, similar information must be on all new tires. Although not covered by the EEC regulations, passenger car and two-wheeler tires may also have a maximum speed code for that tire (applied after the tire's height-to-width ratio or tire width value). On diagonal tires, the code letter is replaced by a horizontal line.

On tires of types VR, VB, ZR and ZB, the code letter is part of the tire size designation.

Information about the number of layers (PR) follows from the designation of the tire size; this designation is used today as the load rating code for different tire variants of the same size.

As operational characteristics, additional designations are used in the form of a combination of the load index (LI) and the nominal speed symbol (GSY). The UNECE rules consider these designations as a replacement for the code for the number of plies PR or the letter code for the nominal speed of movement in the dimensional characteristics of the tire (with the exception of tires of types VB, VR, ZB and ZR).

Table 10.3 – Examples of tire designations

Group of tires	Type of transport tool	Dimensions tires	Numeric PR[3]	Operating codes tray characters-joint		Tire diameter A	Tire width B	Height ratio to width H/W %	Rim diameter d
				LI[4]	GSY[5]				
MS	Mopeds	21/4-16	-	-	-	-	code	-	code
	Mopeds with an engine with a working volume of 50 cm^3	3-17 reinforced[2]	-	51	J	-	code	-	code
	Scooters	3.00-17.00 reinforced[2]	-	50	P	-	code	-	code
		110/80 R 18	-	58	H	-	mm	80	code
		120/90 H 18	-	65	H	-	mm	90	code
	Scooters	3.50-10	-	51	J	-	code	-	code
	Passenger cars	165R14M+S	-	84	Q	-	mm	-	code
		165 R 14 reinforced[2]	-	88	R	-	mm	-	code
		200/60 R 365	-	88	H	-	mm	60	mm
		205/60 ZR15	-	-	-	-	mm	60	code
		CT 235/40 ZR 475	-	-	-	-	mm	40	mm
CV	Delivery vehicles	185 R 14 S[1]	8 PR	102/100	M	-	mm	-	code
	Light cargo. a/m	8 R 17.5 C[1]	-	113/112	M	-	code	-	code
	Cargo vehicles	11/70R22.5	-	146/143	K	-	code	70	code
	Trailers	14/80 R 20	-	157	K	-	code	80	code
	Buses	295/80R 22.5	-	149/145	M	-	mm	80	code
MPV	Multi-purpose vehicles	10.5R20 MPT[8]	14 PR	134	G	-	code	-	code
EM	Vehicles	18.0-25EM[9]	32 PR	-	-	-	code	-	code
	Loaders	29.5-29 EM[9]	28 PR	-	-	-	code	-	code
IT	Industrial trucks a/m	6.50-10[6]	10 PR	-	-	-	code	-	code
	Trailers	21 x 4[6]	4 PR	-	-	code	code	-	-
	Industrial trucks a/m	28x9-15[7]	14 PR	-	-	code	code	-	code
		300x15[7]	18 PR	-	-	-	mm	-	code
AS	Tractors	480/70 R 34	-	143	A8	-	mm	-	code
		7.50-60 AS[10] front	6 PR	-	-	-	code	-	code
	Moving equipment[11]	11.0/65-12	6PR	-	-	-	code	-	code

[1] tires for light (delivery) trucks (also for scooters which carrying capacity); [2] reinforced – additional designation for reinforced tires of two-wheeled vehicles and passenger cars; [3] PR – load range class; [4] the code of the loading range for single and double tires; [5] speed code for the nominal value of the vehicle speed; [6] pneumatic tires; [7] solid rubber tires; [8] MRT - multipurpose tires; [9] EM - tires for earthmoving machines; [10] AS - c.-g. tractors; [11] tires for mobile equipment and trailers.

For car tires: the rated speed is equal to the maximum speed. Cars designed for movement at a maximum speed of 60 km/h and less have additional carrying capacity.

For most tires, reducing the tire's load capacity can provide higher maximum speeds. On the other hand, tires characterized by increased load capacity are intended for trailers for passenger cars moving at speeds of up to 100 km/h, as well as for special trucks used for short-distance transportation.

The internal pressure in tires of certain sizes, PR treads or operational characteristics is regulated by the standards or recommendations of the tire manufacturer.

Correspondence between the nominal speeds for M+S tires of passenger cars, trucks and motorcycles and the maximum speed values is not regulated. However, a table indicating the speed at which this tire can operate must be fixed inside the car within the driver's field of vision. The tire categories below may be marked with additional data regulated by US road safety regulations. These data, applied to the sidewall of the tire near its rim, are used in Canada, as well as in Israel: FMVSS (Federal Motor Vehicle Safety Standard) 109 for passenger car tires; FMVSS 119 for two-wheeler and commercial vehicle tires.

These data are stamped near the letters "DOT" and contain the identification code of the tire and the date of its manufacture, as well as information about the maximum load capacity (load rating), maximum internal pressure and the number of cord layers in the frame and breaker belt.

Australian safety regulations APR 23 apply to passenger car tyres; these standards use identification codes (Table 10.4) borrowed from the US Federal Motor Vehicle Safety Standards (FMVSS) 109 and EEC-R30 standards.

Table 10.4 – Tire performance codes (examples)

Load index									
LI	50	51	88	89	112	113	145	149	157
kg	190	195	560	580	1120	1150	2900	3250	4125

Nominal speed symbols for the tire																
GSY	F	Q	J	K	L	M	N	P	Q	R	S	T	H	V	W	Y
km/h	80	90	100	110	120	130	140	150	160	170	180	190	210	240	270	300

10.2.4 Application of tires

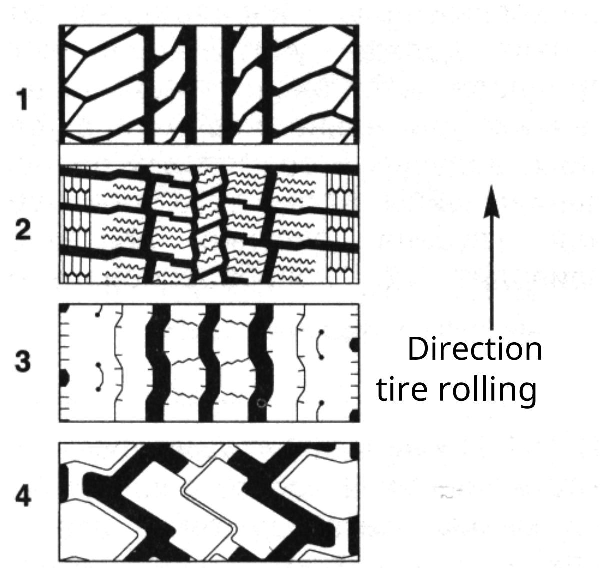
Optimum performance can be obtained when tires of the same design (for example, radial tires) are installed on all wheels of the vehicle.

Particular attention should be paid to the installation of tires on wheel rims. Tires should never be installed on a damaged, deformed rim with traces of corrosion (only minimal rim wear is allowed).

New valves and, if possible, new inner tubes and rims should be used with new tires. Precautions must be taken when reinstalling tubes in tires that have been expanded during previous use, which can lead to dangerous creases in the tube after it is reinstalled in the tire.

Tire protector

It is forbidden to restore the tread (Fig. 10.6) on the tires of two-wheeled vehicles and passenger cars; it is necessary to comply with the requirements of the manufacturing plant when restoring tires of other categories.



1 – car tire; 2 – tire type M+S for a passenger car; 3 – truck tire; 4 – a truck tire with a tread that provides
reliable grip on the road

Figure 10.6 – Examples of tread patterns

It is recommended to replace tires characterized by different degrees of tread wear according to the scheme, when the tire is replaced from one axle of the car to another axle. A decrease in the depth of the tire tread pattern is accompanied by a decrease in the protective layer covering the breaker belt and tire frame. This aspect must be taken into account when the tire is expected to be used for a long time in conditions that lead to possible damage to the tire. In addition, the depth of the drawing is reduced of the tread leads to a disproportionate increase in the car's braking distance. In the table 10.5 shows the data that allows you to compare the braking distances at a speed of 100 km/h for small-class (front-wheel drive) and medium-class (rear-wheel drive) cars with different levels of tire tread wear (braking distances also largely depend on the condition and type of road surface , tire profile and the quality of the rubber material of the tire).

Table 10.5 – Dependence of the braking distance of the car on the depth of the tire tread pattern (with an initial braking speed of 100 km/h)

Car		Small-class passenger car with front driving wheels					Mid-class passenger car with rear-wheel drive (equipped with ABS)			
The depth of the pattern is which, mm		8	4	3	2	1	8	3	1.6	1
Braking distance	in the city	76	99	110	129	166	59	63	80	97
	in %	100	130	145	170	218	100	107	135	165
Increase in braking of travel for every 1 mm of tire tread wear, %			7	15	25	48		1.4	20	50

Determining the exact characteristics of tires is an important condition for optimizing handling, driving qualities, comfort of the car and reducing vibrations in the transmission.

Questions for self-control

1. What types of wheels are used on modern cars?
2. Name the main parts of a car wheel.
3. What are the requirements for car tires?
4. What are the differences between chambered and chamberless, radial and diagonal tires?
5. How are car tires classified?
6. Describe the construction of automobile tires.
7. How are car tires marked?
8. How to choose tires for cars?

11 STEERING

11.1 Requirements for steering

Steering is designed to change the direction of the car. In accordance with the requirements of European Directives 70/311/EWG, the steering system must provide:

- high maneuverability (maneuverability, mobility) of a car;

− ease and convenience of management (easiness of steering), including mi-zero transmission of shocks from the road to the steering wheel (high efficiency);

- high reliability;
- minimal side slip of the wheels when turning;
- minimal vibrations, including the absence of self-oscillations of the wheels;
- kinematic compatibility with the suspension.

The maximum permissible activation time and working effort for a fully operational steering system are given in table 11.1.

Table 11.1 - Norms of work effort in the steering system

Car class la	A fully functional system steering control			Defective steering system bathing		
	Maximum work effort la, N	Chas, p	Radius turning m	Maximum work effort N	Chas, p	The radius of gate, m
M_1	150	4	12	300	4	20
M_2	150	4	12	300	4	20
M_3	200	4	12	450	6	20
N_1	200	4	12	300	4	20
N_2	250	4	12	400	4	20
N_3	200	4	12[1)]	450[2)]	6	20

1)If there is a turn limiter, this value may be smaller.

2)500 N for a vehicle without or with two or more controlled axles, except for friction-controlled axes.

11.2 Classification of steering

1. By the method of turning:

a) controlled wheels; b)
controlled axes;
c) assembly of links;
d) side turn.

2. According to the location of the driver's seat:

a) right - when moving to the left; b)
left - when driving on the right.

3. By type of steering mechanism:
a) by type of transmission: mechanical, hydraulic; b) according to the gear ratio: variable, constant; c) by turnover: turnover, on the verge of turnover.
4. By type of steering drive (steering trapeze):
a) by location: front – in front of the axle, rear – behind the axle; b) depending on the type of suspension: one-piece (with dependent suspension), split (with independent suspension).

11.3 Working process of steering

The requirements for the operation of the steering system (Fig. 11.1) are:

- maximum possible damping of vibrations transmitted from co- the car leans on the steering wheel when driving on uneven roads, but vibration damping should not lead to a loss of feedback in the steering;

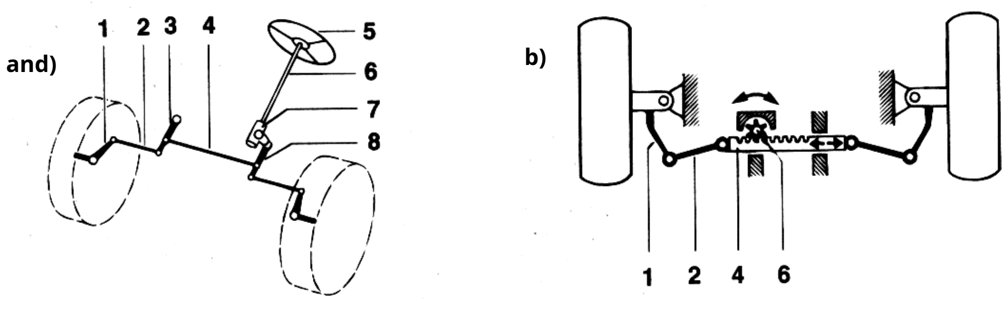
− the main kinematic steering parameters must meet release Akerman's conditions: when turning, the front left and right steering wheel axles must intersect with the rear wheel axle at the same point;

− due to the existing rigidity of the steering system (especially on the left when rubber-metal joints are used) the car must respond to the slightest turns of the steering wheel;

- when the steering wheel is released, the wheels should automatically turn in position of rectilinear movement;

- the steering system must have a minimum gear ratio ratio in the steering drive to ensure steering speed;

- the required force on the steering wheel is determined not only by ratio in the steering drive, but also the load on the front steered axle of the car, the turning radius, wheel suspension parameters and the condition of the tire tread (tire casing).



a) traditional; b) rack-and-pinion steering;
1 – rotary fist lever; 2 – lateral steering pull; 3 – pendulum lever; 4 – transverse steering rod or toothed rack; 5 – steering wheel; 6 – steering shaft;
7 – crankcase of the steering mechanism; 8 – steering bipod

Figure 11.1 - Steering schemes

With the specified values of the radius of the steering wheel rim r_{rk}, transfer number and_{ru} and efficiency η_{ru} steering effort on the steering wheel forests R_{rk} proportional to the moment M_c support of turning wheels

$$R_{rk} = \frac{M_{with}}{r_{rk} \cdot and_{ru} \cdot \eta_{ru}}. \qquad (11.1)$$

Size M_c significantly depends on the weight falling on the controlled wheels sa, from the coefficient of adhesion of the wheels to the road, from the speed of the car V_a, decreasing in 1.5-2 times with an increase V_a from 0 to 20 km/h. For reduction R_{rk} amplifiers are used (booster).

Radius r_{rk}=0.15-0.20 m for cars and 0.2-0.3 m for vans sad ones Gear ratio (by wheel turning angles θ_{to} and steering wheel with θ_{Cr}) and_{ru}=and_{room}+and_{rp}, and usually the gear ratio of the steering gear water and_{rp}=0.9÷1.1 (up to 2 in some models) and changes slightly when turning the steered wheels. Gear ratio of the steering mechanism and_{room}= 15÷20 in passenger cars, 20÷25 in trucks (at BelAZ and_{room}=40.4). Turn controlled wheels usually does not exceed 30÷36° from the average position not what and_{ru}=20 requires turning the steering wheel by 2 revolutions (by 4 revolutions at and_{ru}=40).

efficiency η_{ru}= 0.7÷0.85, and usually the losses in the pins are [l] 40–50%, 10–15% in rod joints, 35–50% in the steering mechanism of the total amount of losses. With two or more steered axles (steering axle) η_{ru}= 0.5÷0.7.

The wear resistance and service life of the steering drive is usually 3-4 times less than that of the steering mechanism. Backlash in the steering wheel (steering wheel play) increases [l] due to wear: rod hinges – by 2–4°, bipod splines – by 10–20°, pivots – by 13–20°, spring shrinkage – by 2–3 °.

Handling characteristics(agility; turnability)

A car with so-called oversteer has a smaller turning radius than corresponds to the steering wheel turning angle; in case of insufficient controllability, a larger turning radius is obtained due to different wheel deflection angles, which occur when centrifugal forces increase; the ratio of lateral forces to the load on the wheel becomes different for the wheels of the front and rear axles. Of course, the car must be driven in neutral. Although this allows for the optimal use of lateral forces (maximum speed when driving on turns), it leads to a decrease in the subjective perception of the car's stability limit. In addition, the moment of separation of the wheel from the road when turning is not amenable to calculation, because this separation can take place simultaneously for both rear and front wheels. For this reason, the goal of most constructors is the achievement of a slight lack of controllability, because in this case the separation of the wheels at the turn ensures the straightening of the course of movement.

Steering kinematics

The kinematic parameters of the steering and the design of the steering axis of the car should be such that the driver can feel the amount of traction between the tires and the road.

The cross fall pivot causes the front part of the car to tilt when the wheels are turned in one direction or another. This, in turn, causes the appearance of a stabilizing moment.

The camber of the steered wheels serves to balance the forces arising due to the misalignment of the wheels even when moving in a straight line; these forces create tension in the rods and steering levers, which leads to the rapid appearance of transverse forces when the car is turning.

The longitudinal inclination of the axis of rotation of the wheel (head fall pivot) creates a shoulder for transverse forces, that is, a stabilizing moment that depends on the speed of movement.

The running-in shoulder determines the degree of influence of disturbing forces. Nowadays, the goal of designers is to obtain a break-in arm equal to zero or a minimal negative value.

11.3.1 Steering drive

From fig. 11.2 it follows that in order for the wheels to move without lateral slippage, it is necessary (without taking into account the lead angles) to fulfill the condition

$$ctg\alpha_{with} - ctg\alpha_{in} = \frac{B'}{L}. \qquad (11.2)$$

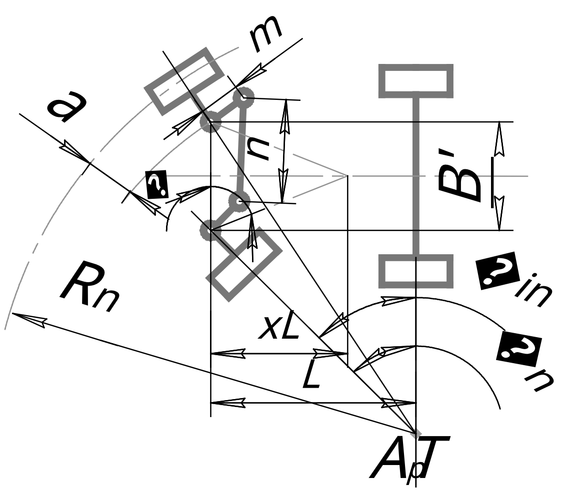


Figure 11.2 – Calculation scheme of kinematic parameters turning the car

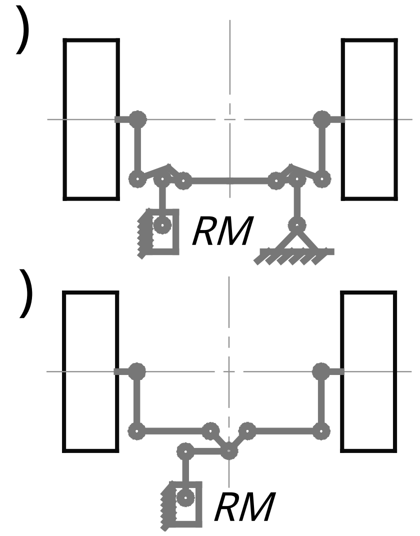


Figure 11.3 – Schemes of articulated trapezoids with independent suspension of controlled wheels

The steering trapezoid establishes (see Fig. 11.2) the following dependence [l] between external turning angles a_{with} and internal a_{in} wheel - $ctg\theta = \frac{B'}{2xL}$:

$$a_{with} = \theta + arctg\frac{m\cos(\theta + a_{in})}{IN' - m\,son(\theta + a_{in})} - \arcsin\frac{m + 2B'sons\,\theta - m\,2\,sons2\theta - B'son(\theta + a_{in})}{\sqrt{B'2 + m2 - 2B'm\,son(\theta + a_{in})}}. \qquad (11.3)$$

Thus, the steering trapezium only approximates the performance conditions (11.2). Of course $\frac{m}{n} = 0.14 \div 0.2$, $\theta = 60 \div 75°$, moreover $x \approx \frac{2}{3}$ - for back trapezius, $x \approx 1$ – for the front trapezium. Turning radius (see rice. 11.2) $R_n = \frac{L}{son\,a_{with}} + a$.

In the case of independent suspension of the steered wheels, to match the kinematics of the suspension, a segmented trapezoid is used, made according to various schemes, for example, according to the schemes of fig. 11.3.

The dimensions and placement of the longitudinal thrust must also be agreed with the kinematics of the suspension. For example, if the spring *1* (rice.11.4) is connected to the frame at the back with an earring *2*, with a hinge at the front, and a steering mechanism *3* get up aligned with the axis of the steered wheels, then the wheel must move in an arc when hitting a bump *MM* together with the spring, but the front end of the longitudinal thrust will move in an arc *NN*. Arcs *MM* and *NN* diverge- which causes the wheels to wobble. A significant reduction in the difference in trajectories *MM* and *NN* is provided by installing the front end of the spring on the ser- front, rear - on a hinge or by placing the steering mechanism in front of the axis of the steered wheels. However, even in this case, some inconsistency is possible, because when the steered wheels turn, the position of the bipod changes (point 4 in Fig. 11.4).

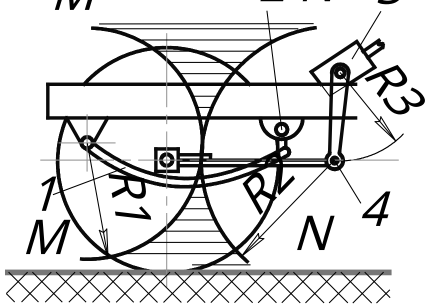


Figure 11.4 – Scheme of selecting the position and method of fastening longitudinal thrust

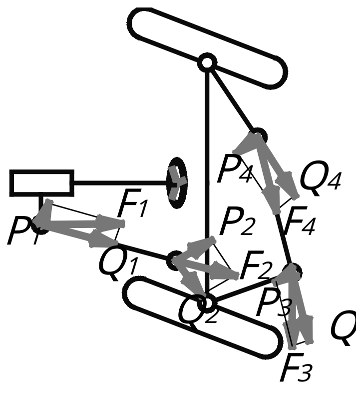


Figure 11.5 – Scheme of forces acting in the steering drive (without taking into account vertical components)

The forces acting in the steering drive, without taking into account the vertical components, are shown in fig. 11.5. At every point F – total force, Q – pos- long and R – transverse component. So,

$$F_1 = \frac{P_{pc} r_{pc} i_{p.m}}{\eta_{p.m} r_1}, F_2 = Q_1, P_2 r_2 = M_{c1} + F_3 r_3, F_4 = Q_3, P_4 r_4 = M_{c1} \quad (11.4)$$

where M_{c1} – moment of resistance of the turning wheel.

When turning the steered wheels of a stationary car, the stabilizing moments can be neglected. In this case, various empirical formulas are used, for example,

IN. Gauha $$M_{with1} = \frac{\phi}{3}\sqrt{\frac{G_{to}^3}{P_{sh}}}, \quad (11.5)$$

I. Taboreka $$M_{1c} = \phi G_{to}\sqrt{e_o^2 + \frac{I_o}{F_{to}}} \quad (11.6)$$

M. AND. Lysova $$M_{c1} = G_{to}(0.132\phi r + fe)\frac{1}{\eta_1}, \quad (11.7)$$

where G_{to} - weight per wheel in kg,

P_{sh} – tire pressure in kg/cm2,

ϕ – coefficient of adhesion of the wheels to the road,

I_o is the polar moment of inertia of the impression,

F_{to} – footprint area,

there are is the distance from the pivot axis, r – free wheel radius,

f – coefficient of rolling resistance (coefficient of rolling resistance),

η - Efficiency, which takes into account losses in the pivots and hinges of the steering trapezium. The stiffness of the steering drive must be high, otherwise significant oscillations may occur in the steering, which impairs the stability of the car. However, too much stiffness contributes to the transfer of shocks from the road to the steering wheel. With independent suspension, the stiffness of the steering drive is 1.5-2 times lower than with dependent suspension. It decreases with an increase in the number of steered wheels.

11.3.2 Steering mechanism

The steering mechanism should be characterized by:

- lack of backlash (looseness; play) in all elements when moving along straight;
- low friction;
- high rigidity;
- the possibility of making adjustments.

Such steering mechanisms became the most widespread.

1. Worm: with a globoid worm and a roller (ZAZ, VAZ, AZLK, GAS); with a cylindrical worm and sector (Ural, KrAZ).

2. Screws: screw-nut and rack (steering rack) - sector (ZIL, MAZ, BelAZ); screw-nut-crankshaft (MAZ-525).

Crank (screw–crank) and gear (gear–gear or gear–rail) steering mechanisms are also used.

Obtaining large gear ratios is ensured by the use of helical and worm gears. However, their efficiency due to sliding friction becomes 0.5÷0.7. Replacing sliding friction with rolling friction (a roller instead of a sector, a nut with balls in helical grooves) made it possible to increase the efficiency to 0.8 ÷0.85.

The reversibility of the steering mechanism is of great importance – the ability to transmit forces from the bipod to the steering wheel. In this case, the greater the friction moment in the steering mechanism, the lower its inverse efficiency $\eta_{room.call}$, the less shocks from the road are transmitted to the steering wheel so, but the stabilization of steered wheels is worse. Therefore, it is desirable to have an inverse efficiency of the order of 0.5÷0.6.

Usually, the gear ratio of steering mechanisms does not change or changes little depending on the angle of rotation of the steering wheel. It is believed that the use of a variable gear ratio with $i_{p.m}>20$ with $\theta_{pc}<90°$ and $i_{p.m}<15$ at $\theta_{pc}>180°$ can improve car handling due to increase in control accuracy and decrease in the transmission of shocks from the road (due to a decrease in efficiency in the middle position) at high speeds and due to increased maneuverability at low speeds.

Steering mechanism with globoid worm and roller (Fig. 11.6). Gear ratio for the average position

$$i_{p.m} = \frac{2\tau r_2}{t z_r}, \qquad (11.8)$$

where t is the pitch of the helical line,

z_r is the number of steps taken by the worm,

r_2 is the initial radius of the globoid of the worm.

From the middle position to the extremes $i_{p.m}$ increases by 5-7%. efficiency η_{room}=0.77÷0.82.

Forces acting on the worm: circumferential $P_1 = \frac{M_{pc}}{r_1}$, axial $Q_1 = \pm P_1 \mathrm{tg}\beta_1$, radial $R_1 = P_1 \cos\beta_1 \frac{tga}{}$, a– engagement angle, β is the spiral angle.

Forces acting on the roller (Fig. 11.6): $P_2 = -Q_1$, $Q_2 = -P_1$, $R_2 = -R_1$.

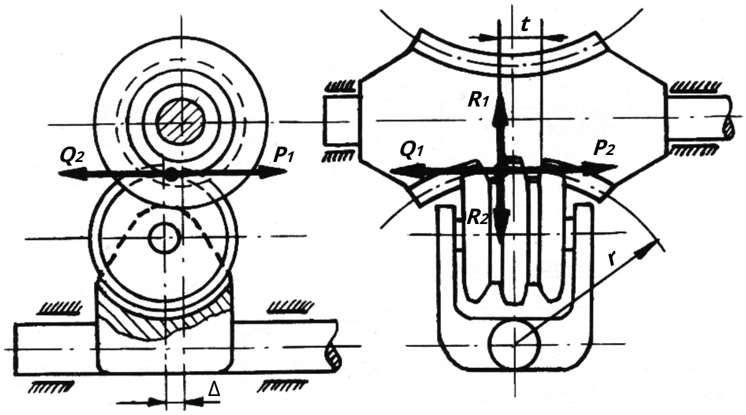


Figure 11.6 - Forces acting on a steering mechanism with a globoid worm and roller

Screw-nut and rail-sector steering mechanism (Fig. 11.7).

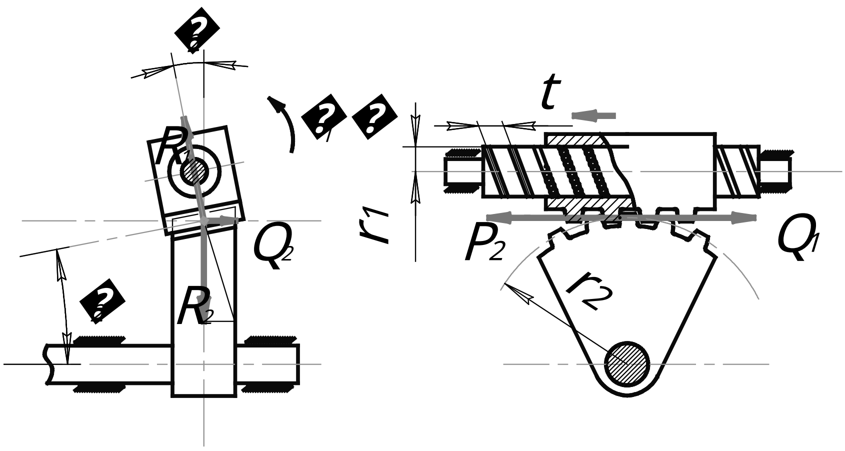


Figure 11.7 – Screw-nut and rail-sector steering mechanism

Gear ratio

$$i_{p.m} = \frac{2\pi \cdot r_2}{t}, \qquad (11.9)$$

where r_2 is the radius of the initial circle of sector teeth,

t - pitch of the screw.

efficiency η_{room} = 0.8÷0.9. The teeth of the sector are straight, but $Q_2 \neq 0$ due to the inclination of the teeth - cone with δ_2 = 7÷9° to adjust the engagement: $Q_2 = P_2\, tg\alpha\, son\, \delta_2$,

$P_2 = \pm \frac{M_{pc}\, i_{p.m},}{r_2}$ $\quad R_2 = P_2 \mathrm{tg} a \cos\delta_2$. For screw: $P_1 = \frac{M_{pc},}{r_1}$

$Q_1 = \pm P_1 \frac{2p}{t_1} = -P_2$Cr in addition, a radial is transmitted to the screw from the sector power $R_1 = -\sqrt{R_2^2 + Q_2^{\ 2}} = -P_2 \mathrm{tg} a$.

Screw-crank steering mechanism (Fig. 11.8).

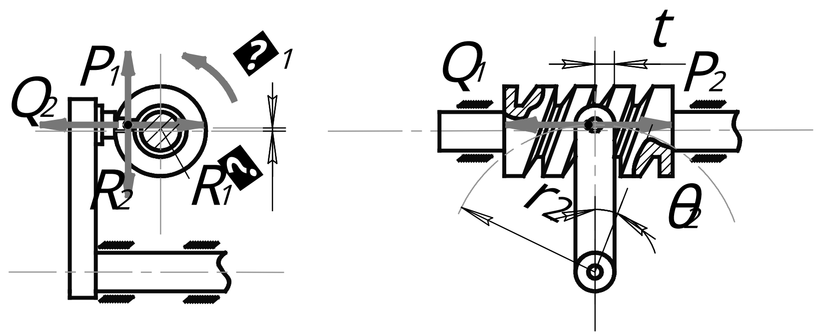


Figure 11.8 – Screw-crank steering mechanism

Gear ratio at t=const.

$$i_{p.m} = \frac{2\pi r_2}{t}\cos\theta_2, \qquad (11.10)$$

where θ_2 – angle of rotation of the bipod shaft.

The spike, made together with the crank, slides, and the spike, installed in the crank on bearings, rotates and rolls on the helical surface. Having made a screw with a variable pitch t, it is possible to obtain a variable gear ratio close to the optimal one.

The same dependences as for the globoid worm can be used to determine the forces acting on the screw. Besides, $P_2 = Q_1$, $Q_2 = -R_1$, $R_2 = -P_1$.

Geared steering mechanisms:

– gear-gear type $i_{p.m} = Z_2 / Z_1$;

– gear-rail type $i_{p.m} = r_{pc} / r_1$, where r_1 is the radius of the initial circle.

11.4 Steering amplifiers

Hydraulic boosters are best suited for steering. They consist of an energy source (pump H with tank B, Fig. 11.9), a distribution device (RP – control valve), executive mechanism (VM – power cylinder).

According to the arrangement with the RM steering mechanism, the following options are possible. 1. RP–VM–RM in one unit (ZIL).

2. VM separately, RP–RM in one block (Urals).

3. RM separately, RP–VM in one block (MAZ).

4. RP, VM, RM separately (gas) - fig.

11.9. Features of layout schemes.

1. Compactness, two short pipelines to the pump. Disadvantages: the drive and shaft of the bipod perceive full loads and shocks.

2. The shaft of the bipod is unloaded, but the pipelines between RP–VM have been added.

3. The bipod shaft is unloaded, only two long pipelines.

4. If the VM affects the steering trapezoid (Fig. 11.9), then the dismount broken shaft of the bipod and part of the drive parts when the pipelines are lengthened and there is a danger of oscillations in the amplifier.

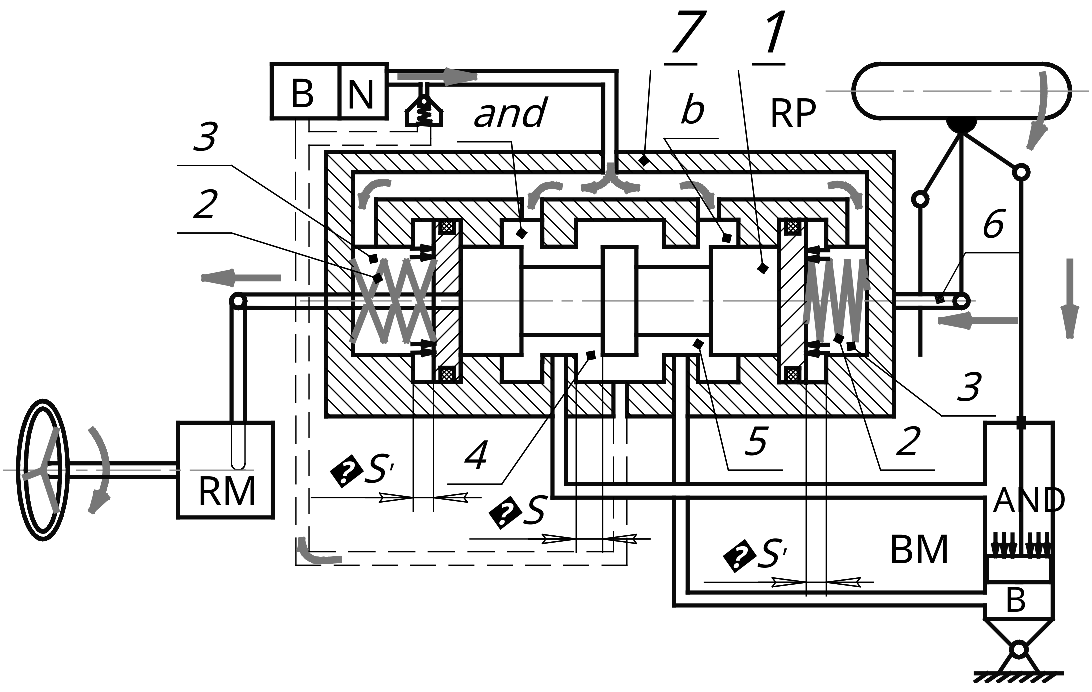


Figure 11.9 – Scheme of steering with a hydraulic amplifier

Requirements for power steering.

1. With a faulty amplifier, the car should not lose its roughness

2. Absence of self-activation of the amplifier from shocks from the side of horns and the ability to maintain the desired direction of movement when braking with a damaged tire.

3 Minimal delay in activation and ensuring the proportionality of the turning angles of the wheels and the steering wheel.

4. The effort on the steering wheel should be small, but proportional resistance to turning the wheels (road feel).

*Work process*hydraulic amplifier. When turning the steering wheel wheels, for example, to the right, (Fig. 11.9) the driver moves the valve 1 RP (left in Fig. 11.9), overcoming the forces of the left centering spring 2 and the pressure of the liquid in the left jet cavity 3. Channels 4 and 5 overlap. Liquid under pressure (pressure) passes from the cavity*and*RP into the cavity A of the VM, moving the piston of the VM (down in Fig. 11.9), since the cavity A of the VM is now connected only to the pumping line of the pump. Cavity B VM - only with a drain line.

Through the rod 6, which acts as a mechanical feedback of the wheel from the RP of the amplifier, the movement of the housing 7 of the RP is ensured - to the left on rice. 11.9 (with the spool in the extreme left position). The amplifier ensures movement monitoring: the angle of rotation of the steering wheel is proportional to the angle of rotation of the steered wheels.

As soon as the steering wheel stops turning, the spool will stop 1, but the housing 7, continuing to move, will take an average position relative to the spool, in which all the channels will be connected to each other again. The pressure in cavity A will decrease because the liquid will be able to pass freely from the injection line to the drain line. The steered wheels will stop turning.

RP according to fig. 11.9 has an open center: in the middle position, the liquid flows freely from the injection line to the drain line. RP with a closed center and a shut-off pump are rarely used.

*Energy source*consists of a pump that is usually driven from the car engine, oil tank and corresponding hoses and pipes.

Pump- usually of the rotary type with an internal passage channel – must have a performance that would ensure the supply of such an amount of oil to the system (even when the engine is idling) to obtain an angular velocity during the rotation of the steering wheel for at least 1.5 s^{-1}.

At higher speeds of rotation of the crankshaft, the further increase in oil supply to the system is stopped with the help of a bypass valve. Also, a valve that limits the maximum oil pressure is usually built into the pump.

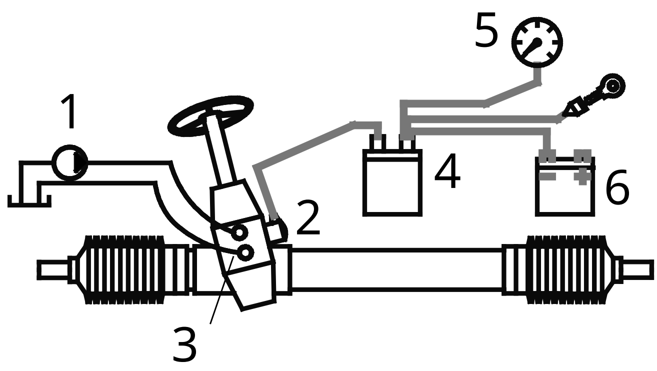
The design of the pump should ensure such a mode of operation that the working temperature of the oil does not rise above 100°C, there is no noise during the operation of the pump and no foam is formed in the used oil.

Distribution valve -serves to inject mass into the power cylinder la under such a pressure that corresponds to the angle of rotation of the steering wheel. The elastic element ensures the transformation of the torque on the steering wheel in the absence of backlash into the smallest possible control movement of the spool. The channels of the spool, which are made in the form of a bevel or a cone, as a result of the control movement, will form an opening of the appropriate cross-section for the passage of oil. Distribution valves usually work according to the so-called "open center" principle, that is, when the distribution valve does not operate, the oil supplied by the pump is passed back to the oil tank at zero pressure.

*Power cylinder*steering turns the oil pressure into additional mutual force affecting the rail and increasing the driver's influence on the steering wheel. This cylinder is usually located inside the crankcase of the steering mechanism and is characterized by low friction.

Steering with an amplifier, the operation of which is modulated depending on the speed of movement

Growing requirements for the comfort and safety of driving a car lead to the need to use a steering system equipped with an amplification system with pressure modulation capabilities (Fig. 11.10). It works depending on the speed, that is, the speed of the car, which is measured using an electronic speedometer, controls the force affecting the steering (Fig. 11.11). The control unit evaluates the signals corresponding to the speed of movement and determines the level of reaction of the hydraulic booster.



1 – injection oil pump; 2 – electro-hydraulic converter;
3 – distribution valve body; 4 – electronic control unit;
5 – electronic speedometer; 6 – battery

Figure 11.10 – Scheme of steering with an amplifier, the operation of which is modulated depending on the speed of movement

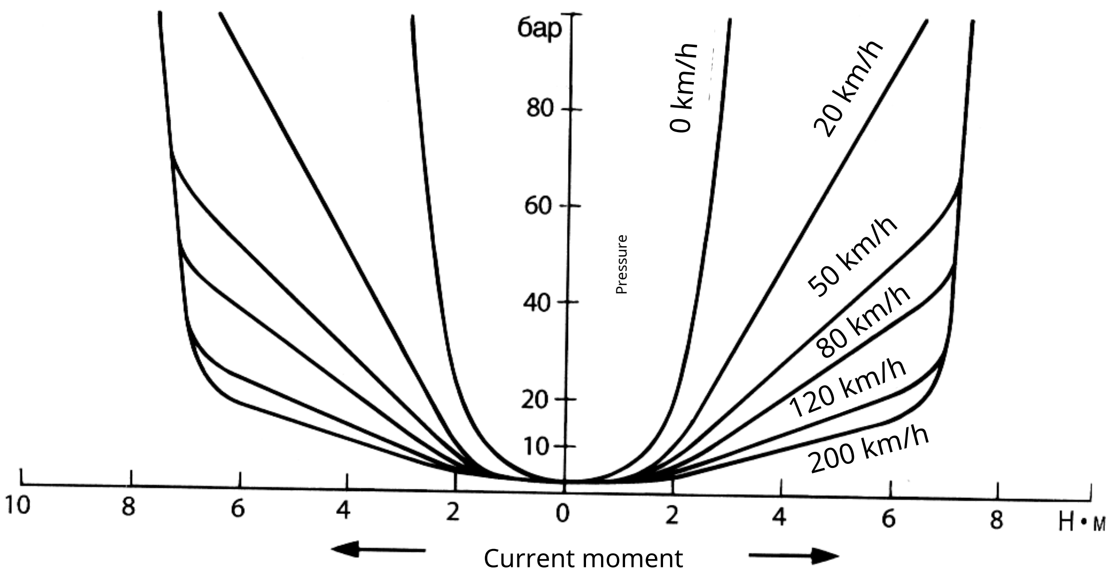
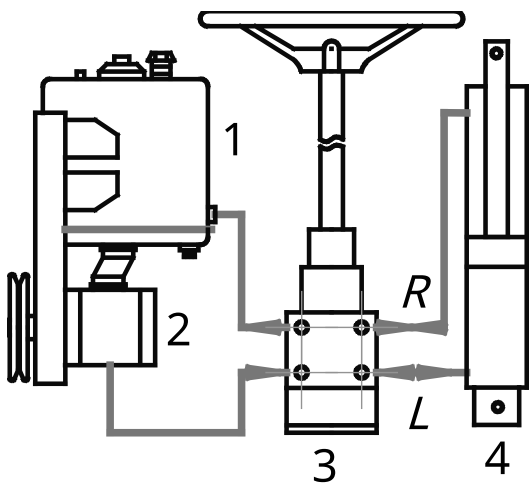


Figure 11.11 – Characteristic curves of the power steering system (the shape of the curves may change according to changes car parameters)

This level of response is transmitted to the distribution valve in the steering system via an electro-hydraulic converter. With such a pethe reaction level changes depending on the speed of the car. The special characteristics of the power steering make it possible to turn the steering wheel with minimal effort when the car is stationary or while it is moving at a low speed; the degree of amplification decreases with increasing speed. Thus, when driving at high speeds, it is possible to control the turns of the car in the optimal mode. With such a system, it is important that the oil pressure and flow never decrease and therefore these parameters can be immediately required in critical control situations.

The steering is completely hydraulic

When using hydraulic steering (Fig. 11.12), there are no mechanical connections between the steering wheel and the wheels of the car. The increase of forces in the steering drive is carried out hydraulically, and these forces are transmitted exclusively with the help of hydraulic devices. The dosing pump located in the control unit ensures the supply of pressurized oil to the power cylinder; oil consumption corresponds to the angle of rotation of the steering wheel. Due to the inevitable loss of oil in the metering pump, problems may arise with setting the steering wheel in a position corresponding to the movement of the car in a straight line. This is the reason for the limited use of such systems.



1 – oil tank; 2 – pump;
3 – control unit with dosing pump; 4 – power cylinder.
Connection: *R*– direction of oil supply when turning the car to the right; *L*- the direction oil supply when turning the car to the left

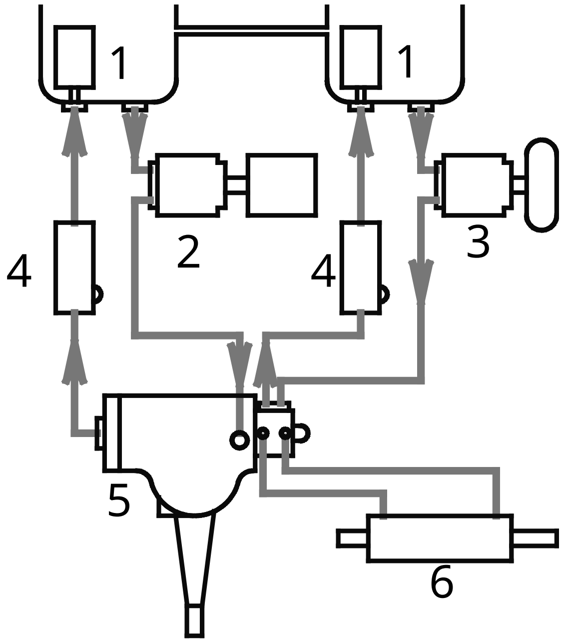
Figure 11.12 – Hydraulic steering

In many European countries, the maximum permissible speed of vehicles equipped with hydraulic steering is is 25 km/h, in Germany – 50 km/h, and with a two-circuit system this speed is increased to 62 km/h.

Two-circuit steering system designed for heavy-duty trucks

Two-circuit steering systems (Fig. 11.13) are necessary where the force on the steering wheel, in case of failure of the amplifier, exceeds 600 N.

These steering systems are characterized by hydraulic redundancy. The operation of each steering circuit in such a system is monitored using flow indicators.



1 – oil tank; 2 – pump driven by the engine; 3 – pump driven by car wheels; 4 – oil consumption indicator; 5 – double-circuit hydraulic turn control system; 6 – power cylinder

Figure 11.13 – Two-circuit steering system

Pumps in the steering circuits must have different drives (for example, from the engine, from a device whose operation depends on the speed of the car or an electric drive). According to the established rules, if the engine or one of the circuits of the steering system fails, it should be possible to drive the car with the help of the remaining functioning circuit.

Single-circuit power steering for trucks

Trucks are usually equipped with a screw-ball-nut-sector steering mechanism with a hydraulic booster.

In modern systems, the distribution valve is built into the steering wheel to make the structure compact and reduce its weight.

In some modifications (modification) of distribution valves, there are components for adjusting the gain depending on the speed of the vehicle or such parameters as lateral acceleration or load (load).

Estimated parameters of the amplifier

In fig. 11.14 shows the dependence of the force on the steering wheel without an amplifier P_{rk} and with an amplifier P_{RCP} from the moment of resistance of the controlled rotation wheel M_{with}.

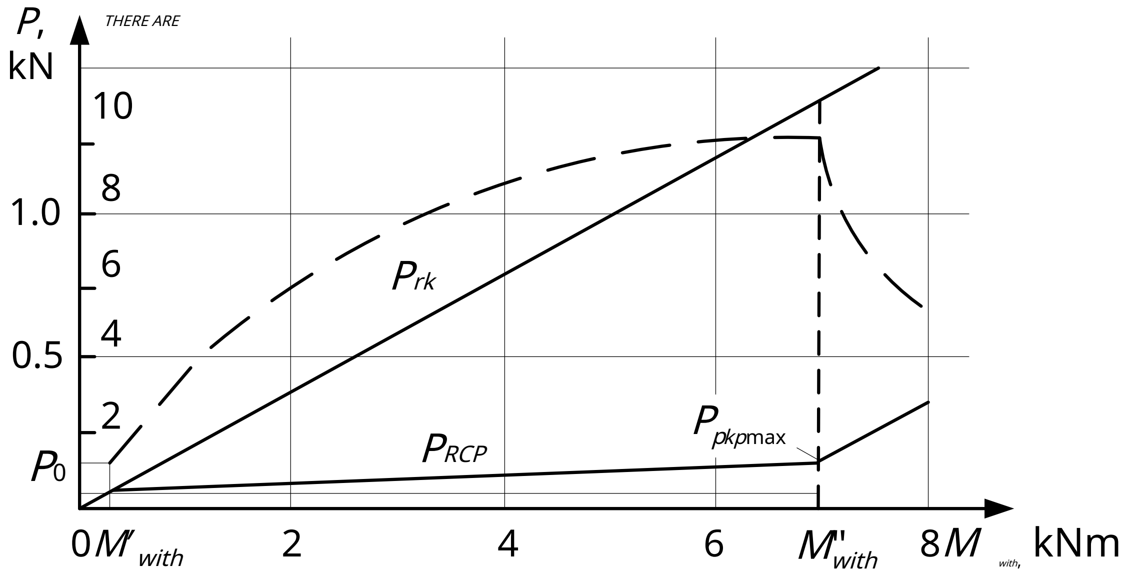


Figure 11.14 – Dependence of the force on the steering wheel on the moment turning resistance of the steered wheels

1. Operational efficiency

$$E = \frac{R_{rk}}{R_{rkn}} = 1 + \frac{R_n}{R_{rkn}}. \quad (11.11)$$

From fig. 11.14 it follows that at small resistances ($M_c < M_c'$) the amplifier does not turns on if there are centering springs. With the increase M_{with} increases- sia *THERE ARE* (to value M_{with}'' , at which the pump's safety valve is triggered), and of course $E_{\max}$ = 5÷10 (in BelAZ: 25). The power of the VM amplifier,

driven to the steering wheel, $P_n = \frac{p \cdot F 10^6}{i' \cdot \eta'}$, where p is the fluid pressure, and

Of course $p_{\max}$ = 6÷10 MPa, F – piston area, i' and η' is a transmission or layer from the steering wheel to the VM and KD.

2. Sensitivity index

$$P_o = \frac{Q+S}{i'' \cdot \eta''},\qquad (11.12)$$

where Q – force of centering springs;

S – friction forces in the RP and pressure in the reactive cavity;

i'' and η'' - gear ratio and efficiency from the steering wheel to the RP.

Of course P_o=20÷30 N for passenger cars and 40÷60 N for trucks.

Sensitivity to the angle of rotation of the steering wheel ±3÷6° at ignition backlash of 10÷15°. The power-on time of the amplifier is about 0.05 s (up to 0.5 s for pneumatic amplifiers).

3. Indicator of reactive influence of the amplifier on the steering wheel

$$\rho = \frac{dP_{rkn}}{dM_c}.\qquad (11.13)$$

From the experience of operation, it is necessary to ρ = 0.02÷0.05 N/N·m.

Other evaluation parameters are also used, for example, the sensitivity indicator when the amplifier is switched on in reverse (from the wheels), the maneuver indicator slowness in turns $\Delta M = \frac{t_n}{t_{bn}}$, where t_n and t_{bn} – turning time with amplifier and without an amplifier.

The use of an amplifier significantly facilitates driving, but at the same time, increased tire wear is possible due to excessive frequent turns on the spot. In addition, power steering is more prone to self-oscillations.

The maximum force on the steering wheel is taken according to the table. 11.1, which usually corresponds to the turn of the steered wheels of a stationary car with a full load on dry asphalt. Values are chosen for these conditions E_{max}. Minimum effort on the steering wheel P_o at THERE ARE=1 recognize is mainly due to the efforts of the centering springs. The less P_o, the less P_{RCPmax}. On the other hand, if $Q+S$ less applied force friction in the steering mechanism, the amplifier will turn on under the influence of the stabilizing moment (except for the case when the RP is installed between the steering mechanism and the steering wheel) and prevent the wheels from returning to the neutral position. That's why power $Q+S$ should be minimal, but greater than the reduced force of friction in the steering mechanism.

When hitting a bump, due to the short duration of the action, there is a significant change in pressure in the cylinder, which prevents the movement of the drive parts and reduces the force of the push transmitted to the steering wheel.

RPs are used: with centering springs and reactive cavities - ZIL, Ural; only with centering springs - ZIL, BelAZ; only with reactive cavities - KrAZ, MAZ, GAZ.

Calculation of the amplifier, in addition to strength calculations, usually covers the following stages.

1. Selection of the type and layout of the amplifier.

2. Static calculation - determination of forces and displacements, dimensions of the lindra and RP, centering springs and areas of the reactive cavities of the RP.

3. Dynamic calculation - determination of the time of turning on the amplifier, analysis of oscillations and stability of amplifier operation.

4. Hydraulic calculation - determination of pump performance, diameters of pipelines, etc. others

Dynamic calculation. Time t_1 turning on the amplifier is practically is indicated by the movement of the spool ΔS (rice.11.9) before channel overlap:

$$t_1 = \frac{\Delta S\, i''}{2\pi r_{pc}\omega_{pc\max}}, \qquad (11.14)$$

where $\omega_{pc\max}$ - the maximum calculated speed of the steering wheel rotation, accepted $\omega_{pc\max} \approx 9$ c^{-1}.

In time t_1 the speed of rotation of the controlled wheels with the help of an amplifier the valve should be slightly larger than without the amplifier, during sharp turns when driving on dry asphalt (approximately at a fluid pressure of $0.5p_{\max}$), which corresponds to an increase in the volume of liquid in the system by ΔV in time t_2. Therefore, should be

$$t_1 > t_2 = \frac{\Delta V}{Q_H\left(\eta_0 - \frac{\Delta Q_3}{Q_H}\right)}, \qquad (11.15)$$

where Q_H - pump performance in m^3/with;

$\eta_0 \approx 0.8$ – volume efficiency of the pump,

$\Delta Q_3/Q_H \approx 0.05 \div 0.1$ - leaks in the RP spool. For hydraulic boosters, of course $t_2 = 0.01 \div 0.02$ s.

When the RP is located between the steering mechanism and the steered wheels, self-oscillations of the wheels with a small amplitude of 1-2° may occur. Self-oscillations can occur when driving in a straight line at high speed due to a shock from the side of the road, but they are more likely when driving slowly on a turn. In the latter case, if you stop the steering wheel while turning the wheels, the wheels will turn another angle under the action of inertial forces and residual pressure in the cylinder Δa. Then under the action of forces the elasticity of the wheel tires will begin to turn in the opposite direction. When the wheels reach the starting position, the RP spool will shift and the pressure in the cylinder will begin to increase. However, due to the elasticity of the hoses and parts, the melting pressure to the required value p_x will happen in time Δt, at the same time the wheels will turn on the corner -Δa. The cycle repeats itself.

The condition of the absence of self-oscillations, according to L. L. Ginsburg, there may be inequality

$$\frac{D}{2I}-\frac{Fl_n}{\pi}\cdot\frac{\sin(\nu t_2)}{I\cdot\nu}\cdot\frac{p_x\cdot 10^6}{\Delta a}>0, \tag{11.16}$$

where D is a coefficient characterizing the friction in the steering gear and tires in N·m·s;

I – the moment of inertia of the masses rotating around the pivots, N·m·s^2; F is the area of the cylinder piston in m^2;

l_n – arm of the force of the cylinder, m;

- the frequency of oscillations of the steered wheels in rads·p^{-1};

t_2 – cylinder delay time relative to RP, s;

$p_x/\Delta a$ – the ratio of the pressure in the cylinder to the angle of rotation of the controlled co-wood causing this pressure, MPa·rad^{-1}.

These values, except F and l_n, can be obtained only experimentally. For example, for MAZ: D=2960 N·m·s, I=66.8 N·m·s^2, = 47 rad·p^{-1}, $p_x/\Delta a$=1.11·10^3 MPa·rad^{-1}.

The occurrence of self-oscillations can be prevented by reducing the capacity of the pipelines, increasing the rigidity of the parts connecting the cylinder to the steered wheels, and the mandatory introduction of reactive cavities when installing the RP and the cylinder separately.

Hydraulic calculation. The selection of the pump performance is carried out on the condition that the filling time of the cylinder must be less than the time of turning the steering wheel by the driver

$$\frac{FS}{Q_N\eta_0-\Delta Q}<\frac{\phi}{\omega_{pc\max}}, \tag{11.17}$$

where S – stroke of the piston over time Δt, m;

ϕ – angle of rotation of the steering wheel during time Δt, rad.; ΔQ - leaks in the system in^3/s, should be $\Delta Q/Q_H$<0.1.

From equation (11.17)

$$Q_H \geq \frac{FS\cdot\omega_{pc\max}}{\phi\cdot\eta_0}+\frac{\Delta Q}{\eta_0}. \tag{11.18}$$

The calculated performance of the pump must correspond to the angular velocity of the motor shaft ω_1=(1÷1.5)·ω_{1xx}. The volume of the tank is taken as equal 10÷15% of the minute performance of the pump for cars and 15÷20% for trucks. The temperature of the liquid in the system should not exceed 100 °C. In addition, for passenger cars, the noise from the pump should not be heard against the background of the engine noise.

Pipeline sections are selected based on head loss Δp_p speed fluid flow V_p

$$\Delta p_p=\lambda_{tr}\rho\cdot\frac{l_T}{d_T}\cdot\frac{V_p^2}{2}, \tag{11.19}$$

where $\lambda_{tr} \approx 0.025$ – coefficient of friction losses;

$\rho \approx 88$ kg·m^{-3}– liquid density;

l_T and d_T - length and diameter of pipelines.

$$V_p = \frac{4Q_H}{\pi \cdot d_T^2}. \qquad (10.20)$$

It should be $\Delta p_p \leq 0.1 \div 0.3$ MPa for cars, 0.2÷0.5 MPa for trucks, $V_p \leq 2 \div 3$ m·p^{-1} for lines under pressure and 1÷2 m·p^{-1} for drain lines.

The working fluid is industrial type 20 or 12 oil with anti-seize and stabilizing additives.

11.5 Strength calculations

Design loads

1. According to the maximum calculated torque on the steering wheel $M_p = P_{rk\max} \cdot r_{rk}$ at $P_{rk\max}$ = 500 N (for cars you can take 250 N).

2. According to the maximum braking force applied to one steering wheel wheel leg ($\phi_{\max}$ = 0.8÷1.0; for another wheel ϕ = 0) $P_p = G_{with}' {}_h \cdot \phi_{\max}$.

3. According to the impact force of the controlled wheel on the limit obstacle [l]. For details of the pivot pin, the calculated loads are the same as for semi-unloaded half-axles.

Materials. The steering wheel shaft and tie rods are usually made of seamless pipes (steel 20, 35, 45). Joint fingers - steel 18XHT, 15XH, etc. with a surface hardness of HPC 56-63.

Details of the steering mechanism: crankcases - malleable cast iron KCh35-10, KCh37-12; worms - steel 35X, 30XH3A with a surface hardness of HPC 45-52; rollers - steel 20X, 18XHT with HRC 52-56; screws, screw nuts, sectors and bipod shafts - 18KHT steel with HRC 58-64.

Details of the hydraulic amplifier: cylinder body and RP spool - gray cast iron СЧ24-44, ductile iron КЧ35-10; separately installed cylinder - often made of steel 35, 45; spool - steel 20, 18XH, 15X with HRC 52-56; piston rod - steel 35, 45.

Calculations. In steering, the strength of the parts of the steering mechanism and steering drive is counted on.

The load in the parts of the steering mechanism and steering drive can be calculated by setting the maximum force on the steering wheel or determining this force by the maximum resistance of the steering wheels of the car in place. These loads are static. However, when the car is moving on an uneven road or when braking on a road with different coupling coefficients in the steered wheels, the steering wheel parts can perceive dynamic loads. Therefore, dynamic loads must be taken into account using the dynamism coefficient k_d=1.5-3.0, which is selected depending on the type and purpose of the car mobile, as well as the conditions of its operation.

The working pair of the steering mechanism (worm-roller, screw-nut, etc.) is checked for crumpling and bending, the bipod for complex stress (bending and torsion), the bipod shaft for torsion.

The power steering rods and the power cylinder rod are designed for compression and longitudinal stability with a margin of stability n_{with}=1.5÷2.5.

The details of the rotary trunnion are designed for bending, and the pivot - for bending, crumpling, shearing.

In the steering drive, the shaft of the steering bipod, the steering bipod, the longitudinal and transverse steering rods, the rotary lever and the levers of the rotary pins are calculated.

The steering bipod shaft is designed for torsion.

The steering bipod is designed for bending and twisting from the maximum force acting on the ball joint from the longitudinal steering rod.

The spherical finger of the bipod is designed for bending and cutting in a dangerous section and for crumpling between the longitudinal steering tie rods.

The calculation of the ball fingers of the longitudinal and transverse steering rods is performed similarly to the calculation of the ball finger of the steering bipod, taking into account the operating loads on each finger.

Longitudinal and transverse steering rods are designed for compression and longitudinal bending. The stability margin should be 1.5...2.5.

The pivot arm and pivot pin arms are designed for bending and torsion.

Questions for self-control

1. Basic requirements for car steering.
2. How is steering classified?
3. Describe the working process of steering.
4. Explain the principle of operation of the steering drive.
5. Types of steering mechanisms, their advantages and disadvantages.
6. Principle of operation and purpose of the power steering.
7. Types of power steering, their advantages and disadvantages.
8. Features of calculating the strength of the steering?

12 BRAKES

Brakes are designed to effectively slow the car down to a stop and keep it stationary. The following braking modes are distinguished.

1. Emergency - it corresponds to the minimum braking distance and maximum significant deceleration due to road adhesion (j_c≈8÷9 m•s^{-2}at ϕ=0.8÷0.9).

2. Official - slow downs correspond to itj_c≤3÷3.5 m•s^{-2} , which are not cause unpleasant sensations in passengers.

3. Service long on protracted descents up to 10 km atj_c = 0÷2 m•s^{-2}.

4. Braking of a stationary car, including significant ones inclinations

Braking is provided by brake systems consisting of brake mechanisms and drives (brake gear).

The brake system consists of a power supply device; control device; brake drive for transmission of braking effort and for activation of auxiliary and parking brake systems; additional tractor equipment for trailer braking; wheel brake mechanisms.

Each of the components affects the forces that determine the braking efficiency of the car.

Working (main) brake system(primary braking system) – equipment that allows the driver to reduce the speed of the vehicle and stop it during normal operation. It should provide the maximum possible deceleration (5.8 m•s^{-2}for vehicles of the category$M_1$and 5.0 m•s^{-2}for vehicles of other categories). At the same time, it is necessary to monitor the duration of the brake system activation (t_c), which for an accident with a hydraulic drive should be no more than 0.5 s, and for an accident with other types of drive - no more than 0.8 s.

Spare brake system(emergency brake system) – equipment that allows the driver to reduce the speed of the vehicle and stop it in the event of a malfunctioning brake system. It should provide at least 40% efficiency compared to the working system.

Parking brake system(parking brake system) – equipment that allows you to keep the vehicle stationary on an inclined surface and in the absence of the driver. It should provide a stationary state when the engine is disconnected from the transmission:

- vehicles with a full load - no less on the slope 16%;

− passenger cars, their modifications for cargo transportation, and also buses in equipped condition - on a slope of at least 23%;

- lorries and road trains in the equipped condition - in storage not less than 31%.

Auxiliary braking system(secondary brake system) – equipment that allows the driver to maintain the speed of the car or reduce it on long road descents. It should operate normally, ensuring (without the use of a working brake system) the stabilization of the speed of a car or a road train in the range of 30-40 km/h on an 8% slope, regardless of its length. Average deceleration values when reducing speed from 40 to 20 km/h should be at least 0.6 m/s^2.

Automatic braking system(automatic braking system) – equipment that automatically brakes the trailer when it accidentally separates from the tractor.

Anti-blocking system(antilock brake system, ABS) – a part of the working brake system that prevents one or more wheels from locking when braking the car. Control of the braking forces on the wheels is carried out on the basis of sensors that control the speed of rotation of each wheel or directly by indirect parameters.

Brake drive– part of the braking system that transmits energy, which is distributed by the control device. Connects the control or power supply device to the brake mechanisms, which create forces directed against the movement of the car. It can be mechanical, hydraulic, pneumatic, vacuum, electric or combined, for example, hydromechanical, hydropneumatic type.

Brake mechanism- a part of the brake system that creates forces, directed against the movement of the vehicle (for example, friction brakes).

Auxiliary device of the tractor for the trailer– part of the brake system we are on a tractor that is designed to transmit power to and control the trailer's braking systems. The auxiliary device contains control and power transmission lines with connecting elements for the brake system of the trailer.

Brake systems of road trains, as well as individual cars, consist of brake mechanisms and a brake drive. Brake mechanisms used on road trains and individual cars are structurally identical. The brake drives of road trains have some features due to the need to provide simultaneous control of the brakes of the tractor and trailer.

12.1 Requirements for brake systems

Basic requirements for brake systems.

1. High and approximately the same for all cars and road trains activity, including:

a) minimum time of operation of braking systems;

b) synchronicity of the increase and decrease of the braking torque of all braking mechanisms of the system (the difference of the largest values is no more than 15%);

c) stable and high values of the friction factor in brake mechanisms in the entire range of operating temperatures and pressures.

2. Ease and convenience of management.
3. High reliability, trouble-free operation during the entire term service under any operating conditions.
4. Good removal of heat from friction pairs and their protection from moisture and dirtying
5. Minimal noise during operation.
6. When the coupling of the road train breaks, the trailer brakes must provide its automatic stop.

In addition, brake systems are subject to requirements to ensure the required braking efficiency, assessed by braking distance and deceleration, as well as the stability of the road train during braking.

According to DSTU 3649:2010 "Wheeled vehicles. Requirements for the safety of the technical condition and methods of control" of the value of the braking distance S vehicles depending on the initial braking speed V_0 (km/h) must correspond to those given in tab.

persons 12.1.

Table 12.1 – The procedure for calculating the normative values of the braking distance

Type of accident	Accident category	Braking distance, m, no more values, calculated by formulas
single	M_1	$V_0 x(0.10 + V_0/150)$
	M_2, M_3, N_1, N_2	$V_0 x(0.15 + V_0/130)$
	N_3	$V_0 x(0.18 + V_0/130)$
Road trains	M_1	$V_0 x(0.15 + V_0/150)$
	N_1, N_2, N_3	$V_0 x(0.18 + V_0/130)$

Stability must be maintained when a part of the brake system fails. When the towing link is detached from the tractor, the brake system installed on the trailer must ensure its stopping. It is allowed not to have a special braking system on single-axle trailers, the mass of which does not exceed 750 kg, provided that the mass of the tractor in the equipped state, with which this trailer is operated, exceeds two times or more the mass of the trailer. In special cases, one-axle and two-axle trailers weighing up to 2.5 tons can be used without braking systems, provided that their weight does not exceed 65% of the tractor's weight in the equipped state. The brake system of a trailer (semi-trailer) necessarily includes a working and stop

Yankee system. A trailer with a gross vehicle weight of more than 10 tons must also have an auxiliary system.

The main international European order regulating requirements for brake systems is UNECE Regulation No. 13, according to which:

– *category vehiclesL*(Less4 wheels) - two- and three-wheeled motor-driven vehicles must have two mutually independent braking systems. On heavy three-wheeled vehicles of the category L_5 both brake systems must act on two wheels and a parking brake system must be additionally installed;

– *vehicles of categories M andN*must have working, spare and parking brake systems, which may have common components. Such vehicles must have at least two mutually independent control devices (equipment) and a specified distribution of braking force on the axles. Vehicles categories M_2 and N_2 must be equipped with ABS. Auxiliary braking systems can be used on vehicles of categories M_3 and N_3. Vehicles category M_3 are used for long-distance and long-distance trips and must meet the requirements for speed stabilization on descents exclusively due to the operation of the auxiliary braking system;

– *trailers of category O*may not be equipped with a brake system, however, there are requirements for safely attaching the trailer to the tractor vehicle. Starting with the category AT_2, trailers must be equipped with a working braking system, which may have common components with the tractor's braking system. Control of the parking brake system must be accessible to the person behind the wheel of the vehicle. Accordingly, the redistribution of braking force on separate axes is set. It is also suggested to use an anti-lock brake system for some trailers in the category AT_3 and above Inertia braking systems are allowed for trailers in the category AT_2 including;

– *vehicles equipped with anti-lock braking systems brakes (ABS),*must meet application requirements13 of UNECE Rules No. 13 (for vehicles of categories M_2, M_3, N_2, N_3 on the roads of the 1st categories). Basic requirements:

- no locking of the controlled wheels at high speeds more than 15 km/h, regardless of the road surface;
- preservation of stability and controllability during braking; optimal the use of tire grip with the road with equal generated efforts on the road surface of the 1st category or with different grip coefficients in the left and right wheels;
- use of a visual means of controlling electrical appliances ABS;

– *tractors and trailers with pneumatic braking systems moms*must have two- or multi-wire means of supplying compressed news rya When the working or spare brake system of the tractor vehicle is operating, the brake system of the trailer compatible with it must work smoothly. In the event of any malfunction in the working brake system of the tractor vehicle, the brake components of the trailer, which do not work in the event of such a failure, must brake the trailer smoothly. If one of the pipelines between the tractor and the trailer breaks or there is a significant leak in it, the trailer must brake automatically. The braking effect of the car is determined by the pressure of the control line. The working brake system of the trailer can function only together with the working spare or parking brake system of the tractor.

12.2 Classification

*Classification of brake systems*by purpose (working, parking, spare, auxiliary) is given above.

Classification of brake mechanisms

1. According to the principle of action (according to the nature of the connection between movable and immovable we in parts):

a) friction (disc, drum; pad and belt); b) hydraulic (hydrodynamic);
c) electric (induction, generator); d) compressor - back pressure in the engine; e) aerodynamic (flaps, parachutes). 2. By location:
a) wheeled;
b) transmission;
c) on the body (flaps, parachutes);
d) near the engine (retarder).

Classification of brake drives

1. According to the method of energy transfer to braking mechanisms: a) mechanical (parking brake systems);
b) hydraulic (at full weight G_a<8 tons);
c) pneumatic;
d) combined (hydropneumatic).
2. By energy source:
a) driver;
b) driver and amplifier (at 8 t > G_a>4 t, for passenger cars at G_a>1.5 tons);
c) almost entirely an amplifier (at G_a>8 tons).

Important criteria for design are: norms and rules regulating the minimum braking force before the initial appearance of blocking and the procedure for blocking the front and rear wheels; distribution of weight load;

engine braking; malfunction of the brake line; regulator of braking forces; amplifier; ABS; systems based on ABS.

*Design of the brake system*mainly concentrated on parameters of wheel brake mechanisms and on the control device.

Calculation criteria for wheel brake mechanisms: type of brake mechanism; service life; space; acceptable pressure levels; rigidity.

Calculation criteria for the control device: travel and effort on the pedal during normal braking, during emergency stops, with a malfunction of the brake line, with a malfunction of the amplifier; compliance with traffic comfort; installation space; connection with brake pressure control systems.

12.3 Working process of braking mechanisms

12.3.1 Drum pad brake mechanisms

Assuming a constant pressure distribution$p$=const (true for made overlays) along the arcβlining and arrangement of uniform$X$in center of the arc (Fig. 12.1), we obtain

$$X = p \cdot in \cdot r \cdot b, \qquad M_{Mr} - X - - r, \quad b \qquad (12.1)$$

where$b$– the width of the pad,r -drum radius.

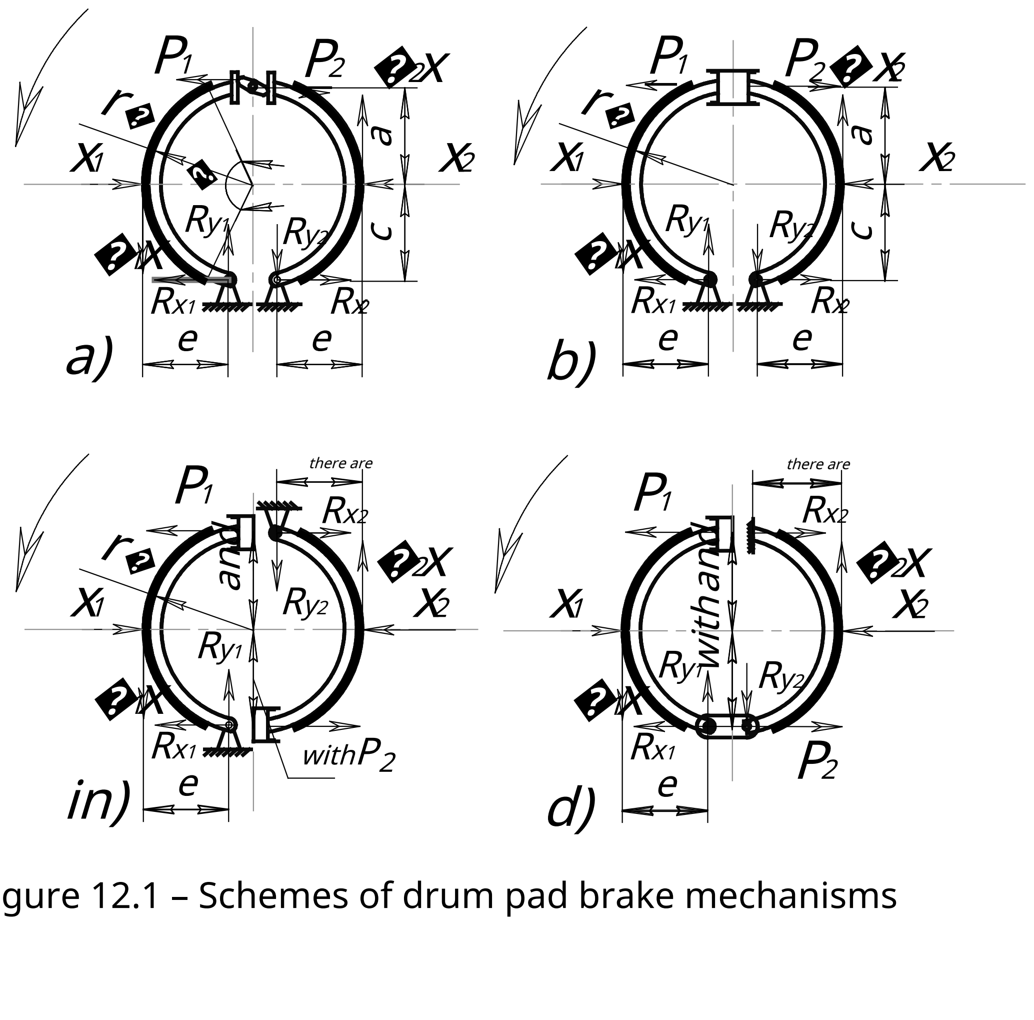


Figure 12.1 – Schemes of drum pad brake mechanisms

Of course β = 1.6÷1.8 rad, p_{max} < 2.5 MPa during emergency braking, friction coefficient μ=0.3÷0.35, r_b- d_{to}/2– (20÷30) mm (d_{to}– rim diameter wheels). The magnitude of the braking torque M_{Mr} depends on the location pad supports and the design of the clamping device. The most common designs are described below.

Brake with one-sided supports (Fig. 12.1, *and*, *b*). When the direction of rotation of the drum is counter-clockwise, the left pad will be active - it is pressed against the drum by the moment of friction $X_1 \cdot \mu \cdot r_b$, rights will be pa-gray - it is pressed by the moment of friction $X_2 \cdot \mu \cdot r_b$. From the moment equations relative to the points AT_1 and AT_2

$$H_1 - p_1 \frac{and\text{-}with}{with - e}, \qquad H_2 = p_2 \frac{and + with}{with + \mu \cdot e}. \tag{12.2}$$

Braking torque of two pads

$$M_{Mr} = \mu \cdot r_b \cdot (X_1 + X_2) = \mu \cdot r_b \cdot (a + c) \cdot \left(\frac{p_1}{c - \mu \cdot e} + \frac{p_2}{c + \mu \cdot e} \right). \tag{12.3}$$

Brake with one-sided supports and equal movements of the pads: $S_1 = S_2$ –rice. 12.1, *and* (the clamping fist is fixed in the supports - ZIL, MAZ, KrAZ), $P_1 < P_2$, $X_1 = X_2$, then from (12.2) $\frac{p_1}{c - \mu \cdot e} = \frac{p_2}{c + \mu \cdot e}$ and

$$M_{Mr} = 2\mu \cdot r_b p_1 \frac{and + with}{c - \mu \cdot e} = M_{MrS}. \tag{12.4}$$

This brake is balanced by radial forces ($X_1 = X_2$), wear of both the pad is the same.

Brake with one-sided supports and equal driving forces: $P_1 = P_2$ - Fig. 12.1, *b* (hydraulic drive cylinder - rear wheels of passenger cars mobiles, GAZ-53A, as well as floating fist, wedge), $X_1 > X_2$ and

$$M_{Mr} = 2\mu \cdot r_b \cdot p_1 \frac{(a + c) \cdot with}{c^2 - \mu^2 \cdot e^2} = M_{gS} \cdot \frac{with}{c + \mu \cdot e}. \tag{12.5}$$

This brake is unbalanced by radial forces ($X_1 > X_2$), wear active pad is higher than passive.

Brake with spread supports (Fig. 12.1, *in*) – both pads are active (for reverse – both are passive), if $P_1 = P_2$ (hydraulic drive - front car wheels), then

$$M_{Mr} = \mu \cdot r_b \cdot p_1 \cdot (a + c) \cdot \left(\frac{1}{c - \mu \cdot e} + \frac{1}{c + \mu \cdot e} \right) \approx M_{MrS}. \tag{12.6}$$

This brake is balanced, pad wear is the same.

A brake with a large self-reinforcement in one direction (Fig. 12.1, *Mr*). $R_x = X_1 - P_1$, P_2 = 0 and

$$M_{Mr}=2\mu\cdot r_b\cdot p\cdot\frac{(a+c)^2}{(c-\mu\cdot e)^2}=M_{gS}\cdot\frac{and+with}{2(c-\mu\cdot e)}. \quad (12.7)$$

This brake has a significant imbalance and significantly greater wear of the second (right in Fig. 12.1,*Mr*) pads.

At $c-\mu\cdot e\to 0$ the active pad is pressed against the drum even at $P\approx 0$ (jams), this is not acceptable. It should be $c>\mu\cdot e$.

For comparative assessment of braking mechanisms, coefficients are used efficiency ratio $R_e=M_{Mr}/M_p$, where M_p – the moment calculated for the drive-by them

$$R_e=\frac{M_{Mr}}{(R_1+R_2)\cdot r_b}. \quad (12.8)$$

In the table 12.2 the given values R_e at $e_1=e_2=0.85\cdot r_b$, $a=c=0.8\cdot r_b$, $\mu=0.35$ [1].

Table 12.2 – Calculated values of the efficiency factor R_e

Brake mechanism	R_e
Brake mechanism with $S_1=S_2$ (Fig. 12.1,*a*)	0.70
WITH $P_1=P_2$ (Fig. 12.1,*b*)	0.81
With spread supports (Fig. 12.1,*in*)	1.11
With a large self-reinforcement (Fig. 12.1,*Mr*)	3.54

An important feature of the working process of the brake mechanism is the phenomenon of self-amplification, which is estimated [6] by the coefficient of self-amplification $R_c=M_{Mr}/M_{Wed}$, where M_{Wed} – static (an unloaded wheel that does not rotate is) braking torque:

$$R_c=\frac{M_{Mr}}{(R_1+R_2)\cdot r_b}\cdot\frac{c}{\mu\cdot(a+c)}. \quad (12.9)$$

In the table 12.3 the given values R_c [6].

Table 12.3 – Coefficients of self-reinforcement

Brake mechanism	Front passage	Reverse gear
Brake mechanism with $S_1=S_2$ (Fig. 12.1,*a*)	1.0	1.0
Z $P_1=P_2$ (Fig. 12.1,*b*)	1.2-1.4	1.2-1.4
With spread supports (Fig. 12.1,*in*)	1.8-2.2	0.5-0.7
With a large self-reinforcement (Fig. 12.1,*Mr*)	4.0-4.5	0.35-0.45

The stability of the braking torque value in operation depends on many factors and, first of all, on the change in the friction coefficient, to which brake mechanisms with higher self-reinforcement have the greatest sensitivity. However, sensitivity is reduced for brake pads with two degrees of freedom, for example, with a non-fixed axis of rotation (see Fig. 12.1,*Mr*).

Brake drums, pads and their supports must have high rigidity to reduce wear and noise (creaking). Brake screeching is caused by high-frequency vibrations (over 500-1500 Hz) of the drum and pads, as well as a significant difference in the static and dynamic coefficients of friction of the pad material.

12.3.2 Disc brake mechanisms

There are two types of disc brake mechanisms: a) with a rotating disc (open), b) with a rotating body (closed).

A brake with a rotating disc (Fig. 12.2,*and*) has the following features in comparison with drum pad brakes:

a) less weight;

b) good heat dissipation;

c) balance of axial, but not circumferential forces with one pair of flat pads;

d) small clearances between friction surfaces (0.1-0.15 mm instead of 0.35-0.5 mm in pad brakes) and piston stroke (0.35-0.4 mm instead of 1.5-2 mm in pad brakes) - of brakes);

e) poor protection against pollution and 2-3 times greater wear and tear due to pressures on the linings reaching 5 MPa during emergency braking.

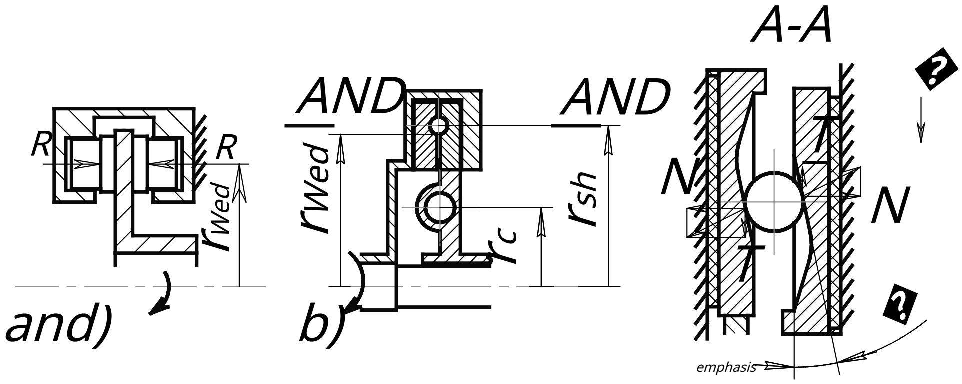


Figure 12.2 – Schemes of disc brake mechanisms

The magnitude of the braking moment (at n flat pads)

$$M_{Mr}=\mu\cdot r_{with_p}\cdot R\cdot n \qquad (12.10)$$

does not depend on the direction of rotation, a $R_e=\mu$, $R_{with}=1$.

Brake with a rotating body (Fig. 12.2,*b*) has two pairs of friction surfaces and several balls in the holes. One of the disks has a stop. When stationary body $P\cdot r_c\cdot n_c - N\cdot r_{sh}\cdot tg\alpha=0$ (total and moments about the brake axis) and

$$M_c = 2\mu\cdot r\cdot N_{cp} = \frac{2\mu\cdot r_{cp}}{r_{sh}\cdot tg\alpha}\cdot P\cdot r_c\cdot n_c, \qquad (12.11)$$

where r_{cp} – average radius of the friction surface;

r_c and r_{sh} is the distance from the axis of rotation to the axis of the working cylinder and to the axis balls;

n_c=2 – the number of working cylinders.

When rotating the case

$$M_c = 2\mu \cdot r_{cp} \cdot N_1 = \frac{2\mu \cdot r_{cp} \cdot P\, r_c n_c}{r_{sh} \cdot tg\alpha - \mu \cdot r_{cp}}, \tag{12.12}$$

since in this case $P \cdot r_c \cdot n_c - N \cdot r_{sh} \cdot tg\alpha + N_1 \cdot \mu \cdot r_{cp} = 0$. To brake it didn't jam, it should have $tg\alpha > \mu \cdot r_{cp} / r_{sh}$. Of course a=30÷35°.

Thus, for this brake mechanism $R_c = \dfrac{1}{1 - \dfrac{\mu \cdot r_{cp}}{tg\alpha \cdot r_{sh}}} \approx 2.5$ and

$$R_e = \frac{M_{Mr}}{P \cdot r_{cp} \cdot n_c} = \frac{2\mu \cdot r_c}{r_{sh} \cdot tg\alpha - \mu \cdot r_{cp}} \approx 2 \text{ with } \mu = 0.35, a = 30°, r_{cp} = r_{sh} = 1.5.$$

12.3.3 Calculation of the brake mechanism for heating Energy balance of the car during braking ($V_{a\ horse}$=0) has the view

$$\frac{d' \cdot G_a V^2_{a\ beginning}}{2g} = A_t + A_f + A_\phi + A_w, \tag{12.13}$$

where d' ≈1.05 – coefficient of consideration of rotating masses when the engine is disconnected;

A_t – operation of frictional forces in brake mechanisms;

A_f, A_ϕ, A_w – the work of the rolling resistance forces (covering the losses in the transmission seeds), tire slippage, air.

If Δ is the slip coefficient of the braked wheel, then

$$A_m = P_{Mr} \cdot S_{Mr} \cdot (1-\Delta), A_f = f \cdot G_a \cdot S_{Mr} \cdot (1-\Delta), A_\phi = \phi \cdot G_a \cdot S_{Mr} \cdot \Delta, A_w = P_{wcp} \cdot S_{Mr}, \tag{12.14}$$

where S_{Mr} – braking path;

R_{Mr} – braking force of braking mechanisms;

P_{wcp} is the force of air resistance.

When locking the wheels Δ =1 and $A_t = A_f = 0$, then

$$\frac{G_a V^2_{a\ beginning}}{g} = \phi \cdot G_a \cdot S_{Mr} + P_{wcp} \cdot S_{Mr} \tag{12.15}$$

The wear of friction linings is indirectly assessed by the value of the specific work of friction during a single braking to $V_{a\ horse}$=0.

$$\varepsilon = \frac{G_a \cdot V^2_{a\ beginning}}{2g \cdot 3.6^2 \cdot F_\Sigma}, \tag{12.16}$$

where F_Σ – the total surface of the friction linings (friction facing).

When braking with $V_{a\ beginning}$ = 60 km/h should be ε < 0.04÷0.1 Jm^{-2} in depending on the type of car.

An increase in the temperature of the brake drum during a single intensive and short-term braking, neglecting the dissipation of heat into the environment

$$\Delta t^\circ_b = t^\circ_b - t^\circ_n = \frac{P_{hk} S_{Mr}}{427\, G_b \cdot c} = 0.922 \cdot 10^{-5} \frac{G_a V^2_{a\ beginning}}{Z \cdot c \cdot G_b}, \qquad (12.17)$$

where Z – number of brake wheels;

t_p – air temperature;

G_b – drum weight;

with = 500 J/kg·K (0.125 kcal/kg·°C) – heat capacity of cast iron (*with* = 840 J/kg·K for aluminum) drum.

It should be Δt°_b < 15 °C at $V_{a\ beginning}$ = 30 km/h

During prolonged braking

$$S_{Mr} = -\frac{c \cdot G_b}{F_b \cdot k_{TV}} \ln\left[\frac{P_{hk}}{427} - F_b \cdot k_{TV} \cdot (t^0_b - t^0_p)\right] + C,$$

where

$$\Delta t^0_b = t^0_b - t^0_p = \frac{P_{hk}}{427\, F_b \cdot k_{TV}} \cdot \left(1 - e^{-a_{Mr}}\right), \qquad (12.18)$$

where $a_{Mr} = \frac{S_{Mr} F_b \cdot k_{TV}}{c \cdot G_b}$;

F_b – drum cooling surface;

k_{TV} is the coefficient of heat transfer from the drum to the air, k_{TV} ≈ 20÷30 kcal/m^2·h·°C.

It follows from equation (12.18) that for reduction Δt°_b it is necessary to to sew F_b. Practically, when finning the drums, the temperature drops by 35÷40%. It should be taken into account that the braking mechanisms of the front and rear axles may have different energy loads during braking. It is believed that the maximum allowable temperature for friction materials with a rubber binder is 400°C. During long-term braking, the temperature of the drums can reach 250°C, the discs - 500°C.

12.4 Brake drives

To increase reliability, the drive, in addition to the mechanical one, must have at least two independent circuits, to increase efficiency - an amplifier, a brake force regulator and an anti-lock system.

Hydraulic, pneumatic and combined brake actuators are most widely used in cars. Combined drives usually include hydropneumatic and hydraulic with vacuum amplifiers.

Mechanical brake actuators nowadays are used, as a rule, only in emergency or parking brake actuators. Electric brake drives are used extremely rarely and only for braking trailers.

When designing a brake drive, regardless of its type, it is necessary to ensure: the necessary forces on the pads of the brake mechanisms of all wheels of the car and the following action of the drive, i.e. direct proportionality between the force applied to the brake pedal and the acting braking torque on the wheels of a car.

The most effective braking of the car will be in the case in which the braking moments on the wheels are proportional to the weight of the wheel. Under the same condition, the best stability of the car during braking will be ensured.

12.4.1 Calculation of the hydraulic brake drive

With the hydraulic brake drive (Fig. 12.3), which is widely used on passenger cars and light trucks, the force on the brake pads is determined by the formula

$$P = P' = \frac{P_p \cdot i_p \cdot d_2^2}{d_1^2}, \tag{12.19}$$

where P_p – effort on the brake pedal;

i_p – gear ratio of the brake pedal;

d_1 and d_2 – diameters, respectively, of the main and wheel brake cylinders ditch.

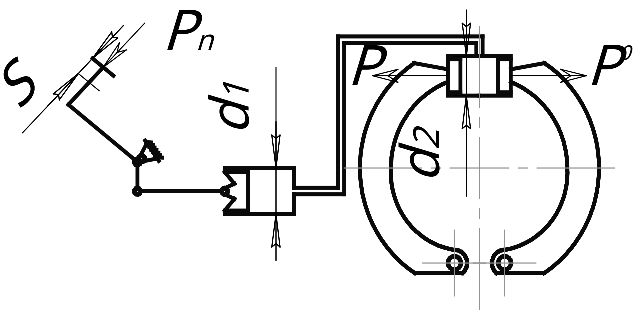


Figure 12.3 – Scheme of the brake hydraulic drive

When choosing the transmission ratio of the pedal and the diameters of the brake cylinders, two factors should be taken into account:

a) the diameter of the main brake cylinder and the transmission ratio of the pedal should be such that with the maximum effort on the brake pedal, the pressure in the brake line does not exceed the permissible value (of course, 10-12 MPa);

b) the working travel of the pedal must be sufficient to create the necessary pressure in the brake line.

The full stroke of the brake pedal (until it hits the floor), based on ease of control, usually does not exceed 170-200 mm. However, the working stroke of the pedal should be less than the maximum, because a certain margin of pedal stroke is necessary. The working stroke of the pedal (usually 40-60% of the full) can be determined by the formula

$$S = S_1 + S_2 + S_3, \qquad (12.20)$$

where S_1 – the pedal stroke required to select clearances in the brake actuator and braking mechanisms;

S_2 – the pedal stroke, necessary for the deformation of the brake actuator and brake speech mechanisms;

S_3 – the pedal stroke required to compensate for the thermal expansion of the lm drums.

The stroke of the pedal may change periodically due to the beating of the brake drums, which should also be taken into account when choosing the stroke of the brake pedal. Stroke size S_1 is determined by the formula

$$S_1 = i_p \cdot \delta_1 + i_p \cdot \sum i_x \cdot \delta_x, \qquad (12.21)$$

where $\delta_1 = \delta_2 + \delta_3$;

δ_2 – the gap between the rod and the piston of the main brake cylinder;

δ_3 – idling of the main brake cylinder piston;

p – the number of wheel brake cylinder pistons;

δ_x – idling strokes of pistons of wheel brake cylinders;

i_x – transmission ratio between the main and wheel brake cylinders dreams $i_x = d_2^{\,2} / d_1^{\,2}$.

The value of the idling stroke of the piston of the wheel brake cylinder can be approximately determined by the formula

$$\delta_x = 2\delta_k, \qquad (12.22)$$

where δ_k - the gap between the brake drum and the pad in its middle part-No.

The deformation of the brake actuator consists of the deformation of its individual parts

$$S_2 = x_1 + x_2 + x_3 + x_4 + x_5 + x_6, \qquad (12.23)$$

where x_1 – deformation of the brake pedal;

x_2 –deformation of flexible brake hoses and pipelines;

x_3 – deformation of friction linings;

x_4 – brake pad deformations;

x_5 –deformation of brake drums;

x_6 – volumetric deformation of the liquid.

The deformation of the brake pedal can be calculated according to known formulas. When determining the value x_2 it is possible (neglecting the deformation of the straight pipelines) take into account the deformation of flexible hoses only. In this case

$$x_2 = a_k \cdot l_{sum} \cdot \frac{i_n^2 \cdot P_n}{F_n^2}, \qquad (12.24)$$

where a_k – coefficient of deformation of flexible hose;

l_{sum} – total length of flexible hoses in the brake actuator;

F_n – piston area of the master brake cylinder.

Coefficient a_k expresses the volume deformation of one linear millimeter tra flexible hose when the pressure in the system increases by 1 MPa.

Size x_3 is determined by the formula

$$x_3 = \frac{2 \cdot i_n}{E} \sum_1^n p_i \cdot y_i \cdot i_{x_i}, \qquad (12.25)$$

where E – modulus of elasticity of the first type of friction pad material;

y_i – thickness of friction linings;

p_i – specific pressures in the middle part of the block (usually maximum pressures);

i_{x_i} – transmission ratios from the wheel cylinders to the main brake cylinder.

Deformation of brake pads can be calculated using the formula

$$x_4 = 2 \cdot i_n \cdot \sum_1^n f_i \cdot i_{x_i}, \qquad (12.26)$$

where f_i is the real deformation of one pad in its middle part. Size f_i can be determined to a first approximation if the load the pad as a two-support beam with an evenly distributed load along the length of the friction pad. For more accurate calculations, it is necessary to take into account the curvature of the block and the law of distribution of specific pressures.

Deformation of brake drums x_5 and volumetric deformation of the brake liquid x_6 is much more difficult to take into account. Nowadays, there are no reliable solutions Hunkov methods for their determination, as a result of which it is necessary to use experimental data to determine these values or not take them into account when calculating the pedal stroke.

Normal operation of the hydraulic brake actuator can be ensured only when using brake fluids of a certain quality. The requirements for brake fluids are determined not only by the conditions of their operation, but also by the material scrap brake drive parts. Brake fluid must have the following basic qualities:

- high boiling point (at least 200°C during the entire service of the car mobile phone) so that gas-air lines do not disrupt the operation of the brakes when they overheat;

– stable viscosity in a wide range of temperatures, at low temperatures temperatures, the viscosity should not exceed $12 \cdot 10^{-4} m^2/c$ (at – 40°C);

– high flash point (at least 80 °C);

– chemical stability, good lubricating and anti-corrosion properties;

- should not cause deterioration of paint coatings, even with prolonged exposure.

In order to increase the reliability of the hydraulic brake actuator, it must have two independent circuits. In the event of failure of one of them, the efficiency of the car's brakes must be maintained. In fig. 12.4 shows diagrams of possible variants of two-circuit brake actuators. The choice of one or another scheme is determined by three factors: the degree of loss of braking efficiency; permissible asymmetry of braking forces; the complexity of the drive.

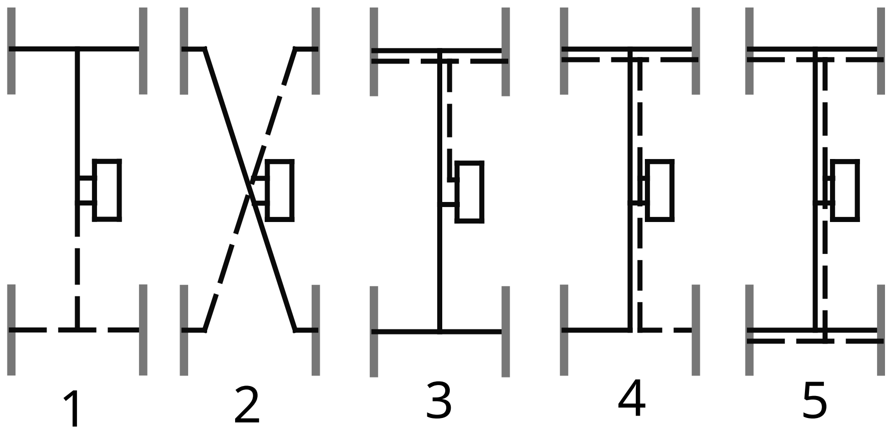


Figure 12.4 – Basic diagrams of two-circuit brake hydraulic drives

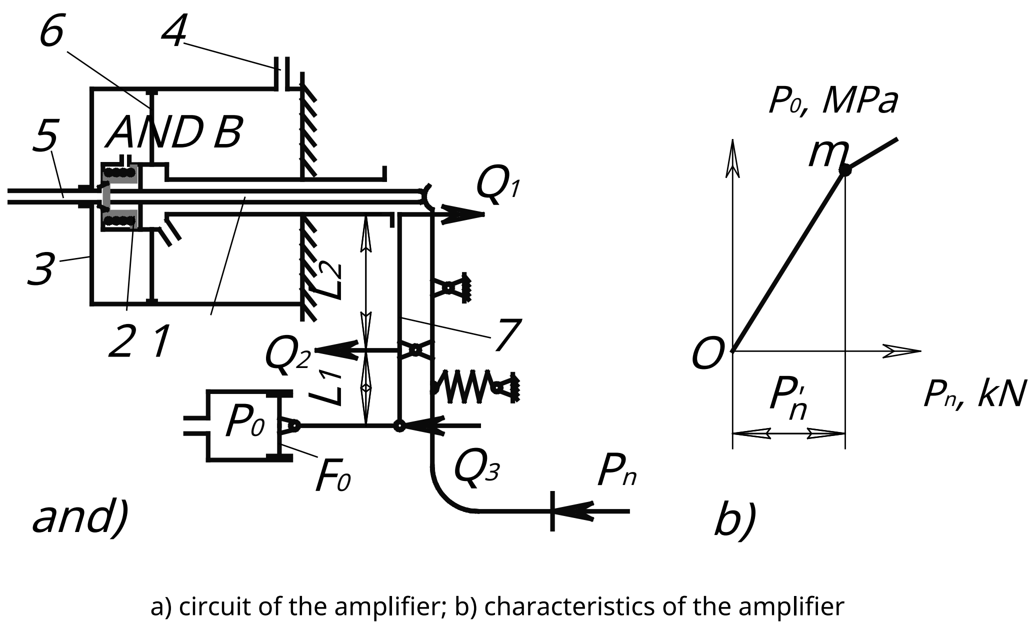
Scheme 1 is characterized by a strong decrease in braking efficiency when the front circuit fails. Schemes 2, 3, 4 maintain significant braking efficiency (at least 50%) in case of failure of any circuit, however, in the case of applying schemes 2 and 4, the braking forces will be asymmetrical. Scheme 5 is free from these shortcomings, but structurally complex.

12.4.2 Design and calculation of hydrovacuum amplifiers brakes

On passenger cars and light trucks with a hydraulic brake drive, in cases where the driver's muscle power is not sufficient to ensure the required braking efficiency, a vakuum brake amplifiers, the main advantage of which is that, using the vacuum in the engine intake pipeline, they do not require additional energy sources. Their significant drawback is that power cylinders or chambers of vacuum amplifiers have large overall dimensions due to small pressure differences created by the engine intake system.

Nowadays, there are many different types of vacuum amplifiers. Regardless of the scheme and design of the vacuum amplifier, it must have two mandatory structural elements - a power chamber (or cylinder) and a tracking device. Depending on the location of the tracking device, power chamber and master brake cylinder, all existing vacuum amplifiers can be divided into three main types.

The peculiarity of the first type of vacuum amplifiers is that the tracking device of the lifting type is not directly connected to the hydraulic system of the brake drive. The scheme of such a vacuum amplifier installed on GAZ and ZIL cars is shown in Fig. 12.5,*and.*The peculiarity of the second type of vacuum amplifiers is that the main brake cylinder, power chamber, and in some cases the tracking device are located separately from each other. Schemes of such amplifiers installed on GAZ-3307 and GAZ-3310 cars are shown in fig. 12.6 and 12.7. The peculiarity of the third type of vacuum amplifiers is that the power chamber, the main brake cylinder and the tracking device are located coaxially and are combined in one unit. Schemes of amplifiers of this type are shown in fig. 12.8 and 12.9.



a) circuit of the amplifier; b) characteristics of the amplifier

Figure 12.5 – Vacuum brake amplifier

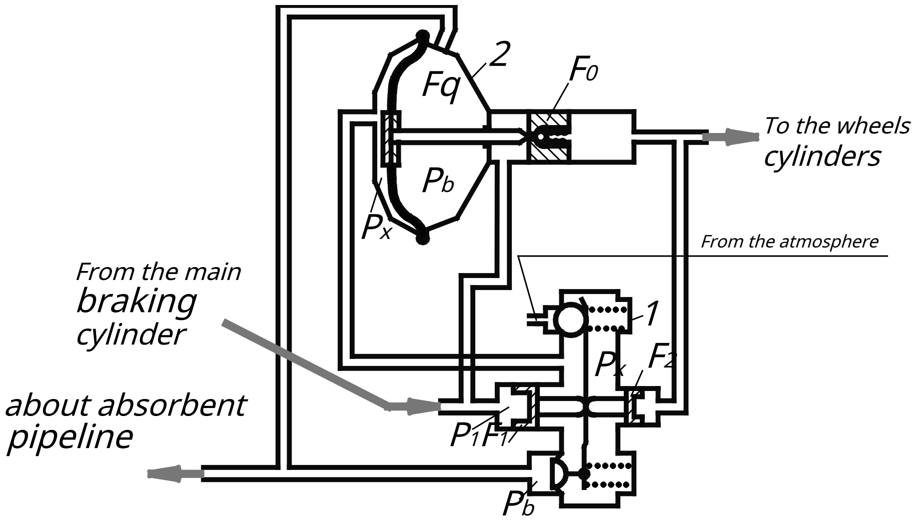
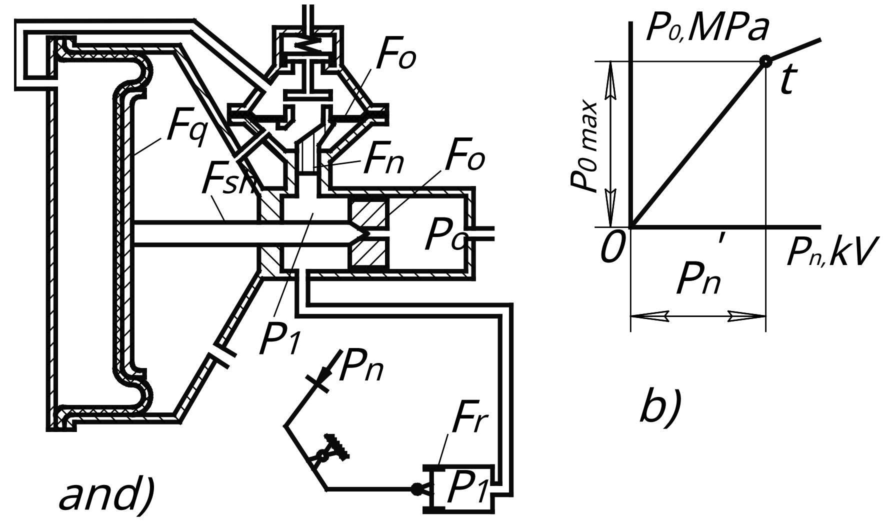


Figure 12.6 – Scheme of a hydrovacuum brake amplifier with separately a hidden tracking device



a) circuit of the amplifier; b) characteristics of the amplifier

Figure 12.7 – Hydrovacuum brake amplifier

The vacuum amplifier shown in Fig. 12.5, works in this way. In the absence of braking, when P_n=0 , rod *1* under the influence of ha-lm pedal moves the valve *2* to the extreme left position. At the same time cavity *AND* power cylinder *3* connects to the cavity *B,* and empty Nina *B* through the pipeline *4* connects to the suction pipeline engine. Thus, in the cavities *A* and *B* power cylinder is created the same pressure, equal to the vacuum in the suction pipeline - p_{in}. When braking the car, the upper end of the pedal moves to the right, allows the valve *2* close the opening connecting the cavities *A* and *B,* and connect the cavity *A* through the pipeline *5* with atmosphere Pressure in vacuum now *A* increases to the value p_{and}, as a result of which the piston *6,* alternating leaning to the right, presses the two-armed lever 7 with force Q_1. At the same time, force is transmitted to the same lever Q_2 from the brake pedal. The lower end is a lever 7 with force Q_3 acts on the piston of the master brake cylinder. Having considered the balance of the lever 7, it is possible to establish the ratio of forces Q_1, Q_2 and Q_3.

Given that $Q_2 = P_n \cdot i_n$, we will get

$$Q_3 = \frac{Q_2 \cdot l_2}{l_1 + l_2} = \frac{P_n i_n l_2}{l_1 + l_2}. \tag{12.27}$$

The fluid pressure in the brake system can be determined by the formula

$$p_0 = \frac{Q_3}{F_0} = \frac{P_n \cdot i_n \cdot l_2}{F_0 \cdot (l_1 + l_2)}, \tag{12.28}$$

where F_0 – piston area of the main brake cylinder.

Thus, the role of the tracking device in this vacuum amplifier is performed by a two-armed lever 7, which sets a certain pressure p_{and} in cavity *AND*, proportional to the magnitude of the force Q_1.

A characteristic feature of the considered vacuum amplifier is that the force created by it Q_3 on the piston of the master brake cylinder is smaller, than strength Q_2, created by the muscular power of the driver. The role of vacuum support The trick here is that the stroke of the piston of the main brake cylinder due to the rotation of the two-armed lever 7 about the point *and* much larger pedal stroke at a point *and.* This makes it possible to have the required volume of displaced fluid in the master brake cylinder at a high pedal ratio i_n and small area of the piston F_0.

In fig. 12.5, *b* the characteristic of the vacuum amplifier is shown, showing the relationship between the force applied to the pedal and the fluid pressure in the master brake cylinder. Characteristic line *Om* built for by equation (12.28). Nadlam characteristics in point *m* due to the fact that with an equal effort on the pedal P_n', pressure p_{and} in the cavity *AND* power cylinder ra 3 increased to atmospheric, as a result of which the force on the amplifier rod reached its limit value. When designing a vacuum brake amplifier, it is necessary to ensure such dimensions of the power cylinder (or chamber) of the vacuum amplifier, at which the termination of the operation of the amplifier would correspond to the maximum permissible effort on the brake pedal. For the considered case, the working area of the piston of the power cylinder or the active area of the camera diaphragm is determined from the expression

$$F_q = \frac{k_n \cdot P_n' \cdot i_n \cdot l_1}{(l_1 + l_2) \cdot p_z}, \text{ mm}^2 \tag{12.29}$$

where p_z – the maximum possible pressure difference in the chambers *AND* and *B* power cylinder (p_z = 0.05 MPa);

P_n' – the maximum allowable force on the brake pedal;

k_n – reserve ratio (k_n=1.0-1.2).

Working stroke of the piston of the power cylinder x_n is determined approximately by formula

$$x_n \approx \frac{l_1 + l_2}{i_n \cdot l_1} \cdot S_n, \qquad (12.30)$$

where S_n – maximum stroke of the brake pedal (S_n=150–170 mm).

For the vacuum amplifier shown in fig. 12.6, the fluid pressure in the brake system can be determined by considering the balance of the two-armed lever of the tracking device *1*

$$p_0 = p_1 \frac{F_1}{F_2} = \frac{P_n \cdot i_n \cdot F_1}{F_2 \cdot F_V}, \qquad (12.31)$$

where p_1 – fluid pressure created by the driver's muscle power in the first to the main brake cylinder;

F_1 and F_2 is the area of the pistons of the tracking device *1*;

F_V – area of the primary master brake cylinder.

The active area of the diaphragm of the vacuum amplifier *2*, which provides the necessary effort, can be determined from the expression

$$F_q = \frac{k_n \cdot p_{0\,max} \cdot F_0 \cdot \left(1 - \frac{F_2}{F_1}\right)}{p_Z}, \qquad (12.32)$$

where $p_{0\,max}$ – the maximum pressure in the wheel cylinders, created under amplifier at a point *t* (Fig. 12.7, *b*);

F_0 – secondary brake master cylinder piston area.

The maximum pressure in the wheel cylinders of the brake system is determined from the expression

$$p_{0\,max} = \frac{P_n' \cdot i_n}{F_V} \cdot \frac{F_1}{F_2}. \qquad (12.33)$$

The maximum stroke of the diaphragm rod of the power chamber s_0 can be approximately determined by the formula

$$s_0 = \frac{F_p s_n}{F_0' \cdot i_n}. \qquad (12.34)$$

For the vacuum amplifier shown in fig. 12.7, the amount of pressure in the brake line p_0 can be determined by considering the equilibrium dia-fragments F_a, according to the formula

$$p_0 = \left(\frac{F_n \cdot F_q}{F_a} + F_0'\right) \cdot \frac{P_n i_n}{F_V \cdot F_0}, \qquad (12.35)$$

where F_n – piston area of the tracking device;

F_v and F_0 – areas of the pistons of the primary and secondary master brake cylinders;

F_a and F_q – active areas of diaphragms, respectively, of the tracking device and power chamber;

F_0' is the area of the piston of the secondary power cylinder, excluding the area of the stem.

The size of the active area of the diaphragm of the vacuum amplifier, which provides the required pressure value p_0 on the entire section of the characteristics $0t$ (Fig. 12.7, *b*), determined by the formula

$$F_q = \frac{p_{0\,max} \cdot F_0 - p_{1\,max} \cdot F_0'}{p_Z}, \qquad (12.36)$$

where $p_{0\,max}$ and $p_{1\,max}$ – pressure in the primary and secondary main brake th cylinder at the maximum effort on the brake pedal.

The full stroke of the diaphragm of the power chamber of the vacuum amplifier is determined by the formula (12.34).

In fig. 12.8 and 12.9 show the schemes of coaxial hydrovacuum amplifiers, in which all the mechanisms of the amplifier are arranged in one block, which simplifies the design of the hydrovacuum amplifier, but slightly complicates its placement on the car.

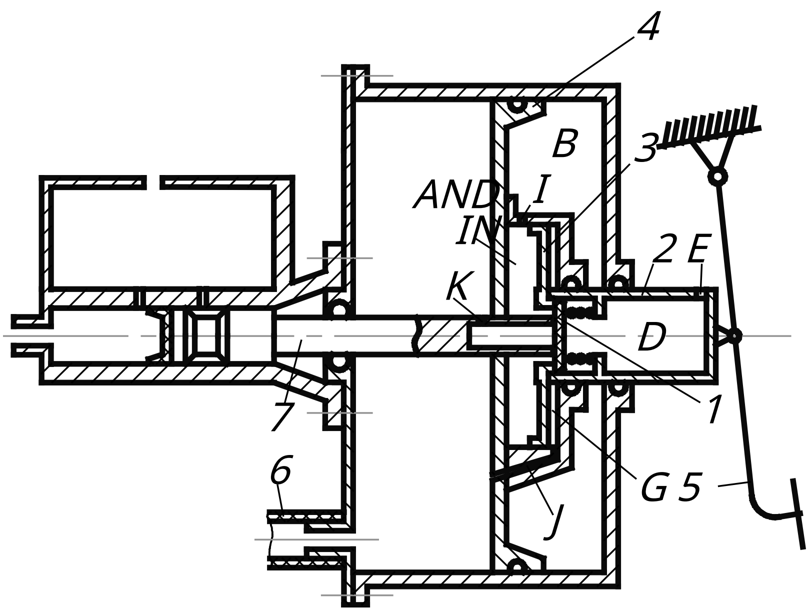
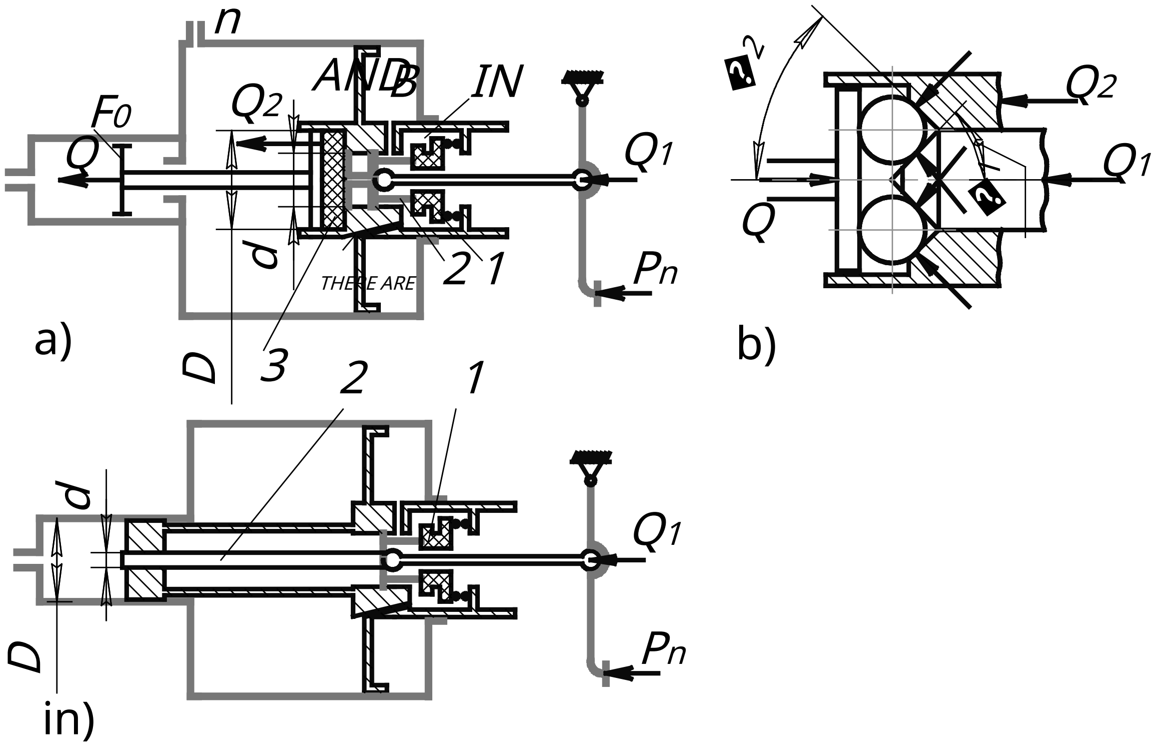


Figure 12.8 – Schematic diagram of coaxial hydrovacuum amplifier

Hydrovacuum amplifier, the scheme of which is shown in fig. 12.8, works like this. In the absence of braking, the valve *1* leans against the saddle piston *3,* that together with the stem *2* is in the far right married In this position, the cavity *AND* power cylinder (connected tube by airship *6* with the engine intake pipe) through the hole *K* connected with cavity *IN* tracking device and further through the hole *I* connected to the cavity *B* power cylinder. Cavity *G* tracking device is connected to the cavity *AND* through the hole *J*. Thus, in the uninhibited state in the cavities *AND*, *B*, *IN* and *G* rarefaction is created, and a cavity *D* through the hole *THERE ARE* connected to the atmosphere.



a) scheme of an amplifier with a reactive washer; b) scheme with a ball tracer device; c) scheme with a hydraulic tracking device

Figure 12.9 – Basic schemes of coaxial hydrovacuum amplifiers

When braking, when to the pedal *5* applied force P_n, stock *2*, re- moving to the left forces the valve *1* rest against the seat of the rod *7* and move away from piston seats *3*. In this position of the cavity *IN* and *B* will connect with poro- harvest *D*, which will lead to an increase in their pressure (close to atmospheric). Then the piston *4* of the power cylinder, moving to the left, will press through the stem *7* on the piston of the main brake cylinder.

With this scheme of the tracking device, the force on the rod *7* mainly- of the brake cylinder is determined by the formula

$$P_{sh} = P_n \cdot i_n \left(\frac{F_q}{F_a} + 1 \right), \qquad (12.37)$$

where F_q – piston area *4*;

F_a – piston area *3*.

The pressure in the brake line can be determined by the formula

$$p_0 = \frac{P_{sh}}{F_0}. \tag{12.38}$$

Piston stroke *4* is from the ratio $s_0 = s_n \cdot i_n$.

The vacuum amplifier, the scheme of which is shown in fig. 12.9, *and*, works in this way. In the absence of braking, the cavity *AND* (which connects through the hole *p* with the engine suction line) through the channel *THERE ARE* and cavity *IN* are connected to the cavity *B*. Thus, in the cavities *AND* and *B* rarefaction is created. When braking, the valve *1,* moving to the left, separates the cavities *B* and *IN*, while connecting the cavity *B* with atmosphere The amount of pressure in the cavity *B* will depend on the relative position of valve 1 and its seat *2.* And this mutual position, in turn, depends on the deformation of the jet washer *3*, on which three forces act: Q_1 – the force produced on the pedal stem, force Q_2, created by the vacuum piston of the amplifier, and power *Q* on the master brake cylinder piston.

Under the influence of these three forces, the jet puck *3* should be located in a certain deformed state, which ensures the proportionality of the forces on the brake pedal and on the piston of the vacuum amplifier. Given the fact that the specific pressures on the surface of the jet washer must be the same at each of its points, we will find the ratio of forces on the pedal rod and the piston of the master brake cylinder

$$Q = P_n \cdot i_n \frac{D^2}{d^2}. \tag{12.39}$$

Fluid pressure in the brake line is determined by the formula

$$p_0 = \frac{Q}{F_0} = \frac{P_n i_n D^2}{F_0 \, d^2}. \tag{12.40}$$

The required size of the piston area of the vacuum amplifier can be found using the formula

$$F_q = \frac{P_n \cdot i_n}{p_Z} \left(\frac{D^2}{d^2} - 1 \right). \tag{12.41}$$

The stroke of the vacuum amplifier piston is determined in the same way as in the previous case.

The jet washer of the vacuum amplifier shown in Fig. 12.9, *and*, can be replaced by a ball tracking device, the diagram of which is shown in Fig. 12.9, *b*. In the case of using such a device, the force on the piston of the master brake cylinder is determined by the formula

$$Q = P_n \cdot i_n \cdot \left(1 + \frac{tg\alpha_2}{tg\alpha_1} \right), \tag{12.42}$$

where are the corners a_1 and a_2 shown in fig. 12.9, *b*.

The fluid pressure in the brake line, the area and stroke of the vacuum booster piston are determined in the same way as in the previous case.

In fig. 12.9,*in*a hydrovacuum amplifier is shown, where the functions of the monitoring device, together with valve 1, are performed by the piston*2*, which has a diameter*d*.In this case, the fluid pressure in the brake master cylinder is determined by the formula

$$p_0 = \frac{4 P_n \cdot i_n}{\pi \cdot d^2}. \qquad (12.43)$$

The necessary dimensions of the piston area of the vacuum amplifier are determined by the formula

$$F_q = \frac{0.785 \cdot \left(D_2 - d_2\right) \cdot p_{0\,\max}}{p_Z}. \qquad (12.44)$$

where*D*and*d* –dimensions marked in fig.12.9,*in*.

The stroke of the vacuum amplifier piston is determined in the same way as in the previous case.

12.4.3 Design of a pneumatic brake actuator The pneumatic brake drive is used for braking heavy trucks, road trains and large-capacity buses. It provides high braking efficiency regardless of the weight of the vehicle. However, the pneumatic brake actuator has a relatively long operating time. Therefore, during its design, special constructive measures are needed to reduce the activation time and reduce the braking time. For this purpose, it is necessary to choose the optimal values of the cross-sections of pipelines and valves, to use accelerating valves and rapid release valves.

The pneumatic brake drive of road trains is performed according to two schemes - one-wire and two-wire.

A brake valve is used as a control body for the brakes of the road train, which supplies compressed air to the working bodies of the brakes of the front and rear wheels through no less than two separate lines.

All cars designed to work with semi-trailers and dumpers must be equipped with an additional brake valve (faucet, valve) with manual control of the pneumatic brake actuator.

The activation time of the brake actuator should be no more than 0.6 s. At the same time, the pressure should increase to 90%, which corresponds to full braking.

Permissible asynchronous braking of the tractor and trailer in terms of the activation time of the working bodies is from + 0.1 to - 0.2 s.

The air pressure in the pneumatic drive to the brakes of cars and tractors should be:

a) nominal (calculated) pressure 0.6 MPa;

b) the pressure due to the setting of the pressure regulator, from 0.67 to 0.73 MPa;

c) the maximum pressure due to the installation of the safety valve, 0.85 MPa.

The air pressure in the connecting line of the brake system of trailers, semi-trailers and trailers (braking pressure) must be maintained within 0.67–0.73 MPa. The beginning of an increase in air pressure in the pneumatic drive of brakes of trailers, semi-trailers and trailers must correspond to a decrease in air pressure in the connecting line by a value of not less than 0.06 MPa and not more than 0.08 MPa.

In the case of using compressed air for additional consumers (door mechanisms, pneumatic suspension, etc.), an additional receiver of sufficient capacity must be introduced into the system and a shut-off valve must be installed that lets compressed air into the additional receiver when the pressure is reached in the main receivers more than 0.60–0.63 MPa.

The total volume of the receivers should be such that when the pedal is fully pressed and the compressor is not working, the pressure drop in the receivers of the car and the tractor (relative to the maximum, limited by the regulator) should be no more than 0.03 MPa, and in the receivers of the trailer, semi-trailer and dissolution 0.03–0.05 MPa.

Tractors and cars must be equipped with a type coupling head*AND*, trailers, semi-trailers and trailers - with a connecting head type *B*. In cars and tractors, the connecting head must be located behind, in tractors - behind the cab.

A pneumatic drive can be used to actuate the spare brake, provided that the spare brake is fully independent from the pneumatic drive to the main brakes.

The driver's dashboard must have a control device that shows the pressure in the receiver and in the brake line of the car, as well as a signal (light or sound) that warns of a decrease in air pressure in the receivers below the permissible level.

The braking system of the car and the tractor must be equipped with regulators of braking forces that change the pressure of the air supplied to the brakes of the rear axles, depending on the change in weight falling on these axles.

The trailer must be equipped with a valve that allows you to change the pressure of the compressed air supplied to the brakes of the axles of the trailer depending on the degree of loading, as well as to perform complete brake release of the trailer.

The pneumatic brake drive must be hermetic, the reduction in air pressure when the compressor is not working is allowed no more than 0.03 MPa from the nominal during 30 minutes with the controls in a free position and during 15 minutes when braking (except for pressure reduction due to filling the brake line) .

In fig. 12.10 shows a diagram of a single-wire brake system with a single brake valve. The principle of its action is as follows. When pressing the pedal brake diaphragm brake valve *1* under the influence of force P_{Mr} moves down and the two-armed lever connected to it, closing the atmospheric valve with one end, opens the valve with the other, which connects the cavity located under the diaphragm with the air receiver. Compressed air (with pressure p_x) from this cavity enters the brake chambers of the tractor and additionally which brake valve *2*, which controls trailer braking. At the same time, the moving piston of the crane *2* moves to the left, connecting the cavity *B* with the atmosphere Pressure reduction p_y causes movement in this cavity up the moving piston of the valve *3*, resulting in pressure p_z in the cavity *IN*, connected to the brake chambers of the trailer, will begin to increase.

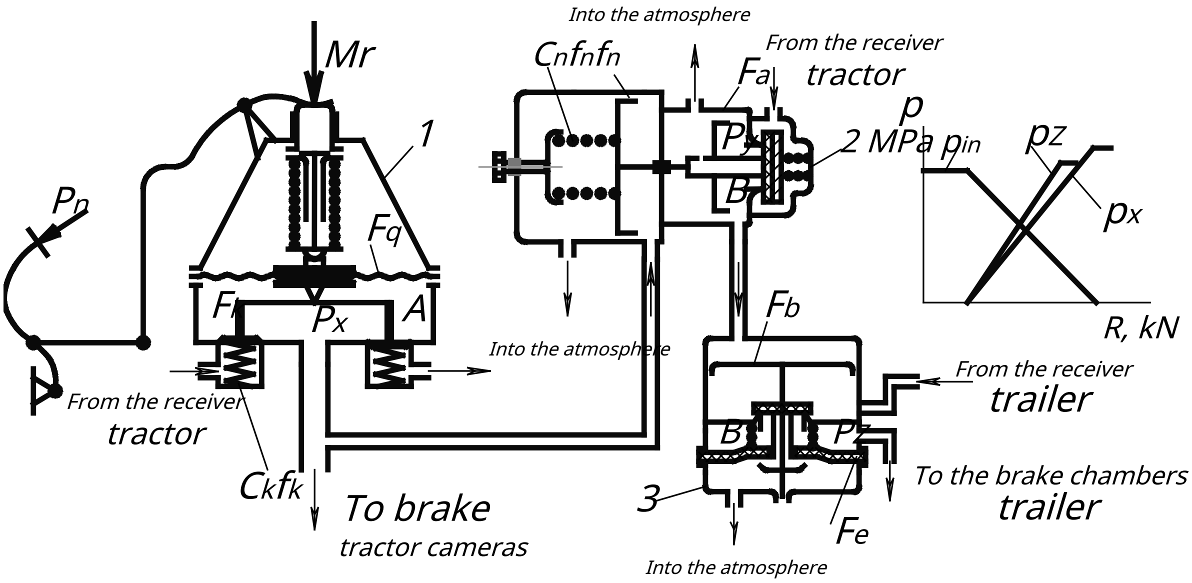


Figure 12.10 – Scheme of a pneumatic brake single-wire system

Air pressure in the cavity of the brake valve *AND*, connected to the brake chambers of the tractor, depends on the force applied to the pedal and is determined by the expression (when the arms of the two-armed lever are equal)

$$p_x = \frac{P_{Mr} - 2\left[C_k \cdot f_k + F_k\left(p_k - p_x\right)\right]}{F_q}, \tag{12.45}$$

where P_{Mr} – the force applied to the valve diaphragm is equal to the force on the pedal, as multiplied by the transmission ratio of the pedal and the crane lever;

C_k and f_k – accordingly, the stiffness and deformation of the intake valve spring Mr. Crane;

F_k and F_q – areas of the intake valve and diaphragm;

p_k – pressure in the tractor receiver.

Cavity pressure *B* trailer brake valve is determined by the form tallow

$$p_y = \frac{C_p \cdot f_p - F_p \cdot p_x}{F_a}, \qquad (12.46)$$

where C_n and f_n – respectively, stiffness and deformation of the crane spring;

F_n and F_a - respectively, the areas of the left and right pistons. Maximum air pressure in the cavity B to be established at p_x=0 (unbraked state of the crane). With a very small error, it can be assumed that the same pressure will be in the trailer receiver connected to the cavity B through the air distribution valve.

Air pressure in the cavity of the air distribution valve IN, connected to the brake chambers of the trailer, is determined by the formula

$$p_z = p_x \frac{F_p}{F_a} \frac{F_b}{F_{there\ are}}, \qquad (12.47)$$

where F_b and $F_{there\ are}$ – respectively, the area of the piston and the active area of the diaphragm increase three-distribution crane of the trailer.

A sample graph of pressure changes p_x, p_y and p_z shown in fig.12.10.

Simultaneous and proportional braking of the tractor and trailer must also be ensured by the appropriate calculation of brake chambers (or cylinders) and brake mechanisms.

The given static calculation allows to determine the main parameters of the brake drive only in a first approximation. More complete and reliable data can be provided by a dynamic calculation, the basics of which are outlined in [2].

In fig. 12.11 shows the combined brake valve of the single-wire system. When the brake pedal is pressed, the upper end of the tap lever moves to the left. The piston of the lower tracking mechanism, while moving to the right, disconnects the cavity AND from the atmosphere and, opening the valve, connects it to the receiver. From the cavity AND compressed air enters the brake chambers (or cylinders) of the tractor.

The rod of the upper tracking mechanism, moving to the left, compresses the spring 2, which allows the piston to move to the left and connect the cavity B with the atmosphere, as a result of which the pressure in it begins to decrease. In the state of deceleration, the cavity B connected to the receiver. Compressed air from the cavity B enters the air distribution valve of the trailer (valve 3 in fig.12.10).

Air pressure in the cavity AND (Fig. 12.11) is determined by the formula

$$p_x = \frac{P_1 - C_1 f_1}{F_1} = \frac{P\frac{a}{b} - C_1 f_1}{F_1}, \qquad (12.48)$$

where F_1 – piston area;

R – the force applied to the crane lever;

$C_1 f_1$ –the force created by the spring *1*; this force can change regularly saliva ring *3*, while the ratio between the pressures will change p_x and p_z.

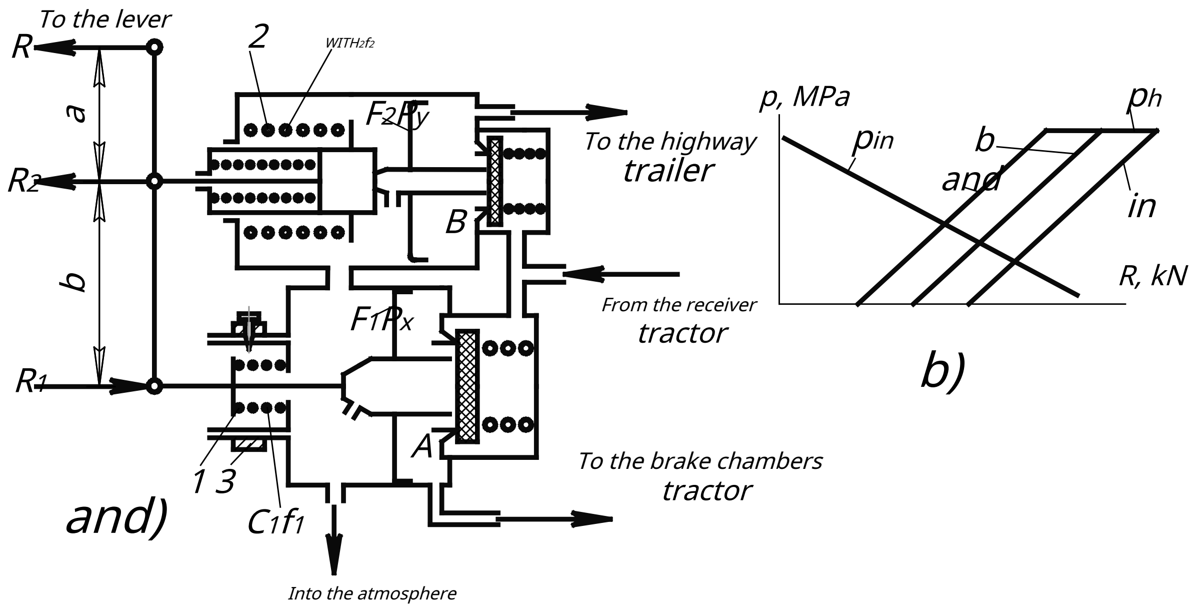


Figure 12.11 – Diagram of a double pneumatic brake valve

Cavity pressure *B* is determined by the formula

$$p_y = \frac{C_1 f_1 - P_2}{F_2} = \frac{C_2 f_2 - P\dfrac{a+b}{b}}{F_2}, \qquad (12.49)$$

where $C_2 f_2$ –the effort with which it presses on the rod of the tracking mechanism spring *2*;

F_2 –the area of the piston of the tracking mechanism.

Maximum pressure p_y will be in the unbraked condition of the crane, when the force applied to the crane lever is zero.

Pressure change diagram p_y and p_x depending on the strength *R* and spring compression *1* shown in fig.12.11, *b*. Lines *a, b* and *in* depict the nature of the pressure change p_x with different spring compression *1*. line *and* -at minimum compression of the pru-wives *1* (the trailer is unloaded); line *in* –at maximum compression of the wives *1* (the trailer is loaded).

In fig. 12.12 shows a diagram of a hydropneumatic drive, the main advantage of which is to reduce the brake activation time. When pressing the pedal, force P_n transmitted to the rods of the tracking mechanism mu and the piston of the power cylinder in a relationship that is inversely proportional to the arms of the lever *2*.

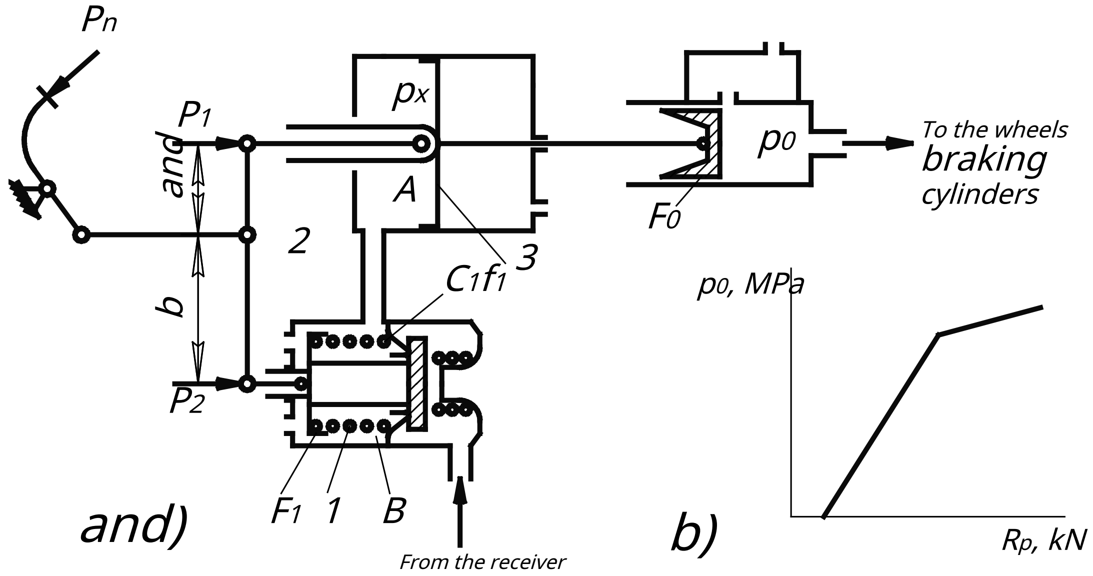


Figure 12.12 – Schematic diagram of hydropneumatic brake system

Cavity pressure *B* of the valve of the tracking mechanism can be determined from the expression

$$p_x = \frac{P_2 - C_1 f_1}{F_1} = \frac{P_p \cdot i_p \dfrac{a}{a+b} - C_1 f_1}{F_1}, \tag{12.50}$$

where $C_1 f_1$ is the compression force of the spring *1*;

F_1 – piston area of the tracking mechanism;

i_n – gear ratio of the brake pedal;

and and *b* – lever arms *2*.

The pressure in the master brake cylinder is determined by the formula

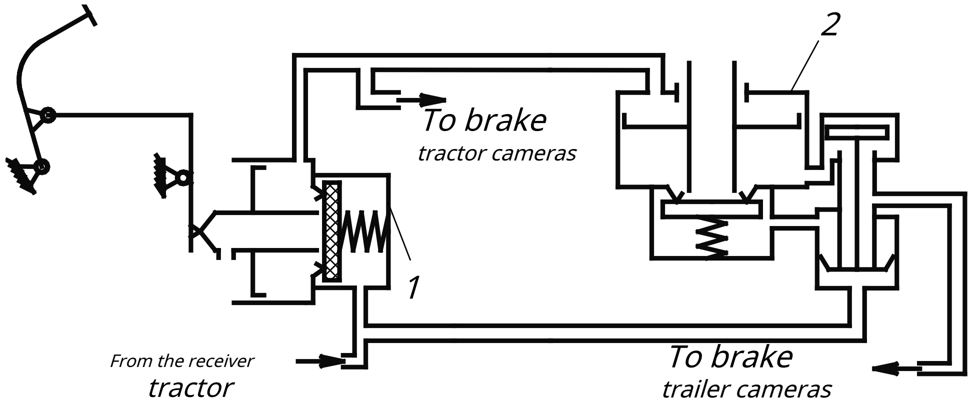
$$p_0 = \frac{p_x \cdot F_2 + P_p \dfrac{b \cdot i_p}{a+b}}{F_0}, \tag{12.51}$$

where F_0 and F_2 – areas of the pistons of the main brake and power cylinder ditch.

If the pneumatic brake system is faulty, braking is possible only due to the muscular strength of the driver. In this case, the fluid pressure can be determined by the formula (12.51), assuming that $p_x = 0$. The change diagram no pressure p_0 depending on the strength P_p shown in fig.12.12, *b*.

In fig. 12.13 shows a diagram of a two-wire pneumatic brake system, which has operational advantages over a single-wire system.

1. On protracted descents with long-term braking in single-track the brake system of the trailer does not have time to refill. In a two-wire system, this can be avoided, because the trailer receiver is constantly replenished during braking.



1– brake valve;*2*– accelerator emergency valve

Figure 12.13 – Schematic diagram of pneumatic two-wire brake system

2. With a single-wire system, in the case when the air flow in the car mobiles (tractors) is greater than in the trailer, there is a danger that when braking, the control line will not receive the necessary filling and the brake valve of the trailer will not move to the braking position.

In a two-wire system, reliable debraking is ensured by the fact that the debraking position corresponds to increased pressure in the brake valve.

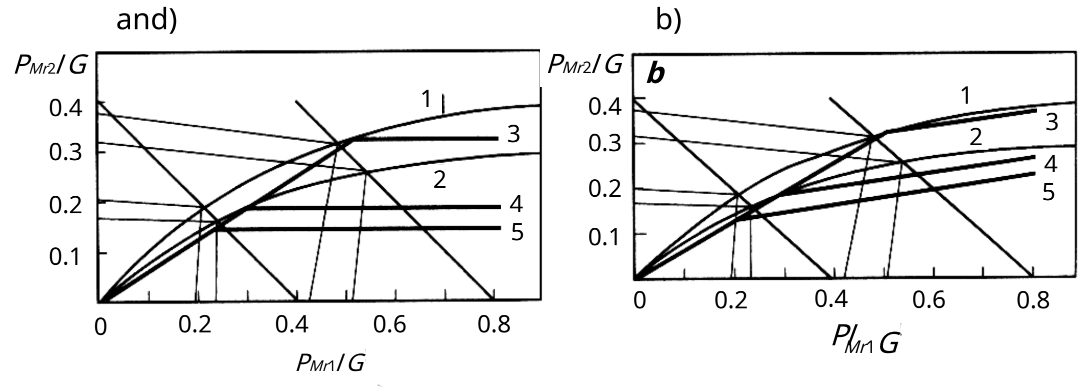
3. Constancy of pressure in the trailer receiver with a two-wire safety system hears a better coordination of braking of the car (tractor) with the trailer.

12.5 Regulator of braking forces

The regulator (controller, tuner) of braking forces is a valve - a control element with an open connection. The valves are static (brake force limiters) and dynamic (proportional, operating according to such parameters as pressure in the brake actuator, axial load, and the amount of deceleration).

The regulator of braking forces measures the ratio of braking forces between the front and rear axles to approach the ideal parabolic curve (Fig. 12.14). Lines of constant braking force have the form of straight lines with a negative slope. The diagram in fig. 12.14,*and*illustrates the effect of static, and the diagram in fig. 12.14,*b*– dynamic regulator.

Pressure-sensitive valves allow you to achieve braking force ratios close to ideal for empty cars. For a loaded car (upper parabola), the distribution is far from ideal as soon as the valve starts to act, that is, the percentage of the final braking force falling on the rear axle decreases, and on the front axle it increases.



P_{Mr2}– braking force of the rear axle; P_{Mr1}– braking force of the front axle;

G– car weight; *1*– an ideal curve for a loaded one; *2*– for empty; *3*– loaded with a load-activated valve; *4*– for an empty z a valve triggered by load; *5*– for loaded with a valve that activated by pressure

Figure 12.14 – Brake force distribution diagram with regulator braking forces: a – static; b - dynamic

When using a load-sensing valve, the actuation point is shifted in the upper direction, bringing the brake force distribution closer to ideal at all operating loads.

The deceleration-sensitive valve is activated at a certain amount of deceleration of the vehicle, regardless of the load.

The valve should be calculated so that its braking force distribution curve is no higher than the ideal distribution curve. The conditions for changing the coefficient of friction of the linings and engine torque, as well as the tolerances of the valve itself, must be taken into account to prevent overbraking of the rear axle. In practice, this means that the curve of the actually established distribution should be much lower than the ideal curve.

Criteria according to which brake force regulators are developed:

- compatibility with the systemABS;

− additional costs due to the separation of the brake line on the rear motor stu (for example, a scheme of type X);

− provision of a bypass or bypass function in the event of a malfunction of the lm main line, especially for static regulators;

- means of testing during manufacture and operation. Vehicles with always evenly loaded axles do not need to be equipped with a brake force regulator, as the disadvantages in determining its malfunction outweigh its minimal advantages.

Since the maximum braking force is limited by road adhesion ϕ, then for the case of simultaneous braking of all wheels to the front, when $P_{Mr}=G_a \cdot \phi$, we obtain for the front and rear axles:

$$R_{Mr1\max} = \frac{\phi \cdot G_a}{L}(b + h\cdot\phi) \text{ and } P_{Mr2\max} = \frac{\phi\cdot G_a}{L}(a - h\cdot\phi), \tag{12.52}$$

where L – vehicle base;

a, b, h – the distance of the center of gravity to the front axle, to the rear axle, to the horns

In this case, we will obtain the optimal value of the coefficient of distribution of braking forces

$$\beta_{gO} = \frac{b + h\phi_0}{a - h\phi_0}. \tag{12.53}$$

VAZ and AZLK regulators (Fig. 12.15, *and*) have the following construction. Body with a piston *1* mounted on the body. On the piston from the adjusting one screw *3* there is a force F_{Mr} from torsion *5*, located perpendicular to the plane of the picture. The other end of the torsion beam is connected by rods *6* and *7* with beam behind of the bridge A force also acts on the piston F_n from the spring 8.

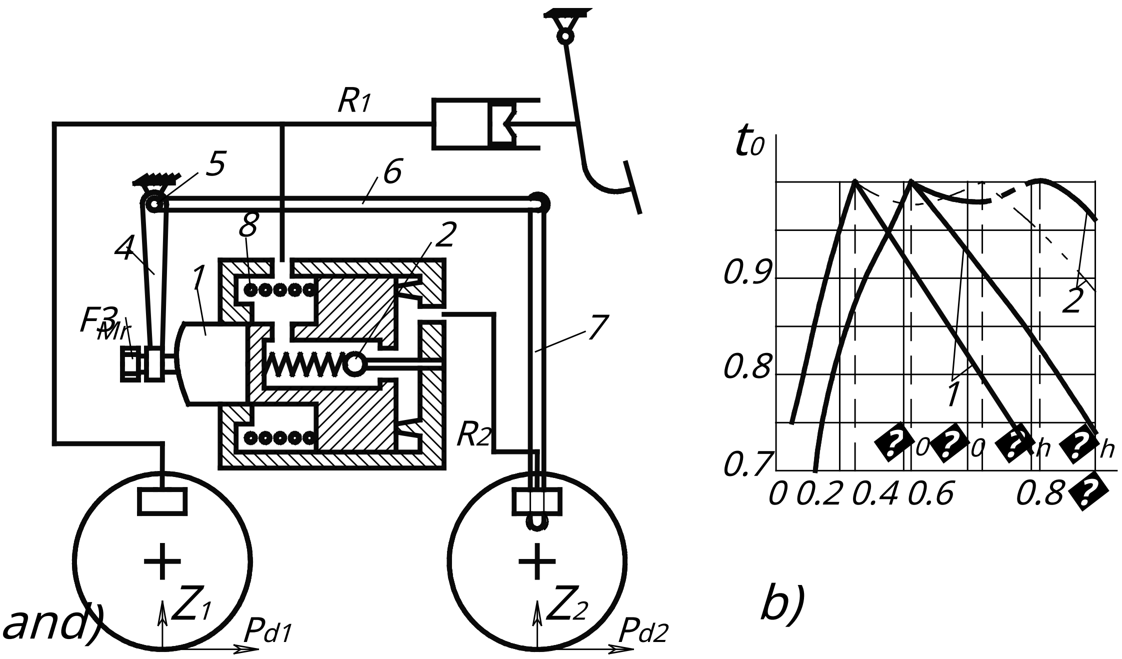


Figure 12.15 – VAZ and AZLK braking force regulator and characteristics its effectiveness

Pressure regulation system p_2 depending on the pressure p_1 and reactions Z_2 is by influence Z_2, expressed through deflection h suspension, closed. She composed is given by three elements: car–torsion–regulator. Equation of the regulator

$$\begin{cases} p_2 = p_1 & \text{at} \quad p_1 < p_0 \\ p_2 = \left[\frac{1}{f_2}\left(F_{Mr} + F_n + (f_2 - f_1)p_1\right)\right] & \text{at} \quad p_1 > p_0 \end{cases} \tag{12.54}$$

Having applied the coupling weight utilization factor to evaluate the effectiveness of the regulator m_G, which shows how many times the slowing down of the of a car with this brake system is less than with the optimal distribution of braking forces, when the entire weight of the car is used, we obtain

$$m_G = \frac{C_1 \cdot p_1 + C_2 \cdot (p_2 - p_2')}{G_a \cdot \phi}, \qquad (12.55)$$

where C_1 and C_2 – constructive constants;

p_2' is the pressure proportional to the force of the return spring.

In fig. 12.15, *b* this dependence with the regulator is presented (curves *2*) and without it (curves *1*) at 100% – bold lines and 20% load – thin lines. From fig. 12.15, *b* it follows that the regulator significantly increases the coefficient m_G, approaching reducing it to one, but only in the zone $\phi > \phi_0$.

12.6 Anti-lock systems

The first ABS began to be used in aviation in 1949, the first samples of ABS appeared on cars in 1969. As a result of the continuous increase in requirements, the improvement of ABS has led to its worldwide recognition as the most radical means of increasing active safety, especially on wet and slippery roads, where the largest number of road accidents occur.

Anti-locking systems of cars are systems equipped with control devices with feedback that prevent the wheels from locking during braking and preserve the controllability and directional stability of the car. The main components of ABS are: hydraulic modulator; wheel speed sensors; electronic control unit (ECU).

When developing the ABS system, the following are taken into account:

– options for coupling between the tire and the road;

- unevenness of the road surface, which causes vibrations of the wheels and axles;

– braking hysteresis;

– pressure changes in the main brake cylinder under the influence of the driver on brake pedal;

– wheel radius changes, for example, when installing a spare wheel.

Management quality criteria:

– support of directional stability while driving the car through to ensure a sufficient amount of transverse traction force on the rear wheels;

- maintaining vehicle controllability by providing sufficient lateral traction force on the front wheels;

– reduction of the braking distance in comparison with braking with locking forged wheels;

– quick change of braking torques for different clutch ratios nya, for example, when the car moves through small areas of ice on the road surface;

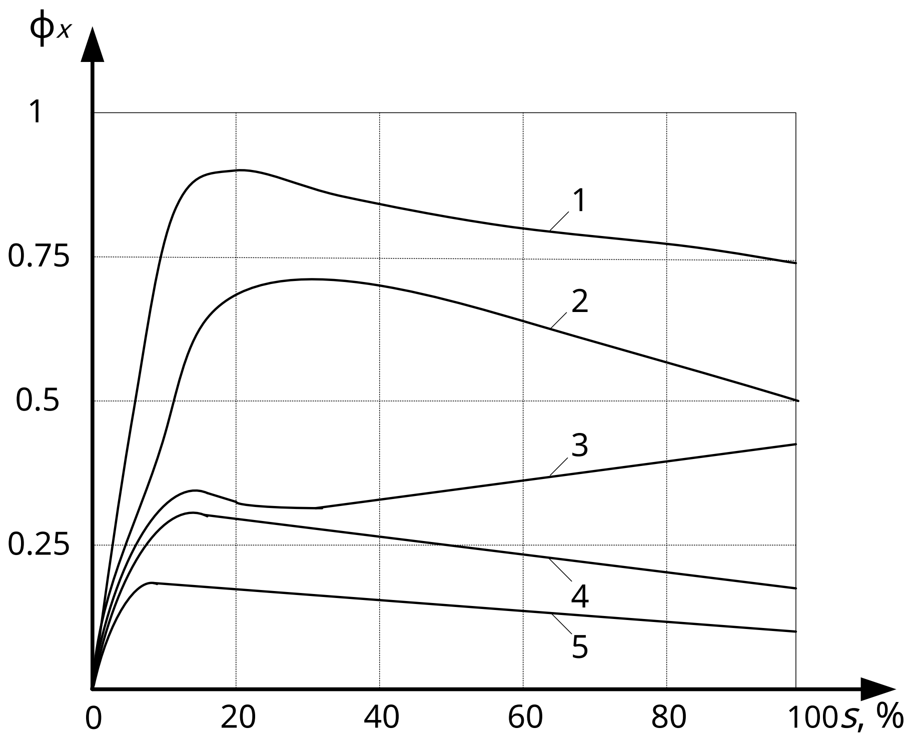
– control of low amplitudes of changes in braking torque in order to transmission of vibrations in gears;

– a high level of driving comfort as a result of minor environmental impact oral communication on the brake pedal and the use of silent executive mechanisms.

12.6.1 Functioning of anti-lock systems

It is known that in the calculations of the braking dynamics of the car, in most cases, table values of the coupling coefficients are used, which are determined experimentally when the wheel is locked, that is, when the wheel slips relative to the road surface at 100%. However, it is known that the coefficient of adhesion of an elastic wheel depends not only on the state of the road surface, but also on the degree of slippage of the wheel relative to this surface during braking and on other factors.

In fig. 12.16 shows the graph of the dependences of the wheel adhesion coefficients on the support surface ϕ_x from relative sliding s at gal- driving on different roads.



1 – dry asphalt; 2 – wet asphalt; 3 – unrolled snow; 4 – frozen snow; 5 - ice

Figure 12.16 - Dependencies of coupling coefficients on the coefficient slip ratio

According to [12], the relative slip is determined by the formula

$$s = \frac{v_a - \omega_k r_0}{v_a}, \qquad (12.56)$$

where v_a – vehicle speed;

ω_k – angular speed of the wheel during braking;

r_0 – free radius of the wheel.

It can be seen from the graph that at a certain value of the relative slip, the longitudinal coefficient of adhesion ϕ_x has a maximum The magnitude of the relative slip corresponding to the maximum is called critical s_{Cr}. For most road surfaces s_{Cr}=0.1...0.3. From the schedule you can also make you conclude that when bringing the wheels to the curb in the process of braking (s=1) significantly decreases ϕ_x, and therefore braking efficiency, stability and steering stability of the car when braking.

The main task of ABS is to maintain relative slip within narrow limits in the vicinity during the braking process s_{Cr}. In this case, provide optimal braking characteristics are baked. To do this, it is necessary to automatically adjust the braking torque applied to the wheels during braking.

Regardless of the design, any ABS must consist of the following elements:

− sensors, the function of which is to issue information, depending on the of the control system in question, about the angular speed of the wheel, the pressure of the working body of the brake actuator, the deceleration of the car, etc.;

- a control unit, usually electronic, where information is received from the sensors, which, after logical processing of the received information, gives a command to the executive mechanisms;

− executive mechanisms (pressure modulators), which, depending on the Commands from the command control unit reduce, increase or maintain the pressure in the brake drive of the wheels at a constant level.

The process of adjusting wheel braking using ABS is cyclical. This is due to the inertia of the wheel itself, the drive, as well as the ABS elements. The quality of adjustment is evaluated by the extent to which ABS ensures the braked wheel slips within the specified limits. With a large range of cyclical pressure fluctuations, comfort during braking (jerking) is disturbed, and car components perceive additional loads. The quality of the ABS work depends on the adopted regulation principle (functioning algorithm), as well as on the speed of the system as a whole. The speed determines the cyclic frequency of the brake torque change. An important property of ABS should be the ability to adapt to changes in braking conditions (adaptability) and, first of all, to changes in the clutch coefficient during braking.

A large number of principles (algorithms of operation) have been developed, according to which ABS work. They differ in complexity, cost of implementation, and the degree of satisfaction of the requirements.

Let's consider the process of ABS operation with the algorithm of functioning when the braked wheel is decelerating.

The equation of motion of the braked wheel has the form:

$$J_k \varepsilon_{hk} = M_{Mr} - M_\phi, \tag{12.57}$$

where J_k – moment of inertia of the wheel;

ε_{hk} – angular deceleration of the wheel;

M_{Mr} – the moment created by the braking mechanism;

M_ϕ is the moment possible when the wheel engages with the support surface.

Using equation (12.57), it is possible to plot the ABS operation process by deceleration (Fig. 12.17). The figure shows the following dependencies: the dependence of the moment on the braked wheel, realized by the clutch, on the relative slip $M_\phi = f(s)$; moment dependence, which is created by the brake mechanism on the braked wheel, from the relative slip during the automatic adjustment process $M_{Mr} = f(s)$.

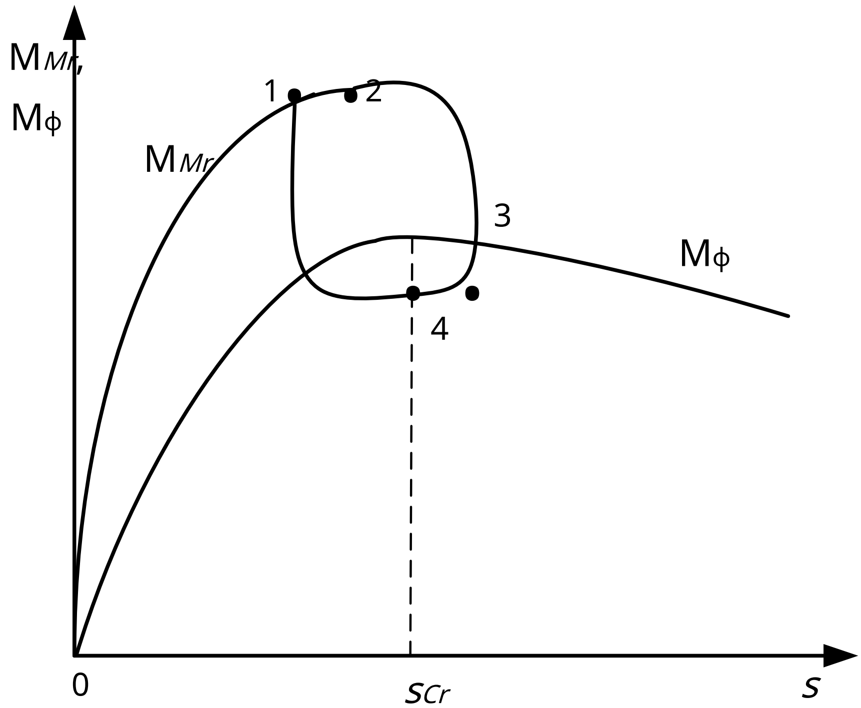


Figure 12.17 – The diagram of the ABS work process by deceleration

Pressing the brake pedal causes an increase in braking torque (section *0 - 1 - 2*) over the entire area, which causes the wheel to slow down with an increase in relative slippage. Especially fast slowing of growth stands on the site *1 - 2*, where the difference $M_{Mr} - M_{\phi}$ suddenly increases as a result decrease M_{ϕ}, and the deceleration is directly proportional to this difference

$$\varepsilon_{hk} = \left(M_{Mr} - M_{\phi} \right) / J_k. \qquad (12.58)$$

A sharp increase in the deceleration indicates that the relative slippage has become somewhat greater s_{Cr}. This becomes the basis for submitting the control unit at a point *2* commands of the modulator to reduce the pressure in the brake water Point *2* corresponds to the first "insert" command. According to the given command braking torque M_{Mr} decreases at the point as well *3* becomes equal to the torque on the clutch Lena: $M_{Mr} = M_{\phi}$, and slowing down $\varepsilon_{hk} = 0$. Zero deceleration value serves as the second "insert", according to which the control unit commands the modulator to maintain constant pressure in the brake actuator, and, therefore, a constant braking torque M_{Mr}. In this phase $M_{\phi} > M_{Mr}$ and $\varepsilon_{hk} = \left(M_{\phi} - M_{Mr} \right) / J_k$, i.e ε_{hk} changes sign and the wheel begins to accelerate. The maximum value of the acceleration corresponds to the maximum difference $M_{\phi} - M_{Mr}$, which has mis-it's on point *4*, which is the third "insert". On point *4* the control unit gives co-mandu modulator to increase the pressure in the brake actuator, and the described cycle is repeated, allowing to maintain the relative slip in the interval, which ensures high values of the longitudinal and transverse coupling coefficients.

The ABS operation process can be implemented in a two- or three-phase cycle. With a two-phase cycle: the first phase of pressure growth; the second phase is pressure relief. With a three-phase cycle: the first phase is an increase in pressure; the second - pressure relief; the third phase – maintaining the pressure at a constant level.

12.6.2 Main design features of ABS

Let's consider some variants of ABS systems (Table 12.4), which are the most common [5].

Four-channel system (options 1, 2).

Allows separate pressure control in two-circuit systems with a bridge connection (diagram ||) and with a diagonal connection (diagram x). At when braking on a road surface with different left and right "mixed" traction coefficients, measures should be taken to ensure the absence of a moment relative to the vertical axis, which could lead to the loss of the car's directional stability.

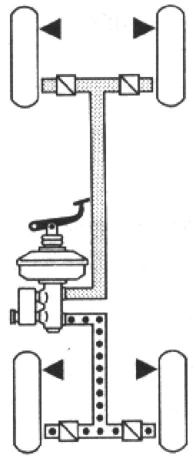
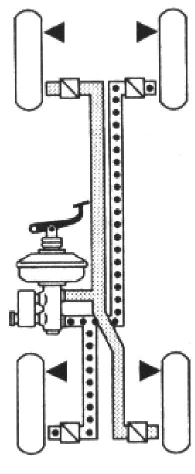
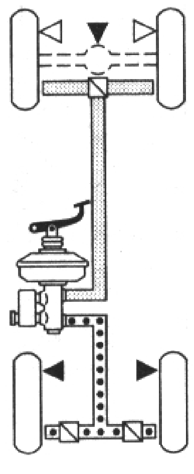
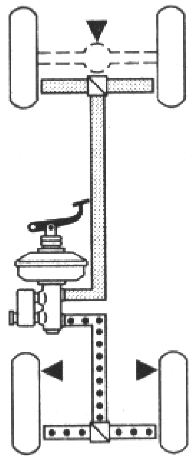
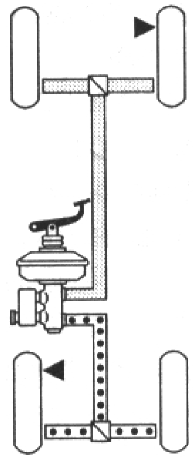
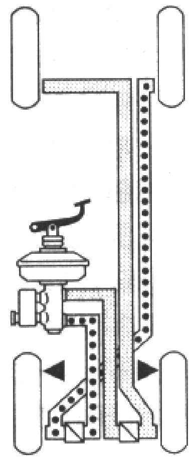
Three-channel system (version 3).

The turning moment during braking on mixed road surfaces is reduced so that passenger cars with a long wheelbase and a large moment of inertia relative to the vertical axis do not lose directional stability and controllability.

Two-channel system(options4, 5, 6).

These systems, on the one hand, have fewer components than three-channel and four-channel systems, which makes them less expensive. On the other hand, there are some functional limitations. In option 4, with high-threshold adjustment, the front wheel with a higher clutch ratio determines the pressure applied to both front wheels. In this case, during emergency braking, blocking of one of the front wheels occurs. This is accompanied by an increase in tire wear and deterioration of controllability. When using option 5, this happens when the controlled wheel of the front axle has a higher coefficient of adhesion than the uncontrolled one. In option 6, the pressure supplied to the front wheels is regulated separately, and on each rear wheel - together with the front ones. Due to the need to redistribute the braking force from the rear axle to the front, in order to prevent the rear wheels from locking, this system provides lower levels of deceleration than three- or four-channel.

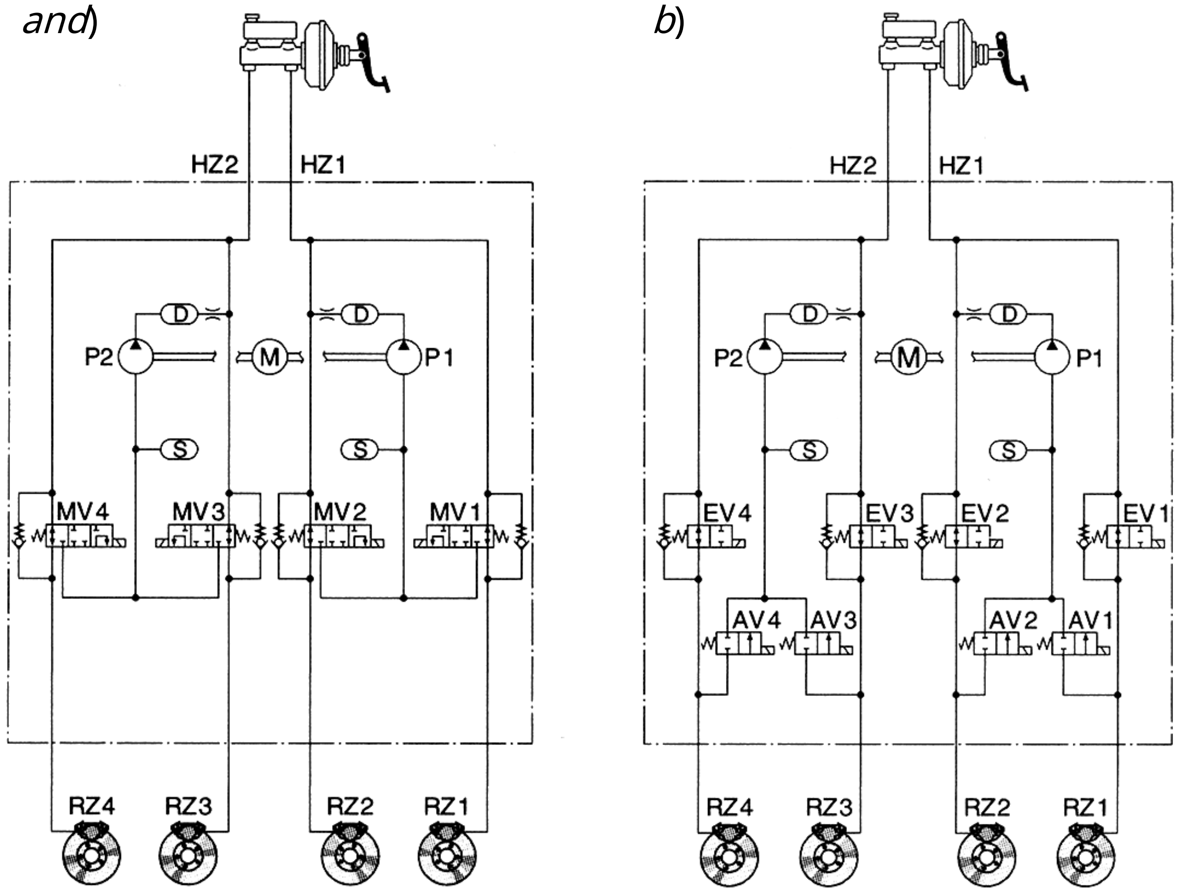
Table 12.4 – Options for ABS systems

Four-channel with 4 sensors		Three-channel with 3 sensors	Two-channel		
			with 3 sensors	with 2 sensors	
front and back	diagonally	front and back	front and back	front and back	diagonally
option 1	option 2	option 3	option 4	option 5	option 6
					

► – sensor;
▷ – a sensor alternative to the differential sensor;
⊠ – control channel

The main components of the Bosch ABS (Fig. 12.18).

1.*Inductive wheel speed sensor*provides the electronic control unit (ECU) with the necessary information about the speed of rotation of the wheel.



and) – ABS 2S scheme; *b*) – ABS 5S scheme; HZ – master brake cylinder; M – electric motor; P - piston pump; D – damper; S – battery; MV - solenoid valve; EV – intake valve; AV – outlet valve; RZ is a wheel brake cylinder

Figure 12.18 – Diagram of a hydraulic modulator

2.*Control unitECU with LSI*(large integrated circuit) The ECU receives, filters and amplifies signals from the wheel speed sensor before using them to determine wheel slip and acceleration.

3.*Input unit*

It consists of a low-pass filter and an input amplifier. 4. *Digital controller*

It consists of two identical mutually independent LSI digital integrated circuits. These blocks work in parallel, processing the information coming from two wheels (channels 1+2 and 3+4), carry out logical calculations. The controller logic converts the control signals into position commands for the solenoid valves. The serial interface, connected to the input stage of the logic device and the controller logic via a data link, supports communication and data transfer between two digital LSIs.

Another functional block contains a control circuit to ensure error recognition and analysis. As soon as a malfunction appears in the ECU, a warning light informs the driver that the ABS system is not functioning. However, the braking system remains fully functional even when the ABS system is turned off.

5.*Output blocks*

The two output units function like current regulators for channels 1+2 and 3+4 when receiving position commands from the LSI used to control the solenoids.

6.*Output cascade*

Uses the data from the current regulators of the two output units to excite the current of the solenoid valves.

7.*Voltage stabilizer*

The function of this block includes voltage stabilization within the tolerance required for reliable ECU operation. The unit also reacts to insufficient on-board voltage by turning off the device, controls the operation of the relay and the circuit of the signal lamp.

8.*Control unit with microprocessors*

In this ECU unit, instead of LSI, two microprocessors are used, which perform signal processing, controller program execution and ABS auto-control function. The unit also performs diagnostics according to ISO standards, which makes it possible to track faulty ABS components using a signal lamp or a measuring device.

9.*Hydraulic modulator for systemsABS 2S and ABS 5S with brake drive according to the schemes*|| andx

For each brake circuit, the hydraulic modulator includes (see Fig. 12.18): pump P with an electric motor; battery S; damper D; solenoid valves.

*Pump*R. Returns the brake fluid from the wheel brake cylinders to the master brake cylinder.

*Battery*S. Provides temporary accumulation of a large amount of brake fluid under low pressure.

*Damper*D. Throttle dampers serve to smooth out the high levels of pulsation that occur when brake fluid returns to the brake master cylinder. They ensure that the noise level is kept to a minimum.

10.*Solenoid valves forABS2*(MV)

A solenoid valve is connected to each wheel brake mechanism. The valve serves to modulate the pressure in the wheel cylinders during active ABS control. Modulation is carried out according to three modes: increase, maintenance and decrease of pressure.

11.*Solenoid valves forABS5*(EV, AV)

An EV and AV valve is connected to each wheel brake mechanism. The same modes of pressure modulation as indicated above are carried out by controlling these valves.

The "Bosch" system of the 2S model works in the mode of the so-called three-phase cycle according to the principle of reverse injection. During normal anti-lock braking, the solenoid valve connects the wheel brake cylinder to the master cylinder, and the brake system works as if there was no anti-lock system. When blocking, the electronic control unit sends a signal in the form of an electric current to the electrovalve solenoid. The core begins to move longitudinally and closes the main line with the end valve, simultaneously opening the way for the drain brake fluid from the brake system to the special damping chamber. (Later, the pump will drive it back into the reservoir of the master brake cylinder). This leads to a decrease in pressure in the main line. Then there is a waiting phase, when the electrovalve disconnects all the lines of the system before the pressure increase phase, which occurs when the electrovalve returns to its initial position, closes the drain hole, and the brake fluid pressure increases again. The advantage of the considered ABS is the possibility of using three-position electric valves instead of the six two-position ones that were used earlier. The next step in the development of ABS was the unification of the master brake cylinder together with the amplifier and modulator into a single unit, that is, the creation of an integrated anti-lock system, the assembly of which on the conveyor became much easier. An example of such ABS can be the system described below. Nowadays, the development of ABS goes in two diametrically opposite directions. For high-class cars, the most effective integrated four-channel ABS is created, and for mass cheap models, simplified options are being developed that are built into serial brake systems as additional equipment. ABS has changed the perception of the level of traffic safety. Today, this system is included in the list of additional equipment of almost every new model. In all developed countries, the majority of motorists are convinced: you cannot save money on ABS. The mandatory use of ABS is not far off. To date, international and national requirements (in particular, Directive 71/320 EEC and Annex 13 to UNECE Regulation 13) provide for the mandatory presence of ABS only on trucks with a total weight of more than 16 tons, trailers and semi-trailers with a gross weight of more than 10 t and international buses with a gross weight of more than 12 t, as the consequences of accidents of these vehicles can be the most tragic.

12.7 Parking brake system

Brake mechanisms equipped with spring-type energy accumulators, which are the usual equipment of trucks with a gross weight of more than 7.5 tons, are convenient for use in the parking brake system.

In the off position, the four-circuit safety valve and the parking brake valve connect the receivers of the working brake system with the chamber of the energy accumulator for spring compression. In trailers of road trains on a spare cylinder is located on this line of the brake system. When the parking brake tap is actuated, the pressure under the accumulator piston decreases. At first, partial braking occurs, later, when the pressure decreases to atmospheric, complete braking (triggering of the backup system) takes place. Parking mode is set. In road trains with semi-trailer tractors, it is possible to brake only the tractor, but not the entire road train. In such a situation, EU/EEC rules and regulations provide for a guaranteed air supply under extreme conditions, nine-time switching on and off by using stored energy, the presence of a spring-operated brake activation control device, and an auxiliary switch-off device.

12.8 Brake systems with retarder (auxiliary brake systems topics)

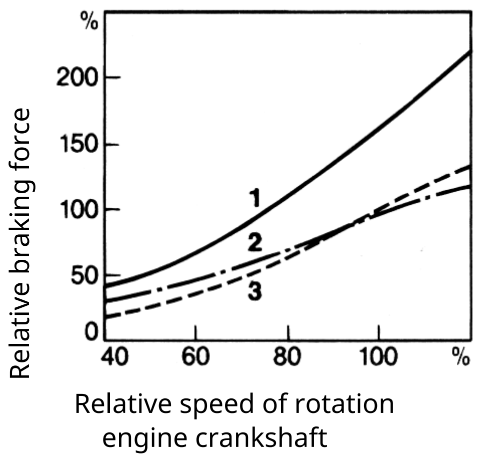
Wheel brakes used in cars and trucks are not designed to provide long-term braking (for example, when the car is moving downhill), because they can overheat, which causes a decrease in the braking effect. In emergency cases, this can lead to a complete failure of the braking system. Therefore, cars with a large gross weight began to be more often equipped with non-wearing auxiliary braking systems in addition to the conventional wheel brakes. Such a system must be independent of the wheel brakes, and must also take into account speed limits. This reduces the wear of wheel brakes, and also increases comfort (easy riding) when braking.

12.8.1 Exhaust (engine) braking system

The power spent on braking includes the power spent on engine rotation with gas flow throttling in the exhaust tract. The maximum braking power of modern standard engines is 5-7 kW/l, depending on the working volume of the engine. Engines with output limiters achieve braking power of the order of 14-20 kW/l. An increase in the maximum braking force is possible only with the help of additional modifications. Decompression reduction of output (for example, "C-brake", "Jake-brake", "Dynatard" or "Powertard"), as well as a brake based on constant throttling of exhaust gases can significantly increase the braking force (Fig. 12.19).

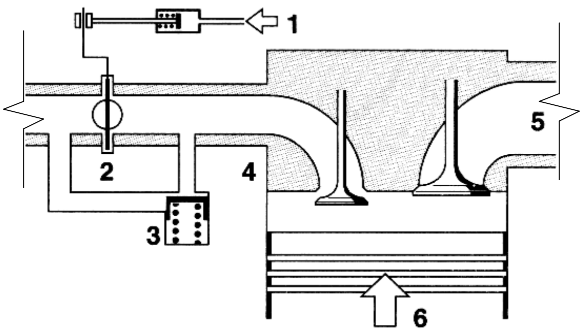
Motor brake with exhaust valve

The brake with exhaust valve is still the most common nowadays. In this system, the driver can close the exhaust duct with a damper using a pneumatic actuator. As a result, back pressure will form in the exhaust gas exhaust system, which must be overcome by each piston during the exhaust stroke (Fig. 12.20).



1 – exhaust valve and constant throttle; 2 – constant throttle; 3 – exhaust valve

Figure 12.19 – Curves of braking forces (V8 engine)



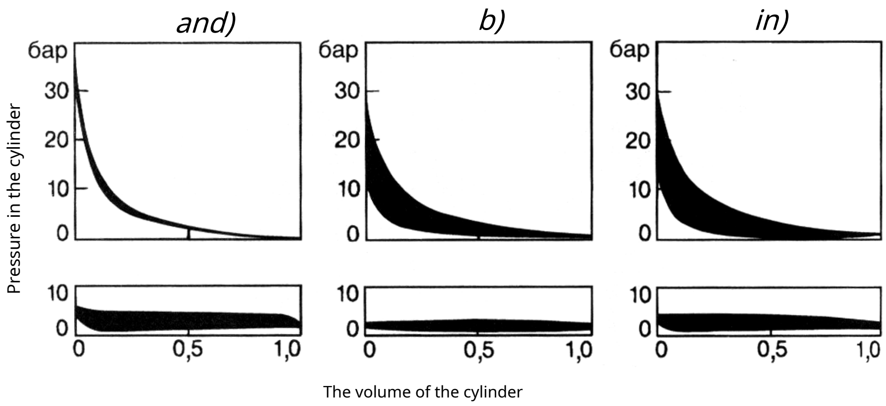
1 – exhaust damper drive (compressed air); 2 – exhaust valve; 3 – pressure regulator valve; 4 – outlet channel; 5 – intake channel; 6 – piston (fourth stroke)

Figure 12.20 – Motor brake with additional pressure regulator

When using the pressure regulator valve in the passage channel (bypass) together with the exhaust valve, the braking force can be increased at low and medium crankshaft rotation frequencies. At high speed, the pressure regulator valve prevents the pressure from increasing beyond the limit, which could damage the damper and its drive.

Engine brake with constant throttle

A standard exhaust valve brake uses energy absorbed during the fourth (exhaust) and first (intake) strokes. Special decompression during the second and third strokes allows you to use part of the energy of the compression stroke of the engine. The pressure-volume diagrams show the pressure changes in the cylinder when using the exhaust valve and constant throttle, as well as when connecting both devices (Fig. 12.21).

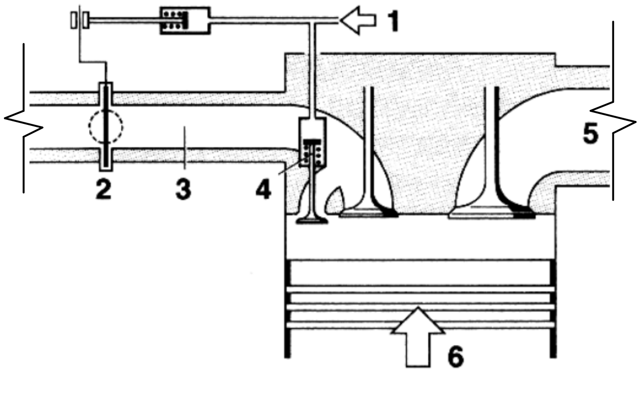


a)– closed position of the brake outlet valve;
b)– working position of the constant throttle;*in)*– the closed state of the exhaust valve gal-
ma and the working position of the constant throttle

Figure 12.21 – The principle of operation of exhaust brake systems, respectively
to the pressure–volume diagram,n_M=1700 min^{-1}

The braking power when a constant throttle system is used, as opposed to a throttle gate system, is primarily determined by the compression stroke.

Installation of a limiting valve in the bypass of the exhaust valve of the engine allows to increase the braking force. This valve is actuated by compressed air from the brake exhaust valve pneumatic actuator. During operation of the brake with the exhaust valve, it can be constantly open, providing a constant cross-section of the throttle (Fig. 12.22).



1 – compressed air; 2 – exhaust valve; 3 – release; 4 – constant throttle;
5 – inlet; 6 – piston (second stroke)

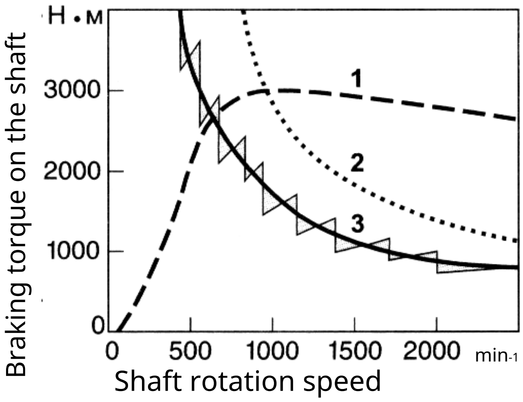
Figure 12.22 – Motor brake with exhaust valve and permanent draw
Salem (Mercedes-Benz)

12.8.2 Retarders

Retarders are increasingly used as non-wearing auxiliary braking systems for trucks and buses. Active the safety of cars increases as a result of reducing the loads acting on the working brake system, the economy of car operation also increases due to higher average driving speeds and the wear of brake linings decreases.

Retarders in a car can be installed between the engine and the gearbox (primary retarders) or between the gearbox and the drive axle (secondary retarders).

The disadvantage of primary retarders is the unwanted interruption of the braking moment when switching the mechanical gearbox. Primary retarders can have variable power. A significant difference between the performance characteristics of primary and secondary retarders emphasizes the need for further development in this area (Fig. 12.23).



1 – secondary retarder; 2 – primary retarder

Figure 12.23 – Braking moments of hydrodynamic retarders

Nowadays, there are two main designs of retarders: hydrodynamic and electrodynamic.

Hydrodynamic retarders

They work similarly to the "Foettinger" hydraulic clutch (Fig. 12.24). The rotor converts mechanical energy into kinetic energy of the liquid, which, in turn, is converted into heat in the stator (therefore, the used liquid must be cooled). Braking is controlled by the lever handle or the brake pedal (integrated retarder). During the operation of the electronic control circuit, the necessary air pressure is created, under the influence of which a certain amount of liquid (oil) is pumped into the working cavity between the rotor and the stator. The flow of oil formed as a result of the movement of the rotor is inhibited by the stationary blades of the stator. This causes braking of the rotor and the entire car.

Characteristics:

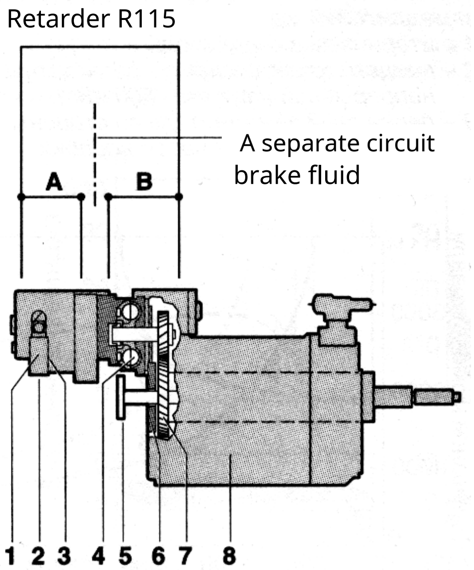
– relative complexity of the design; small mass of the retarder, which indirectly attached to the gearbox;

– the need to provide a cooling circuit for the purpose of dissipating heat generated by the retarder in the engine cooling system, for which an oil-water heat exchanger is used;

– high specific braking force;

– ultra-sensitive braking torque control;

- when developing a retarder, the valve should be taken into account tion losses of the non-activated retarder.



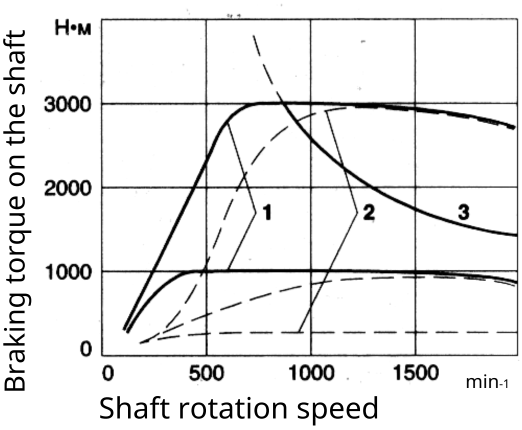
1 – wire connector; 2 – distribution valve; 3 – air receiver; 4 – stator; 5 – rotor; 6 – gearbox flange; 7 – cylindrical spur gears (gear ratio of the transmission); 8 – transmission

Figure 12.24 - Hydrodynamic retarder

In the hydrodynamic secondary retarder, it is possible to obtain an almost constant braking torque within a wide range of transmission shaft rotation speed (Fig. 12.25). A little below 1000 min^{-1}the braking torque decreases sharply. As a result of this characteristic, standard hydrodynamic retarders are particularly suitable for use on high-speed vehicles.

In modern designs of secondary retarders, the characteristic of the braking torque is improved by providing higher values of braking torques at low speeds of rotation of the shaft using a gear with a gear ratio of approximately 1:2 (reinforced retarder).

The maximum cooling power of modern diesel engines is approximately 300 kW. Due to the connection of the cooling systems of the engine and the retarder, there is some risk of overheating of both units, if additional safety measures are not provided. For this reason, thermal switches are used to limit the braking force of the retarder in order to ensure thermal equilibrium.



1 – reinforced retarder; 2 – standard retarder; 3 – power limit cooling at continuous load (300 kW)

Figure 12.25 – Operating characteristics of the hydrodynamic retarder

Electrodynamic retarders

Nowadays, the most widespread electrodynamic retarders have a stator with excitation windings (Fig. 12.26). The rotors are installed on both sides of the stator and have a ribbed surface for better heat dissipation. In order to brake the car, voltage is applied to the excitation coils (from the battery or from the generator), which generates a magnetic field that induces eddy currents in the rotors as they pass through the field. This creates a braking torque that depends on the excitation of the stator, as well as on the air gap between the rotor and the stator.

Characteristics:

– atmospheric dispersion of the produced heat;

– relative simplicity of construction;

– relatively large weight;

– non-stop work with sufficient power supply;

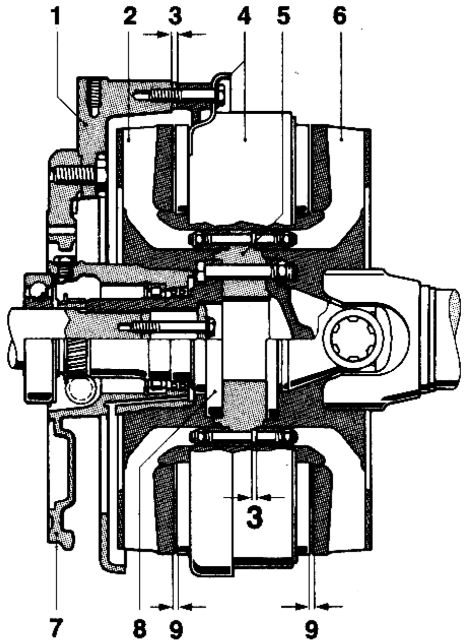
– reduction of braking torque when heating the retarder;

- ensuring high braking force even at low speed car;

– the braking power depends on the rotor blades, namely on air cooling conditions of the brake and the temperature of the surrounding air.

In contrast to standard hydrodynamic secondary retarders, electrodynamic retarders create relatively high values of braking moments at low speeds of rotation of transmission shafts.

A significant reduction in the braking torque of the electrodynamic retarder when the operating temperature of the rotor is increased occurs due to thermal protection measures (Fig. 12.27). The car's deceleration is reduced, because the thermal stresses of the electrodynamic retarder increase.



1 – support; 2 – front rotor; 3 – adjusting gaskets (establishment of the air gap); 4 – stator with coils; 5 – intermediate flange; 6 – rear rotor; 7 – cover gearboxes; 8 – transmission shaft; 9 – air gap

Figure 12.26 – Electrodynamic retarder

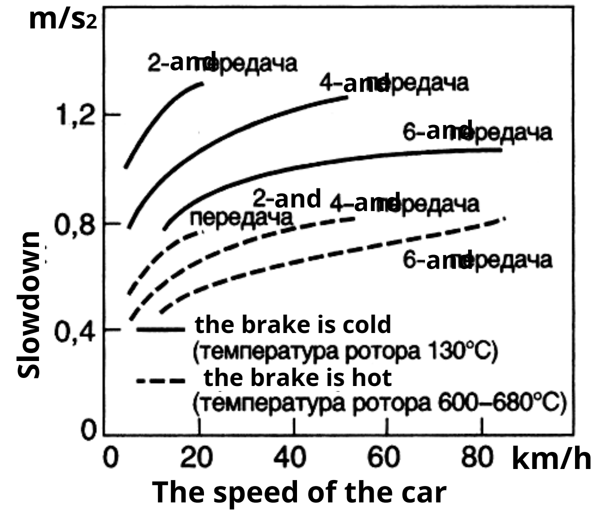


Figure 12.27 – Influence of gear ratio and rotor temperature on operation of electrodynamic retarders (17-ton loaded truck)

To prevent thermal destruction of the retarder during car braking, a bimetallic switch is used, which limits the current supply to half of the eight used coils when the stator temperature reaches approximately 250 °C.

12.9 Strength calculations

Design loads

For details of brake mechanisms, the amount of loads is determined by the maximum traction with the road

$$M_{Mr\max} = \phi_{\max} \cdot G_a \cdot r_{to}, \text{ with } \phi_{\max} = 0.8 \div 1.0.$$

For details of the drive - according to the calculated force of 1500 H on the pedal (800 N on the lever) or according to the maximum calculated pressure in the system (with a pneumatic drive).

Materials

Brake drums are usually cast from cast iron alloyed with copper, nickel, molybdenum, sometimes from steel with the addition of copper. Often they are made in the form of a combined design with a steel disk or an aluminum body. Brake blocks – welded steel or cast from cast iron or light alloys. Casings of taps, valves, etc. others often made of zinc or aluminum alloy.

Calculations

Pads are designed for bending and stiffness, fingers for bending and cutting. Friction surfaces - for heating and specific work of friction. Pedal - for bending. Pipelines - for breaking with a margin of about 2.

Questions for self-control

1. What is the car's brake system for?
2. What are the types of brake systems, their advantages and disadvantages?
3. Name the main requirements for brake systems.
4. By what features are brake systems classified?
5. Describe the working process of brake mechanisms for drums block and disk mechanisms.
6. How is the brake mechanism calculated on the heating?
7. Purpose and principle of operation of the brake actuator.
8. How is the brake actuator calculated?
9. Describe the design and calculation of hydrovacuum amplifiers.
10. Design features of a pneumatic brake actuator.
11. What is a brake force regulator?
12. Purpose and design of anti-blocking systems.
13. Working process of anti-lock systems.
14. Purpose and principle of operation of the parking brake system?
15. What types of retarders do you know?
16. Peculiarities of calculation of brake system strength.

13 BODY AND FRAME, DRIVER'S WORKPLACE

The frame, and often the body, are load-bearing systems that accept weight and shock-overloading loads. The car body is used to accommodate cargo, passengers and the driver.

13.1 Requirements for bodies and frames

1. Minimum weight with durability, taking into account corrosion resistance bone corresponding to the service life of the car.
2. Rigidity sufficient for the operation of units and components of the car.
3. The shape of the frame (body) should ensure the convenience of mounting the regattas, low height of the center of gravity and low loading height.
4. The shape and construction of the body should provide the necessary com-portability, injury safety, as well as fashion requirements.

13.2 Classification of bodies and frames

Bearing systems are classified depending on what perceives weight loads:
a) frame (body unloaded);
b) body – load-bearing body (frame, with load-bearing stand, panel);

c) the body combined with the frame is an integral supporting system (body frame).

Frames are divided into longitudinal (peripheral, X-shaped, ladder, with X-shaped crossbars) and spinal (central).

Bodies are divided into:
a) passenger (buses and cars); b) freight;

in) cargo and passenger;
d) special

Depending on the number of doors and the structure of the roof, passenger car bodies are divided into closed (sedan, limousine, coupe, etc.) and open (phaeton, convertible, etc.).

Truck bodies are divided into: a) general purpose – on-board platform;
b) specialized (dump trucks, vans, tanks, etc.).

13.3 Work process and basis of calculation of bodies and frames

In a static position (when resting on all wheels), weight loads act symmetrically along the longitudinal axis and cause the frame (body) to bend, and the stresses are small.

Calculation of the frame. When moving, dynamic loads act on the frame (body). At the same time, two regimes are usually considered for calculation.

*1. Movement at high speed on a road with small bumps.*Under-sored parts make intensive vertical oscillations. Dynamic loads can be expressed through static loads

$$P_i = P_{and\ cm} \cdot k_d, \tag{13.1}$$

where $P_{and\ cm}$ – static load;

$k_d = J_i / g$ is the dynamism coefficient for i of the calculated section.

Approximately vertical dynamic loads can be considered symmetric, and therefore, the spars work for bending, but significant torques are possible in the places where the units are attached. For multi-axle cars, the most dangerous case is considered: support on the wheels of only the extreme axles. When calculating the bending, the frame is considered as a beam on the supports - the axles of the car.

*2. Overcoming large bumps with suspension of individual wheels.*Opposite the reactions are not symmetrical, and the frame works in torsion at the design the moment

$$M_p = \frac{h}{B} \cdot \frac{C_n \cdot C_p}{C_n + C_p}, \tag{13.2}$$

where h - the height of the unevenness; IN – track;

C_n and C_p – angular stiffness of suspension and frame. In normal conditions, corners twisting of the frame is no more than 3-4°, off-road – up to 13-16°.

Spars and crossbars are thin-walled profiles that bend when twisted, become non-flat - deplan. In addition to tangential stresses, there are normal stresses of compressed torsion, so the calculations of frames for torsion are usually based on the theory of thin-walled profiles B. WITH. Vlasov. There are also approximate calculation methods. Table 13.1 shows the data of various profiles with the same wall thickness δ and cross-sectional areas (with equal weight per unit length).

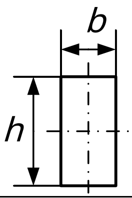
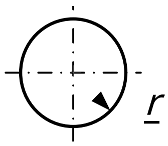
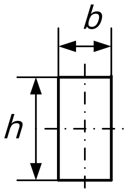
Calculation of the body. It is much more difficult to calculate the stresses in the load-bearing body, which is a complex shell structure with various slots and risers. Until recently, the main method of evaluating body strength was bending and torsion tests with stress measurement at many points.

The tension in the body of a passenger car during torsion is 2-4 times higher than during bending, and individual elements work for bending, stretching, compression, and torsion. Rigidity at the twist angle is considered normal θ' per meter of length no more $0°15'$.

In recent years, body calculations have been significantly improved. In the development of the theory of V. WITH. The Vlasov body is presented in the form of thin-walled bistructures consisting of two shell-type formations (roof and floor) and rods (window and door risers). With the help of the matrix apparatus of linear algebra, loads, moment diagrams, displacements, and geometric dimensions are described. According to M. B. Shkolnikov did not use an elementary rod as a basis, but a bistructure with one pair of risers, and the construction of the theory was carried out on the basis of the method of forces of construction mechanics, and not displacements. All this made it possible to significantly reduce the order of the matrices and to work out calculation methods using computers for both bending and torsion, but without taking into account random dynamic loads from the road and vibrations.

Methods of approximate calculation of deformations or damage to the body from an impact during an accident (from the front, rear, side, or top – overturning) have been developed.

Table 13.1 – Profile parameters of spars and crossbars

Profile scheme	W_{in} - bending	W_{in}, %	W_{Cr} - torsion	W_{Cr}, %
 $\frac{h}{b}=2$	$\frac{\delta \cdot h}{3}\cdot(h+3\cdot b)$	53	$2\cdot\delta\cdot b\cdot h$	70
	$\pi\cdot\delta\cdot r^2$	47	$2\cdot\pi\cdot\delta\cdot r^2$	100
 $\frac{h}{b}=2.5$ $\frac{h}{b}=3.5$	$\frac{\delta \cdot h}{6}\cdot(h+6\cdot b)$	100 105	$\frac{\delta^2}{3}\cdot(h+2\cdot b)$	3.5 3.5

Buffer calculation. A buffer with an energy-absorbing element – a shock absorber – reduces the force of a frontal impact in light collisions. The movement of the car during the impact can be described by the equation

$$\frac{G_a \ddot{X}}{g} = -F(t), \tag{13.3}$$

where $F(t)$ – change in the force of depreciation over time;

$\ddot{X}$ – shock absorber acceleration. Having accepted $F(t)=F_a=\text{const}$, we obtain from (13.3)

$$X = -\frac{g\cdot F_0}{2G_a}\cdot t^2 + V_0\cdot t. \tag{13.4}$$

Also, the timing of the strike $t_y = \dfrac{G_a \cdot V_a}{g \cdot F_0}$ (at $\ddot{X} = 0$) and shock absorber travel

$X_{max} = \dfrac{G_a \cdot V_0^2}{2g \cdot F_0}$ from (13.4) at $t = t_y$. If X_{max} = 80 mm and $\ddot{X}$ = 4g, then receive we press $V_0 = \sqrt{2X_{max} \cdot \ddot{X}} = 2.5$ m/s = 9 km/h.

13.4 Strength calculations

Design loads. The car frame perceives static and dynamic loads. Static loads arise as a result of the weight of the frame itself, the body, the payload and the reactions of the supports of the elastic element of the suspension. Thus, after assembly of the car, bending stresses occur in its frame, which make up 10...15% of the yield strength of the frame material. At the same time, the stresses in the spars of the frame are 2-2.5 times greater than in its crossbars. Dynamic loads arise during the movement of the car as a result of the action of the inertia of the sprung masses during the oscillations of the car.

When the car is moving, the supporting body bends under the influence of the load, passengers, installed units and mechanisms, as well as its own weight. The body also perceives torsional loads during lateral rolls and skewing of bridges, inertial loads during acceleration and braking, vibrations during its own oscillations.

Materials. Frames are made of thick sheet steel 08KP, 20KP or low-alloy steels 14G2, 30T, etc. Bodies made of sheet steel 08KP, 08FKP, 08Ю with a thickness of 0.8-1.5 mm according to GOST 9045.

Calculations. The frame is designed for bending from vertical loads (static and dynamic) and for torsion that occurs when the car drives over road bumps (ditches, ditches, etc.). The frame calculation is approximate. When calculating, the effect of longitudinal loads arising from units and mechanisms installed on the frame during uneven movement of the car (braking, acceleration) is neglected. The calculations also do not take into account various reactive moments (braking, from the crankcases of the transmission and steering mechanisms), which are perceived by the frame.

The body is a complex spatial system and its calculation for complex bending and torsional stresses is very difficult. The calculation of the body is performed by various approximate methods with simplifications and assumptions. These methods include: the potential energy method, the thin-walled rod method, and the finite element method.

The potential energy method is used in comparative calculations at the initial stage of body design.

The method of thin-walled rods is used after the development of the body structure is completed.

The finite element method is the most accurate when calculating the body. Its essence is that the body structure is replaced by a structural model, which consists of the simplest elements with known elastic properties This makes it possible to evaluate the properties of the body under certain loads.

13.5 Features of the arrangement of the driver's workplace

The driver's working conditions are determined by many factors. One of the main factors is the organization of the workplace: seating, visibility, effort and movement of levers and pedals, safety.

The position of the driver's seat and the typical dimensions of his workplace are regulated by UNECE Rules and national standards (GOST 9734, GOST 12024).

The view of the road from the seat is improved by moving the driver's seat closer to the front of the car, increasing the windshield and removing the opaque pillars.

Maximum permissible forces in N: brake pedal – 700, clutch pedal – 150, steering wheel when driving on asphalt concrete – 60, handbrake lever – 400, gear shift lever – 60.

Car safety includes active measures to prevent accidents and passive measures that protect people from injuries. For active activities, along with increased strength and reliability of steering and brakes; excellent visibility from the driver's seat, comfortable seating for the driver, reduced effort on the levers and pedals are also attributed to the high stability and improved controllability of the car, improved braking properties.

Passive measures (passive safety) include:

a) safety belts, airbags and other devices;

b) less traumatic parts: shatter-free glass, collapsible steering wheel, and panels;

c) reduced loads on people in frontal, rear, side impacts and overturning due to a significant increase in the rigidity of the part of the body where people are located, the use of energy-absorbing bumpers, etc. d.;

d) fire prevention measures, etc. d.

Questions for self-control

1. What is the car’s supporting system and what is it for?

chena

2. Requirements for bodies and frames.
3. How are bodies and frames classified?
4. Describe the work process and the basics of calculating bodies and frames.
5. How is the strength of the supporting system calculated?
6. Name the main features of the arrangement of the driver's workplace.