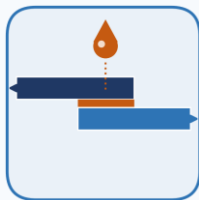


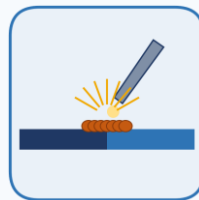
CONTENTS



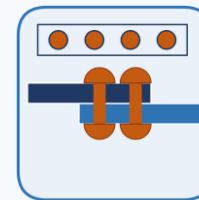
CHAPTER 1

PRC**CONSTRUCTIVE DESIGN OF
MACHINE PARTS***Constructive design of elements of
mechanical technical products*[pages 1-1 ... 1-76](#) >

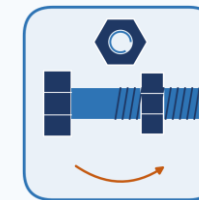
CHAPTER 2

LIP**BONDED JOINTS***Structure and modeling of joints by
bonding (with alloys or adhesives)*[pages 2-1 ... 2-11](#) >

CHAPTER 3

SUD**WELDED JOINTS***Structure and modeling of welded joints*[pages 3-1 ... 3-16](#) >

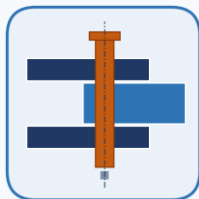
CHAPTER 4

NIT**RIVET JOINTS***Structure and modeling of rivet joints*[pages 4-1 ... 4-12](#) >

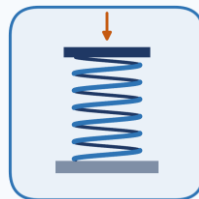
CHAPTER 5

AFS**THREADED AND SCREWED
ASSEMBLIES***Construction and modeling of threaded
assemblies*[pages 5-1 ... 5-50](#) >

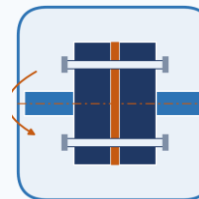
CHAPTER 6

AAB**SHAFT-HUB ASSEMBLIES***Construction and modeling of shaft-hub
assemblies*[pages 6-1 ... 6-96](#) >

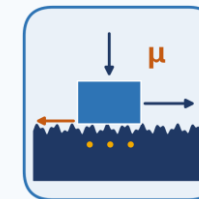
CHAPTER 7

ASB**ASSEMBLIES BY PINS AND
BOLTS***Construction and modeling of
assemblies using pins and bolts*[pages 7-1 ... 7-19](#) >

CHAPTER 8

ARC**ELASTIC ELEMENTS
(SPRINGS)***Construction and modeling of elastic
elements (springs)*[pages 8-1 ... 8-36](#) >

CHAPTER 9

CUP**MECHANICAL COUPLINGS***Construction and modeling of
mechanical couplings*[pages 9-1 ... 9-37](#) >

CHAPTER 10

ETB**ELEMENTS OF TRIBOLOGY***Elements of tribology: friction and wear*[pages 10-1 ... 10-58](#) >

PRC. CONSTRUCTIVE DESIGN OF MACHINE PARTS

**PRC-T. CONSTRUCTIVE DESIGN OF ELEMENTS OF MECHANICAL TECHNICAL
PRODUCTS**

CONTENTS

PRC-T.1 THEORETICAL BASIS	1-4
PRC-T.1.1 DEFINITIONS	1-4
PRC-T.1.2 GENERAL INFORMATION ABOUT MECHANICAL PRODUCTS	1-4
PRC-T.1.2.1 STAGES OF TECHNICAL PRODUCT DEVELOPMENT	1-4
PRC-T.1.2.2 BLOCK DIAGRAM AND FUNCTIONS OF TECHNICAL PRODUCTS	1-5
PRC-T.1.2.3 TYPES OF MECHANICAL TECHNICAL PRODUCTS	1-5
PRC-T.1.2.4 STRUCTURAL-CONSTRUCTIVE DESCRIPTION OF MECHANICAL PRODUCTS	1-7
PRC-T.1.2.5 STRUCTURAL-CONSTRUCTIVE DIAGRAM OF A MECHANICAL TECHNICAL PRODUCT	1-8
PRC-T.1.3 ASPECTS REGARDING THE DESIGN OF MECHANICAL TECHNICAL PRODUCTS	1-9
PRC-T.1.3.1 THE DESIGN PROCESS OF MECHANICAL TECHNICAL PRODUCTS	1-9
PRC-T.1.3.2 PRINCIPLE AND FUNCTIONAL DESIGN	1-10
PRC-T.1.3.3 CONSTRUCTION DESIGN	1-11
PRC-T.1.3.4 COMPUTER-AIDED DESIGN	1-14
PRC-T.1.4 OBJECTIVES OF THE MACHINE PARTS DISCIPLINE	1-15
PRC-T.2 MATERIALS AND HEAT TREATMENTS	1-16
PRC-T.2.1 MAIN CHARACTERISTICS AND PROPERTIES FOR USE OF MATERIALS	1-16
PRC-T.2.1.1 PHYSICAL CHARACTERISTICS AND PROPERTIES	1-16
PRC-T.2.1.2 CHEMICAL CHARACTERISTICS AND PROPERTIES	1-16
PRC-T.2.1.3 MECHANICAL CHARACTERISTICS AND PROPERTIES	1-17
PRC-T.2.1.4 TECHNOLOGICAL CHARACTERISTICS AND PROPERTIES	1-21
PRC-T.2.2 STEELS FREQUENTLY USED FOR THE CONSTRUCTION OF MACHINERY ELEMENTS	1-21
PRC-T.2.3 MAIN HEAT TREATMENTS APPLIED TO STEELS	1-23
PRC-T.2.3.1 ANNEALING	1-23
PRC-T.2.3.2 HARDENING	1-23
PRC-T.2.3.3 TEMPERING	1-23
PRC-T.2.3.3 THERMOCHEMICAL TREATMENTS	1-24
PRC-T.3 CALCULATION OF STRENGTH AND RIGIDITY (DEFORMATION) OF MACHINE ELEMENTS	1-25
PRC-T.3.1 CALCULATION OF RESISTANCE OF MACHINE ELEMENTS	1-25
PRC-T.3.1.1 GENERAL STRUCTURE OF INTERACTING BODIES SYSTEMS	1-25
PRC-T.3.1.2 GEOMETRIC MODELING OF MECHANICAL STRUCTURE ELEMENTS	1-26
PRC-T.3.1.3 MODELING OF CONNECTIONS (SUPPORTS) BETWEEN BODIES	1-28
PRC-T.3.1.4 LOAD MODELING	1-29
PRC-T.3.1.5 REQUEST MODELING	1-31
PRC-T.3.1.6 ALLOWABLE STRESS AND SAFETY COEFFICIENTS	1-39
PRC-T.3.2 CALCULATION OF RIGIDITY (DEFORMATIONS) OF MACHINE ELEMENTS	1-40
PRC-T.3.2.1 MODELING OF DEFORMATIONS AND VOLUME DISPLACEMENTS	1-40

PRC-T.4 TECHNOLOGICAL DESIGN OF SHAPES 1-43

PRC-T4.1 DESIGN OF TECHNOLOGICAL FORMS 1-43

PRC-T.4.1.1 GENERAL ASPECTS 1-43

PRC-T.4.1.2 RECOMMENDATIONS ON THE DESIGN OF THE SHAPE OF CASTINGS..... 1-45

PRC-T.4.1.3 RECOMMENDATIONS ON THE DESIGN OF THE SHAPE OF PARTS PROCESSED BY DEFORMATION 1-46

PRC-T.4.1.4 RECOMMENDATIONS ON THE DESIGN OF THE SHAPE OF MACHINED PARTS 1-47

PRC-T.4.2 ADJUSTMENTS AND TOLERANCES 1-48

PRC-T.4.2.1 DEFINITIONS 1-48

PRC-T.4.2.2 DIMENSIONS, DEVIATIONS AND TOLERANCE FIELDS 1-49

PRC-T.4.2.3 BORES AND BORE SYSTEMS 1-49

PRC-T.4.2.4 ISO TOLERANCE AND ADJUSTMENT SYSTEMS..... 1-50

PRC-T.1 THEORETICAL BASIS

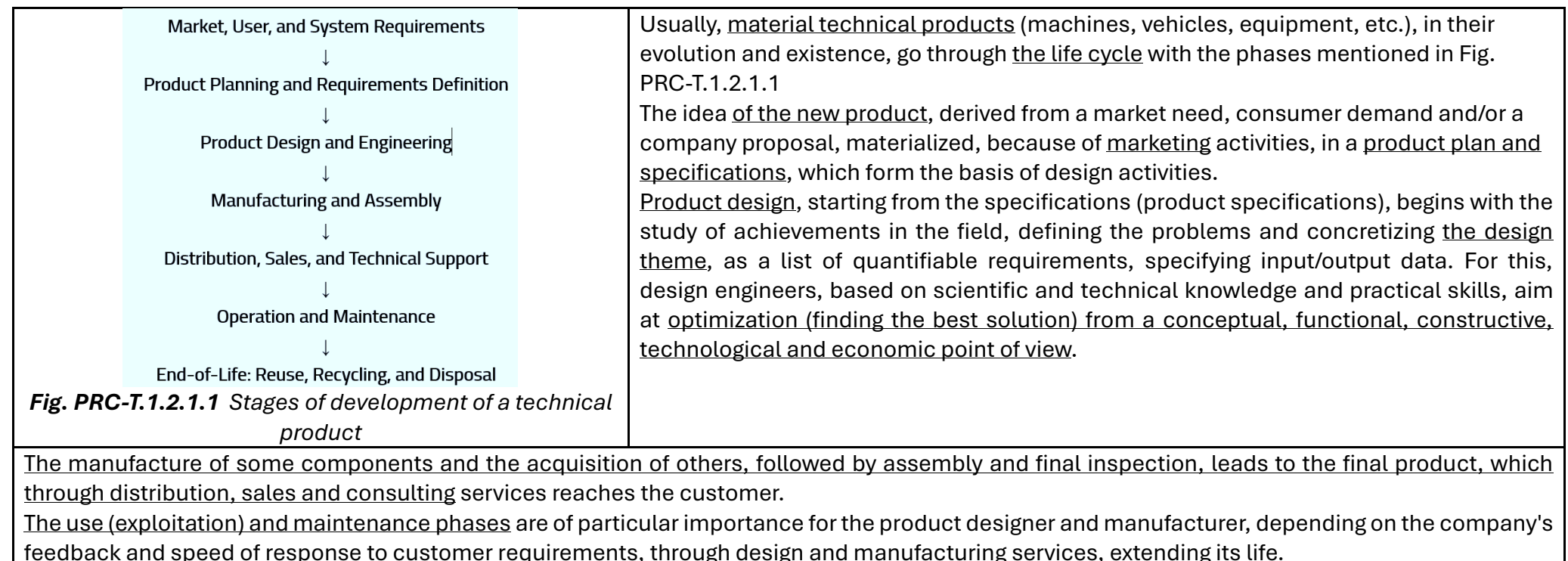
PRC-T.1.1 DEFINITIONS

A *mechanical technical product* is a physical, tangible good that can satisfy human desires and/or needs regarding the development of useful mechanical work.

Engineering design is the intellectual creative activity that aims to design and develop technical documentation for obtaining new products, with high performance, at low costs. Due to its complexity, importance and implications, design activity can be considered as being in direct relation with various fundamental fields of human activity (philosophy, artistic design, economic sciences).

PRC-T.1.2 GENERAL INFORMATION ABOUT MECHANICAL PRODUCTS

PRC-T.1.2.1 STAGES OF TECHNICAL PRODUCT DEVELOPMENT



Once the product is no longer in use, the last stage begins, which aims to further use its elements with identical or subsequent functions (reuse, revalorization) or reuse of materials after disassembly, as well as the efficient recovery of waste.

PRC-T.1.2.2 BLOCK DIAGRAM AND FUNCTIONS OF TECHNICAL PRODUCTS

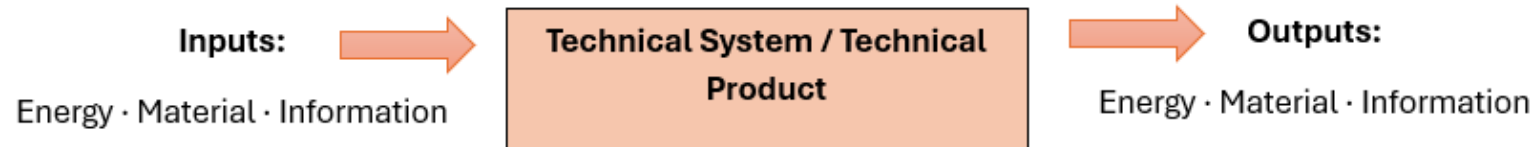


Fig. PRC-T.1.2.2.1 Block diagram of a technical product

Technical products, in general, involve the qualitative and quantitative conversion of energy, materials and/or input information into similar quantities corresponding to the output (fig. PRC-T.1.2.1.2).

For a technical product (material), the qualitative description of its purpose, as a relationship (connection) between the input and output parameters, is achieved by the global purpose function. By dividing the global function into independent sequences, partial purpose functions are obtained. Both the global function and the partial ones, usually, are descriptively expressed as energy transformation, material transmission, etc.

Energy quantities (forces, speeds, power), material quantities (mass, weight, volume, flow rate) and/or information quantities (signals) are transmitted and transformed into the technical product through flows. The functions that connect the output and input parameters (energy, material or information) of a technical product are main functions, with mathematical expressions that deterministically describe the correlations between specific physical parameters.

By dividing the technical product into functionally independent subsystems, the main functions are divided into their associated partial functions. In addition, for certain products there may be subsystems and accompanying flows, indirectly participating in the fulfillment of the main functions, to which auxiliary functions are associated.

PRC-T.1.2.3 TYPES OF MECHANICAL TECHNICAL PRODUCTS

Mechanical technical products as systems of several interacting solid material bodies are made in various configurations.

Tab. PRC-T.1.2.3.1 Classification of mechanical technical products

Criterion	Technical product type	Description
The existence of relative motion between the	Mechanical structures	Technical products composed of several <u>material bodies</u> (parts, elements) in interaction WITHOUT relative movements whose main function is <u>the transmission of mechanical loads</u> (forces and/or moments) are called <u>mechanical structures</u> (chassis, bodies, towing devices, etc.)

component bodies	machinery	Technical products composed of several <u>material bodies (parts, elements) interacting WITH determined relative movements</u> whose main function is the transmission and transformation of various forms of energy are called <u>machines</u> (technological equipment, vehicles, etc.)
The existence of electronic information flows	Mechanical	Technical products that are composed only of material bodies (parts, elements) in interaction and characterized by physical flows of forces, speeds and power are called <u>mechanical products</u> (non-automatic gearboxes, drive axles, etc.)
	Mechatronics	Mechanical technical products characterized by energy flows (forces, speeds and power), accompanied by important <i>information flows</i> for regulation, indication and/or control, become <u>mechatronic products</u> (automatic washing machines, modern vehicles, robots, etc.)

Tab. PRC-T.1.2.3.2 Classification of mechanical technical products of machine type

Criterion	Machine type	Description
The purpose for which the mechanical energy from the output is used	engines	Machines that transform different forms of energy (mechanical, electrical, thermal) into <u>mechanical energy in order to generate useful mechanical work</u> (for propulsion or technological processing) are called motor machines (electric motors, thermal engines, etc.)
	Working	Machines that transform different forms of energy (mechanical, electrical, thermal) into mechanical energy used to <u>modify the shape, state, properties and/or positions</u> of materials or objects to be processed are called working machines (machine tools, robots, etc.)
Possibility of movement	Static	Machines that have <u>the chassis (frame) fixed to the foundation</u> are called static machines (machine tools, technological equipment, etc.)
	Mobile (vehicles)	Machines that can <u>move</u> , by rolling (using wheels or tracks) or by sliding (sled type) for the purpose of transporting (goods or people) or for performing industrial, agricultural and service works (cars, tractors, robot cars, etc.)

Tab. PRC-T.1.2.3.3 Classification of vehicle-type mechanical technical products

Criterion	Vehicle type	Description
Runway	Road	Vehicles traveling on public roads (cars, trucks, etc.)
	Railway	Vehicles that move on rails (locomotives)
	Naval	Vehicles that move on water or in the air (river ships, airplanes, etc.).
Propulsion	Externally powered	Vehicles that are driven by static motors fixed to the foundation (cable-driven vehicles)
	Motor vehicles	Self-propelled vehicles (cars, tractors, etc.)

PRC-T.1.2.4 STRUCTURAL-CONSTRUCTIVE DESCRIPTION OF MECHANICAL PRODUCTS

From a constructive point of view, mechanical products are composed of assemblies, subassemblies and constructive elements (including machine parts), easily identifiable in the structural-constructive diagram.

Assemblies are independent constructive entities, which respect the functional structure of the product and are composed of subassemblies and constructive elements with shapes and positions determined by the assembly, maintenance and operation technologies. For a complex product, several assemblies can be highlighted in interaction, through permanent fixed links, between them (internal links) and, sometimes, with the adjacent environment (external links) which, usually, are materialized through demountable assemblies.

Subassemblies are independent structures, which are not always constituted respecting functionality, and which are highlighted by a compact group composed, in minimal configuration, of at least two constructive elements or of other subassemblies and constructive elements, in permanent interaction, formed taking into account, in particular, the assembly, maintenance and operation technologies.

The constructive elements (machine parts) as indivisible component parts of a technical mechanical product, assembly and/or subassembly, are distinct entities in direct fixed, permanent interaction (non-demountable or dismountable), or in direct mobile interaction (without lubrication) or indirect (with lubrication). The types of these connections, from a constructive point of view and depending on the functional needs, are diverse and are treated in detail within the Machine Parts discipline.

Tab. PRC-T.1.2.4.1 Classification of structural elements and subassemblies (parts) of mechanical products

Criterion	Type of elements and subassemblies	Description
Importance within energy flows	Main (vital)	They are positioned on the main energy flows
	Support	They are positioned on auxiliary energy flows
Type of partial or auxiliary functions	Transmission	Partial function of transmitting loads (forces and/or moments); they are positioned on the main energy flows
	For support	Auxiliary functions, for unidirectional, bidirectional, or tridirectional support of other structural elements or assemblies
	Link	Partial connecting function, for making fastening assemblies, usually removable
	Protective	Auxiliary function for human and environmental protection
Field of use	Specific	The structural elements or subassemblies in a mechanical product are specific when <u>they are designed and manufactured only for one product</u> .
	Specialized	The structural elements (assembly elements, kinematic couplings, belts, chains, etc.) or subassemblies (speed reducers, screw-nut transmissions, speed variators, bearings, bushings, guides, etc.) of a mechanical product are specialized when they are designed and produced by established specialized companies, for distinct <u>general functions</u> applicable to several products. For these, established design algorithms and advanced execution and assembly technologies

		have been developed over time, thus ensuring optimal performance from a functional, constructive, technological, and economic point of view.
--	--	--

PRC-T.1.2.5 STRUCTURAL-CONSTRUCTIVE DIAGRAM OF A MECHANICAL TECHNICAL PRODUCT

For the analysis and, especially, for the synthesis of the constructive structure of a mechanical product, a structural-constructive diagram is drawn up, which is a graphic representation based on symbols associated with assemblies, subassemblies and constructive elements, highlighting all the internal connections between them and the external connections with adjacent constructive entities.

Unlike the structural-functional diagram, in the structural-constructive diagram one can also identify passive couplings, bearing systems, fixed connections, etc. The representations in the structural-constructive diagram must allow easy identification of motion flows and, especially, those of internal (main and auxiliary) forces.

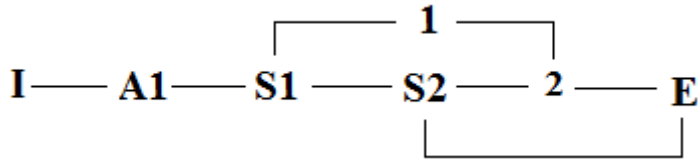
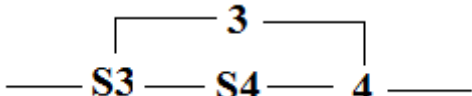
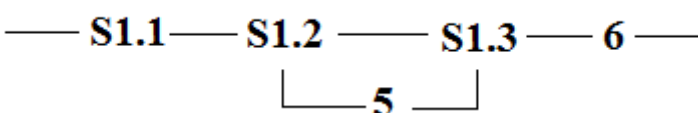
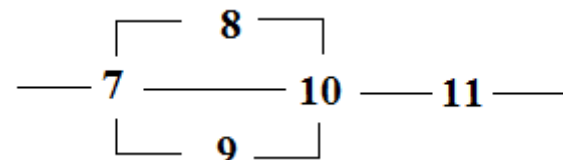
		Division of mechanical product structures Starting from the fact that a subassembly implies the existence of at least two constructive elements or in turn can be composed of other specific or specialized subassemblies, the division of complex assemblies into subassemblies and, further, into other subassemblies and constructive elements, implies, depending on the specific cases, <u>multiple levels of subdivision</u> (fig. PRC-T.1.2.5.1).
<i>a</i>	<i>b</i>	
		
<i>c</i>	<i>d</i>	
Fig. PRC-T.1.2.5.1 Structural construction diagrams: <i>a</i> – of the product; <i>b</i> – of the assembly A1; <i>c</i> – of the subassembly S1; <i>d</i> – of the subassembly S1.1		

Figure PRC-T.1.2.5.1,a shows, in general, a structural and constructive diagram of a product with input I and output E containing assembly A1, subassemblies S1 and S2 and elements 1 and 2; assembly A1 contains subassemblies S3, S4 and elements 3 and 4 (fig. PRC-T.1.2.5.1,b); subassembly S1 contains subassemblies S1.1, S1.2, S1.3 and elements 3, 4 (fig. PRC-T.1.2.5.1,c); subassembly S1.1 from the composition of subassembly S1 contains elements 7,8...11 (fig. PRC-T.1.2.5.1,d). In subchapter PRC-P.1 a concrete example is presented.

PRC-T.1.3 ASPECTS REGARDING THE DESIGN OF MECHANICAL TECHNICAL PRODUCTS

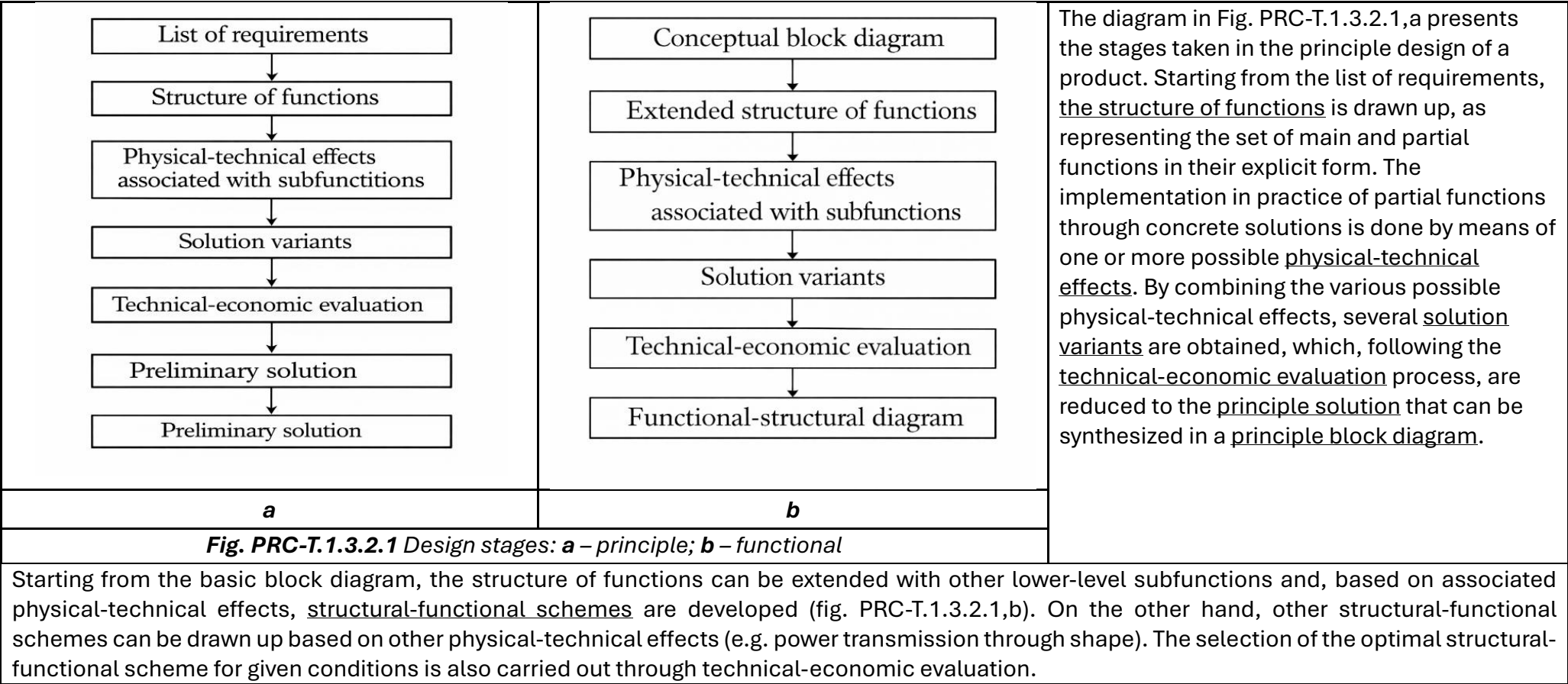
PRC-T.1.3.1 THE DESIGN PROCESS OF MECHANICAL TECHNICAL PRODUCTS

The formulation of the design theme of a technical product is preceded by detailed documentation and critical analysis of global achievements in the field, regarding the requirements for possible technical solutions and the existing practical constraints. Following this analysis, **a list of requirements** is drawn up with a clear mention of the performances at a qualitative and/or quantitative level, indicating the values of specific indicators. In general, the list of requirements includes information on the following general characteristics of the product: number of copies; geometric restrictions (imposed overall dimensions); kinematics (speeds, accelerations, directions of movement); forces (weight, load, directions, application frequencies); energy (powers, efficiencies); safety (labor and environmental); working conditions (noise, comfort, ergonomics); quality (reliability, maintenance, standards to be respected); assembly, operation and maintenance conditions; economic (maximum allowed costs, recycling possibilities); manufacturing (available processes); working environment (temperature, temperature variations, altitude, type of atmosphere), etc.

The general objectives pursued in the design of technical material products can be: fulfilling the main and auxiliary functions; reliability and operational safety; increased ergonomics; ensuring human and environmental safety; simple execution and easy assembly; ensuring efficient maintenance; minimizing costs; recycling possibilities, etc. The final solution adopted in the design will be determined by the general objectives and imposed restrictions, which are obtained as a consequence of the requirements and conditions in the list of requirements, as a document that underlies product development.

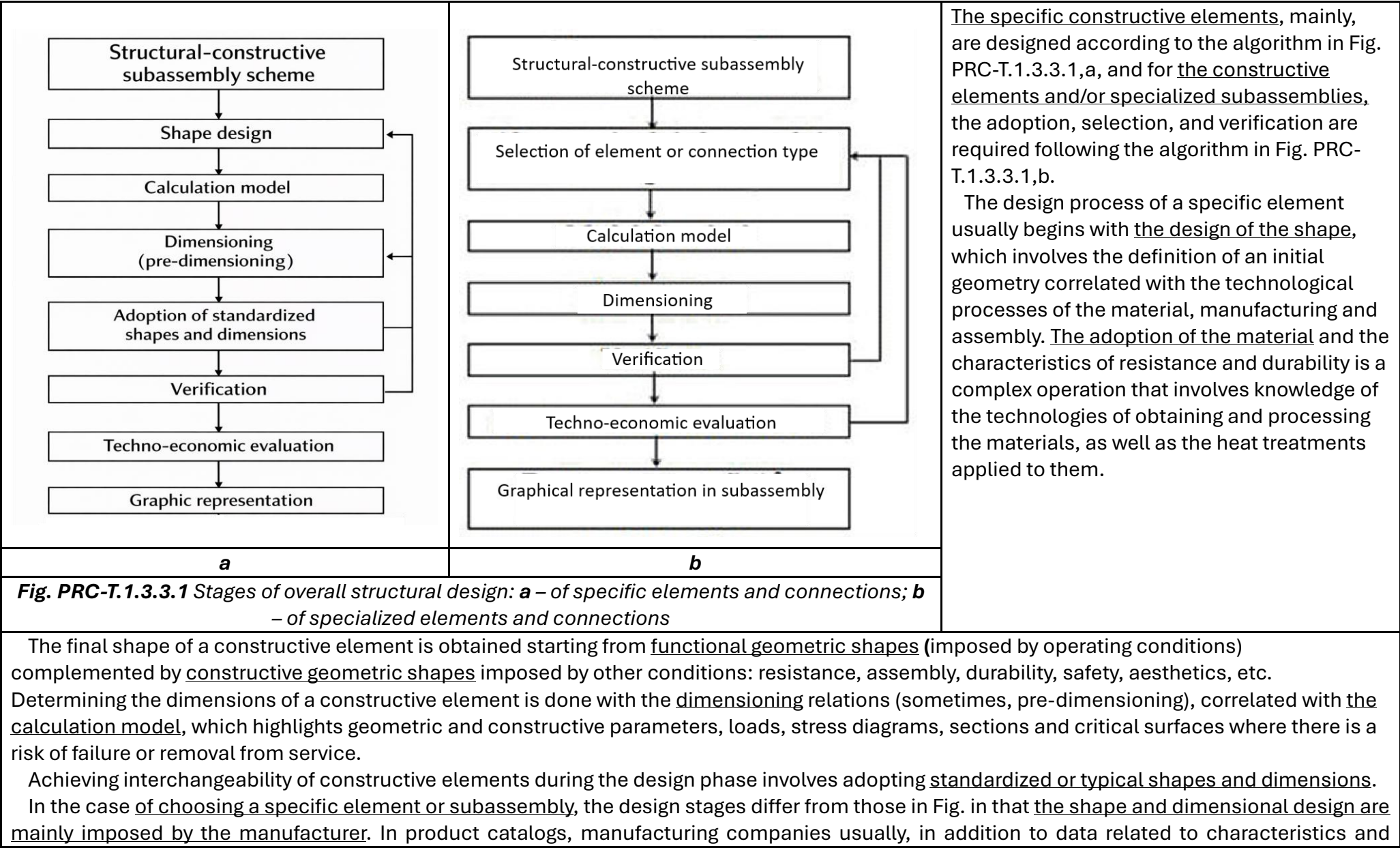
The actual design is based on algorithmic procedures that, starting from the imposed functions and objectives, seek conceptual (principled design), functional (functional design), and constructive (constructive design) solutions.

PRC-T.1.3.2 PRINCIPLE AND FUNCTIONAL DESIGN



PRC-T.1.3.3 CONSTRUCTION DESIGN

Tab. PRC-T.1.3.3.1 Overall structural design



performance, also introduce information regarding the calculation model, sizing and/or verification relationships, recommendations for assembly, as well as operating modes and maintenance methods.

Note.

a. Since most of the structural elements of mechanical systems are in mutual interaction, forming direct mobile (kinematic couplings) or fixed removable connections, in order to determine the dimensional parameters of the surfaces of the structural elements in contact, it is necessary to follow the general algorithm in Fig. PRC-T.1.3.3.1,a. The shapes of the surfaces of the mobile elements in contact are also established, starting from the imposed functional conditions, as well as from the available technological possibilities. The shapes of the surfaces of the structural elements that participate in fixed connections are established, as a rule, depending on the type of assembly. The calculation models and the dimensioning and verification relations of the specialized elements have a pronounced specificity, depending on the type of materialization of the direct mobile connection (gears, friction transmissions, bearings, etc.) or fixed (threaded, shaft-hub, welded, etc.). On the other hand, in the case where for the connections between specific constructive elements there are subassemblies or specialized constructive elements or subassemblies (bearings, bushings, rolling guides, assembly elements, etc.), the design raises the issue of their selection, dimensioning and verification starting from the functional characteristics and performances, according to the algorithm in Fig. PRC-T.1.3.3.1,b.

b. In all phases of the design of a product, assembly, subassembly, constructive element or a connection between them, economic evaluations by identifying expenses and value analysis studies for efficiency are of particular importance in achieving the target cost. In general, the following costs can be identified in the technical-economic evaluation: material (individual, general), manufacturing (individual, general, special), administration, sales and other activities (resespring, insurance, licenses, etc.).

c. Technical products in modular construction have a structure divided into several modules (blocks) that have elements and subassemblies with shapes that allow other combinations and, possibly, by adding other elements, lead to new functional structures with different partial functions.

Tab. PRC-T.1.3.3.1 Detailed structural design

Assembly Drawing ↓ Subassembly Drawings ↓ Technological Form Design ↓ Manufacturing (Execution) Drawings ↓ Product Technical Documentation	Assembly Drawing ↓ Subassembly Drawings ↓ Manufacturing (Execution) Drawings ↓ Operation Plans and/or Process Sheets ↓ Tooling, Fixtures and Inspection Documentation ↓ Product Technological Documentation	Detailed design is the complementary part of the overall constructive design process, in which the constructive and technological form of the component elements is finalized through specifications related to the material, heat treatments, technological form, their dimensions and deviations, surface quality, etc.
Fig. PRC-T.1.3.3.2 Detailed design stages	Fig. PRC-T.1.3.3.3 Technological design stages	

The main stages of the detailed constructive design process are presented in Fig. PRC-T.1.3.3.2. The results of this process are expressed through the technical documentation of the product, in particular, the execution drawings, assembly drawings, parts list, etc. The design of the technological shape of an element involves small changes to the shape of the constructive element, required by the technological processes of its realization, without affecting the critical areas defined in the constructive design.

In order to manufacture the product, starting from the technical documentation, technological design is required, according to the general algorithm in Fig. PRC-T.1.3.3.3 which leads to technological documentation of processing and assembly. To draw up the operation plans or technological sheets, the technological engineer must know the available processing, handling and transfer equipment and installations. In addition, to make a product, specific tools, fixtures, and devices are required, for which specific documentation is required.

PRC-T.1.3.4 COMPUTER-AIDED DESIGN

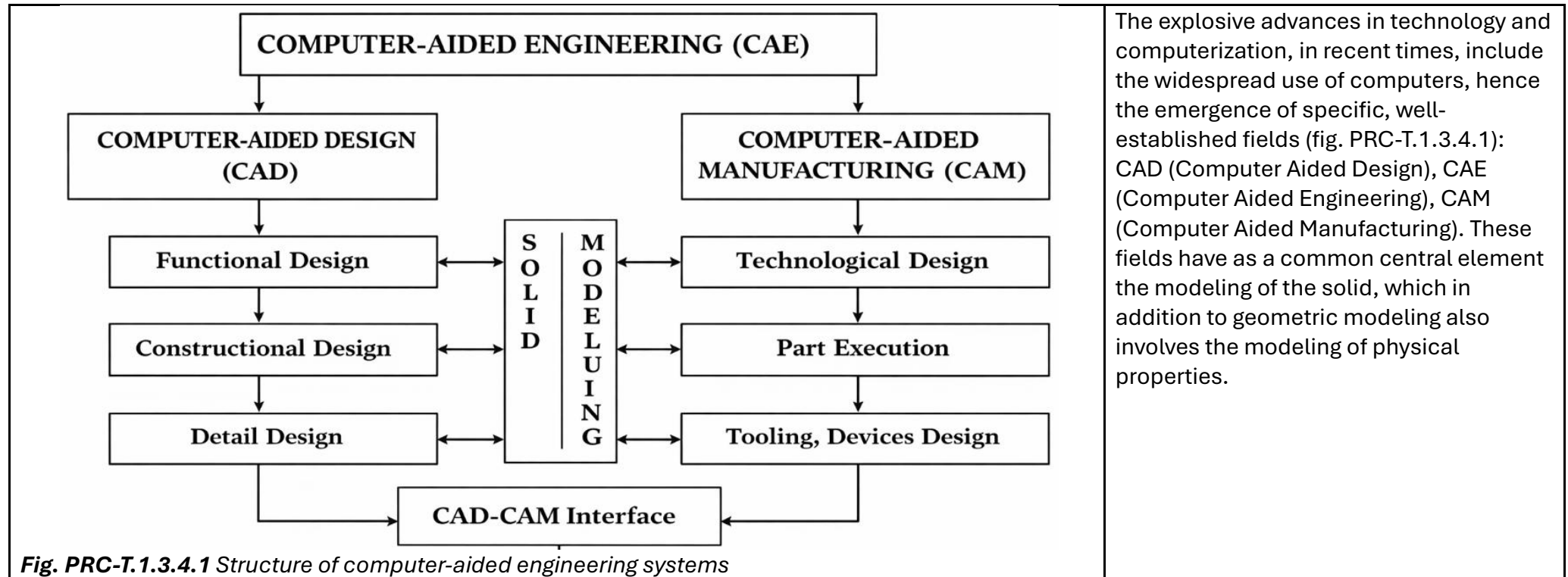


Fig. PRC-T.1.3.4.1 Structure of computer-aided engineering systems

For solid modeling, as a primary stage in the design activity, the following software packages have been noted: AutoCAD, ProEngineering, Euclid and, in recent times, CATIA. For the functional design of products, high-performance software has been developed that, with a high level of idealization of the geometric shape (with point, line and symbol entities), allows the determination, optimization and simulation of some functional parameters (the ADAMS software package is representative). For the constructive design, on the one hand, classical CAE calculation packages (e.g. the MDESIGN package) are used, which have the role of supporting the configuration of a primary shape of the constructive element and, on the other hand, finite element analysis packages that involve the modeling and simulation of physical processes that allow the optimization of the shape and dimensions taking into account the imposed geometric and physical (material) constraints. The following high-performance finite element analysis (FEA) software packages were dedicated to these activities: MSC/NASTRAN, ALGOR, ANSYS, CATIA, COSMOS, etc.

PRC-T.1.4 OBJECTIVES OF THE MACHINE PARTS DISCIPLINE

Main objective

Knowledge and understanding of the structures, calculation models, and design algorithms of specialized elements and subassemblies, as well as the connections between them, based on the optimal correlation of knowledge acquired in the disciplines of Technical Drawing, Mechanics, Mechanisms, Strength of Materials, and Thermotechnics.

Specific objectives

Developing knowledge of identifying and modeling specialized elements and subassemblies in vehicle structures.

Developing skills in using calculation software packages (MDESIGN) for machine elements.

Developing mechanical product design skills.

PRC-T.2 MATERIALS AND HEAT TREATMENTS

PRC-T.2.1 MAIN CHARACTERISTICS AND PROPERTIES FOR USE OF MATERIALS

PRC-T.2.1.1 PHYSICAL CHARACTERISTICS AND PROPERTIES

Physical properties determine the behavior of metallic materials under the action of physical phenomena: gravity, thermal, electric, magnetic, light fields, etc.

Tab. PRC-T.2.1.1.1 Main physical characteristics

Characteristic	Symbol	Definition	Unit of measurement	Note.
Density	ρ	Mass per unit volume of a homogeneous body	kg/ m ³	Varies with temperature and chemical composition
Specific heat	C_p	The amount of heat required to raise the temperature of a mass of 1 kg by 1°C.	J/kg.K, cal/g. °C	Materials with higher atomic weights have lower specific heats
Thermal conductivity	λ	The ability of metallic materials to conduct and transmit heat using free electrons (electron gas)	W/mK, cal/cm. °C	Temperature independent
Thermal expansion	α	The ability of metallic materials to change (increase) their dimensions when temperature changes (increases).	°C ⁻¹	It is the opposite of contraction.

PRC-T.2.1.2 CHEMICAL CHARACTERISTICS AND PROPERTIES

The chemical properties determine the behavior of metallic materials under the action of atmospheric, chemical, electrochemical, and other agents.

Tab. PRC-T.2.1.2.1 Main chemical characteristics

Characteristic	Definition	Note
Corrosion resistance (chemical stability)	The ability of metallic materials to resist the destructive action of atmospheric and/or chemical agents	Corrosion manifests itself through mass reduction and changes in the structure, chemical composition, dimensions, surface color and physical properties. Corrosion resistance is assessed by the corrosion rate $V_{cor} = \Delta m / S \cdot t$, [g/m ² s], where: Δm represents mass loss, S - surface, t - time.
Refractory (heat stability)	The ability of metals and alloys to retain their mechanical	For this, it is necessary that secondary transformations do not occur in the mass of metallic materials that can reduce mechanical resistance. Refractory or thermostable

strength and not increase unacceptably in volume under high temperature conditions

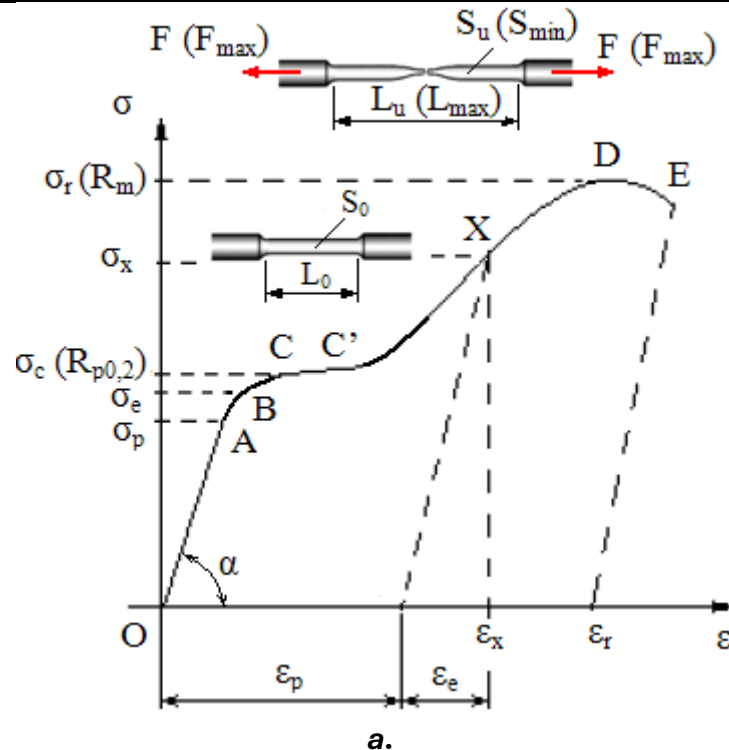
metallic materials are characterized by increased creep limits and resistance to high temperatures.

PRC-T.2.1.3 MECHANICAL CHARACTERISTICS AND PROPERTIES

The main mechanical properties of metallic materials - elasticity, plasticity, toughness, brittleness, creep, wear resistance, hardness, tensile strength, fatigue resistance, resilience, etc. - determine their behavior under static and dynamic mechanical stresses (variable in time) of tension, compression, bending, shear, torsion, penetration, etc. These properties are determined by mechanical tests which, unlike the case of physical and chemical characteristics, lead to mechanical characteristics that are also dependent on the test methods.

The determination of mechanical characteristics (longitudinal modulus of elasticity, conventional elastic limit, transverse shrinkage coefficient, etc.) is practically carried out based on the characteristic curve drawn during the tensile test.

Tab. PRC-T.2.1.3.1 Characteristic curve σ - ε for static tensile load



Test hypotheses:

- the force is applied statically (it increases slowly over time),
- the specimen has standard shapes and dimensions.

Main parameters:

$\sigma = F / S_0$ [MPa] – normal stress in the specimen section; F [N] – loading force; S_0 [mm²] – initial cross- sectional area of the specimen;

$\varepsilon = (L_u - L_0) / L_0 \cdot 100$ [%] – specific deformation (elongation) of the specimen; L_0 [mm] – initial length of the specimen; L_u [mm] length of the specimen after loading with force F ;

$E = \sigma / \varepsilon$ [MPa] = $\tan \alpha$ – the longitudinal modulus of elasticity expresses the material's ability to resist elastic deformations, a property associated with the material's stiffness, quantifiable by the slope of the tangent at the characteristic points, e.g. in the area of linear elasticity (OA), $\tan \alpha = E$;

$A_r = (L_{max} - L_0) / L_0 \cdot 100$ [%] - deformation (elongation) at break; L_{max} [mm] - length of the specimen after loading with the force F_{max} at which the break occurs;

$Z = (S_0 - S_{min}) / S_0 \cdot 100$ [%] - thinning (necking) at break; S_{min} [mm²] – cross-sectional area of the specimen at break after loading with force F_{max} [N]; this factor is frequently replaced in practice with the transverse contraction factor (Poisson) ν .

Areas of the characteristic curve:

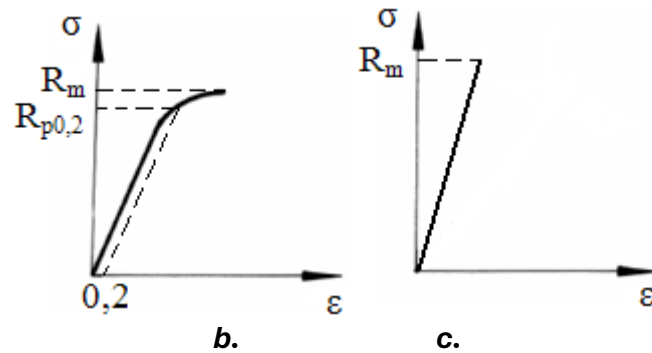


Fig. PRC-T.2.1.3.1 Characteristic curves: **a** – characteristic for materials (steels) with medium hardness (tough, ductile) with yield point; **b** – characteristic of materials without yield point (low toughness); **c** – characteristic of brittle materials

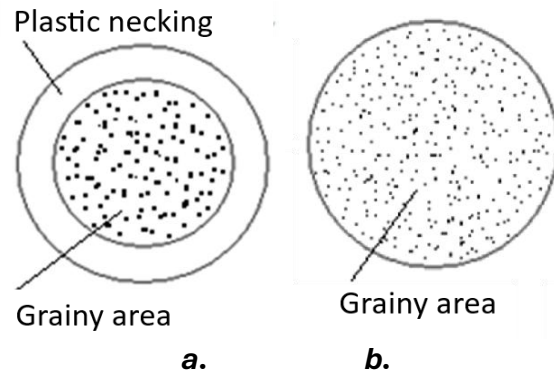


Fig. PRC-T.2.1.3.2 Failure modes: **a** – plastic (tough, ductile); **b** – brittle

OA - zone of linear elasticity, deformations are proportional to stresses, $\sigma = E \varepsilon$ (Hooke's law);
 AB - nonlinear elasticity zone, deformations are not proportional to stresses;
 BC - elasto-plastic zone, small permanent plastic deformations appear;
 CD - yield zone, permanent plastic deformations (CC' - yield point);
 DE - breaking zone

The characteristic curve points:

A - proportionality limit, σ_p ;

B - elastic limit, σ_e or R_{001} – technical elastic limit, the stress at which a residual deformation of 0.01% occurs for materials that are not perfectly elastic);

C - yield strength (resistance), σ_c - yield strength or $R_{p0.2}$ - technical yield strength (the stress at which a 0.2% residual deformation/elongation occurs) for materials that do not have a distinct yield point (fig. PRC-T.2.1.3.1,b);

D - breaking limit (resistance), σ_r or R_m , maximum stress at which breaking can occur;

E – final breaking point

X – current point in the yield zone with the ordinate, the stress σ_x and the ordinate, the strain $\varepsilon_x = \varepsilon_e + \varepsilon_p$ (ε_e – elastic strain, ε_p – plastic, permanent strain)

Note.

a. Materials that have the characteristic in fig. PRC-T.2.1.3.1,a are called tough (ductile) and have high plastic deformation capacity (e.g. low carbon steels; fig. PRC-T.2.1.3.2,a); materials that have the characteristic in fig. PRC-T.2.1.3.1,b have low toughness and rupture occurs after a small plastic deformation (e.g. hard steels); materials that have the characteristic in fig. PRC-T.2.1.3.1,c, called brittle (brittle), are devoid of plasticity and rupture occurs without plastic deformation (e.g. white cast irons, very hard steels, glass; fig. PRC-T.2.1.3.2,b)

b. The area under the characteristic curve represents the measure of the deformation energy stored in the material until rupture; its increased value indicates high toughness and, therefore, increased shock absorption capacity; toughness is more precisely demonstrated by the dynamic resilience test.

c. In the case of heat-treatable steels (especially those for improvement and carburization), different characteristic curves can be obtained following heat treatments.

d. In general, the static mechanical performances of steels are expressed (in norms or standards) by the values of the parameters: σ_c or $R_{p0.2}$ (yield strength); σ_r or R_m (tensile strength); A_r (elongation at break, plasticity); E (longitudinal modulus of elasticity); ν – transverse contraction factor (Poisson)

Metallic materials subjected to time-varying loads in which stresses below the yield strength occur can break due to fatigue, a process that develops over time by starting microcracks that appear in areas of stress concentration.

Tab. PRC-T.2.1.3.2 Durability curve σ - N (Wohler)

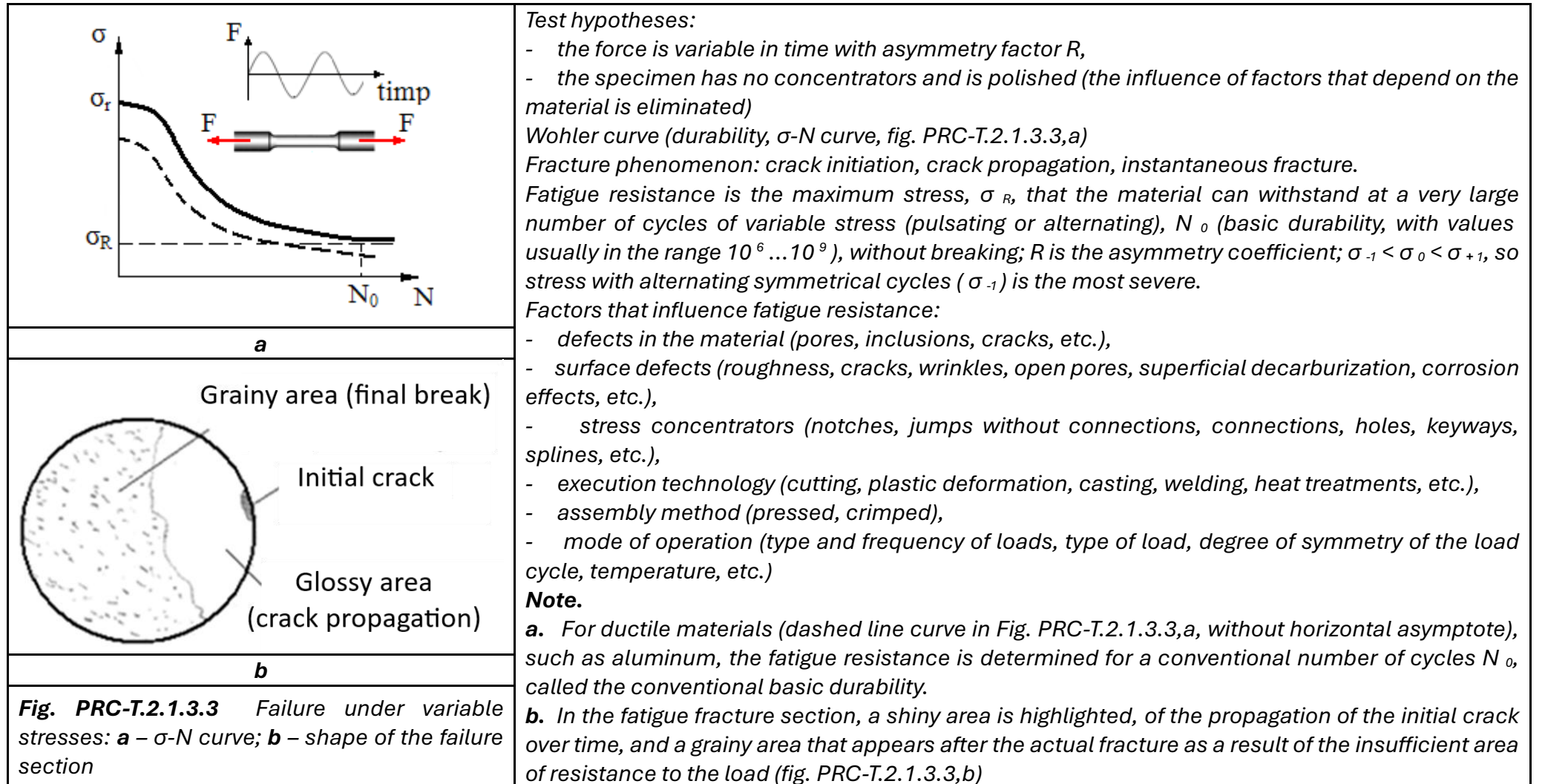


Fig. PRC-T.2.1.3.3 Failure under variable stresses: **a** – σ - N curve; **b** – shape of the failure section

Hardness, as a surface mechanical property, represents the ability of a material to resist a mechanical penetration action exerted from the outside by means of another solid body; Hardness is determined easily and quickly, being frequently used in industrial practice during technological processing (cutting, plastic deformation, heat treatments, etc.), for the indirect evaluation of mechanical properties of volume, highlighting the cutting capacity (materials with higher hardness can cut those with lower hardness), evaluating wear resistance, etc.

Tab. PRC-T.2.1.3.3 Hardness determination methods

The diagrams illustrate three common hardness testing methods: a) Brinell method: A spherical indenter of diameter D is pressed into a material under a constant normal force F. The resulting indentation has a diameter d. The hardness is calculated as $HB = F/S$, where S is the surface area of the indentation. b) Vickers method: A square-based pyramidal indenter with a 136° angle is pressed into a material under a constant normal force F. The resulting indentation is a square with diagonal length d. The hardness is calculated as $HV = F/S$, where S is the surface area of the indentation. c) Rockwell method: A conical indenter is used in three stages: loading with an initial load F_0, bringing it to zero, applying an overload $F_0 + F$ for a period of over 15s, removing the overload F, and measuring the residual penetration depth. The hardness is expressed as HRC or HRB, depending on the indenter used (cone or ball).	<u>The Brinell method</u> (Fig. PRC-T.2.1.3.3, a) involves determining the resistance of materials to the penetration of a hardened steel ball under the action of a constant normal force. The expression is made by the ratio of the force to the area of the cap (indentation) by measuring its diameter ($HB = F/S$). The method is frequently used to determine the hardness of non-heat-treated materials and heat-treated alloys (including steels) with <u>medium or low strengths</u> for which values of 100...450HB can be obtained. The correspondence between hardness and tensile strength, for unalloyed steels with (120-175) HB, $R_m \approx 0.34 HB$ and for alloyed steels with (175-450) HB, $R_m \approx 0.35 HB$ [Suciu, 2008] <u>Vickers method</u>, determination of the resistance of materials to the penetration of a hardened steel penetrator under the action of a constant normal force. The expression is made by the ratio of the force to the area of the square (indentation) by measuring its diagonal ($HV = F/S$). The Vickers method has a greater applicability than the Brinell method because it has the widest range (10...1900HV) but is not as frequently used; it is used to determine the hardness of hard metallic materials (cased, nitrided, chromed steels); it is not recommended for heat-treated materials and alloys with medium or low strengths. Usually, the Vickers method is applied for hardened surface layers. <u>The Rockwell method</u> (Fig. PRC-T.2.1.3.3,c), more operative than the others, involves expressing the hardness by evaluating the depth of the residual trace resulting from the penetration of a penetrator (usually conical). The measurement is made in three stages: loading with the initial load F_0 and bringing it to zero; applying the overload F for a period of over 15s; removing the overload F and measuring the residual penetration depth. The hardness HRC or HRB (cone or ball indenter, respectively) is conventionally established depending on the measured value. For hard materials, the diamond cone-shaped indenter is used, with an angle at the apex of 120. The method is recommended for hardness values between 25...67 HRC, mainly for volume-hardened or superficially hardened and tempered ferrous alloys. The
--	--

Fig. PRC-T.2.1.3.3 *Principal diagrams of hardness determination methods: **a** – Brinell; **b** – Vickers; **c** – Rockwell*

correspondence with the HB values can be made with the relation $1 \text{ HRC} \sim 10 \text{ HB}$ [Suciu, 2008]. For steels hardened by heat treatments, values of 45...65 HRC are obtained.

The Shore (elastic rebound) method is the most widespread apparatus based on elastic repulsion, consisting of a vertical tube in which a 3g metal indenter can move freely, ending with a diamond tip. The vertical tube is graduated in 140 equal divisions called Shore hardness units. The determination of hardness is very operative; the indenter is dropped onto the piece from a height H and the height h at which it rebounds is measured. This height represents the Shore hardness. The Shore method can only be used to determine the hardness of horizontal surfaces. The Poldi method is applied to heavy parts attached to foundations.

PRC-T.2.1.4 TECHNOLOGICAL CHARACTERISTICS AND PROPERTIES

The technological properties of metallic materials determine the behavior of metallic materials under the action of complex stresses developed during processing with different processing procedures (forging, rolling, welding, machining, etc.)

Plastic deformability (workability) is the property of metallic materials to easily and permanently modify their shape and dimensions at constant volume without cracking under the action of external cold and hot stresses; it is characterized by malleability, ductility and/or forgeability.

Castability is the property of metallic materials to be able to be cast and solidified into shapes; it is characterized by fluidity, shrinkage and/or segregation.

Weldability is the property of metallic materials to join permanently through local heating to a plastic or molten state, with or without the addition of filler materials and with or without the application of mechanical pressure.

Machinability is the ability of metallic materials to be machined by cutting operations with minimal tool wear and energy consumption.

Hardenability is the property of metallic materials (especially steels and cast irons) to achieve the required hardness over the greatest possible depth, being characterized by the depth of penetration of the hardening.

PRC-T.2.2 STEELS FREQUENTLY USED FOR THE CONSTRUCTION OF MACHINERY ELEMENTS

Tab. PRC-T.2.1.4.1 *General purpose non-alloy carbon steels*

Group	Symbol	Use
Structural steels - weldable (SR EN 10025/2005)	S185 ($R_p = 185 \text{ MPa}$)	General purpose metal structural elements subject to moderate loads (supports, frames, reinforcements, chains, flanges)
	S235 ($R_p = 235 \text{ MPa}$)	Metal construction elements welded or joined by other processes (welded frames, welded mesh for reinforced concrete, tanks, pillars, bridges, trusses, load-bearing structures of machines and equipment)
	S275 ($R_p = 275 \text{ MPa}$)	Welded construction elements subjected to relatively high mechanical stresses
	S355 ($R_p = 355 \text{ MPa}$)	Highly stressed metal construction elements (poles for overhead power lines, taxiways, cranes, vehicle chassis, high-capacity tanks)

Steels for mechanical construction (SR EN 10025/2005)	E295 ($R_{p02} = 295 \text{ MPa}$)	Mechanical construction elements subject to high stresses (drawbars, straight and crankshafts, pump and turbine shafts, crane hooks, vices, nuts, precision screws, gears for small peripheral gears)
	E335 ($R_p = 335 \text{ MPa}$)	Mechanical construction elements under higher loads (straight and crankshafts, precision screws, gears for moderate peripheral speeds)
	E360 ($R_p = 360 \text{ MPa}$)	Machine elements subject to wear (splined shafts, keys, couplings, worm wheels, transport augers, chain wheels, centering pins)

Tab. PRC-T.2.1.4.2 Quality non-alloy carbon steels

Group	Symbol	Use
Non-alloyed case-hardening (carburizing) steels	C10; C15; C16 (0.15%C)	Carbide-hardened parts with low core strength (ratchets, forks, guide pins, rocker arms, valves, discs, bushings and rollers for traction chains)
Non-alloyed steels for improvement (hardening and high tempering)	C35, C40 (0.40%C)	Medium-duty heat-treated parts (small crankshafts, connecting rods, welded hubs for press wheel cylinders)
	C45 (0.45%C)	Heat-treated parts of high strength and medium toughness (turbine discs, crankshafts, connecting rods, crown gears, ratchet wheels, flywheels, worms, etc.)
	C55 (0.55%C)	Highly stressed heat-treated parts (gears, belts, crowns, shafts, chain pins, etc.)
	C60 (0.60%C)	Heat-treated parts with high strength properties combined with elasticity elements (gears, belts, elastic bushes, eccentrics, etc.)

Tab. PRC-T.2.1.4.3 Quality carbon alloy steels

Group	Symbol	Use
Alloyed case-hardening steels	16MnCr5 (0.16%C, alloyed with Mn and 0.5%Cr); 17CrNi6; 20MnCrS5; 18CrNiMo7; 20NiCrMoS6	Case-hardened parts (carburization followed by quenching and low tempering) with increased surface resistance and soft core (gears, forks, connecting rods, etc.) machinable by grinding after quenching and low tempering
Alloyed steels for improvement	25CrMo4 (0.25%C, alloyed with Cr and 0.4%Mo); 30CrNiMo6; 34CrNiMo8, 41Cr4; 42CrMo4, 42CrMoS4, 51CrV4	High-strength parts (gears, assembly elements, screws, etc.) machinable by cutting after improvement treatment (hardening followed by high recovery)

Tab. PRC-T.2.1.4.4 Special alloy carbon steels

Group	Symbol	Use
Spring steels (EN10089)	38Si7; 46Si7; 56Si7; 60Cr3; 51CrV4	Various elastic elements (springs)
Bearing steels (SR EN 683-17)	100Cr6; 100CrMnSi6-4	Bearings and rolling elements

PRC-T.2.3 MAIN HEAT TREATMENTS APPLIED TO STEELS

PRC-T.2.3.1 ANNEALING

Annealing is the heat treatment applied to cast, welded, plastically deformed or mechanically processed parts, in order to establish an equilibrium structure, by relieving internal stresses and correcting defective microstructures.

Annealing type	Description
Stress relief	<i>It is applied to cast, welded or cold-formed products to remove (partially or completely) residual stresses if it is not followed by another heat treatment that aims at structural changes. It is performed by heating at 550...650 ° C for 6...8 h</i>
Homogenization	<i>It is applied to cast products that show chemical segregation to remove chemical inhomogeneities. Chemical homogenization occurs by diffusion and therefore the heating temperature must be very close to the real solidus temperature: $T_{rec} = 0.9 \cdot T_s$.</i>
Recrystallization	<i>It is applied to eliminate work hardening as an intermediate step in the cold plastic deformation process. The products are cooled in air. The cooling of the products is done in air</i>
Normalization	<i>It is mainly applied to hypereutectoid and hypoeutectoid steels to refine the grain structure.</i>

PRC-T.2.3.2 HARDENING

Hardening consists of a sequence of heating operations as rapidly as possible to temperatures above the critical points of structural transformation, holding at these temperatures (for a duration sufficient for the structural transformations to occur) and high-speed cooling.

Simple hardening is performed in a single cooling medium: water, oil, synthetic media, molten salts, molten metals, etc. It is not recommended for large steel parts or steels with a C content greater than 0.8%, as the risk of distortion and cracking is high.

Interrupted quenching (in two cooling media) is performed first in a medium with high cooling capacity and then in a medium with lower cooling power. Generally, the first medium is water and the second is oil.

PRC-T.2.3.3 TEMPERING

Tempering is applied to hardened parts, regardless of the technology adopted, with the aim of achieving a balanced internal structure by heating at low temperatures (150 ...400 ° C) and very slow cooling.

Tempering type	Description
low	<i>It is applied to volume or superficially hardened products by heating at 150-250 ° C, aiming to obtain tempered martensite with high hardness and reduced internal stress.</i>
average	<i>It is applied mainly to spring steels by heating at 250...400 ° C, aiming to obtain <u>tempering</u> with elastic properties.</i>
high	<i>It is mainly applied to heat-treating steels to ensure increased toughness and machinability.</i>

Note: *Improvement heat treatment (quenching and tempering) involves quenching followed by high-temperature tempering and applies to (0.3...0.6)%C unalloyed and alloyed tempering steels.*

PRC-T.2.3.3 THERMOCHEMICAL TREATMENTS

Thermochemical treatment involves changes in the chemical composition and structure of the surface layer of a part by enriching the surface layer (through heating in chemically active environments) with chemical elements (carbon, nitrogen) and subsequent heat treatment (usually quenching and tempering) with the aim of surface hardening for improved resistance to contact stress, wear, seizure, corrosion, and/or fatigue.

Carburizing involves enriching the surface layer of steel parts with carbon (by 0.25...0.3%C) by heating in a gaseous atmosphere or in salt baths up to 0.8...1%C over a thickness of 0.5...1.5 mm; as a result of quenching and low tempering (carburizing), a hardness of 55...60HRC is obtained in the surface layer.

Carbonitriding (cyanidation) involves enriching the surface layer of steel parts with carbon and nitrogen (with 0.25...0.3%C) by heating in an environment rich in C and N.

Nitriding involves enriching the surface layer of steel parts with nitrogen (with 0.25...0.3%C) by heating in a nitrogen-rich environment.

PRC-T.3 CALCULATION OF STRENGTH AND RIGIDITY (DEFORMATION) OF MACHINE ELEMENTS

PRC-T.3.1 CALCULATION OF RESISTANCE OF MACHINE ELEMENTS

PRC-T.3.1.1 GENERAL STRUCTURE OF INTERACTING BODIES SYSTEMS

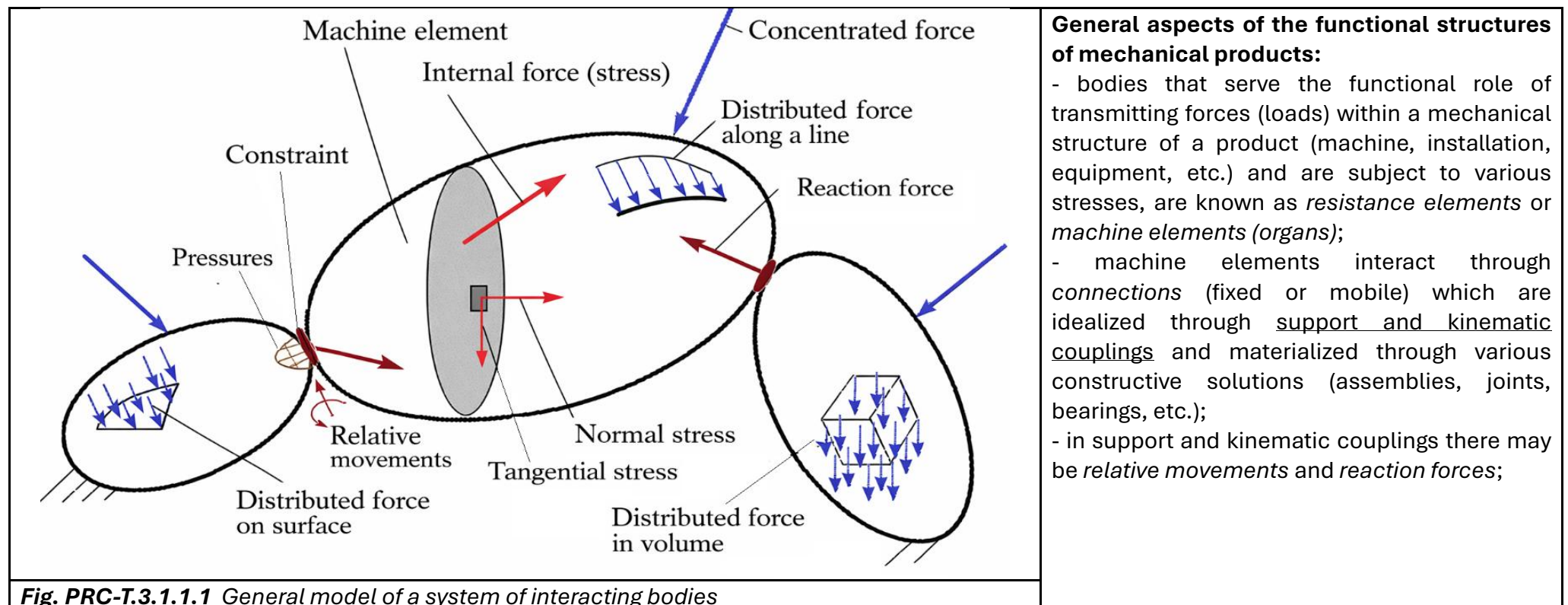


Fig. PRC-T.3.1.1.1 General model of a system of interacting bodies

- because of the action of external and internal loads, *tensions (normal and/or tangential)* appear inside the bodies and *contact pressures* appear on the surfaces.
- machine elements can be *loaded with external loads, forces and/or moments, concentrated or distributed* (online, on surface and/or in volume).
- the stresses in the structural elements are integrated at the interior points into *internal forces (efforts)* and at the contact surface points in the supports into *reaction forces*

PRC-T.3.1.2 GEOMETRIC MODELING OF MECHANICAL STRUCTURE ELEMENTS

The *constructive elements* of mechanical products, considering the ratio between their dimensions, can be reduced (simplified) to three categories: bars (rigid or flexible), with a much larger dimension compared to the other two; plates with two dimensions much larger than the third; massive elements (blocks) with the three comparable dimensions (balls, wheels, flanges, thick-walled tubes, flywheels, etc.).

Tab. PRC-T.3.1.2.1 Classification of bar-type structural elements

Criterion	Type	Scheme	Description
The shape of the average fiber (the locus of the centers of gravity of the sectional shapes)	straight	Fig. PRC-T.3.1.2.2,a	They materialize as shafts, bolts, screws, etc.
	Cranked	Fig. PRC-T.3.1.2.2,b	They materialize as crankshafts
	curve	Fig. PRC-T.3.1.2.2,c,d	They materialize as rigid elements (tow hook) or flexible
Cross-sectional shape	With solid section	Fig. PRC-T.3.1.2.1,a	They can be circular, square, rectangular, hexagonal, etc. in section.
	With open section	Fig. PRC-T.3.1.2.1,b	They can be with semicircular, U, I, L, etc.
	With hollow section	Fig. PRC-T.3.1.2.1,c	With circular, square, rectangular profile, composed of simple shapes, etc.
Main request type	tie	Fig. PRC-T.3.1.2.1,a	The main stress is traction (stretching)
	Wires, cables	Fig. PRC-T.3.1.2.1,d	
	Pillars (columns)	Fig. PRC-T.3.1.2.1,b	The main requirement is compression
	beams	Fig. PRC-T.3.1.2.1,c	The main requirement is bending.
	Shafts	Fig. PRC-T.3.1.2.1,a,b	The main stresses for rigid shafts are: torsion and bending

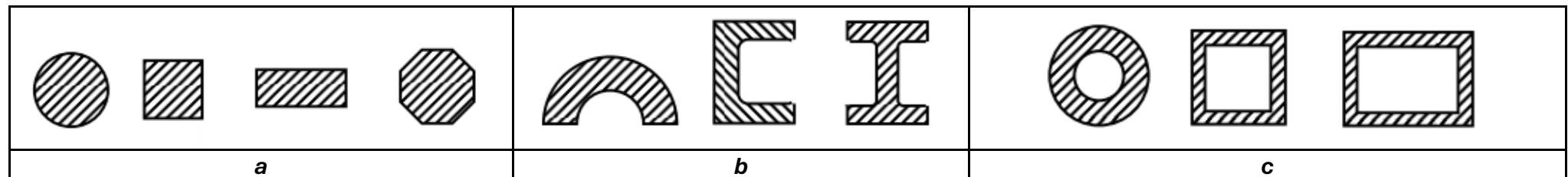
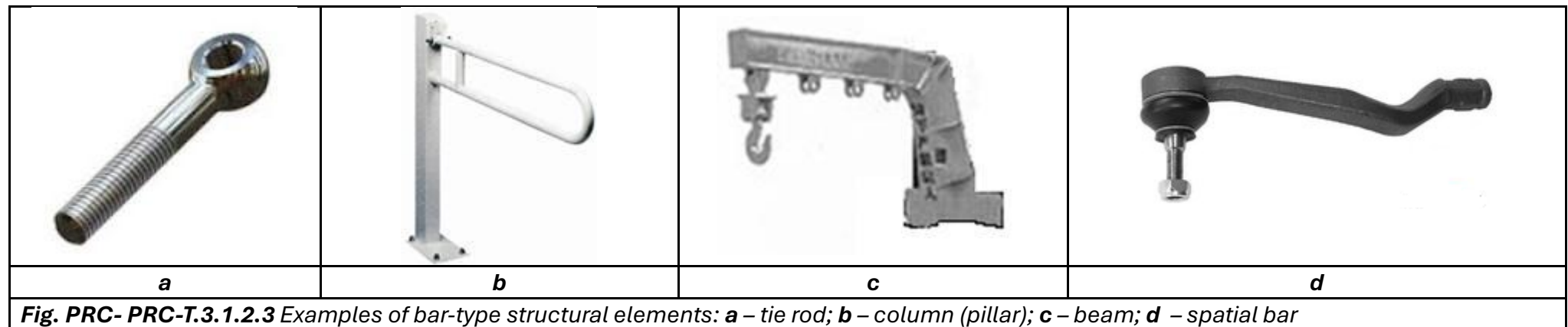
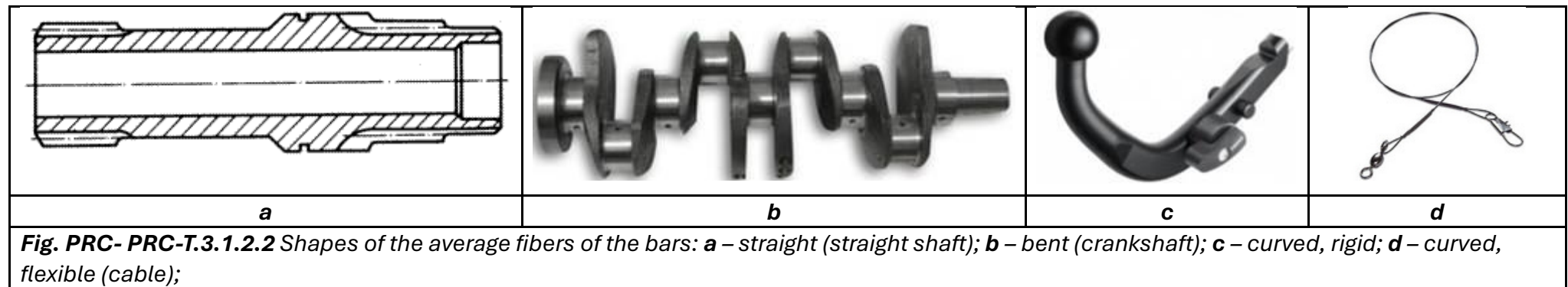


Fig. PRC-T.3.1.2.1 Bar section shapes: **a** – solid; **b** – open; **c** – tubular (pipe)



Tab. PRC-T.3.1.2.2 Classification of plate-type structural elements

Criterion	Type	Scheme	Description	The diagram shows a 3D model of a plate with a curved median surface. The surface is defined by a dashed line. The thickness of the plate is indicated as $g/2$ at two points. The label 'Suprafața mediană' (Median surface) points to the dashed line.
Shape of the median surface	flat	Fig. PRC-T.3.1.2.4	The median surface is flat.	
	curve		The median surface is spatial	
depth	Membranes (thin plates)		They have small thicknesses and do not take on transverse forces.	
	roof		They have very small thicknesses and do not take up bending moments.	
	Tiles (thick slabs)		They have large thicknesses and small median surface area dimensions	

Fig. PRC-T.3.1.2.4 Description of the plates

PRC-T.3.1.3 MODELING OF CONNECTIONS (SUPPORTS) BETWEEN BODIES

Connections model the interactions between two constructive elements through supports which, in turn, are materialized through specific or specialized connection elements or subassemblies (assemblies, bearings, joints); Connections can be fixed, without relative movements (only with reactions) or mobile, with relative movements (kinematic couplings in which reactions occur in the direction of the canceled movements).

Tab. PRC-T.3.1.3.1 *Classification of supports*

Criterion	Type	Scheme	Description
Geometric domain	flat	Fig. PRC-T.3.1.3.1,a,b,c,d	Displacements and rotations along one of the transverse axes are zero; the stiffnesses of the elements in this direction are infinite.
	space	Fig. PRC-T.3.1.3.1,e,f,g	They can be rigid or elastic
Mobility degrees canceled	With a constraint	Fig. PRC-T.3.1.3.1,a,b	Allow 5 relative movements for spatial supports and 2 for planar ones
	With 2 constraints	Fig. PRC-T.3.1.3.1,c,e	They allow 4 relative movements in the case of spatial supports and one for planar ones
	With 3 constraints	Fig. PRC-T.3.1.3.1,d,f	Allow 3 relative movements in the case of spatial supports
	With 4 constraints	-	Allow 2 relative movements in the case of spatial supports
	With 5 constraints	-	Allow relative movement for spatial supports
	With 6 constraints	Fig. PRC-T.3.1.3.1,g	They do not allow relative movements (they are materialized by non-demountable assemblies and joints)
Stiffness of the supported elements	Rigid	Fig. PRC-T.3.1.3.1,b ... g	Elastic deformations in the connections are not taken into account.
	SPRING	Fig. PRC-T.3.1.3.1,a	Elastic deformations are taken into account by means of elastic elements

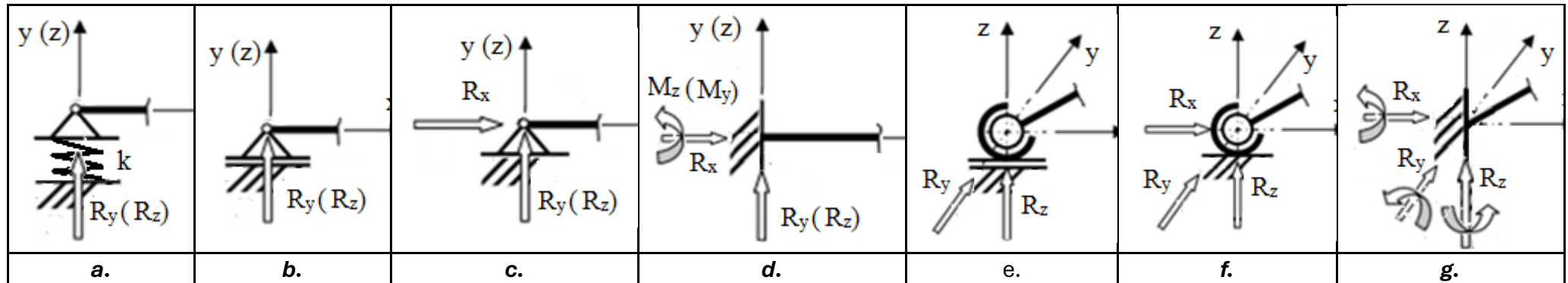


Fig. PRC-T.3.1.3.1 Common types of supports: **a** – plane mobile elastic support; **b** – plane mobile support; **c** – plane fixed support (cylindrical joint); **d** – plane embedment; **e** – spatial mobile support; **f** – spatial fixed support (spherical joint); **g** – spatial embedment

PRC-T.3.1.4 LOAD MODELING

PRC-T.3.1.4.1 MODELING OF EXTERNAL LOADS (LOADS)

Tab. PRC-T.3.1.4.1.1 Classification of external loads

Criterion	Type	Scheme	Description
The field in which it operates	External (tasks)	Fig. PRC-T.3.1.4.1	External loads are the result of the action of physical fields (mechanical, thermal, electromagnetic)
	Interiors (efforts)	Fig. PRC-T.3.1.4.2	It acts inside bodies and is determined by the sum of internal tensions
	Linking (reactions)	Fig. PRC-T.3.1.3.1	They act as reaction forces and are determined by summing the pressures on the contacting surfaces.
The support on which it acts	On point (concentrated)	Fig. PRC-T.3.1.1.1	It acts on very small surfaces that can theoretically be reduced to a point
	On the line		It acts on a long and narrow surface that can be reduced to a line
	On the surface		Acts on external and internal surfaces of construction elements
	In volume		They can act, rarely, only in certain parts of the body volume or, frequently, in the entire volume (gravity, centrifugal, electromagnetic forces, etc.)
Distribution on the supporting material that acts	Evenly distributed	Fig. PRC-T.3.1.1.1	They are valid for loads applied to contours, surfaces and volumes.
	Unevenly distributed		
	Static	Fig. PRC-T.3.1.4.1,a	They are considered to be applied slowly and do not involve dynamic effects.

Application method over time	Dynamics applied suddenly (with shock)	Fig. PRC-T.3.1.4.1,d	They are applied in short times and are considered dynamic (inertial) effects.
	Periodic variables	Fig. PRC-T.3.1.4.1,b	Variations in time, periodic, pulsating or alternating, have effects on fatigue resistance during extended operating periods (durability)
	Random variables	Fig. PRC-T.3.1.4.1,c	Random variations over time have effects on fatigue resistance over extended operating periods (durability)
Mobility of the support on which it acts	fixed	-	The action of these loads remains invariable (linear resistance calculations)
	Mobile	-	For design calculations, of all possible loading conditions, the one that produces the most unfavorable test is considered.

Note: Reactions from supports (fig. PRC-T.3.1.3.1), as indirect external loads, are frequently modeled as concentrated forces and moments.

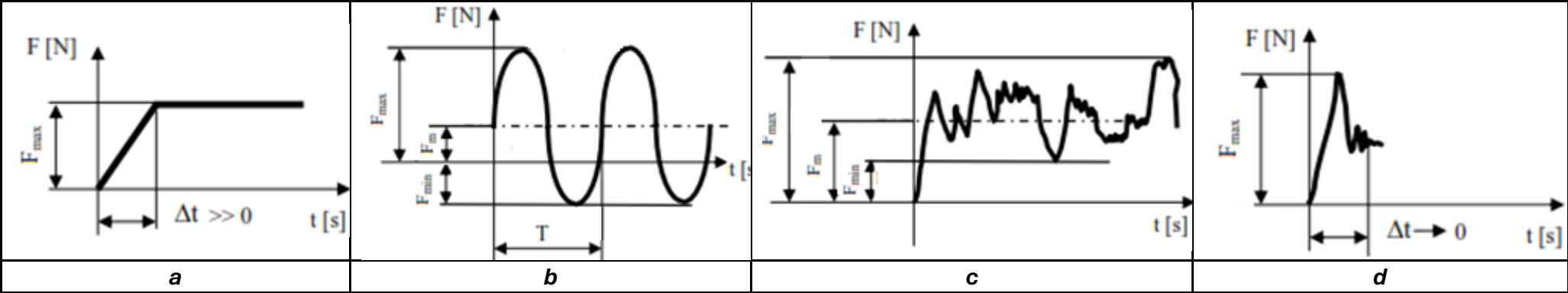


Fig. PRC-T.3.1.4.1.1 Types of external loads (forces or moments): **a** – static; **b** – periodically variable; **c** – randomly variable; **d** – with shock;

PRC-T.3.1.4.2 INTERNAL LOADS (EFFORTS) MODELING

Tab. T.3.1.4.1.2 Classification of sectional forces

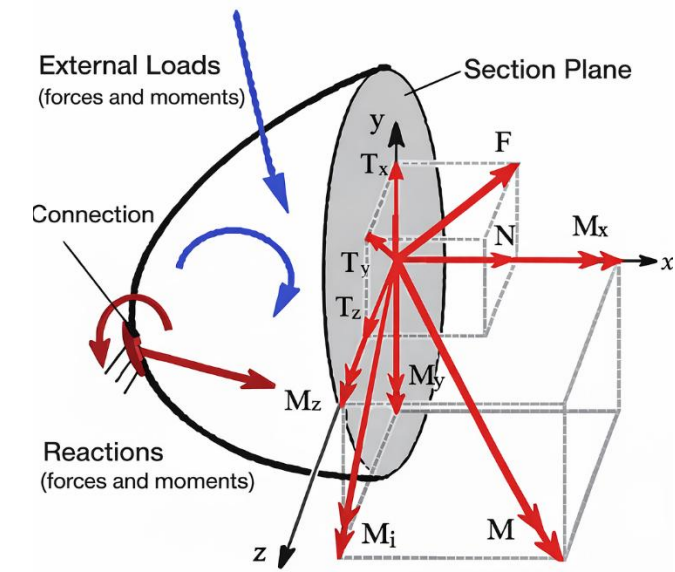
	Criterion	Type of effort	Description
	Position relative to the section plane	Normal (axial)	N, M_x - have the direction along the x-axis, normal to the section plane
		Tangential	$T_{y,z}, M_{y,z}, T, M_i$ - have the direction in the section plane, yz
Related requests		Tensile/compression	N – <u>axial tensile/compressive force</u>
		Shear	$T_{y,z}$ – <u>shear stresses</u> , $T = \sqrt{T_y^2 + T_z^2}$ – <u>resultant shear stress</u>
		Torsion (axial)	M_x – <u>axial torsional moment</u> ;
		Bending	$M_{y,z}$ – <u>bending stresses</u> , $M_i = \sqrt{M_y^2 + M_z^2}$ – <u>resultant bending stress</u>
Consideration as a component or as a result		Partial (component)	$T_{y,z}, M_{x,y,z}$ – components of the forces T and M
		Resultant	$F = \sqrt{N^2 + T_y^2 + T_z^2}$ – <u>inner strength</u> $M_i = \sqrt{M_y^2 + M_z^2 + M_x^2}$ – <u>the inner moment</u>

Fig. PRC-T.3.1.4.2.1 Components of sectional forces

PRC-T.3.1.5 REQUEST MODELING

Assumptions for resistance calculations:

- continuity, according to which bodies are considered continuous, regardless of their crystalline structure;
- isotropy, according to which materials have the same properties in any direction (which passes through a certain point);
- homogeneity, bodies being considered homogeneous, with the same properties;
- small displacements, according to which the displacements of different points of the body are much smaller compared to its dimensions;
- perfect elasticity, according to which bodies return to their initial shape after the external load ceases;

- the linear dependence between force and displacement $F = k \delta$, where k = constant (stiffness);
- superposition of effects, according to which the final state of equilibrium does not depend on the order of application of the loads;
- the constancy of the cross-section, to simplify the calculation relationships;
- negligible demands, to simplify the calculation relationships, demands that prove insignificant are omitted;
- Saint-Venant, according to which in a section far from the loading area of the body, the stress does not depend on the loading mode;
- Bernoulli's, according to which a plane section normal to the axis of the resistance element remains plane and normal to its deformed axis

Tab. PRC-T.3.1.5.1 Classification of mechanical stresses

Criterion	Request type	Scheme	Description
The geometric domain in which it manifests itself	By volume	Table PRC-T.3.1.5.1	They are the subject of study of the strength of materials.
	Surface	Table PRC-T.3.1.5.2	They are the subject of study of material strength and tribology.
By the number of efforts in a section	Simple	Table PRC-T.3.1.5.1.1.1	They have a single effort (simple requests)
	Compounds	Table PRC-T.3.1.5.1.2.1	Have at least two efforts (compound demands)
Type of loads	Static	Fig. PRC-T.3.1.4.1.1,a	The load is applied slowly and is considered time independent.
	Dynamic	Fig. PRC-T.3.1.4.1.1,d	The load is variable in time; it is considered for short periods of time (very short), usually for calculations in transient regimes (shock or impact)
	Variables	Fig. PRC-T.3.1.4.1.1,b,c	The load is variable in time (periodic or random); long periods of time (comparable to the product's lifetime) are considered for the fatigue calculation; depending on the material, failures (ruptures) occur at stresses lower than the stress corresponding to the yield (rupture) limit for static loading

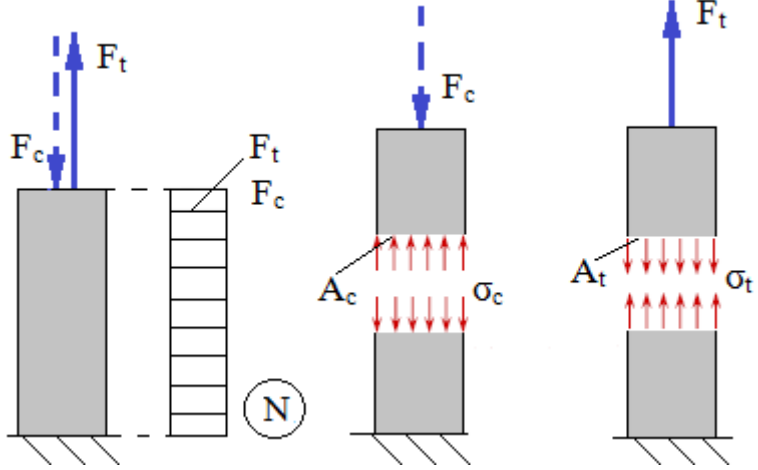
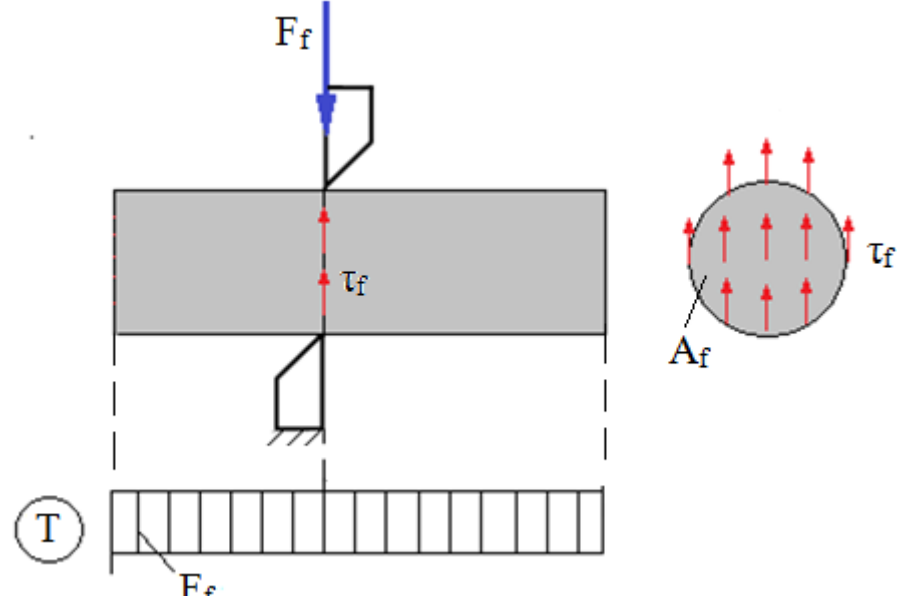
PRC-T.3.1.5.1 MODELING OF VOLUME REQUESTS

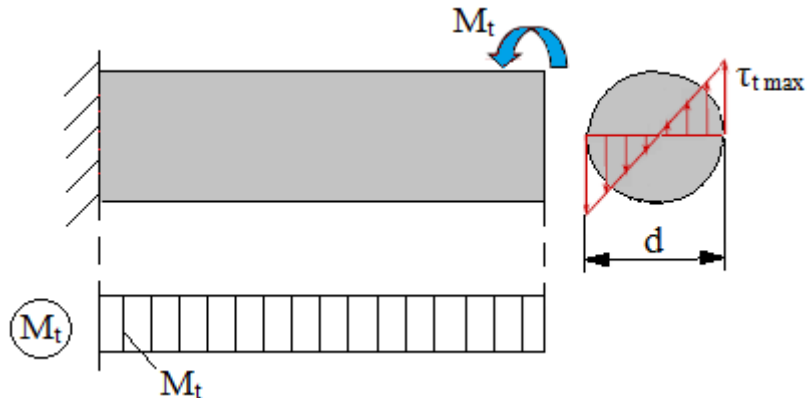
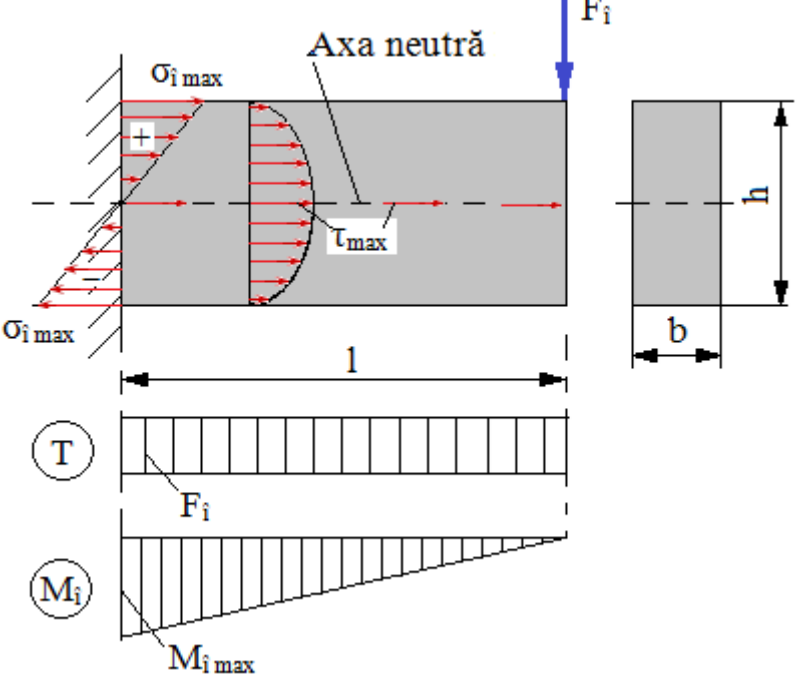
PRC-T.3.1.5.1.1 Modeling simple volume demands

Simple requests are determined by a single effort.

Tab. PRC-T.3.1.5.1.1.1 Simple mechanical loads

Request	Scheme	The mathematical model		Parameter meanings
		Scope	Calculation relationships	

Traction, compression		Verification	$\sigma_{t,c} = \frac{F_{t,c}}{A_{t,c}} \leq \sigma_{at,c}$	$F_{t,c}$ [N] – tensile, compressive force; $A_{t,c}$ [mm ²] – the cross-sectional area of the bar subjected to tension, compression; $\sigma_{t,c}$ [MPa] – tensile stress, maximum compression; $\sigma_{at,c}$ [MPa] – allowable tensile, compressive stress; $F_{t,c\,cap}$ [N] – tensile force, capable compression
Pure shearing (cutting)		Verification	$\tau_f = \frac{F_f}{A_f} \leq \tau_{af}$	
		sizing	$A_f = \frac{F_f}{\tau_{af}}$	
		Capable load	$F_{f\,cap} = A_f \tau_{af}$	F_f [N] – shear force; A_f [mm ²] – shear section area; τ_f [MPa] – maximum shear stress; τ_{af} [MPa] – allowable shear stress; $F_{f\,cap}$ [N] – shear force capable
Torsion (twisting)		Verification	$\tau_{t\,max} = \frac{M_t}{W_p} \leq \tau_{at}$	M_t [Nmm] – torsional moment (twisting);

		sizing	$W_p = \frac{M_t}{\tau_{at}}$	W_p [mm ³] – polar modulus of resistance; $W_p = \frac{\pi d^3}{16}$, for circular section; $\tau_{t \max}$ [MPa] – maximum torsional stress; τ_{at} [MPa] – allowable torsional stress; $M_{t \text{ cap}}$ [Nmm] – capable torque
Bending		Verification	$\sigma_{i \max} = \frac{M_{i \max}}{W_z} \leq \sigma_{ai}$ $\tau_{\max} = \frac{3 T}{2 A} \leq \tau_{af}$	$M_{i \max}$ [Nmm] – maximum bending moment; F_i [N] – bending force; W_z [mm ³] – axial modulus of resistance;
		sizing	$W_z = \frac{M_{i \max}}{\sigma_{ai}}$	$W_z = \frac{\pi d^3}{32}$, for circular section; $W_z = \frac{b h^2}{6}$, for rectangular section;
		Capable load	$M_{i \text{ cap}} = W_z \sigma_{ai}$	$\sigma_{i \max}$ [MPa] – maximum bending stress, from Navier's relation; $\tau_{\max}$ [MPa] – maximum shear stress, from Juravski's relation; $A = bh$ [mm ²] – cross-sectional area of the bar; σ_{ai} [MPa] – allowable bending stress;

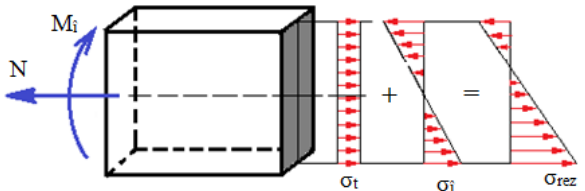
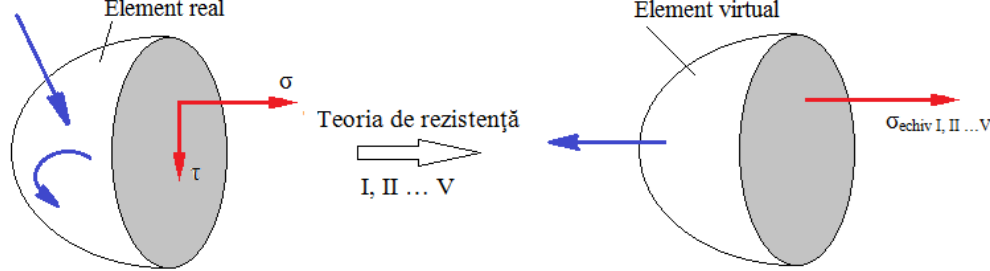
				$M_{i \text{ cap}}$ [MPa] – capable bending moment
--	--	--	--	--

PRC-T.3.1.5.1.2 Modeling of compound volume demands

Compound loads are caused by two or more stresses.

Tab. PRC-T.3.1.5.1.2.1 Classification of compound requests

critereon	Type	Scheme	Description
Type of participating stresss	Of the same nature (σ or τ) with parallel directions	Fig. PRC-T.3.1.5.1.2.1	If the stresses have the same direction, the resultant stress is calculated as the algebraic sum of the component stresses, $\sigma_{rez} = \sum_1^n \sigma_i, \quad (\text{PRC-T.3.1.5.1.2.1}) \quad \tau_{rez} = \sum_1^n \tau_i$ Possible loads: eccentric axial, oblique bending, axial (tension or compression) and bending; shear with torsion (twisting)
	Of different nature (σ and τ) with different directions	Fig. PRC-T.3.1.5.1.2.2	In this case, it is necessary to determine an <u>equivalent stress</u> that appears in an imaginary (virtual) element subjected to tension, made of the same material, and which produces a <u>stress state with stress σ that is as dangerous (equivalent)</u> as the real one with stresses σ and τ . <u>The equivalent stress</u> is determined, based on equivalence criteria called resistance theories (tab. PRC-T.3.1.5.1.2.2) and is compared with allowable stresses (PRC-T.3.1.5.1.2.1) based on limit stresses, easily determined experimentally. <u>Possible compound loads</u> : bending and twisting; tension, bending and twisting

	
Fig. PRC-T.3.1.5.1.2.1 Summation of stresses of the same nature (σ)	Fig. PRC-T.3.1.5.1.2.2 Equivalence of composite stress states

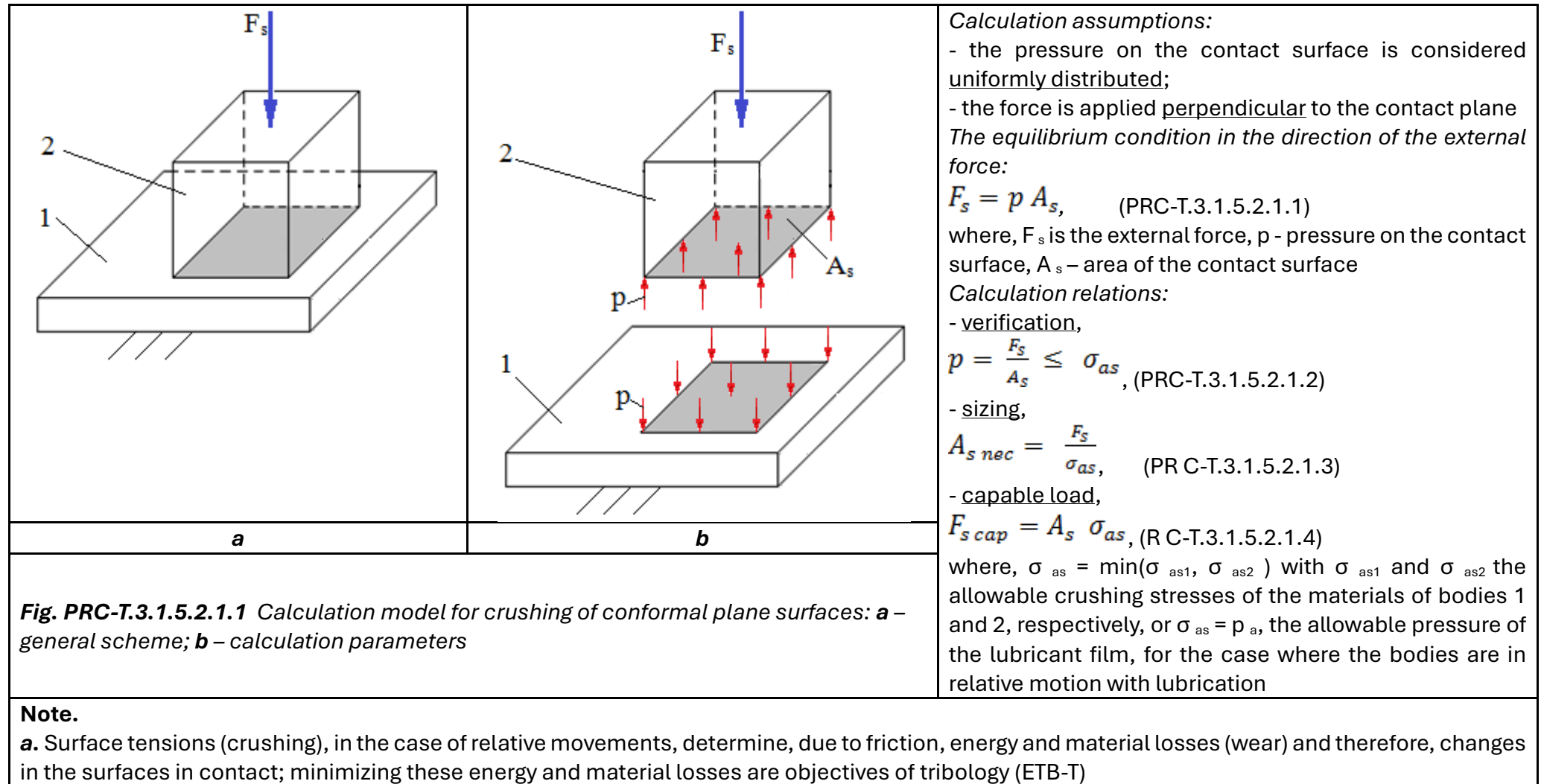
Tab. PRC-T.3.1.5.1.2.2 Resistance theories

Theory no.	Equivalence hypothesis	Equivalence relation	Description
1	Maximum normal stress	$\sigma_{echiv I} = 0,5\sigma + 0,5\sqrt{\sigma^2 + 4\tau^2}$ (PRC-T.3.1.5.1.2.1)	It has been experimentally confirmed in the case of fracture of brittle materials under tensile stress; in the case of composite stress states it leads to oversizing.
2	Maximum linear deformation	$\sigma_{echiv II} = 0,35\sigma + 0,65\sqrt{\sigma^2 + 4\tau^2}$ (PRC-T.3.1.5.1.2.2)	Recommended for fragile materials
3	Maximum tangential stresses	$\sigma_{echiv III} = \sqrt{\sigma^2 + 4\tau^2}$ (PRC-T.3.1.5.1.2.3)	It has been experimentally verified for tough materials under simple and mixed tensile stresses; this theory is the most comprehensive, leads to larger differences (dimensions) and is frequently used for design calculations.
4	Strain energy	$\sigma_{echiv IV} = \sqrt{\sigma^2 + 2,6\tau^2}$ (PRC-T.3.1.5.1.2.4)	-
6	Shape-modifying strain energy	$\sigma_{echiv V} = \sqrt{\sigma^2 + 3\tau^2}$ (PRC-T.3.1.5.1.2.5)	It has been experimentally verified for tough materials and corresponds better to reality than all other theories; this theory has a wide use in design because it leads to smaller differences (dimensions close to those required)

PRC-T.3.1.5.2 MODELING OF SURFACE STRESSES

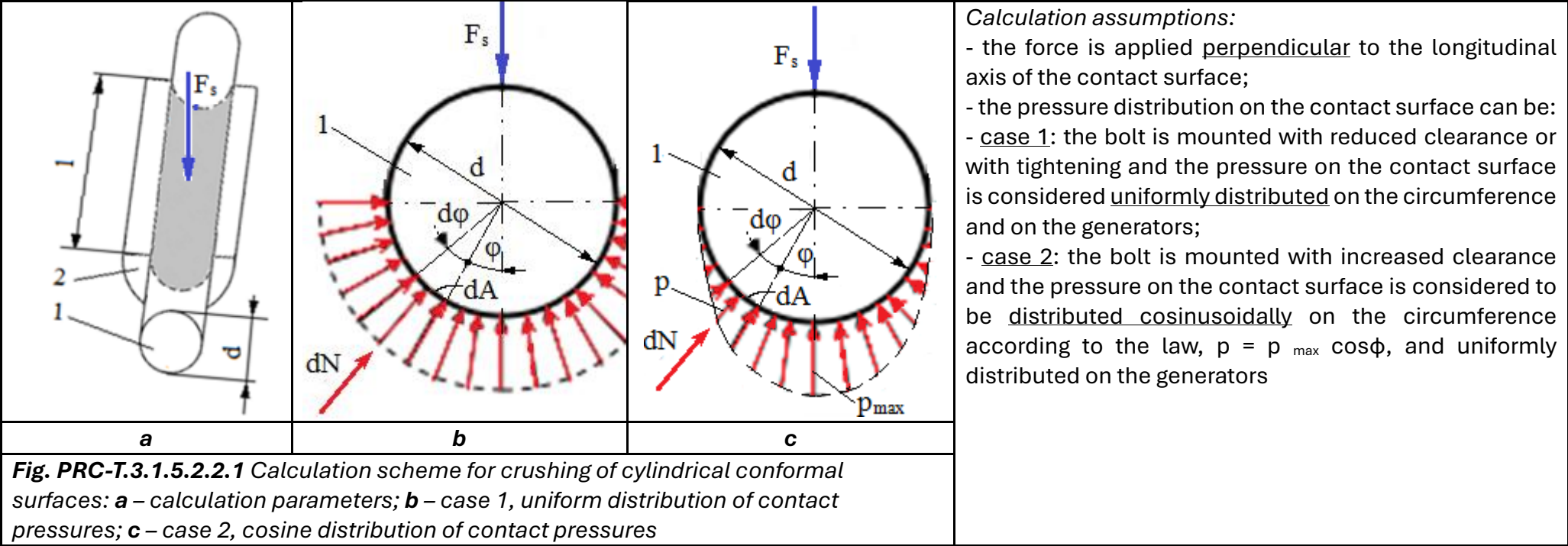
PRC-T.3.1.5.2.1 Modeling of crushing stresses on plane conformal surfaces

Tab. PRC-T.3.1.5.2.1.1 *Crushing calculation of plane conformal surfaces*



PRC-T.3.1.5.2.2 Modeling of crushing stresses on cylindrical conformal surfaces

Tab. PRC-T.3.1.5.2.2.1 Crushing calculation of cylindrical conformal surfaces



Case 1. Force balance condition:

$$F_s = 2 \int_0^{\frac{\pi}{2}} dN \cos \varphi = 2 \int_0^{\frac{\pi}{2}} p \, dA \cos \varphi = 2 \int_0^{\frac{\pi}{2}} p \, l \frac{d}{2} \, d\varphi \cos \varphi = p \, d \, l \int_0^{\frac{\pi}{2}} \cos^2 \varphi \, d\varphi$$

(PRC-T.3.1.5.2.2.1)

Calculation relations:

- verification,

$$p = \frac{F_s}{A_s} = \frac{F_s}{d \, l} \leq \sigma_{as}, \quad (\text{PRC-T.3.1.5.2.2.2})$$

- sizing,

$$d \, l = \frac{F_s}{\sigma_{as}}, \quad (\text{PRC-T.3.1.5.2.2.3})$$

- capable load,

$$F_{s \, cap} = d \, l \, \sigma_{as}, \quad (\text{PRC-T.3.1.5.2.2.4})$$

where, $A_s = d \, l$ is the conventional crushing area of the projection of the contact surface in the axial direction; $\sigma_{as} = \min(\sigma_{as1}, \sigma_{as2})$ with σ_{as1} and σ_{as2} the allowable crushing stresses of the materials of bodies 1 and 2, respectively, and $\sigma_{as} = p_a$ [MPa], the allowable pressure of the lubricant film.

Case 2. Force balance condition:

$$F_s = 2 \int_0^{\frac{\pi}{2}} dN \cos \varphi = 2 \int_0^{\frac{\pi}{2}} p \, dA \cos \varphi = 2 \int_0^{\frac{\pi}{2}} p_{max} \, l \frac{d}{2} \, d\varphi (\cos \varphi)$$

$$p_{max} \, d \, l \int_0^{\frac{\pi}{2}} (\cos \varphi)^2 \, d\varphi = \frac{\pi}{4} \, d \, l \, p_{max}$$

$$= \approx 0.8 \, d \, l \, p_{max}, \quad (\text{PRC-T.3.1.5.2.2.5})$$

Calculation relations:

- verification,

$$p_{max} = \frac{F_s}{A_s} = \frac{F_s}{0.78 \, d \, l} \leq \sigma_{as}, \quad (\text{PRC-T.3.1.5.2.2.6})$$

- sizing,

$$d \, l = \frac{F_s}{0.78 \, \sigma_{as}}, \quad (\text{PRC-T.3.1.5.2.2.7})$$

- capable load,

$$F_{s \, cap} = 0.78 \, d \, l \, \sigma_{as}, \quad (\text{PRC-T.3.1.5.2.2.8})$$

where, $A_s = 0.8 \, d \, l$ represents 0.8 of the area of the axial rectangle as a projection of the contact surface in the axial direction; $\sigma_{as} = \min(\sigma_{as1}, \sigma_{as2})$ with σ_{as1} and σ_{as2} the allowable crushing stresses of the materials of bodies 1 and 2, respectively, and $\sigma_{as} = p_a$ [MPa], the allowable pressure of the lubricant film.

PRC-T.3.1.6 ALLOWABLE STRESS AND SAFETY COEFFICIENTS

Tab. PRC-T.3.1.6.1 Strength conditions to avoid damage and failure

critereon	Relations	Description
Damage by exceeding the yield strength	$\sigma_{max} \leq \sigma_a, \sigma_{rez} \leq \sigma_a;$ $\tau_{max} \leq \tau_a, \tau_{rez} \leq \tau_a;$	$\sigma_a = \sigma_{02} / c$ ($\sigma_a = \sigma_c / c$) – <u>allowable stress related to the yield strength</u> for ductile materials; usually, $c = 1.2...3$ – <u>safety factor</u> to avoid damage by plastic deformation
Damage by exceeding the breaking limit	$\sigma_{echiv} \leq \sigma_a$	$\sigma_a = \sigma_r / c$ – <u>allowable stress related to the tensile strength</u> for brittle materials; usually, $c = 3...4$ – <u>safety factor</u> to avoid failure by rupture

Note.

a. The safety coefficient is a parameter that depends on the type of material, the technology of obtaining the semi-finished product, the size and type of applied loads, the operating mode, the calculation model (simple calculation models involve many simplifying assumptions and therefore a safety coefficient with high values), the working conditions and environment, etc.

b. $\tau_a = 0.8 \, \sigma_a$, for a large category of materials

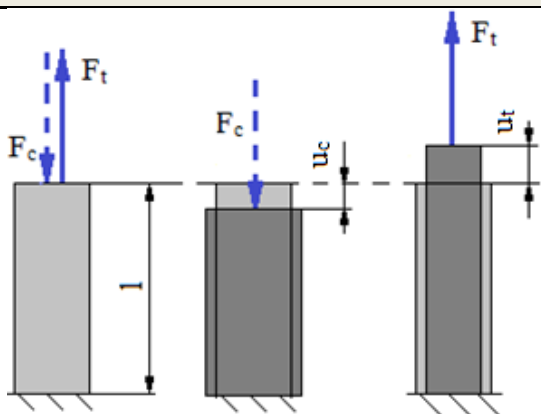
PRC-T.3.2 CALCULATION OF RIGIDITY (DEFORMATIONS) OF MACHINE ELEMENTS

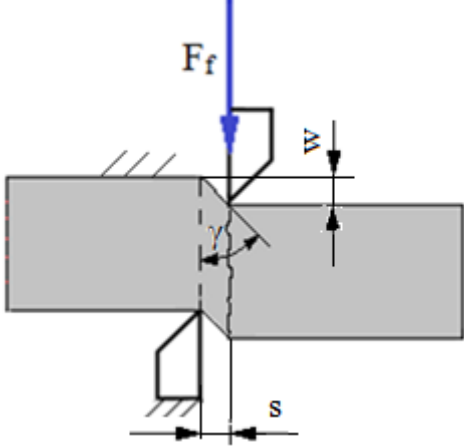
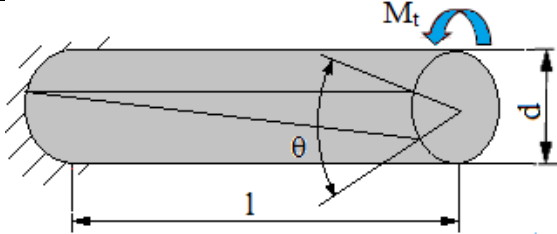
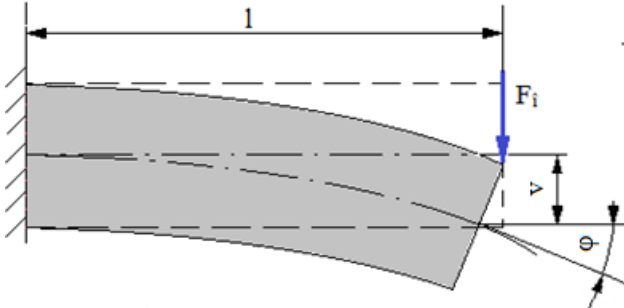
Elastic (with return to the initial state) *and/or plastic* (without return to the initial state) *deformations* represent measures (values) of the change (increase or decrease) of the distance between two points or two sections (linear deformations), respectively of the angles between two sections (angular deformations or slips) of a structure.

Displacements or arrows (linear or angular) represent values of the distance, respectively, of the angles between the positions (usually initial and final) of a point or of a section of a structure; displacements as a consequence of the deformation (change in shape) of the body can be: elastic (disappear after the load is removed), elasto-plastic or plastic.

PRC-T.3.2.1 MODELING OF DEFORMATIONS AND VOLUME DISPLACEMENTS

Tab. PRC-T.3.1.5.1.1.1 Deformations of mechanical structures under simple stresses

Request	Scheme	The mathematical model		Parameter meanings
		Scope	Calculation relationships	
Traction, compression		Verification	$u_{t,c} = \frac{F_{t,c} l}{E A_{t,c}} \leq u_{a,t,c}$	$F_{t,c}$ [N] – tensile, compressive force; $A_{t,c}$ [mm ²] – the cross-sectional area of the bar subjected to tension, compression; $u_{t,c}$ [mm] – <u>axial displacement (arrow)</u> ; $u_{a,t,c}$ [MPa] – <u>permissible axial displacement (deflection)</u> in tension, compression; $F_{t,c \text{ cap}}$ [N] – tensile force, compression capable due to rigidity considerations; E [MPa] – longitudinal modulus of elasticity of the material; $E A_{t,c}$ [N] – <u>modulus of rigidity in tension, compression</u>
		sizing	$A_{t,c} = \frac{F_{t,c} l}{E u_{a,t,c}}$	
		Capable load	$F_{t,c \text{ cap}} = \frac{E A_{t,c} u_{a,t,c}}{l}$	
Pure shearing (cutting)		Verification	$w_f = \frac{F_f s}{A_f G} \leq w_{af}$	F_f [N] – shear force; A_f [mm ²] – shear section area; w [mm] – <u>shear displacement (arrow)</u> ; w_{af} [mm] – <u>permissible shear displacement (deflection)</u> ;

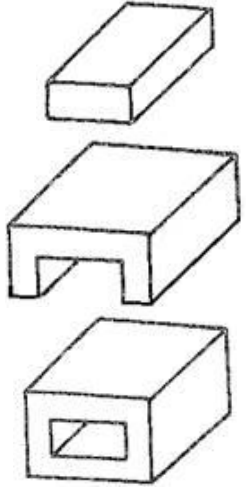
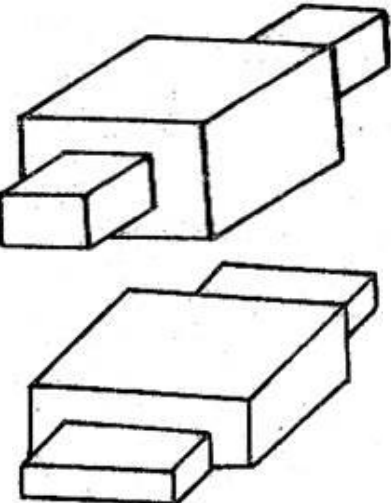
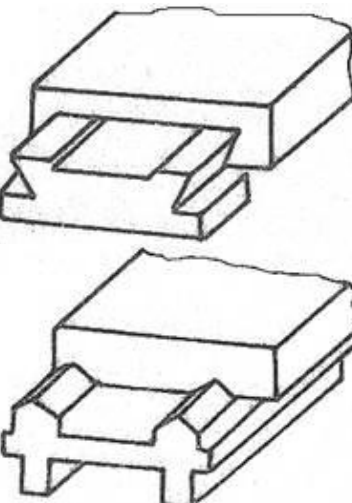
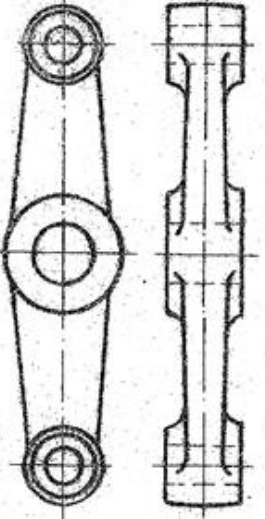
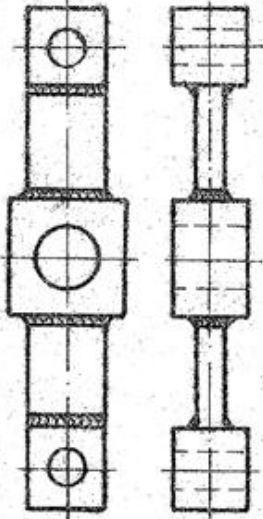
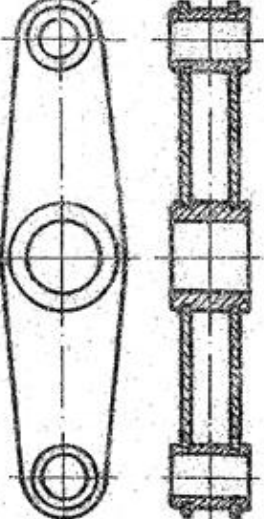
		sizing	$A_f = \frac{F_f s}{G w_{af}}$	γ [rad] – shear rotation; $F_{f \text{ cap}}$ [N] – <u>shear force capable</u> due to rigidity considerations; G [MPa] – transverse modulus of elasticity of the material; GA_f [N] – <u>shear stiffness modulus</u>
		Capable load	$F_{f \text{ cap}} = \frac{GA_f w_{af}}{s}$	
Torsion (twisting)		Verification	$\theta = \frac{M_t l}{G I_p} \leq \theta_a$	l [mm] – bar length; M_t [Nmm] – torsional moment (twisting); I_p [mm ⁴] – polar moment of inertia; $I_p = \frac{\pi d^4}{32}$, for circular section; θ [rad] – angular displacement (rotation); θ_a [rad] – permissible angular displacement (rotation); G [MPa] – transverse modulus of elasticity of the material; GI_p [Nmm ²] – <u>torsional stiffness modulus</u>
		sizing	$I_p = \frac{M_t l}{G \theta_a}$	
		Capable load	$M_{t \text{ cap}} = \frac{GI_p \theta_a}{l}$	
Deflections		Verification	$v = \frac{F_i l^3}{3 E I_z} \leq v_a,$ $\varphi = \frac{F_i l^2}{2 E I_z} \leq \varphi_a,$	l [mm] – bar length; v [mm] – displacement (arrow) during bending; ϕ [rad] – rotation during bending; v_a [mm] – permissible bending displacement (sag); ϕ_a [rad] – permissible bending rotation; F_i [N] – bending force; I_z [mm ⁴] – axial moment of inertia; $I_z = \frac{\pi d^4}{64}$ – axial moment of inertia of the circular section;
		sizing	$I_z^v = \frac{F_i l^3}{3 E v_a}$ $I_z^\varphi = \frac{F_i l^2}{3 E \varphi_a}$	

		Capable load	$F_{i\text{ cap}}^v = \frac{3 E I_z v_a}{l^3}$ $F_{i\text{ cap}}^\varphi = \frac{2 E I_z \varphi_a}{l^2}$	$I_z^{v,\varphi}$ [mm ⁴] – axial moment of inertia required from the condition of displacement, respectively, the imposed rotation; σ_{ai} [MPa] – allowable bending stress; $F_i^{v,\varphi}$ [N] – <u>the force capable</u> of the imposed displacement, respect, rotation condition; E [MPa] – longitudinal modulus of elasticity of the material; $E I_z$ [Nmm ²] – <u>bending stiffness modulus</u>
Note: The values of linear displacements (arrows) and angular displacements (rotations) are determined from the operating conditions imposed in the norms and standards.				

PRC-T.4 TECHNOLOGICAL DESIGN OF SHAPES

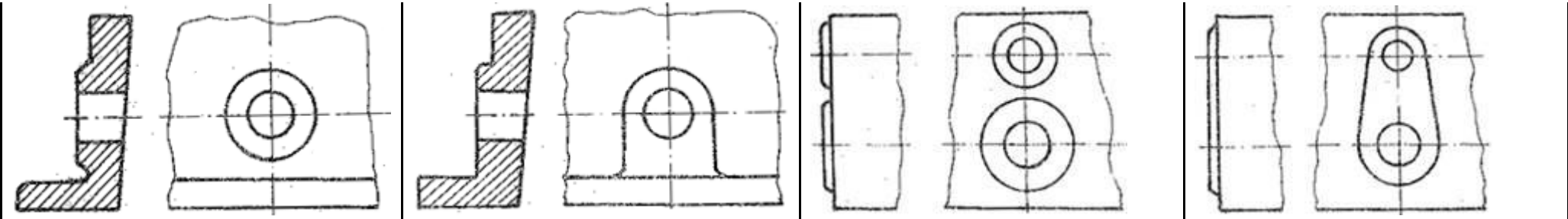
PRC-T4.1 DESIGN OF TECHNOLOGICAL FORMS

PRC-T.4.1.1 GENERAL ASPECTS

					
A	b	c	A	b	c
Fig. PRC-T.4.1.1 Geometric shapes of mechanical product elements: a – geometric; b – functional; c – constructive			Fig. PRC-T.4.1.2 Constructive forms of a mechanical product element (lever): a – cast; b – welded; c – assembled		
The shapes of the elements (parts, components) of mechanical products can be: - <u>geometric</u> (fig. PRC-T.4.1.1,a) - it is necessary to be as few and simple as possible in order to lead to simple and cheap technological processes; - <u>functional</u> (fig. PRC-T.4.1.1,b) - based on geometric shapes, usually ensure the development of physical phenomena (movements and interactions); - <u>constructive</u> (fig. PRC-T.4.1.1,c) - based on functional forms dependent on the technology adopted in principle; - <u>technological</u> (fig. PRC-T.4.1.1,c) - materializes the constructive forms dependent on the available technologies in detail.			The constructive forms may depend on: - the number of pieces to be made (production batch size); - the materials from which it can be made; - technological processes through which they can be obtained: casting, welding, cold or hot plastic deformation (forging), machining, etc. Fig. PRC-T.4.1.1 highlights three possibilities for obtaining a lever-type element: turning, welded from simple semi-finished products and assembled from elements processed by machining and stamping.		

PRC-T.4.1.2 RECOMMENDATIONS ON THE DESIGN OF THE SHAPE OF CASTINGS

TO AVOID	RECOMMENDED	TO AVOID	RECOMMENDED
Wall intersections made with a small connection radius lead to cracks, and those with too large a radius to undercuts; connection radii $r = (1/3...1/5)s$ are recommended, s is the wall thickness		To avoid the appearance of undercuts in areas of material accumulation at the intersections of two walls with large radii, it is recommended to choose appropriate radii and make gradual transitions between the two walls.	
TO AVOID	RECOMMENDED	TO AVOID	RECOMMENDED
The aim is to design parts with quasi-constant thickness, a solution that ensures good cooling, avoiding the appearance of internal stresses due to the uneven and simultaneous cooling of parts with variable thicknesses.		It is recommended to avoid curved surfaces which require much more expensive models compared to those for flat surfaces.	
TO AVOID	RECOMMENDED	TO AVOID	RECOMMENDED



It is recommended that the protrusions (bosses) adjacent to other surfaces or between them, for reasons of simplification of the models, be connected to the surfaces or, respectively, between them.

TO AVOID	RECOMMENDED	TO AVOID	RECOMMENDED

It is recommended to avoid large contact surfaces on other parts due to the increased volume of subsequent processing.

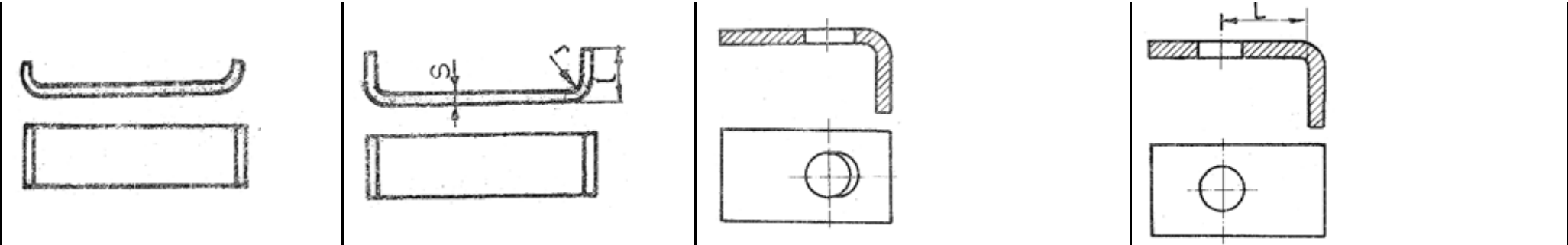
It is recommended, due to the increased resistance of cast materials (especially cast iron) to compression stress, to adopt compressed ribs.

PRC-T.4.1.3 RECOMMENDATIONS ON THE DESIGN OF THE SHAPE OF PARTS PROCESSED BY DEFORMATION

TO AVOID	RECOMMENDED	TO AVOID	RECOMMENDED

Forged parts in dies (die-forged) have inclined walls (larger for the inner walls) to avoid jamming in the dies and it is recommended to avoid strangulated areas in the median plane.

TO AVOID	RECOMMENDED	TO AVOID	RECOMMENDED
----------	-------------	----------	-------------



Cold or hot bent sheet metal parts have bending radii and bent sides dependent on the sheet metal thickness ($l > 3s$) as well as the position of the bent side sufficiently far from existing holes to avoid their ovalization; in the case of holes close to the bent area, these will be drilled after bending.

PRC-T.4.1.4 RECOMMENDATIONS ON THE DESIGN OF THE SHAPE OF MACHINED PARTS

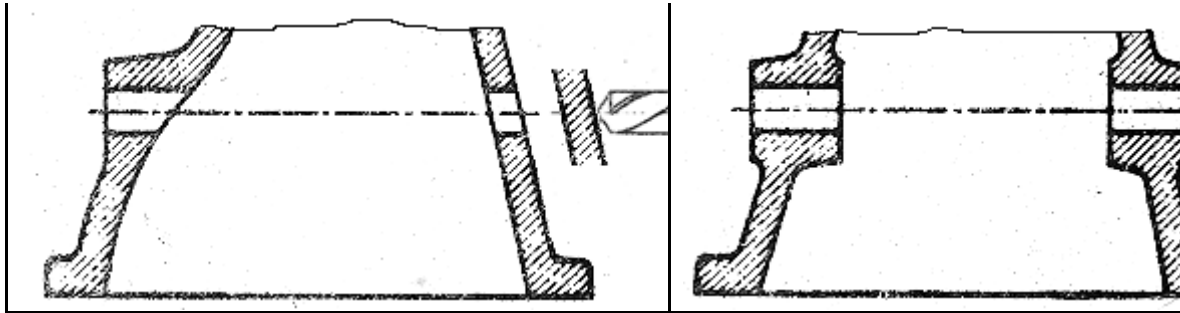
TO AVOID	RECOMMENDED	TO AVOID	RECOMMENDED

Turning of bores is much easier and more precise when it can be done from the same fixture, which implies access to the tools from both sides.

TO AVOID	RECOMMENDED	TO AVOID	RECOMMENDED

For thread machining in areas of diameter jumps, it is necessary to use clearances for the exit of the threading tool.

TO AVOID	RECOMMENDED
----------	-------------



PRC-T.4.2 ADJUSTMENTS AND TOLERANCES

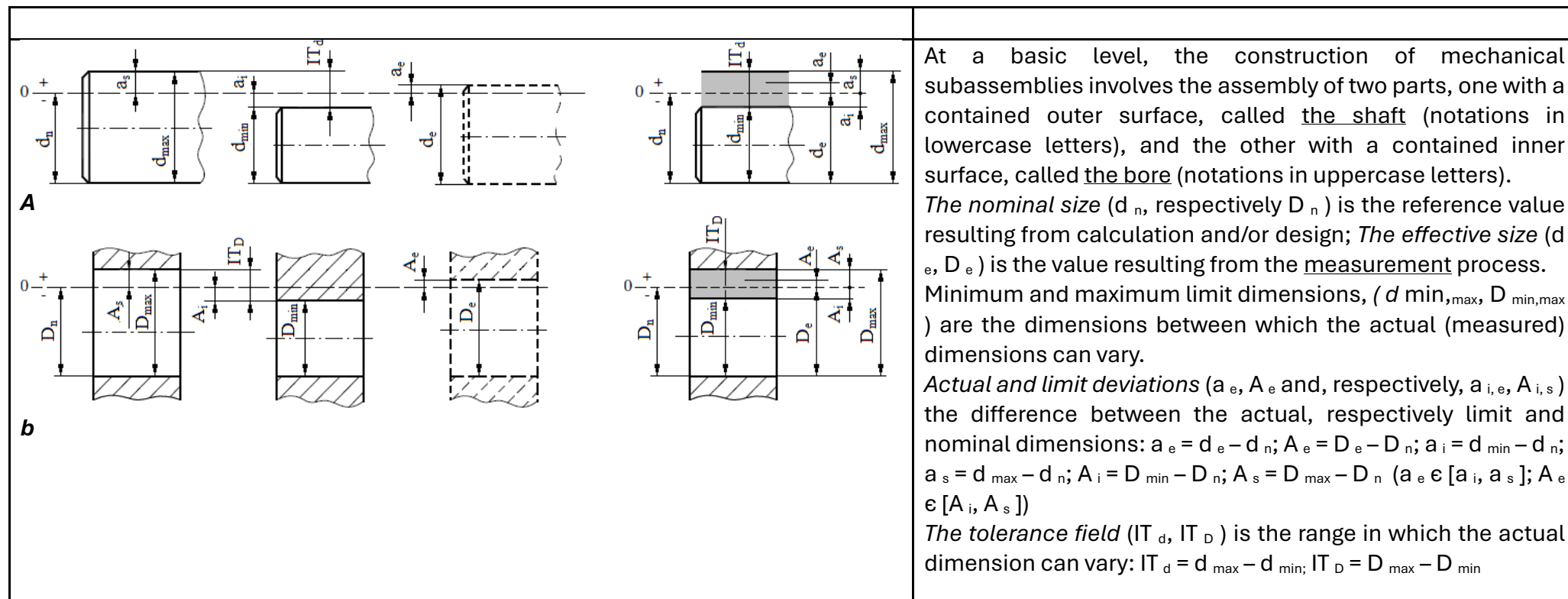
PRC-T.4.2.1 DEFINITIONS

Assembly interchangeability is the property of certain parts (references) with dimensions and shapes between two prescribed limits to be assembled in the respective assembly, without prior selection or additional processing.

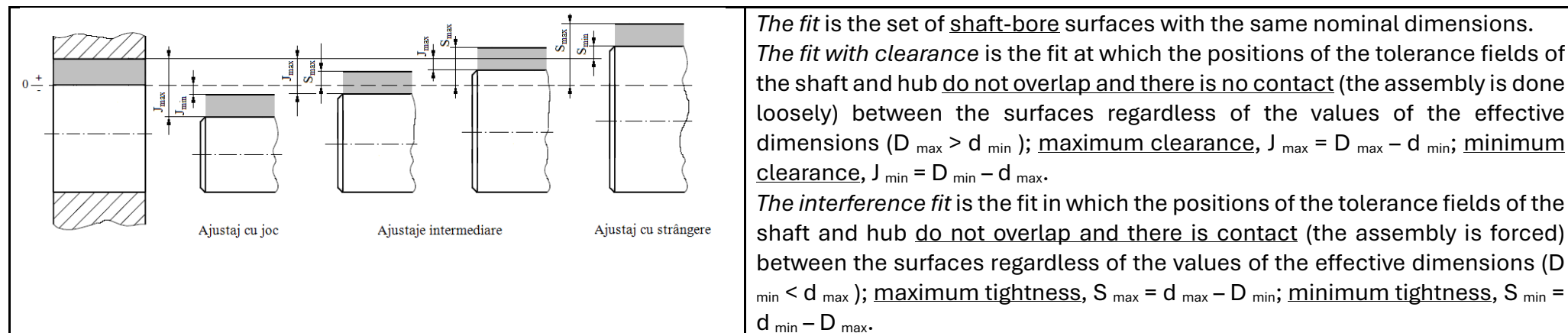
Machining accuracy is the degree of concordance between the physical model (the machined part), with dimensional and shape deviations (errors) (due to imperfections in the machining means and processes), and the theoretical model represented by the execution drawing.

Dimensional accuracy is the agreement between the dimension values of the physical model (machined part) and the theoretical model (execution drawing) under conditions of machining errors.

PRC-T.4.2.2 DIMENSIONS, DEVIATIONS AND TOLERANCE FIELDS



PRC-T.4.2.3 BORES AND BORE SYSTEMS



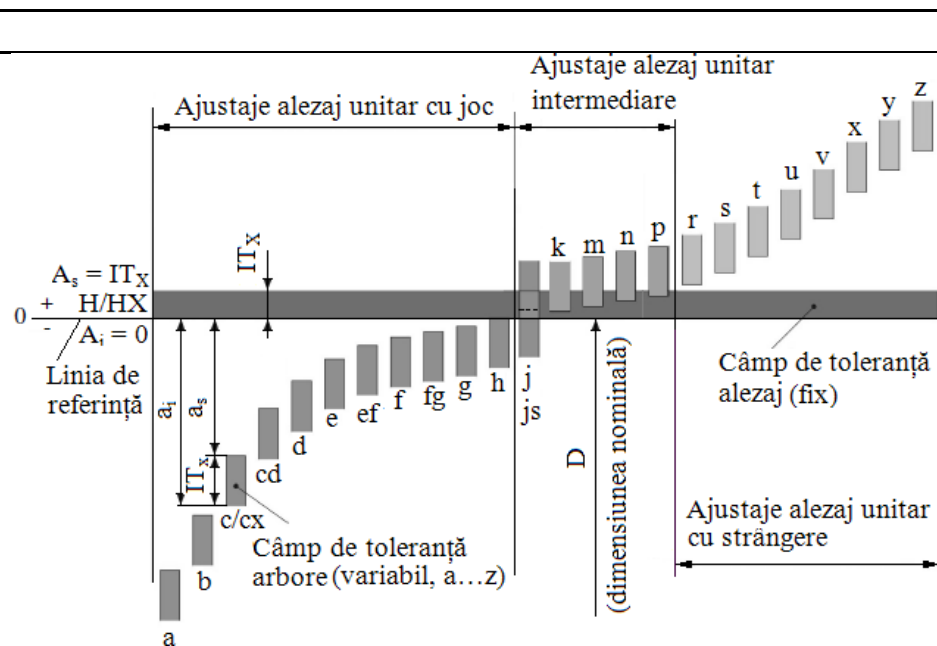
The *intermediate fit* is the fit at which the positions of the tolerance fields of the shaft and hub overlap (the assembly can be free or forced); maximum clearance, $J_{\max} = D_{\max} - d_{\min}$; maximum tightness $S_{\max} = d_{\max} - D_{\min}$.

Unitary bore fits are formed by combining a bore with a fixed tolerance field position and its position relative to the reference line and shafts that have a variable tolerance field position.

Unitary shaft fits are formed by combining a bore with a fixed position of the tolerance field and its position relative to the reference line and bores that have a variable position of the tolerance field.

Note: The unitary bore adjustment system is frequently used in practice because machining the inner surfaces is more difficult than the outer ones and thus by using unitary bores the number of bores to be made as well as the tools (drills, reamers, broaches) for machining is greatly reduced; the unitary adjustment system is rarely used depending on the constructive needs (e.g. some bearing adjustments)

PRC-T.4.2.4 ISO TOLERANCE AND ADJUSTMENT SYSTEMS



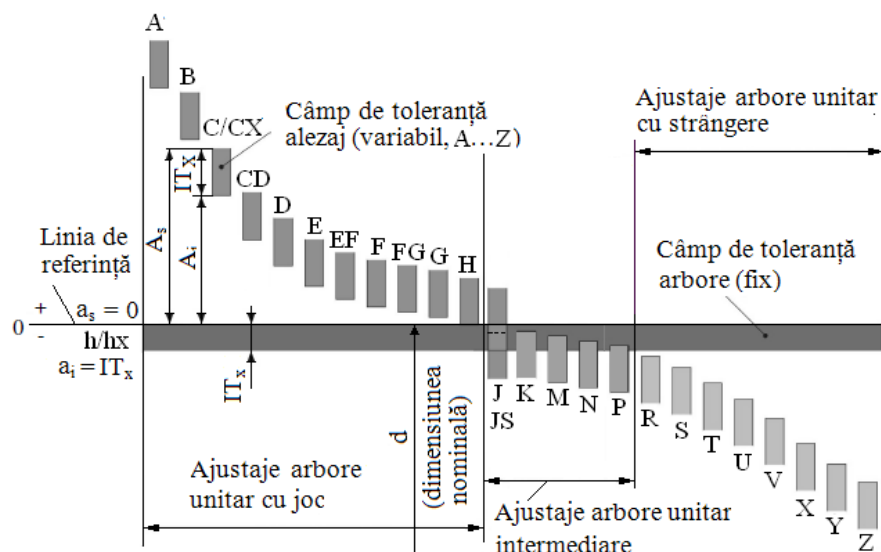
The *ISO tolerance and fit system* is defined by a finite set of tolerance and fit fields that ensure rational and economically efficient manufacturing.

Fundamental tolerances (precision steps) are standardized tolerance fields (see SR EN 20286 or ISO) according to dimensional ranges defined in the range 1...3150 mm. In fer. [PRC-T.4.1](#) fundamental tolerances corresponding to unitary bore fits are presented (extract from SR EN 20286-2).

Areas of use of precision steps: small steps up to IT_4 are used for parts for high-precision mechanisms and verification gauges, IT steps $5...11$ are used for parts in machine construction, steps larger than IT_{12} are used for cast, forged or rolled semi-finished products

The *symbolization of fundamental tolerances* of bores and shafts, according to ISO, is done using alphanumeric codes: uppercase (A, B, C, CD, D, E, EF, F, F, G, H, J, JS, K, M, N, P, R, S, T, U, V, X, Y, Z), for the shaft and lowercase (a, b, c, cd, d, e, ef, f, f, g, h, j, js, k, m, n, p, r, s, t, u, v, x, y, z) for the shaft.

The *positions of the (fundamental) tolerances relative to the reference line* are established using the fundamental deviations: lower, A_i and a_i for the tolerances above the reference line (J, JS, K, M, N, P, R, S, T, U, V, X, Y, Z and respectively a, b, c, cd, d, e, ef, f, f, g, h) and upper A_s and a_s for the tolerances below the reference line (j, js, k, m, n, p, r, s, t, u, v, x, y, z and respectively A, B, C, CD, D, E, EF, F, F, G, H)

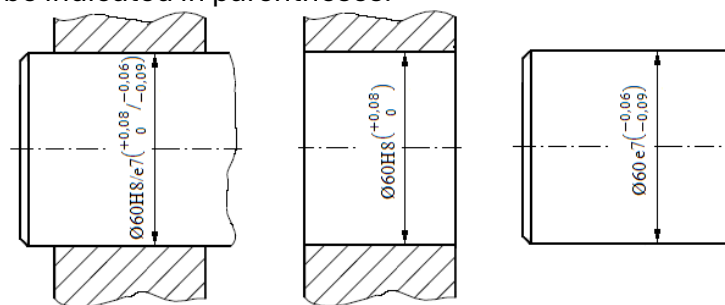


Relationships between tolerances, limit deviations and fundamental deviations: $A_s = A_i + IT_x$ with X the numerical code of the accuracy level of the bore; $a_s = a_i + IT_x$ with x the numerical code of the accuracy level of the shaft; fundamental tolerances are positive, and deviations can be positive or negative.

In fer. it is presented

ISO fit formation. **Unitary bore**, by combining the tolerance fields of the bore H with the tolerance fields of the shaft: a, b, c, cd, d, e, ef, f, f, g, h (clearance fit); j, js, k, m, n, p (intermediate fit); r, s, t, u, v, x, y, z (tight fit). **Unitary shaft**, by combining the tolerance fields of the shaft H with the tolerance fields of the bore: A, B, C, CD, D, E, EF, F, F, G, H (clearance fit); J, JS, K, M, N, P (intermediate fit); R, S, T, U, V, X, Y, Z (tight fit)

The notation (marking) of adjustments and deviations on the execution drawings, according to ISO, is done by indicating the sequence: the shape symbol (e.g. cylindrical Æ), the nominal dimension value, the alphanumeric code associated with the fundamental tolerance position (e.g. H, h, e, f) and the numerical code of the precision grade (e.g. 1...18, for dimensions in the range 3...500); optionally, but very useful for manufacturing, the deviations can be indicated in parentheses.



a b c

Note. From the multitude of possible fits, for reasons of economic efficiency, preferential unitary bore fits are used in practice (STAS 8100): H7/e8, H7/f7, H7/g6, H7/h6, H8/d9, H8/f8, H8/h8, H11/a11, H11/b11, H11/d11, H11/h11 (fit with clearance); H7/j6, H7/k6, H7/m6, H7/n6, H7/p6 (intermediate fit); H7/p6, H7/r6, H7/s6, H7/u6, H7/z7 (fit with tightening).

LIP. BONDED JOINTS

LIP-T. STRUCTURE AND MODELING OF JOINTS BY BONDING (WITH ALLOYS OR ADHESIVES)

CONTENTS

LIP-T.1 DEFINITION 2-3

LIP-T.2 CONSTRUCTION STRUCTURES 2-3

LIP-T.3 AREAS OF USE, ADVANTAGES AND DISADVANTAGES..... 2-4

LIP-T.4 CLASSIFICATION 2-4

LIP-T.5 MATERIALS AND TECHNOLOGIES 2-5

LIP-T.6 FORMS AND CAUSES OF DISPOSAL OR IMPROPER BEHAVIOR 2-6

LIP-T.7 FUNCTIONAL AND CONSTRUCTIVE PARAMETERS2-7

LIP-T.8 CALCULATION MODELS2-7

 LIP-T.8.1 Brazing strength calculation models (strength) 2-7

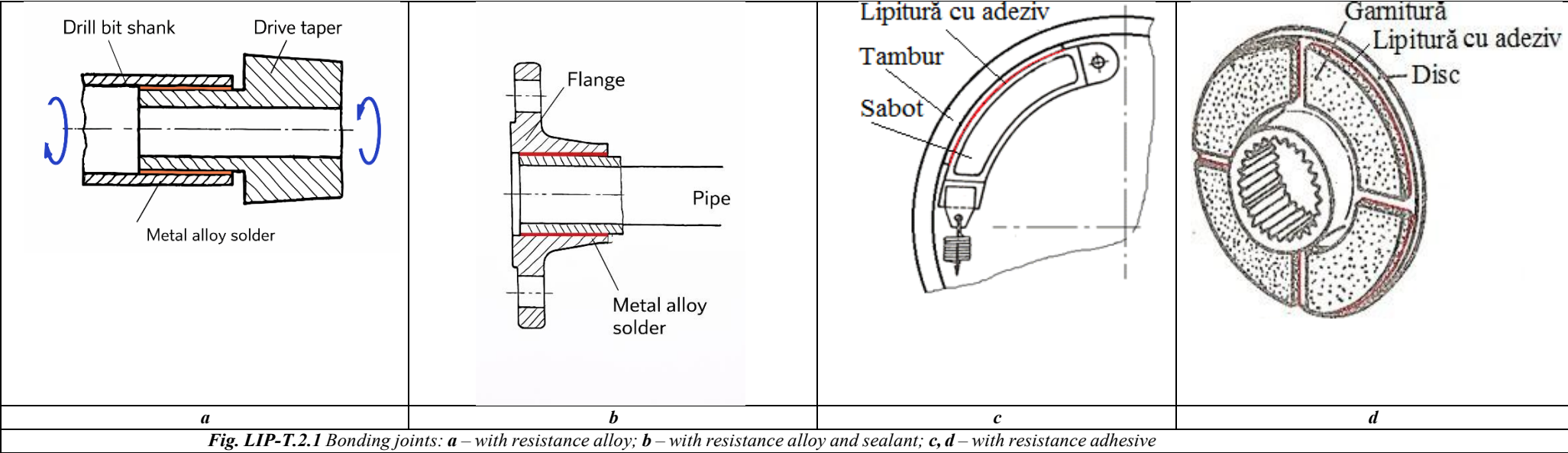
LIP-T.1 DEFINITION

*Brazing is the technological process by which an **irremovable bond** is obtained between two or more parts with the same or similar structures, by means of a metal alloy (usually non-ferrous), melted by heating to a temperature lower than the melting temperatures of the materials of the parts to be joined.*

Bonding with adhesives is the technological process by which an irremovable bond is obtained between two or more parts with the same or similar structures, by means of an adhesive that is introduced.

*Bonded joints are **non-detachable bonds**, made through intermolecular cohesion or adhesion forces of the materials of the joined parts with the bonding material (**metal alloy or adhesive**, respectively) that penetrates and adheres to the interstices between the contacting surfaces of the parts.*

LIP-T.2 CONSTRUCTION STRUCTURES



LIP-T.3 AREAS OF USE, ADVANTAGES AND DISADVANTAGES

They are used in cases **of low mechanical stress** when **other secondary functions are also required** (sealing, electrical and/or thermal conductivity).

Main areas of use: electrotechnical products, measuring instruments, turbine rotors, bicycles, reconditioning of cast parts, hard alloy plating of surfaces subject to wear.

Tab. LIP-T.3.1 Advantages and disadvantages of soldered joints

*Advantages	*Disadvantages
- parts made of different materials (usually non-weldable) can be joined; - good tightness (sometimes, joints are made only for sealing, e.g. on radiators); - reduced damage to the surface treatments of the joined parts (e.g. galvanized); - possibility of automation; - good electrical conductivity (for soldering with alloys); - the occurrence of reduced deformations and residual stresses	- they are uneconomical for joining large parts due to the increased consumption of soldering alloys (e.g. zinc or silver); - lower mechanical resistance; - the need for more laborious preliminary preparation of the contact surfaces of the joined parts; - the danger of electrolytic destruction of the solder due to the potential difference between the materials of the joined parts (e.g. aluminum) and the components of the solder alloy; - bonding materials are expensive

*) especially in relation to welded joints

LIP-T.4 CLASSIFICATION

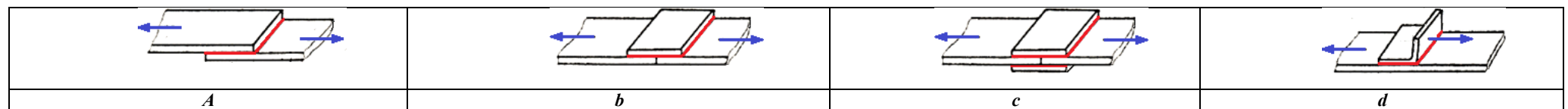
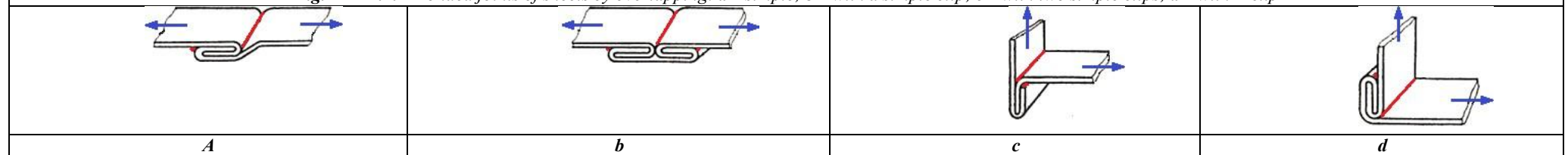
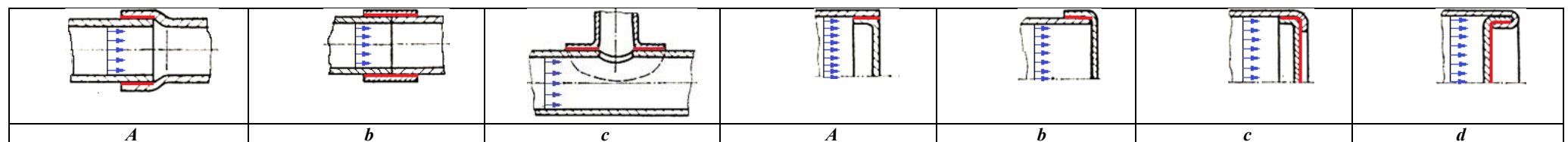
Tab. LIP-T.4.1 Classification of alloy brazing joints

Criterion	Type	Observations
Melting temperature of solder alloy	Smooth	Low melting point (below 450 °C)
	Hard	High melting point (greater than 450 °C, approx. 850 °C)
Technology used	With flame	Soft or hard bonds can be made in the case of large series productions
	By immersion in the bath	Soft or hard solders can be used in the case of one-off productions; salts, metal baths can be used
	By induction	Heating is done by electric field in a protective gas atmosphere or in vacuum.

The physical process	Electrolytic	Soft or hard bonds can be made in short production times
	by layer deposition	The molten filler material is deposited in the joints of the joint
	Capillary	The filler material enters the gaps between the joined parts due to capillary forces (surface tension)

Tab. LIP-T.4.2 Classification of adhesive bonding joints

Criterion	Type	Observations
Scope	With structural adhesives	For joints with good mechanical resistance
	With hot plug	For unstressed (or low-stress) joints
Heating temperature of soldered parts	With cold socket	It is done at ambient temperature; requires long drying time and parts fixing devices
	With hot plug	Heating at lower temperatures; reduced drying time and requires heating installations

Fig. LIP-T.4.1 Bonded joints of sheets by overlapping: *a* – simple; *b* – with a simple clip; *c* – with two simple clips; *d* – with L-clipFig. LIP-T.4.2 Alloy bonding joints of sheets assembled by bending at the end (stress-relieved): *a* – simple; *b* – with flange; *c* – simple corner; *d* – double cornerFig. LIP-T.4.3 Pipe joints by soldering: *a* – simple; *b* – with socket; *c* – with deviationFig. LIP-T.4.1 Glued joints of pipe covers: *a* – inside; *b* – outside; *c* – inside with 90° discharge; *d* – inside with 180° discharge

it

LIP-T.5 MATERIALS AND TECHNOLOGIES

Properties of **soldering alloys**:

- lowest possible melting temperature;
- highest possible fluidity in the molten state;
- lowest possible surface tension in the fluid state to flow as easily as possible into the (often small) gaps between the surfaces of the joined parts, as a result of the capillarity effect.

Alloys for soft joints are frequently **Pb-Sn (tin)** based.

Alloys for resistant (hard) solders are based on **Cu-Zn (brass)**, Cu-Ph, Mg, Ni (for high-temperature resistant solders) or precious metals (silver, gold).

Frequently used adhesives for bonding are **resins**: epoxy-phenolic (for bonding stainless steels, titanium, etc.), epoxy-polyamide (used for metals, rubber, glass, ceramics, plastics), epoxy-silicone (stainless steels), polyurethane (metals, provides resistance to vibrations and shocks), phenolic (for metals and plastics).

The surfaces on which the molten solder will flow must, as a preliminary step, be cleaned of impurities, oxides and grease by one of the following processes: *mechanical* (with a glass fiber or stainless steel brush), *chemical* (using zinc or ammonium chloride pastes), *ultrasonic* (by eroding oxide layers with ultrasonic energy).

LIP-T.6 FORMS AND CAUSES OF DISPOSAL OR IMPROPER BEHAVIOR

Tab. LIP-T.6.1 Forms and causes of inappropriate behavior or removal from use

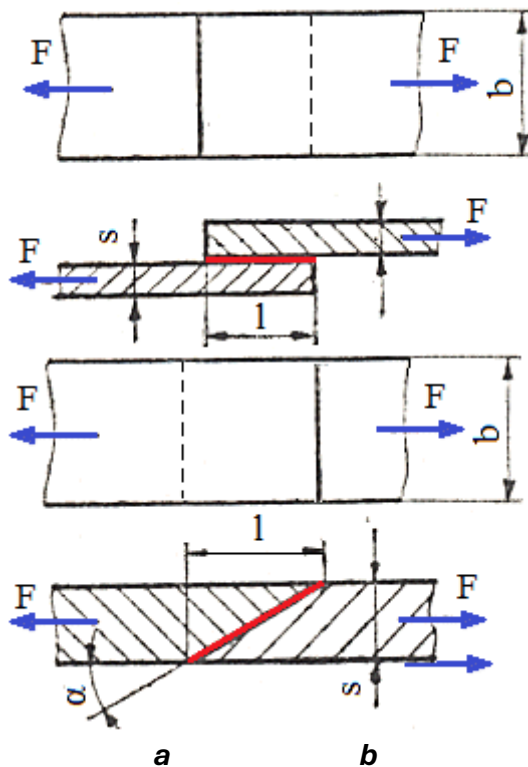
Shapes	Result	Appearance	Cause	Manifestation	Avoidance
Breaking (debonding)	Disposal	The break occurs in the area where the fluid adhesive was introduced	Accidental and/or shock overloads; technological errors	Interruption of load transmission and/or loss of tightness	Avoiding overloads; calculating solder resistance
Corrosion	Disposal	In the case of solders working in corrosive environments	The existence of an environment with corrosive agents	Decrease in the area of bonded surfaces over time	Protective measures to reduce corrosive processes

LIP-T.7 FUNCTIONAL AND CONSTRUCTIVE PARAMETERS

LIP-T.8 CALCULATION MODELS

LIP-T.8.1 BRAZING STRENGTH CALCULATION MODELS (STRENGTH)

Calculation assumptions



Calculation assumptions:

- it is considered that the resistance in the diffusion/ adhesion zone of the material bonded in/to the materials of the joined parts is the same as the resistance of the bonding material;
- the dangerous section is in the alloy/bonding adhesive layer;
- usually, in practice, constructive solutions in which the layer of the bonding material is stressed in tension and/or bending are avoided and joints with the layer stressed in shear are preferred, which have the advantage of linear increase in resistance with increasing surface area;
- the distribution of stresses in the shear layer is considered uniform

Shear bond strength condition:

$$\tau_f = \frac{F}{b l} \leq \tau_{af} \quad (\text{LIP-T.8.1.1})$$

for the overlapping joint of two sheets, subjected to tensile stress (LIP-T.8.1,a);

$$\tau_f = \frac{F \cos \alpha}{b l} \leq \tau_{af} \quad (\text{LIP-T.8.1.2})$$

for the oblique butt joint, stressed in tension (fig. LIP-T.8.1,b);

$$\tau_f = \frac{F}{\pi D l} \leq \tau_{af} \quad (\text{LIP-T.8.1.3})$$

for the pipe-pipe joint, subjected to tensile stress (LIP-T.8.1,c);

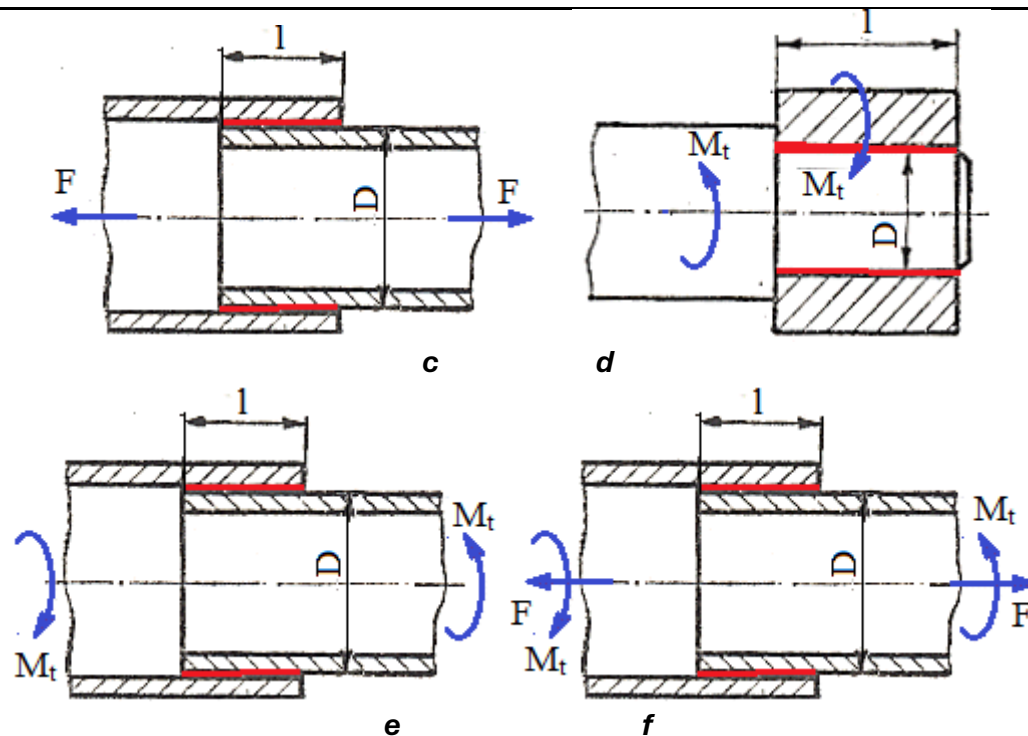


Fig. LIP-T.8.1 Strength calculation models: **a** – overlap pejin; **b** – oblique butt; **c**, **e**, **f** – pipe-pipe; **d** – shaft-hub

$$\tau_f = \frac{F}{A_f} = \frac{\frac{2 M_t}{D}}{\pi D l} = \frac{2 M_t}{\pi D^2 l} \leq \tau_{af} \quad (\text{LIP-T.8.1.4})$$

for pipe-pipe (LIP-T.8.1,e) and shaft-hub (LIP-T.8.1,d) joints, subjected to torsion;

$$\tau_f = \frac{F}{A_f} = \frac{\sqrt{F^2 + \left(\frac{2 M_t}{D}\right)^2}}{\pi D l} \leq \tau_{af} \quad (\text{LIP-T.8.1.5})$$

for pipe-pipe joints (LIP-T.8.1,f), rated for traction and torsion.

The meanings of the parameters in the above relations: τ_f is the effective shear stress; $\tau_{af} = \tau_{rf} / c$ - allowable shear stress; $c = 2 \dots 5$ - safety factor.

Shear rupture stresses of soldered alloys: $\tau_{rf} = 200$ MPa, for brass joints (Cu-Zn alloys); $\tau_{rf} = 120$ MPa, for Zn alloy joints; $\tau_{rf} = 40 \dots 80$ MPa, for Sn alloy joints (tin);

Shear rupture stresses of adhesive bonds: $\tau_{rf} = 9$ MPa, for statically loaded joints; $\tau_{rf} = 6$ MPa, for pulsating loaded joints; $\tau_{rf} = 3$ MPa, for alternately loaded joints.

Note: In design practice, the length of the strip s is determined from the above relationships for dimensioning.

SUD. WELDED JOINTS

SUD-T. STRUCTURE AND MODELING OF WELDED JOINTS

CONTENTS

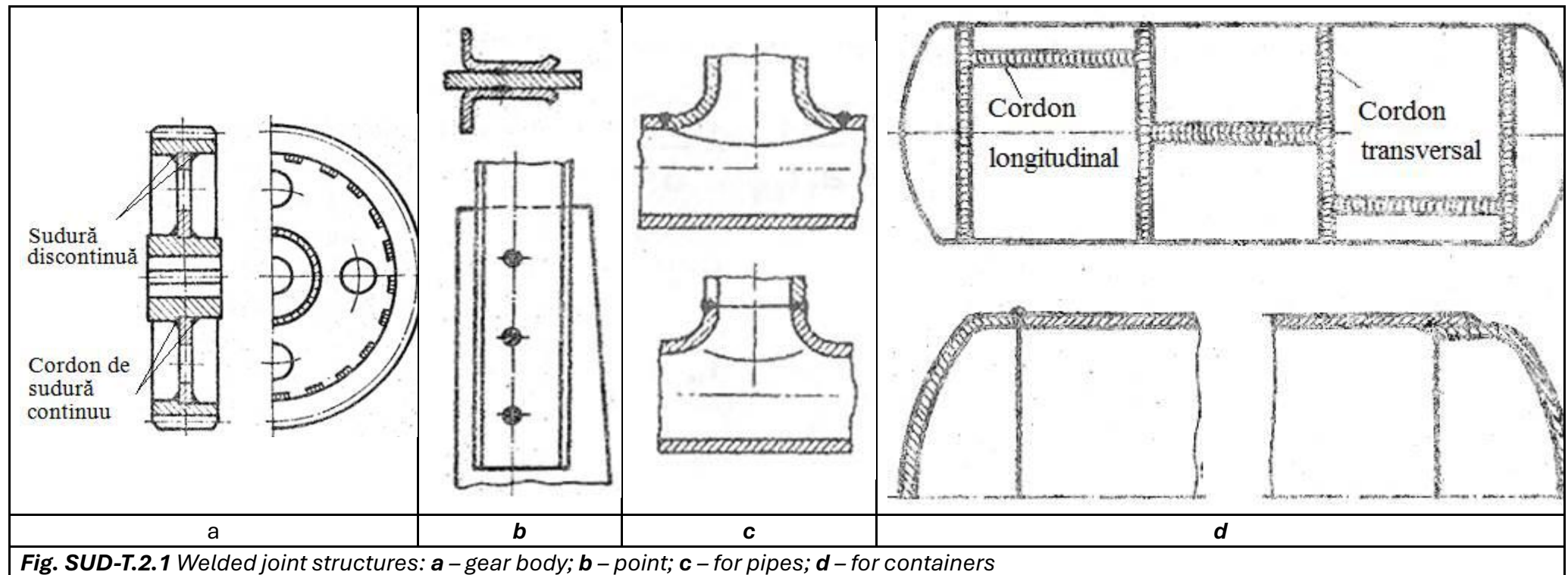
SUD-T.1 DEFINITION	3-3
SUD-T.2 CONSTRUCTION STRUCTURES	3-3
SUD-T.3 AREAS OF USE, ADVANTAGES AND DISADVANTAGES.....	3-4
SUD-T.4 CLASSIFICATION	3-4
SUD-T.5 MATERIALS AND TECHNOLOGIES	3-6
SUD-T.6 FORMS AND CAUSES OF DISPOSAL OR INAPPROPRIATE BEHAVIOR	3-7
SUD-T.7 FUNCTIONAL AND CONSTRUCTIVE PARAMETERS	3-8
SUD-T.8 CALCULATION MODELS	3-8
SUD-T.8.1 Models for calculating stresses in the weld bead for the case of compound stresses	3-8
SUD-T.8.1.1 Calculation model for butt welds	3-9
SUD-T.8.1.2 Calculation model for lateral fillet welds	3-9
SUD-T.8.1.3 Spot weld calculation model.....	3-11

SUD-T.1 DEFINITION

Welding is the technological process by which a **non-detachable connection** is obtained between two or more parts with the same or similar structures, whose joined surfaces have been brought, through local heating, into a liquid or plastic state.

Welded joints are non-detachable connections made by intermolecular cohesive forces from the materials of the joined parts which are processed by local heating to the liquid (melting) state, with or without filler material (usually of the same type as the joined materials) and/or with or without the application of an external pressing force.

SUD-T.2 CONSTRUCTION STRUCTURES



SUD-T.3 AREAS OF USE, ADVANTAGES AND DISADVANTAGES

Main areas of use:

- metal constructions (bridges, construction structures, industrial halls, etc.),
- machine construction (gear and transmission wheel bodies, casings),
- ship construction,
- boilers and containers, etc.

Tab. SUD-T.3.1 Advantages and disadvantages of welded joints

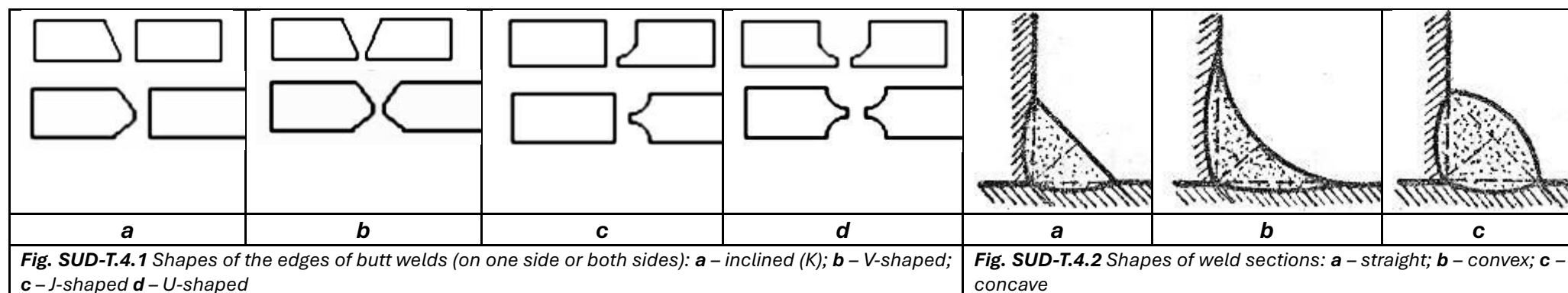
Advantages	Disadvantage
- material savings (up to 40% compared to casting); - good tightness; - increased rigidity; - possibility of automation; - high productivity production processes	- uncertainties related to deformations and residual stresses induced by thermal shock and different contraction upon cooling (this disadvantage is reduced by thermal or mechanical treatments); - for control, expensive modern methods of non-destructive testing are used (with ultrasound, γ rays; magnetic); - the quality of welding, in the case of manual processes, depends on the skills and attention of the welder; - structural changes in the joined materials

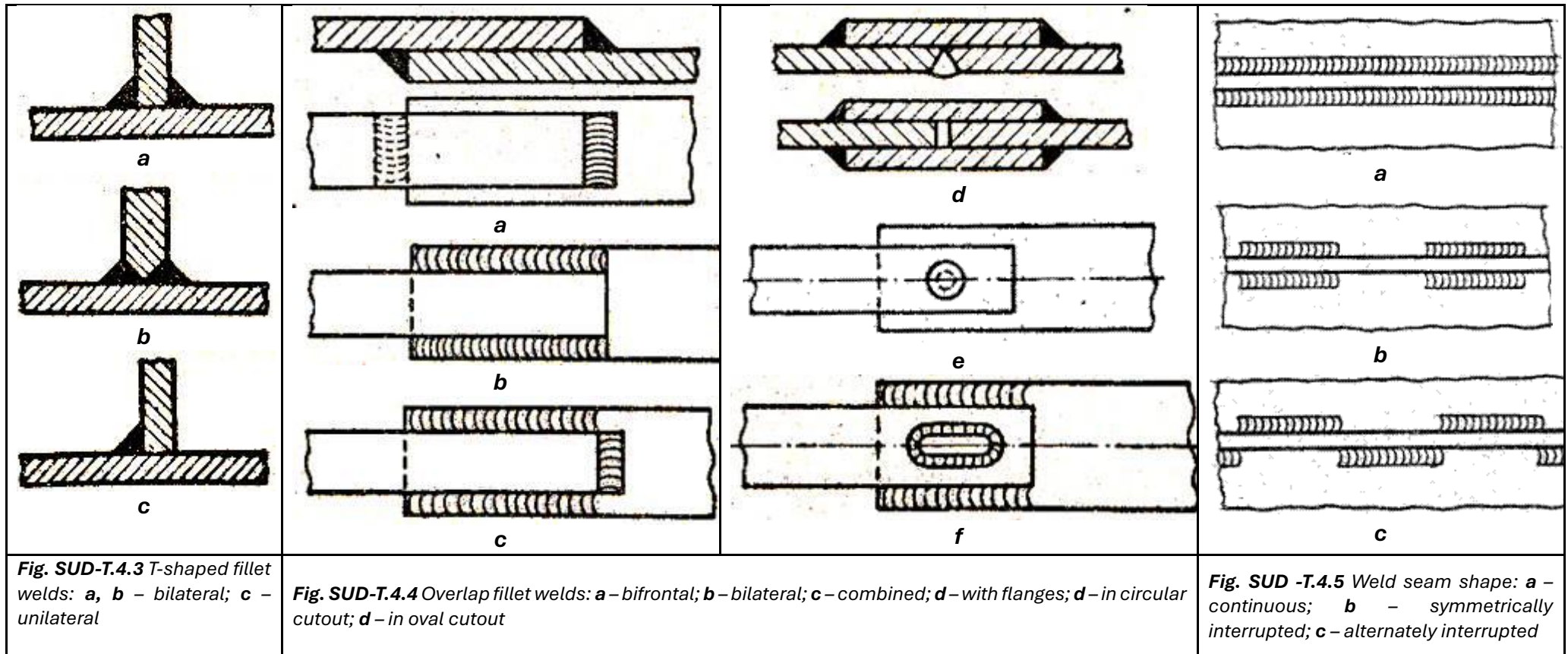
SUD-T.4 CLASSIFICATION

Tab. SUD-T.4.1 Classification of welded joints

Criterion	Type	Scheme	Note
Relative position of joined parts	Head to head	Fig. SUD-T.4.1	When welding thin sheets, the edges are not processed; Processing can be performed on one side or both sides.
	Corner	Fig. SUD-T.4.4	By overlapping (frontal, lateral, inclined, combined, with slats, in cutouts); at an angle: on the edge (on one bilateral side); in T
According to the shape of the weld bead surface	Convex	Fig. SUD-T.4.2	The shapes are dependent on the welding processes
	Right		

	Concave		
Technological process	By melting	Fig. SUD-T.4.3; 4.4; 4.5	It is performed by local melting of the parts to be joined together with the filler material.
	By pressure	Fig. SUD-T.2.1,b	It is performed by pressing and local heating until the parts to be joined are plasticized. Spot or line welds can be performed.
Shape of the edges of the pieces	Inclined; in V, J	Fig. SUD-T.4.1,a,b	Processing can be on one side or both sides
	In J, U	Fig. SUD-T.4.1,c,d	Processing can be on one side or both sides
Cord shape	Continue	Fig. SUD-T.4.5,a	
	Discontinuous	Fig. SUD-T.4.5,a,b	





SUD-T.5 MATERIALS AND TECHNOLOGIES

Weldability is the property of metals to be able to be welded to form strong intermolecular bonds; for steels, weldability is determined by the percentage of carbon (**very good for $C < 0.25\%$**).

Fusion welding is the technological welding process by which the joined parts are **heated locally**, until they melt together with the **filler material** (electrode) of the same type as the base materials; local heating can be obtained by **burning a gas** (acetylene, methane gas, etc.) mixed with oxygen or by means of an **electric arc** formed between the electrode and the part which produces an increased heat concentration.

Press welding is the technological welding process, **without the addition of material**, through which the assembled parts are heated, only until **plasticization**, with the application of a local pressing force that generates strong concentrations of energy: **plasma jet welding**, **laser welding** (with electron beam), **cold press welding**.

SUD-T.6 FORMS AND CAUSES OF DISPOSAL OR INAPPROPRIATE BEHAVIOR

Tab. SUD-T.6.1 Forms and causes of inappropriate behavior or removal from use

Shapes	Result	Appearance	Cause	Manifestation	Avoidance
Overload failure	Disposal	The rupture occurs in the area of the section with the minimum area	Accidental and/or shock overloads	Interruption of the transmission of the load	Avoiding overloads
Fatigue fracture	Disposal	Rupture occurs in areas where there are structural defects or inclusions	Variable loads and the stress concentrator	Interruption of the transmission of the load	Limitation of fatigue stresses to admissible values by calculation; constructive measures to reduce the causes
Corrosion	Disposal	In the case of welds exposed to corrosive environments	Exposure to corrosive agents in the environment	Decrease in the dangerous section of the weld over time	Measures to reduce corrosive processes

SUD-T.7 FUNCTIONAL AND CONSTRUCTIVE PARAMETERS

SUD-T.8 CALCULATION MODELS

SUD-T.8.1 MODELS FOR CALCULATING STRESSES IN THE WELD BEAD FOR THE CASE OF COMPOUND STRESSES

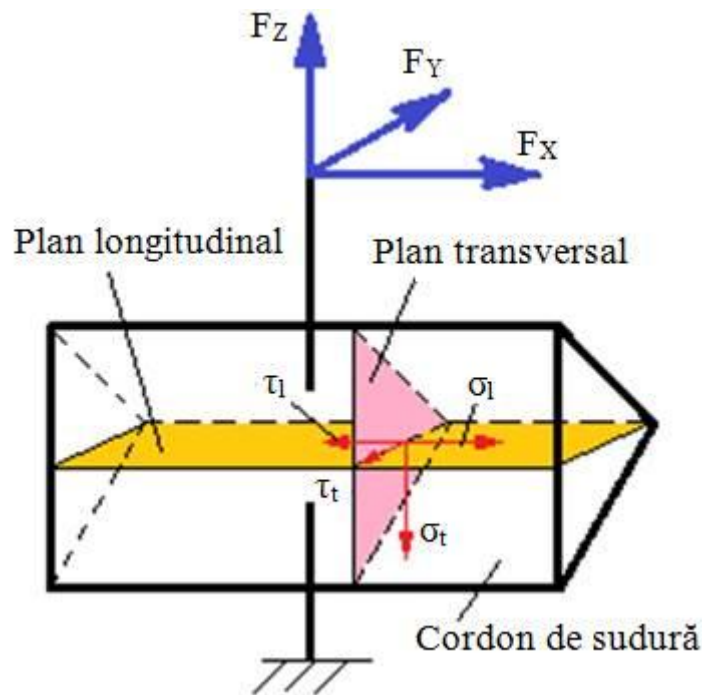


Fig. SUD-T.8.1 General calculation model

Calculation assumptions:

- because determining the stresses and stress states in welded parts and, especially, in weld seams are complex problems, the strength calculation is a conventional calculation;
- the weld is designed in such a way that the resistance of the seam is equal to that of the base material; in the case of joining parts made of different materials, the weld resistance is adopted to be equal to that of the softer material;
- because the weld bead is a structure with a much larger dimension (length) than the other two that describe the cross-section, at a point in the longitudinal plane (critical section), the following stresses are highlighted: σ_l – longitudinal normal stress (perpendicular to the transverse plane); σ_t – transverse normal stress (perpendicular to the longitudinal plane); τ_{fl} – longitudinal shear stress (from the longitudinal plane); τ_{ft} – transverse shear stress (from the transverse plane); for calculations, the longitudinal normal stress, σ_l

Weld seam strength condition:

- According to the 5th resistance theory, the equivalent stress,

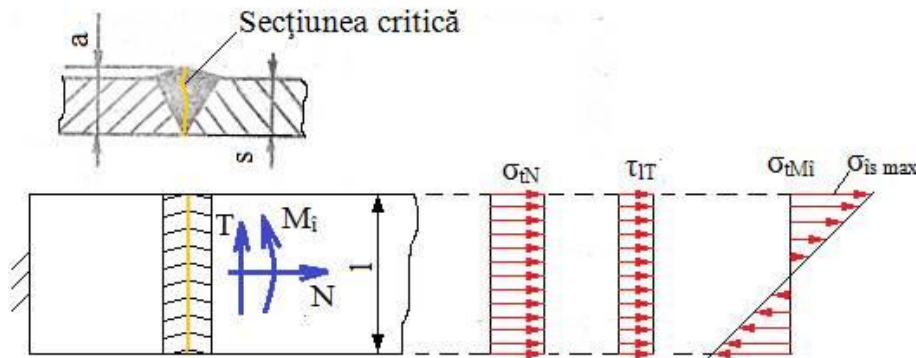
$$\sigma_{es} = \sqrt{\sigma_t^2 + 3(\tau_l^2 + \tau_t^2)} \leq \sigma_{ats} \quad (\text{SUD-T.8.1.1})$$

where: $\sigma_{ats} = \min(\sigma_{r1}, \sigma_{r2})/c$ with $\sigma_{r1,2}$ – the breaking stresses of the joined materials; $c = 1.25 \dots 2$, the safety factor; the stresses (σ_t, τ_l, τ_t) in the weld bead are determined separately for the simple stresses in relation to the longitudinal critical section area.

- In design practice, the relationship is frequently used,

$$\sigma_{es} = \sqrt{\sigma_t^2 + (\tau_l^2 + \tau_t^2)} \leq \sigma_{ats} \quad (\text{SUD-T.8.1.2})$$

SUD-T.8.1.1 Calculation model for butt welds



Calculation assumptions:

- the action of the longitudinal force N determines normal transverse tensile stresses in the weld seam, σ_{tN} ;
- the action of the transverse force T determines longitudinal shear stresses in the weld bead, τ_{IT} ;
- the action of the bending moment, M_i , determines in the weld seam normal transverse tensile/compressive stresses, σ_{tMi} with the maximum $\sigma_{is\ max}$;
- the critical section is considered rectangular with length $l_{cs} = l - 2a$ (the head portions where the weld bead is incomplete are not taken into account) and width, $a \approx s$ (the concavity is not taken into account).

Weld strength condition, (according to rel. SUD-T.8.1.2):

$$\sigma_e = \sqrt{(\sigma_{tN} + \sigma_{is\ max})^2 + \tau_{IT}^2} \leq \sigma_{ats}, \quad (\text{SUD-T.8.1.1.1})$$

where:

$$\sigma_{tN} = \frac{N}{a l_{cs}}, \text{ the tensile stress due to the longitudinal force } N;$$

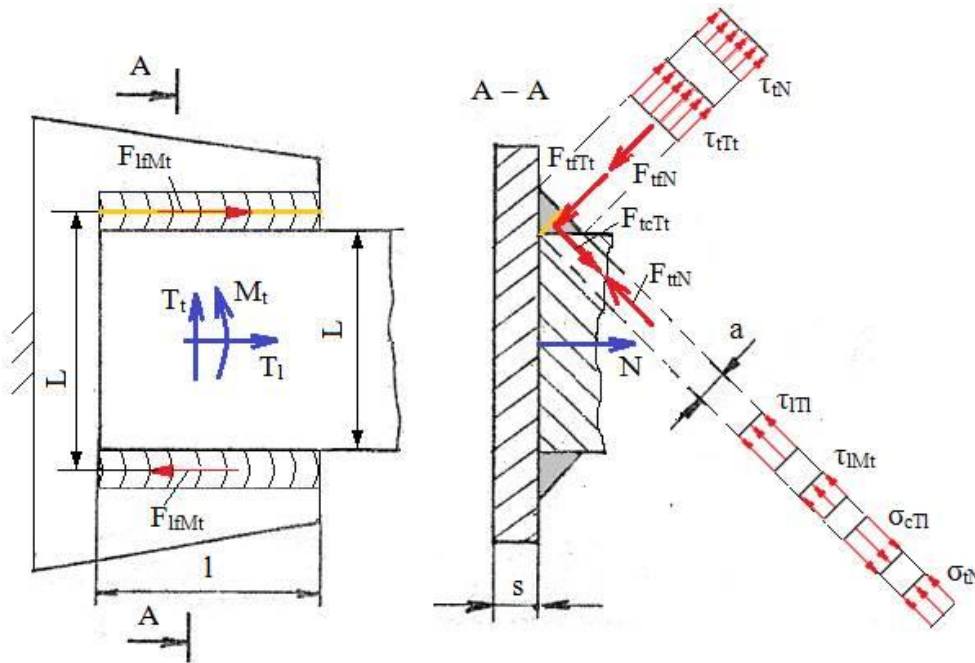
$$\tau_{IT} = \frac{T}{a l_{cs}}, \text{ the shear stress due to the transverse force } T;$$

$$\sigma_{is\ max} = \frac{M_i}{s l_{cs}^2}, \text{ the maximum tensile stress due to the bending moment } M_i;$$

$l_{cs} = l - 2a$ – design length of the weld; $\sigma_{ats} = \min(\sigma_{r1}, \sigma_{r2})/c$ with $\sigma_{r1,2}$ – ultimate stresses of the joined materials; $c = 1.25 \dots 2$, safety factor

Fig. SUD-T.8.1.1.1 Calculation model

SUD-T.8.1.2 Calculation model for lateral fillet welds



Calculation assumptions:

- the action of the longitudinal force T_l determines longitudinal shear stresses in the weld bead, τ_{lT_l} ;

- the action of the transverse force T_t on the weld seams is equivalent to

the transverse forces, and which $F_{tcT_t} = \frac{T_t}{2} \cos 45^\circ = T_t \frac{\sqrt{2}}{4}$

$$F_{ttT_t} = \frac{T_t}{2} \sin 45^\circ = T_t \frac{\sqrt{2}}{4}$$

determines compressive stresses, σ_{cT_l} , and respectively shear stresses, τ_{tT_t} ;

- the action of the normal force N on the weld seams is equivalent to the

transverse forces, and which determines $F_{ttN} = \frac{N}{2} \cos 45^\circ = N \frac{\sqrt{2}}{4}$

$$F_{tN} = \frac{N}{2} \sin 45^\circ = N \frac{\sqrt{2}}{4}$$

tensile stresses, σ_{tN} , and respectively shear stresses, τ_{tN} ;

- the action of the torsion moment, M_t , on the weld seams is equivalent

to the force couple, which determines the stresses $F_{lM_t} = \frac{M_t}{L+s}$ of longitudinal shear, τ_{lM_t} ;

- the critical section associated with the weld bead in which the longitudinal shear stresses have the same direction is considered rectangular with length $l_{cs} = l - 2a$ (the head portions where the weld bead is incomplete are not taken into account) and width equal to the maximum thickness of the weld bead, $a \approx 0.7s$ (the concavity is not taken into account).

Weld strength condition (according to rel. SUD-T.8.1.2)

$$\sigma_e = \sqrt{(\sigma_{tN} - \sigma_{cT_l})^2 + (\tau_{tT_t} + \tau_{tN})^2 + (\tau_{lT_l} + \tau_{lM_t})^2} \leq \sigma_{ats}, \quad (\text{SUD-T.8.1.1.2})$$

where:

$$\sigma_{tN} = \frac{F_{ttN}}{2a l_{cs}}, \text{ the tensile stress due to the normal force } N;$$

$$\sigma_{cT_l} = \frac{F_{tcT_t}}{2a l_{cs}}, \text{ the compressive stress due to the transverse force } T_t;$$

Fig. SUD-T.8.1.2.1 Calculation model of lateral fillet welds

$$\tau_{tTt} = \frac{F_{tTt}}{2a l_{cs}}, \text{ the shear stress in the transverse plane due to the transverse force } T_t;$$

$$\tau_{tN} = \frac{F_{tN}}{2a l_{cs}}, \text{ the shear stress in the transverse plane due to the normal force } N;$$

$$\tau_{lTl} = \frac{T_l}{2a l_{cs}}, \text{ the shear stress in the longitudinal plane due to the longitudinal force } T_l;$$

$$\tau_{lMt} = \frac{F_{lMt}}{2a l_{cs}}, \text{ the shear stress in the longitudinal plane due to the moment of torsion, } M_t; \sigma_{ats} = \min(\sigma_{r1}, \sigma_{r2})/c \text{ with } \sigma_{r1,2} - \text{ the breaking stresses of the joined materials; } c = 1.25 \dots 2, \text{ the safety factor}$$

SUD-T.8.1.3 Spot weld calculation model

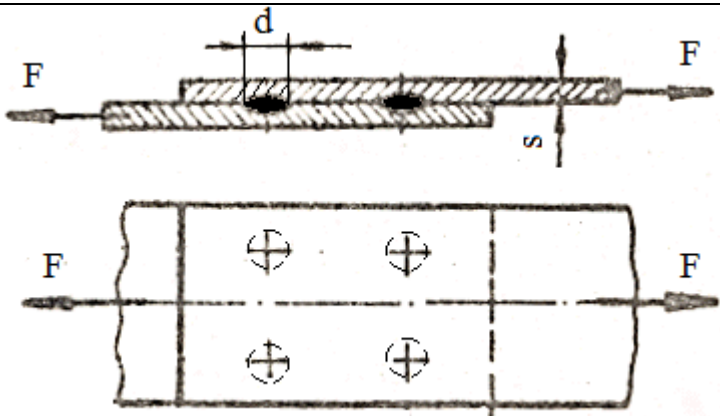


Fig. SUD-T.8.1.3.1 Spot weld calculation model

Calculation assumptions:

- the welding points are considered to be identical and equally loaded;
- conventional, shear calculation is adopted.

Shear stress from a weld point,

$$\tau_{fs} = \leq, \frac{F}{A} \frac{T}{n} \frac{\pi d^2}{4} \tau_{afs}$$

where: F represents the external force; τ_{afs} is the allowable shear stress of the weakest material; n - number of weld points; d - diameter of the weld point

NIT. RIVET JOINTS

NIT-T. STRUCTURE AND MODELING OF RIVET JOINTS

CONTENTS

NIT-T.1 DEFINITION..... 4-3

NIT-T.2 CONSTRUCTION STRUCTURES 4-3

NIT-T.3 AREAS OF USE, ADVANTAGES AND DISADVANTAGES 4-3

NIT-T.4 CLASSIFICATION 4-4

NIT-T.5 MATERIALS AND TECHNOLOGIES 4-5

NIT-T.6 FORMS AND CAUSES OF DISPOSAL OR IMPROPER BEHAVIOR 4-5

NIT-T.7 FUNCTIONAL AND CONSTRUCTIVE PARAMETERS 4-6

NIT-T.8 CALCULATION MODELS 4-6

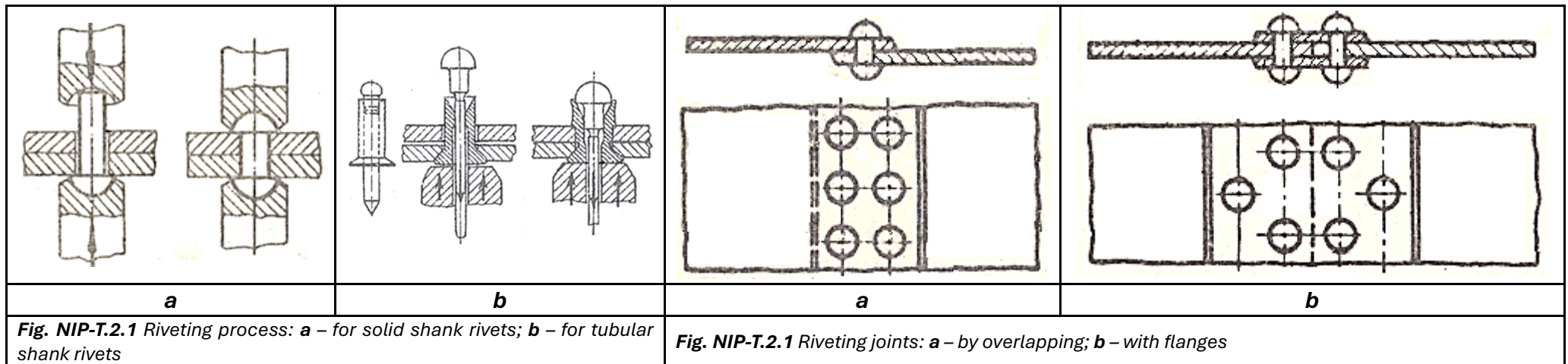
 NIT-T.8.1 Strength calculation models 4-6

NIT-T.1 DEFINITION

Riveted joints are non-detachable connections, made by means of intermediate elements called rivets.

The rivet, in general, has a cylindrical shape (fig. NIT-T.2.1) with one of the ends initially formed, and the second, closing end is formed by plastic deformation (cold or hot) between the tools with which the assembly is carried out.

NIT-T.2 CONSTRUCTION STRUCTURES



NIT-T.3 AREAS OF USE, ADVANTAGES AND DISADVANTAGES

Main areas of use: metal construction, machine construction; joining of high-value elements subject to variable loads and/or shocks (cranes, transport installations, etc.); boilers, pipes and pressure tanks; vehicle bodies

Tab. NIT-T.3.1 Advantages and disadvantages of riveted joints

Advantages *	Disadvantages *
- joining non-weldable materials; - joining materials that cannot be heated; - better resistance to variable stresses and/or shocks.	- high cost price; - lower mechanical resistance. - reduced productivity.

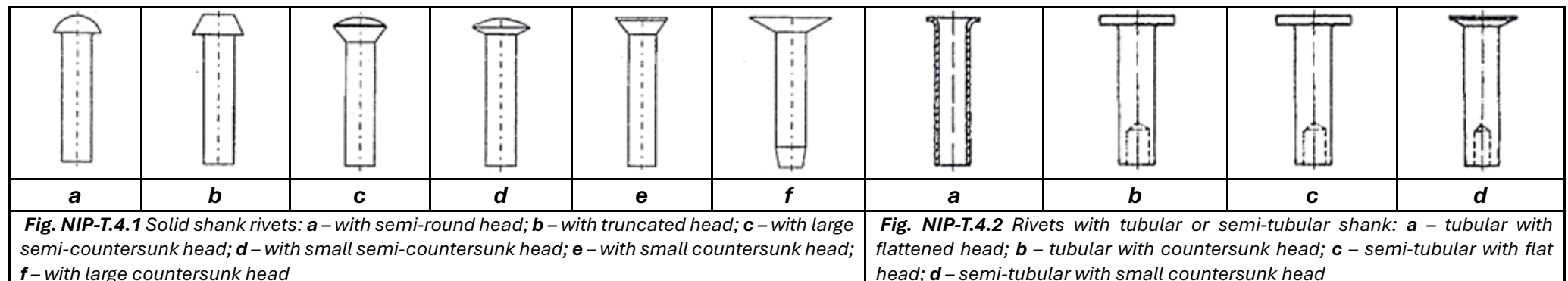
- | | |
|--|---|
| - joining heat-treated parts that no longer allow heating.
- increased operational safety | - the to ensure the tightness of the assembled parts;
- higher metal consumption |
|--|---|

*) especially in relation to welded joints

NIT-T.4 CLASSIFICATION

Tab. NIT-T.4.1 Classification of riveted joints

Criterion	Type	Observations
Purpose	Of resistance	They take on external tasks.
	Sealing and resistance	They take on external loads and ensure tightness (boilers, tanks and/or pressure pipes)
Joint temperature	Hot	By heating the rivet head before riveting in order to reduce the resistance to deformation
	Cold	At the working environment temperature
Rivet type	With semi-round head rivets	Fig. NIT-T.4.1,a
	With truncated cone head rivets	Fig. NIT-T.4.1,b
	With countersunk head rivets	Fig. NIT-T.4.1,c,d
	With countersunk head rivets	Fig. NIT-T.4.1,e,f
	With tubular/semi-tubular rivets	Fig. NIT-T.4.2



NIT-T.5 MATERIALS AND TECHNOLOGIES

The rivet materials must be tough (ductile), not hardenable and have a coefficient of thermal expansion close to that of the parts to be assembled; ordinary rivets can be made of construction steels (S185, S235); in cases of increased stress, rivets made of high-quality non-alloyed or even alloyed steels can be used; for reduced loads, rivets made of brass, copper, aluminum and aluminum alloys, etc. can be used.

Riveting can be done cold, for steel rivets with a diameter of up to 10...12 mm as well as for brass, copper or aluminum rivets (all diameters), or hot by heating the rivet head (1000...1100 ° C), for steel rivets with large diameters.

NIT-T.6 FORMS AND CAUSES OF DISPOSAL OR IMPROPER BEHAVIOR

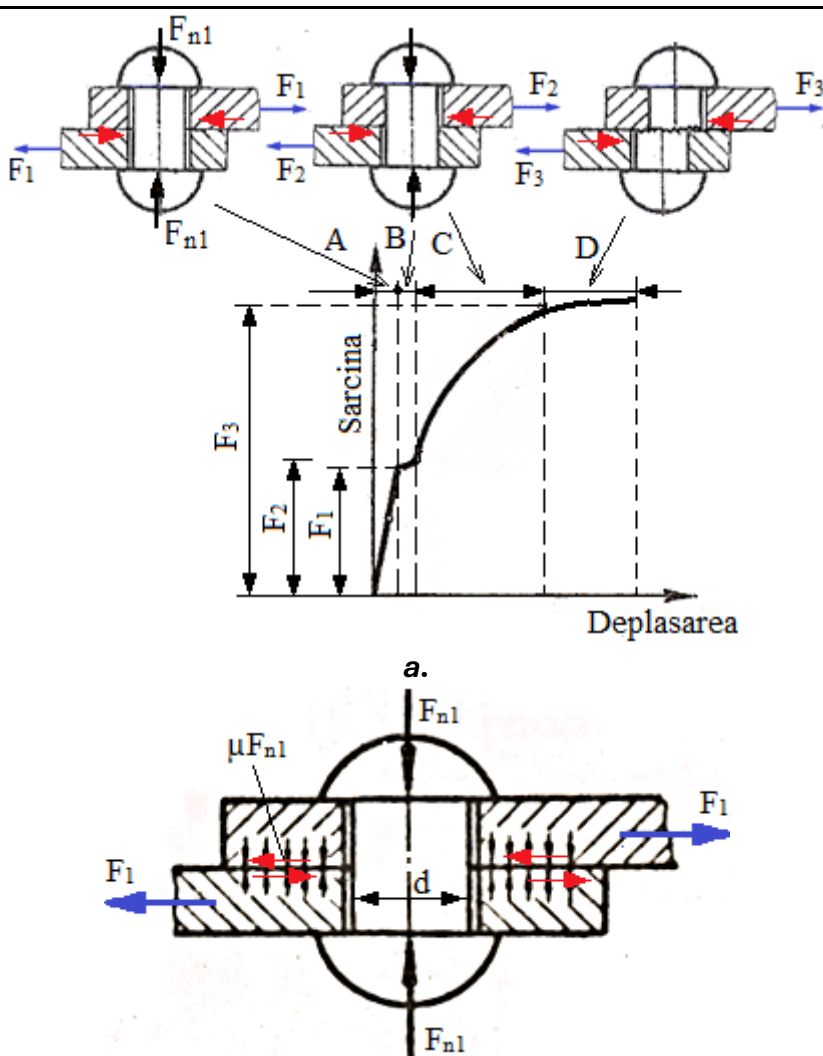
Tab. NIT-T.6.1 Forms and causes of inappropriate behavior or removal from use

<i>shapes</i>	<i>result</i>	<i>Appearance</i>	<i>CAUSE</i>	<i>Manifestation</i>	<i>Avoidance</i>
Crushing the rod	Unsafe operation	Plastic deformation of rivet rods in semi-cylindrical contact areas with sheets	Accidental and/or shock overloads	Damage to the rivet shank	Avoiding overloads; crushing resistance calculation
Rod breakage	Disposal	The breakage occurs in the areas between the assembled sheets	Accidental and/or shock overloads	Interruption of load transmission and/or loss of tightness	Avoiding overloads; shear strength calculation

NIT-T.7 FUNCTIONAL AND CONSTRUCTIVE PARAMETERS

NIT-T.8 CALCULATION MODELS

NIT-T.8.1 STRENGTH CALCULATION MODELS



The process of transmitting load through riveting

After riveting, due to the tightening of the parts and/or the cooling of the rivet (in the case of hot riveting), an axial force F_{n1} appears in its rod, which stresses the rod in tension.

When operating a riveted joint, four stages can be encountered:

- in the first, the external force being lower than the friction forces ($F_1 < \mu F_{n1}$), the external load is transmitted entirely through friction (case A in fig. NIT-T.8.1,a; fig. NIT-T.8.1,b) and the joined parts deform elastically linearly;
- in the second stage, when the external force becomes equal to or slightly greater than the friction forces ($F_1 \geq \mu F_{n1}$), the relative displacement with friction of the assembled sheets takes place within the clearance between the rivet and the hole (case B in fig. NIT-T.8.1,a; fig. NIT-T.8.1,c);
- in the third stage the external force increases ($F_2 > \mu F_{n1}$) and, in addition, from the frictional forces between the sheets the external force is taken over by the rivet rod which is stressed in crushing and shearing (case C in fig. NIT-T.8.1,a; fig. NIT-T.8.1,b); in the last stage the external force being much increased ($F_3 \gg \mu F_{n1}$), the tension in the rivet rod exceeds the yield point and consequently large relative displacements occur and finally breakage (case D, in fig. NIT-T.8.1,a).

In conclusion, in the cases of real riveted joints, **the external force can be transmitted** by friction, by friction and shape or only by shape (the rivet rod).

Calculation assumptions:

- the external force is distributed uniformly over the rivets and therefore the most stressed rivet will be calculated;
- the partial transmission of the external force through friction is neglected (this being difficult to evaluate), in design practice it is considered included in the values of admissible stresses which are superior to those frequently used for other elements;

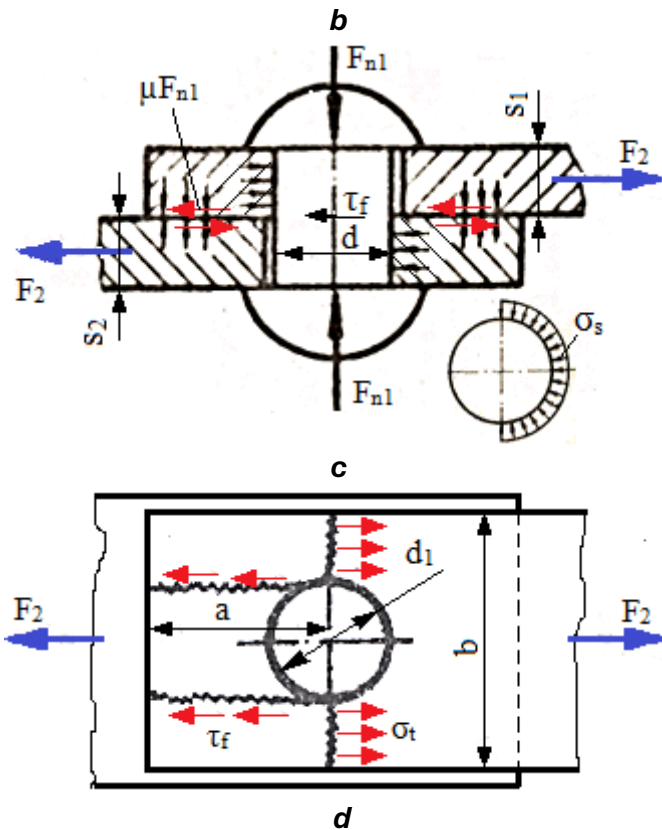


Fig. NIT-T.8.1 Strength calculation models: **a** – operation stages; **b** – friction force transmission model; **c** – shape force transmission model **d** – plate calculation model

- the stresses are distributed uniformly in the rivet section and on the contact surfaces between the rivet and the parts;
- the bending stress of the rivet rod that occurs as a result of the sliding of the sheets is neglected;
- the tensile stress of the rivet rod is neglected.

Rivet shear strength condition:

$$\tau_f = \frac{F_2}{i z \frac{\pi d_1^2}{4}} = \frac{4 F_2}{i \pi d_1^2} \leq \tau_{af}, \quad (\text{NIT-T.8.1.1})$$

where, d_1 represents the design diameter (equal to the hole diameter); i - the number of shear sections, z - the number of rivets; τ_{af} - the conventional allowable shear stress of the rivet material ($\tau_{af} = (100...100)$ MPa).

Rivet crushing strength condition:

$$\sigma_s = \frac{F_2}{z d_1 \min(s_1, s_2)} \leq \sigma_{as}, \quad (\text{NIT-T.8.1.2})$$

where, $s_{1,2}$ are the thicknesses of the assembled sheets; $\sigma_{as} = \min(,)$ - the minimum allowable, conventional, crushing stress of the rivet or sheet material ($\sigma_{as} = (240...320)$ MPa, for rivets; $\sigma_{as} = (210...280)$ MPa, for sheet metal); $\sigma_{as}^b \sigma_{as}^t$

Tensile strength condition of the joined sheet:

$$\sigma_t = \frac{F_2}{(b-d_1)s} \leq \sigma_{at}, \quad (\text{NIT-T.8.1.3})$$

where b is the width of the sheet; F_2 - the force that loads the sheet; s - the thickness of the sheet; σ_{at} - the allowable tensile stress of the sheet material ($\sigma_{at} = (110...160)$ MPa);

Shear resistance condition of the joined sheet:

$$\tau_f = \frac{F_2}{2(a-d_1)s} \leq \tau_{af}, \quad (\text{NIT-T.8.1.4})$$

where a is the length of the sheet edge; F_2 - the force loading the sheet; s - the thickness of the sheet; τ_{af} - the allowable shear stress of the sheet material ($\tau_{af} = (80...120)$ MPa);

Note: For the design calculation of a resistance riveted joint, the following steps are recommended:

- | | |
|--|--|
| | <ul style="list-style-type: none">- choosing the diameters of the rivet and the hole in the sheets depending on the thickness of the sheets;- determining from the shear resistance condition the force that a single rivet can take;- checking the rivet and the sheets for crushing;- determining the number of rivets by dividing the total external force by the force taken by a rivet;- tensile and shear testing of joined sheets |
|--|--|

AFS. THREADED AND SCREWED ASSEMBLIES

AFS-T. CONSTRUCTION AND MODELING OF THREADED ASSEMBLIES

CONTENTS

AFS-T.1 DEFINITION.....	5-3
AFS-T.2 CONSTRUCTION STRUCTURES	5-3
AFS-T.3 AREAS OF USE, ADVANTAGES AND DISADVANTAGES	5-3
AFS-T.4 CLASSIFICATION	5-4
AFS-T.5 MATERIALS AND TECHNOLOGIES	5-5
AFS-T.6 FORMS AND CAUSES OF DISPOSAL OR IMPROPER PERFORMANCE OF THREADED ASSEMBLIES.....	5-6
AFS-T.7 FUNCTIONAL AND CONSTRUCTIVE PARAMETERS	5-9
AFS-T.8 CALCULATION MODELS	5-10
AFS-T.8.1 Models for calculating forces and moments in threaded assemblies	5-10
AFS-T.8.1.1 Models for calculating forces and moments in non-prestressed threaded assemblies	5-10
AFS-T.8.1.2 Model for calculating forces in prestressed threaded assemblies	5-16
AFS-T.8.2 Threaded assembly strength calculation model	5-20
AFS-T.8.3 Strength calculation models for bolted joints	5-22
AFS-T.8.3.1 Strength calculation models for axially loaded bolted assemblies	5-22
AFS-T.8.3.2 Calculation models for the strength of transversely loaded bolted joints	5-27
AFS-T.9 CONSTRUCTION STRUCTURES	5-36
AFS-T.9.1 Construction forms of bolts, nuts and flat washers	5-36
AFS-T.9.2 Solutions for constructive optimization of threaded and bolted assemblies	5-41
AFS-T.9.3 Securing threaded assemblies against self-loosening	5-43

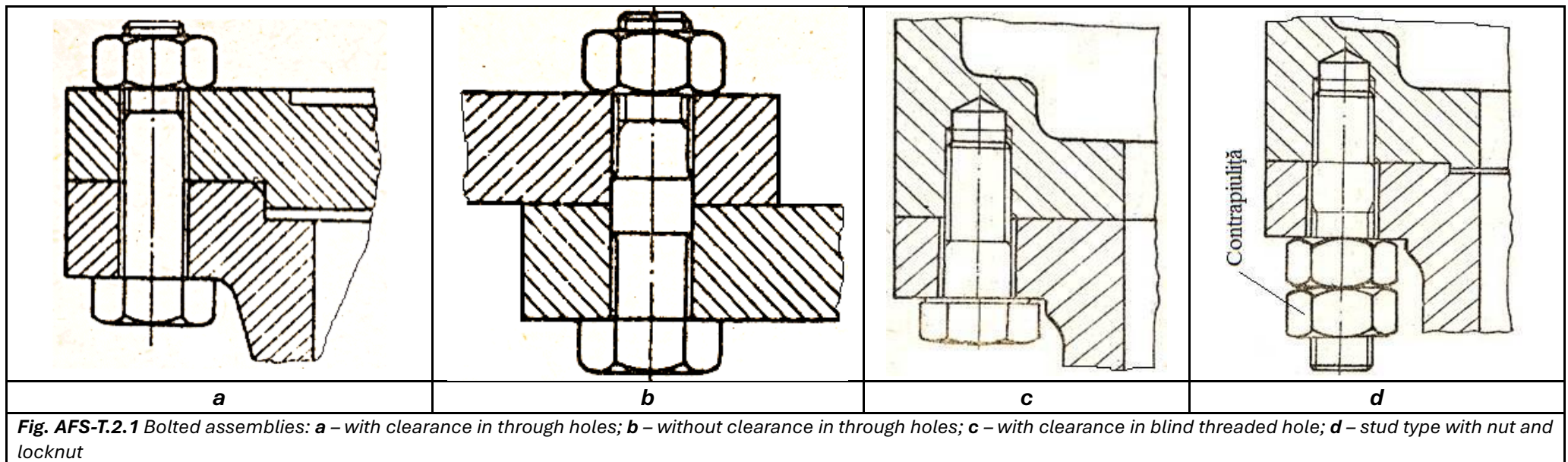
AFS-T.1 DEFINITION

The *thread* is presented in the form of **helical spirals** (protrusions) uniformly arranged on a cylindrical, conical or other type of surface on the outside or inside.

Threaded assemblies are direct, **removable connections** made through direct contact between mating threads, internal (of the nut) and external (of the bolt).

Bolted assemblies are **removable assemblies** that position and secure two or more elements (parts) by means of one or more threaded assemblies (bolt-nut).

AFS-T.2 CONSTRUCTION STRUCTURES



AFS-T.3 AREAS OF USE, ADVANTAGES AND DISADVANTAGES

Main areas of use: construction of machines, equipment, technological installations, constructions, etc.

Tab. AFS-T.3.1 Advantages and disadvantages of threaded assemblies

Advantages *	Disadvantages *
- high clamping (fixing) forces; - easy to assemble and disassemble with reduced forces; - safe in operation; - cheap because they are made in specialized enterprises, in mass production; - they are interchangeable, on an international scale; - ensures self-fixing condition, especially under static loading conditions	- the thread, due to its shape, is a strong stress concentrator; - requires additional insurance against self-unlocking, especially in the case of dynamic loads (vibrations)

AFS-T.4 CLASSIFICATION

Tab. AFS-T.4.1 Thread classification

<i>criterion</i>	<i>Type</i>	<i>Scheme</i>	<i>Description</i>
Destination	Fixing		Used in the construction of threaded assemblies
	Measuring/adjusting		They are used in the construction of measuring instruments (micrometers)/they are used for the relative positioning of elements in the construction of mechanical systems
	Of movement		It is used in the construction of helical couplings (screw-nut transmissions)
Number of beginnings	With a beginning	Fig. AFT-T.2.2,a	
	With two or more beginnings	Fig. AFT-T.2.2,c	It is used for large displacements (strokes) corresponding to small angles of rotation; there is a danger of not respecting the self-braking condition.
The direction of the thread's winding	On the right	Fig. AFT-T.2.2,c	Angular velocity, ω has the same meaning as linear velocity, v .
	On the left		Angular velocity, ω has the opposite direction to linear velocity, v .
By the measurement system	metric		They have dimensions in mm and are standardized; profile angle, $\alpha = 60^\circ$
	In inches		They have dimensions in inches and are rarely used; profile angle, $\alpha = 55^\circ$.
The surface on which the thread is generated	Cylindrical (external, internal)		
	Tapered (external, internal)		

Thread pitch size	With a big step		There is a danger of not respecting the self-braking condition
	At a normal pace		
	With a fine step		They are used for measuring and regulating threads; they meet the self-locking condition very well.
Spiral profile shape	Triangular with angle, $\alpha = 60^\circ$		Fastening assemblies
	Triangular with angle, $\alpha = 55^\circ$		Fastening and sealing assemblies
	Square, $\alpha = 0^\circ$		For higher efficiency but imprecise helical couplings
	Trapezoidal, $\alpha = 30^\circ$		For lower efficiency but more precise helical couplings

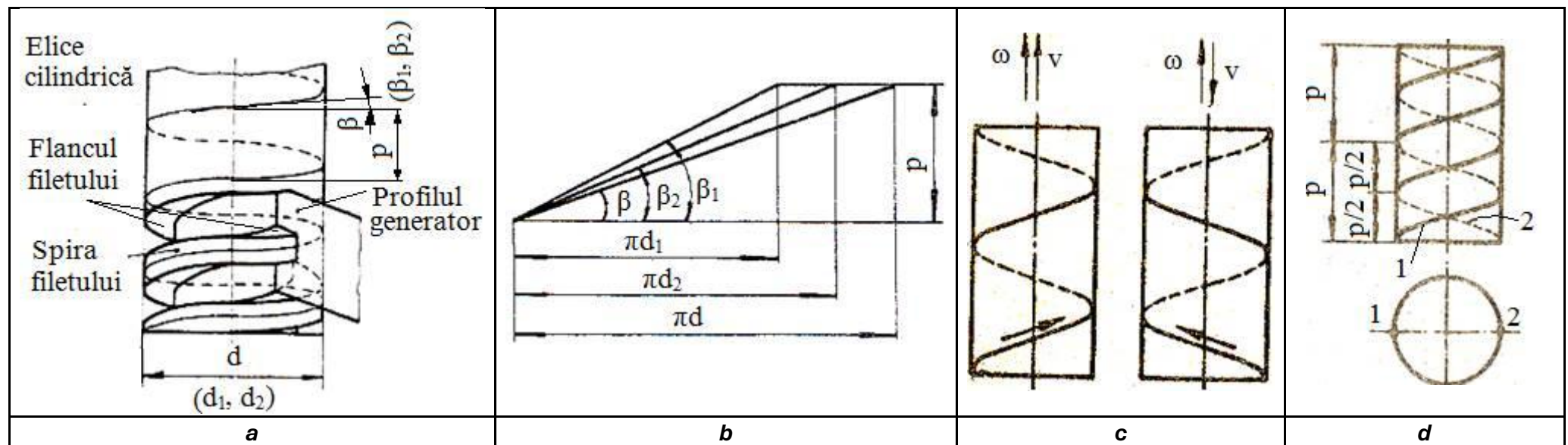


Fig. AFS-T.4.1 Thread generation features: **a** – overview of the generation process; **b** – variation of the thread helix angle with diameter; **c** – helix direction (right, left); **d** – multi-helix threads (with two starts)

AFS-T.5 MATERIALS AND TECHNOLOGIES

Steels (S235, E285, C35, C45, 40Cr10, 33MoCr11, 13CrNi30, etc.) for standardized fastening screws and nuts are divided into *mechanical property groups*, each group being associated with materials with appropriate properties. The symbols of the common mechanical property groups for screws are: 3.6, 4.6, 4.8, 5.6, 5.8, 6.8, 8.8, 10.9, 12.9; *meaning*: first number x 100 = minimum tensile strength (σ_{rmin}) in MPa and first number x second number x 10 = minimum yield strength (σ_{cmin}) in MPa (e.g. for group 6.8, $\sigma_{rmin} = 6 \times 100 = 600$ MPa, $\sigma_{cmin} = 6 \times 8 \times 10 = 480$ MPa). The symbols of the usual

mechanical characteristics groups for nuts are: 4, 5, 6, 8, 10, 12; *meaning*: number x 100 = minimum tensile strength (σ_{rmin}) in MPa (e.g. for group 8, $\sigma_{rmin} = 8 \times 100 = 800$ MPa).

The thread execution technology depends on the production series: in the case of unique pieces and small series, it is done by threading with a die, for the screw, and with a tap, for the nut, or by turning with threading knives (the fiber is interrupted, fig. AFS-T.5.2, a) in the case of large production series, the threading is done by cutting with special tools (comb knife or disc) or by plastic deformation by rolling (the continuity of the material fibers is maintained, fig. AFS-T.5.2, b), on automatic machines.

	 a	 b
Fig. AFS-T.5.1 Marking of the material characteristic group	Fig. AFS-T.5.2 Fiber shape of the laminated semi-finished product in the thread area: a – processed by machining (turning); b – cold plastic deformation	

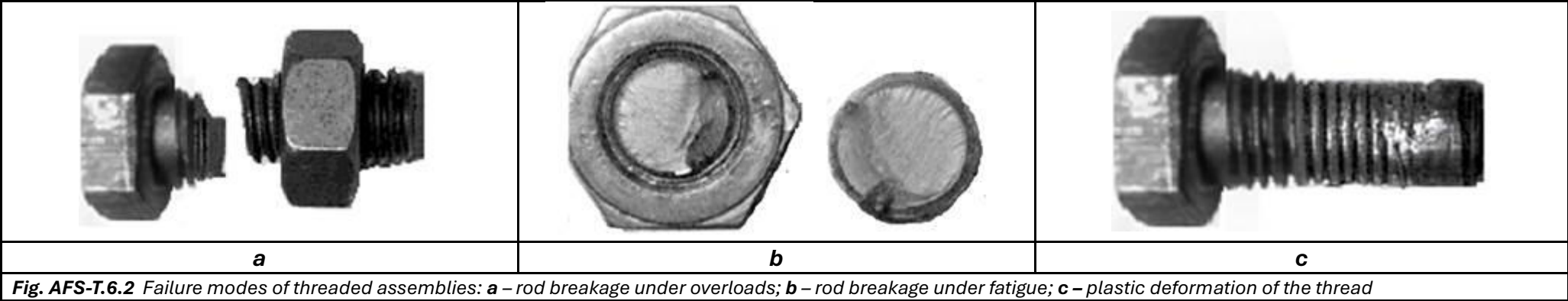
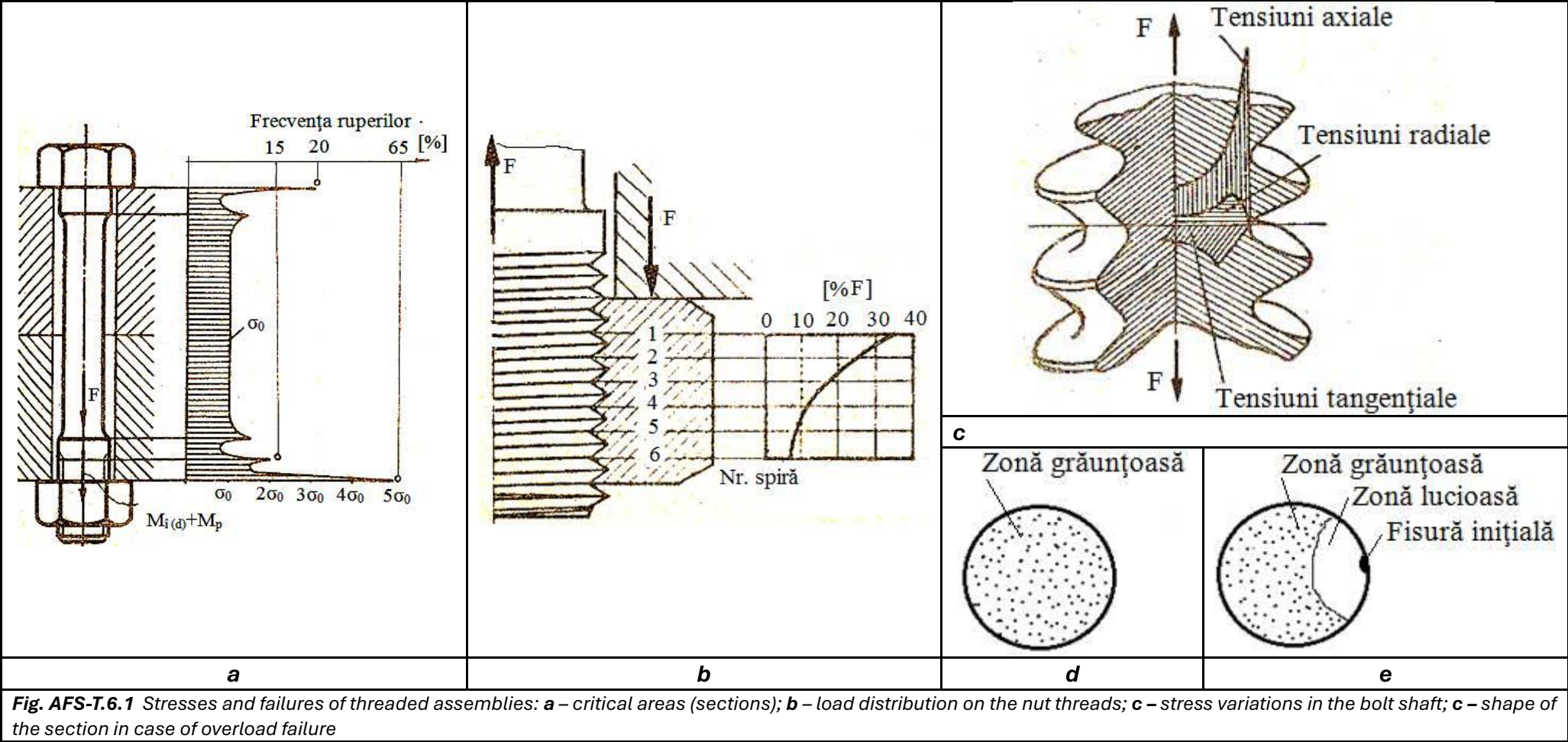
AFS-T.6 FORMS AND CAUSES OF DISPOSAL OR IMPROPER PERFORMANCE OF THREADED ASSEMBLIES

Tab. AFS-T.6.1 Forms and causes of inappropriate behavior or removal from use

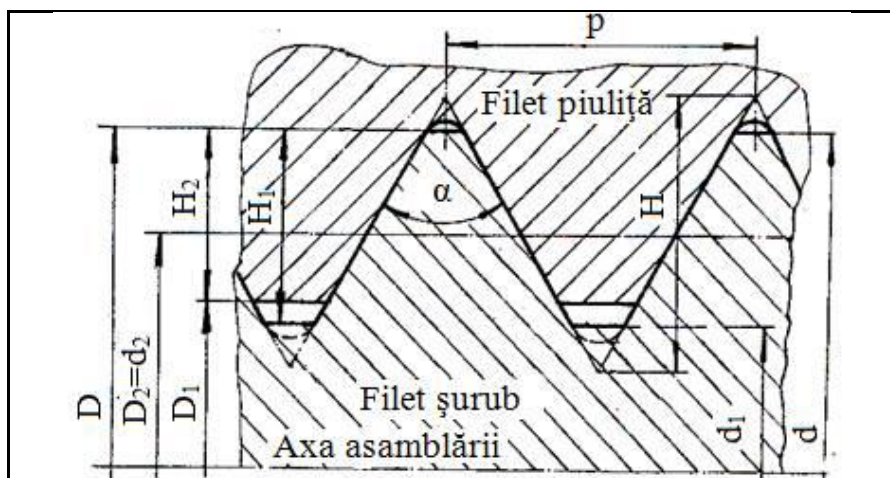
Shapes	Result	Appearance	Cause	Manifestation	Avoidance
Screw rod breakage due to overload	Disposal	The rupture occurs in the section with the minimum area (fig. AFS-T.6.2,a,b)	Accidental and/or shock overloads	Interruption of the transmission of the load	Avoiding overloads and/or mounting with pre-tightening so that no play and therefore shocks occur during operation.
Fatigue failure of the screw shaft	Disposal	The breakage (fig. AFS-T.6.1,e; AFS-T.6.2,b) occurs frequently (approx. 65%,) in the area of the first turn of the nut in contact with the turn as a result of the	Variable loads and stress concentrator at the base of the tooth (fig. AFS-T.6.2,c; stresses in the inner	Interruption of the transmission of the load	Limitation by calculation of fatigue stresses to admissible values; constructive measures to reduce the causes

		maximum load (approx. 30%F, fig. AFS-T.6.1,a,b)	area of the thread are maximum)		
Damage to the screw and/or nut thread	Difficult disassembly and impossibility of reassembly	In the contact area of the thread flank of the nut and bolt	Exceeding the crushing strength of the nut and/or bolt material	Plastic deformation of the thread flanks and modification of its pitch in the tight zone	Limitation by calculation of crushing stresses in the areas of the coils in contact to permissible values

The high frequency of fatigue failures, usually in the area of the first turn in contact with the nut, can be explained by the uneven distribution of the load on the turn - the first turn takes on approximately 35% of the load (Fig. AFS-T.6.1,b) and the stress in this area is 5 times higher than in the rod (Fig. AFS-T.6.1,a) - and by the uneven distribution of stresses (Fig. AFS-T.6.1,c) in the section of the bolt with maximum values, especially of the axial stress, in the area of the stress concentrator, the connection at the bottom of the thread.

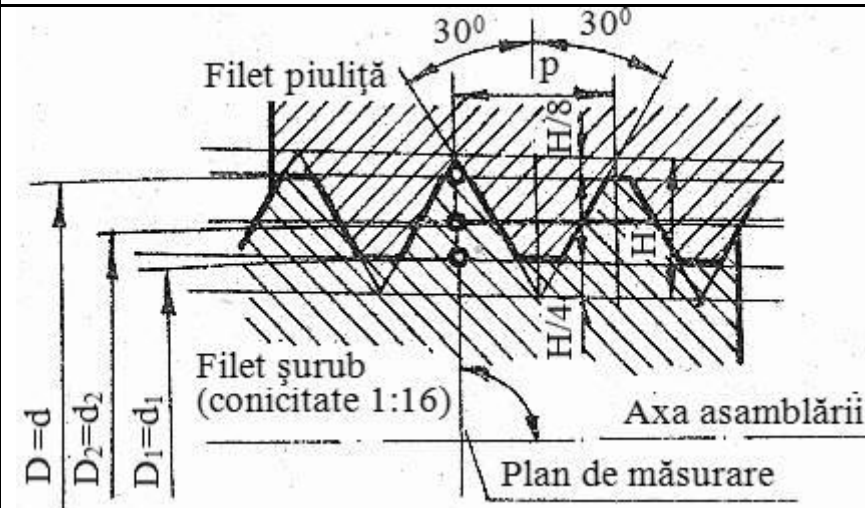


AFS-T.7 FUNCTIONAL AND CONSTRUCTIVE PARAMETERS



Main geometric parameters:

- nominal thread diameter, d is the outer diameter of the screw thread;
- the outer diameter of the nut thread, D ;
- the average diameter of the screw and nut thread, $d_2 = D_2$ (the diameter of the cylinder on whose generator the thicknesses of the solid and hollow are equal);
- the inner diameter of the screw thread, respectively the nut, d_1 , D_2 ;
- thread pitch, p (distance between homologous points of two consecutive turns);
- height of the thread generator profile, H ;
- effective height of the thread profile, H_1 ;
- the useful height of the thread profile, H_2 (the height of the profile in contact between the threads of the screw and the nut);

A

- thread profile angle, α ($\alpha = 60^\circ$, for metric threads; $\alpha = 55^\circ$, for Whitworth threads, in inches);

- the angle of inclination of the thread helix, β ; it is variable, depending on the cylinder considered (fig. AFS-T.4.1,a,b); in the calculations the angle of inclination corresponding to the average diameter, d_2 is used

$$\beta_2 = \arctg \frac{p}{\pi d_2} \quad (\text{AFS-T.7.1})$$

Conical threads are used, in particular, for assembling parts that also require good sealing; they can have the bisector of the profile apex angle perpendicular to the assembly axis (fig. AFS-T.1,c) - they can also be assembled with cylindrical threads - or on the generator of the conical surface (AFS-T.1,d) which is recommended at high pressures and temperatures; the geometric elements of conical threads depend on the measurement plane

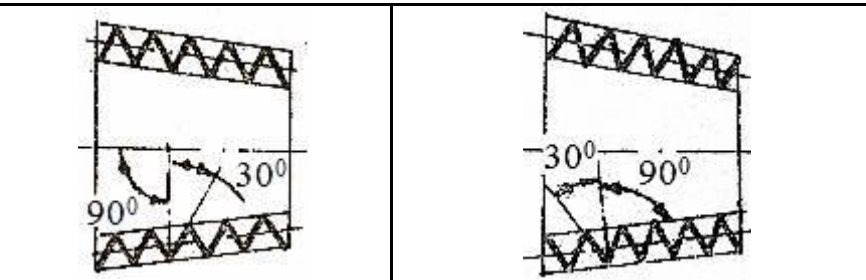
b**c****d**

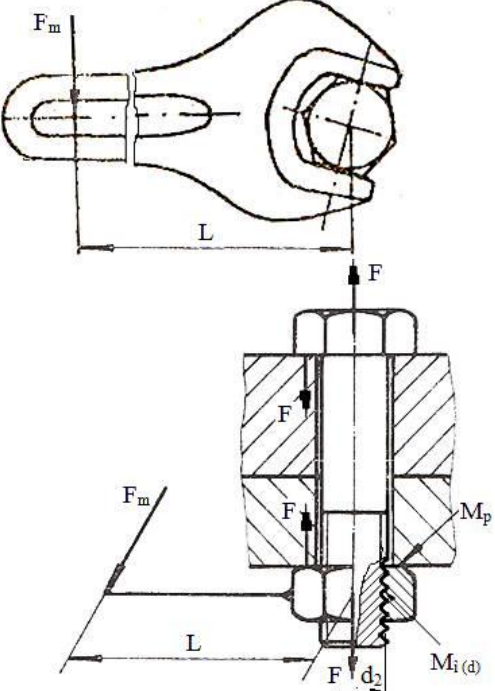
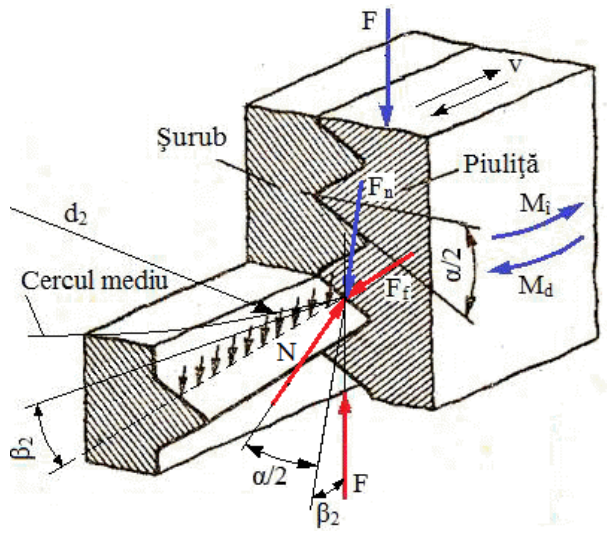
Fig. AFS-T.7.1 Geometric parameters of fastening and sealing threads: **a** – cylindrical; **b** – conical; **c** – conical perpendicular to the screw axis; **d** – conical perpendicular to the generators

AFS-T.8 CALCULATION MODELS

AFS-T.8.1 MODELS FOR CALCULATING FORCES AND MOMENTS IN THREADED ASSEMBLIES

AFS-T.8.1.1 Models for calculating forces and moments in non-prestressed threaded assemblies

Tab. AFS-T.8.1.1.1 Screwing (unscrewing) torque calculation model

		Calculation assumptions: - the movement of the nut is assimilated to the movement of an angular sector on an inclined plane tangent to the mean circle; - the pressure distribution on the contacting surfaces is uniform and the resulting force is considered concentrated at a point at the level of a medium circle. Forces and moments in thread ($\alpha > 0$): - <u>external axial force</u>, F; - <u>the normal reaction force</u>, N, resulting from the contact pressures between the threads of the nut and the bolt; - <u>the normal force in the tangent plane</u>, F_n, is the component of the force N, $F_n = N \cos \frac{\alpha}{2} ; (\text{AFS-T.8.1.1.1})$
---	--	--

a	b	- the friction force (opposite direction of the relative speed between the threads of the screw and the nut), $F_f = \mu N$ which, taking into account (AFS-T.8.1.1.1), becomes, $F_f = \mu N = \frac{\mu}{\cos \frac{\alpha}{2}} F_n = \mu' F_n = \operatorname{tg} \varphi' F_n$ (AFS-T.8.1.1.2) where, μ is the friction coefficient; $\mu' = \mu / \cos(\alpha/2)$ - the apparent friction coefficient (reduced); $\varphi' = \operatorname{arctg} \mu'$ - the apparent friction angle (reduced); - the radial reaction force, F_r, is the component of the force N in the radial plane perpendicular to the axial plane, xx/yy; - the pushing force of the nut body on the inclined plane during screwing, H and
c	d	

Fig. AFS-T.1.1.1 Calculation schemes of forces and moments in threaded assembly: **a** – general scheme; **b** – threaded assembly detail; **c** – screwing (climbing the inclined plane); **d** – unscrewing (descenting the inclined plane);

unscrewing, H' is determined from the force balance relations according to those xx and yy :

$$H - F_f \cos \beta_m - F_n \sin \beta_2 = 0,$$

$$F + F_f \sin \beta_m - F_n \cos \beta_2 = 0,$$

which after taking into account the friction force relationship, F_f become

$$H - F_n (\operatorname{tg} \varphi' \cos \beta_2 + \sin \beta_2) = 0,$$

$$F - F_n (-\operatorname{tg} \varphi' \sin \beta_2 + \cos \beta_2) = 0$$

and after processing it results,

$$H = F \operatorname{tg}(\varphi' + \beta_2). \quad (\text{AFS-T.8.1.1.3})$$

Similarly, for unscrewing we obtain,

$$H' = F \operatorname{tg}(\varphi' - \beta_2) \quad (\text{AFS-T.8.1.1.4})$$

the screwing and unscrewing torque,

$$M_{i(d)} = H(H') \frac{d_2}{2} = F \frac{d_2}{2} \operatorname{tg}(\varphi' \pm \beta_2) =$$

$$= F \frac{d_2}{2} \operatorname{tg}(\operatorname{arctg} \frac{\mu}{\cos \frac{\alpha}{2}} \pm \beta_2)$$

(AFS-T.8.1.1.5)

Obs. a. In fig. AFS-T the calculation model of forces and moments is presented for the case of square thread ($\alpha = 0$), not used for fastening threads but found in helical coupling threads; the calculation relations are particular cases of the general case $\alpha > 0$ above with $\alpha = 0$.

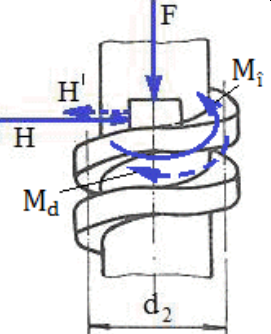
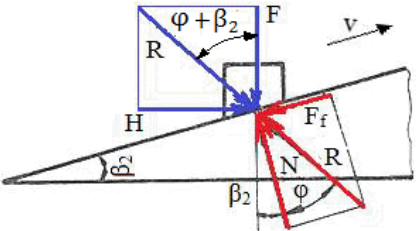
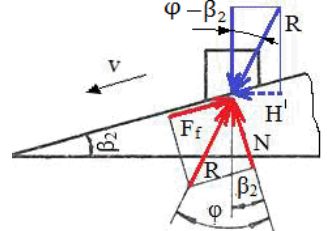
			Forces and moments in the square thread (particular case, $\alpha = 0$): - external axial force, F; - normal reaction force, N; - action/reaction force, R - friction force, $\phi = \operatorname{arctan} \mu$; $F_f = \mu N = \operatorname{tg} \varphi N$ - the pushing force of the nut body on the inclined plane when screwing and unscrewing; $H = F \operatorname{tg}(\varphi + \beta_2)$ $H' = F \operatorname{tg}(\varphi - \beta_2)$ - screwing and unscrewing torque, respectively $M_{i(d)} = H(H') \frac{d_2}{2} = F \frac{d_2}{2} \operatorname{tg}(\varphi \pm \beta_2)$
A	b	c	

Fig. AFS-T.1.1.2 Calculation schemes for forces and moments in square profile threaded assembly ($\alpha = 0$): **a** – general scheme; **b** – screwing (climbing the inclined plane); **d** – unscrewing (descenting the inclined plane);

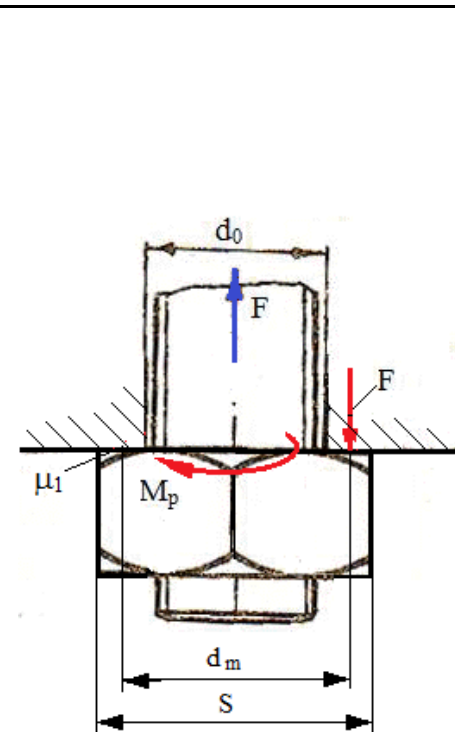
Note.

Self-locking condition (self-locking): the nut must not loosen by itself under the action of the force F in the direction of its unscrewing; from a mathematical point of view this condition is equivalent to one of the inequalities, $M_d > 0$ or $H' > 0$, from which it follows, $\beta_2 < \varphi'$ and, therefore,

$$\beta_2 < \arctg \frac{\mu}{\cos \frac{\alpha}{2}}, \quad (\text{AFS-T.8.1.1.6})$$

for the thread with $\alpha > 0$ or $\beta_2 < \varphi$ and $\beta_2 < \arctg \mu$, for the square thread $\alpha = 0$.

Tab. AFS-T.8.1.1.2 *Pivoting moment calculation model (friction between nut and assembled part)*



A

Calculation assumptions:

- the coefficient of friction between the nut and the piece on which it sits, μ_1 , is constant;
- the pressure is considered to be uniformly distributed on the contact surface (fig. AFS-T.1.1.2, b),

$$p = \frac{F}{\frac{\pi}{4}(S^2 - d_0^2)}, \quad (\text{AFS-T.8.1.1.6})$$

where d_0 represents the diameter of the through hole and S – the opening (size) of the wrench for tightening the nut.

Pivoting moment - simplified model (case I, fig. AFS-T.1.1.2, a)

The pivoting friction moment, considering the normal force on the contact surface acting at the level of the circle with the average diameter d_m , is determined with the relation,

$$M_p = \mu_1 F = \mu_1 F. \quad (\text{AFS-T.8.1.1.7}) \quad \frac{d_m S + d_0}{2} \quad 4$$

Pivoting moment - more precise model (case II, fig. AFS-T.1.1.2, b)

The pivoting friction moment is determined, taking into account the uniform pressure distribution ($p = \text{ct.}$) on the contact surface (according to the above assumption), with the relationship,

$$M_p = \int dM_f = \int dF_f r = \int r \mu_1 dF_n = \int r \mu_1 p dA = \int_{\frac{d_0}{2}}^{\frac{S}{2}} r \mu_1 p 2\pi r dr = 2\pi \mu_1 p \int_{\frac{d_0}{2}}^{\frac{S}{2}} r^2 dr = \frac{\pi}{12} \mu_1 p (S^3 - d_0^3)$$

(AFS-T.8.1.1.8)

which, taking into account the relationship (AFS-T.8.1.1.6) becomes,

$$M_p = \frac{1}{3} \mu_1 F \frac{S^3 - d_0^3}{S^2 - d_0^2} \quad (\text{AFS-T.8.1.1.9})$$

Moment and force acting on the key (crank):

From the equilibrium equation of the nut (fig. AFS-T.1.1.1,a) $M_m = F_m L$, where

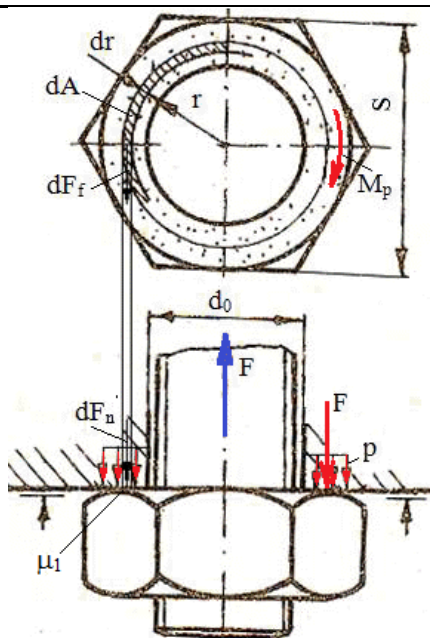
**b**

Fig. AFS-T.1.1.2 Calculation scheme of the friction pivot moment: **a** - with concentrated force (case I), **b** - with uniformly distributed pressure (case II)

$$M_m = M_i + M_p = F \left[\frac{d_2}{2} \operatorname{tg}(\beta_m + \varphi') + \frac{1}{3} \frac{s^3 - d_0^3}{s^2 - d_0^2} \right], \quad (\text{AFS-T.8.1.1.1.10})$$

or, considering the simplified relationship (AFS-T.8.1.1.1.7) of the pivot moment,

$$M_m = F, \quad (\text{AFS-T.8.1.1.1.11}) \left[\frac{d_2}{2} \operatorname{tg}(\beta_2 + \varphi') + \mu_1 \frac{s + d_0}{4} \right]$$

the calculation relationship for the action force of the key results (fig. AFS-T.1.1.1,a),

$$F_m = \frac{F}{L} \left[\frac{d_2}{2} \operatorname{tg}(\beta_m + \varphi') + \frac{1}{3} \mu_1 \frac{s^3 - d_0^3}{s^2 - d_0^2} \right]. \quad (\text{AFS-T.8.1.1.1.12})$$

or, respectively,

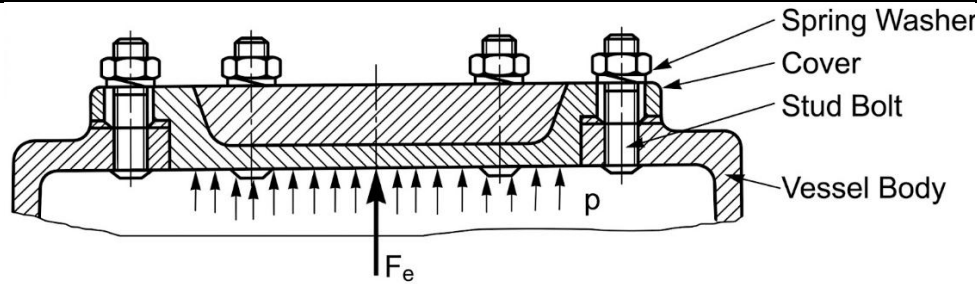
$$F_m = \frac{F}{L} \left[\frac{d_2}{2} \operatorname{tg}(\beta_2 + \varphi') + \mu_1 \frac{s + d_0}{4} \right] \quad (\text{AFS-T.8.1.1.1.13})$$

Note. For average statistical values of the parameters in the relations (AFS-T.8.1.1.1.12, (AFS-T.8.1.1.1.13)) and in

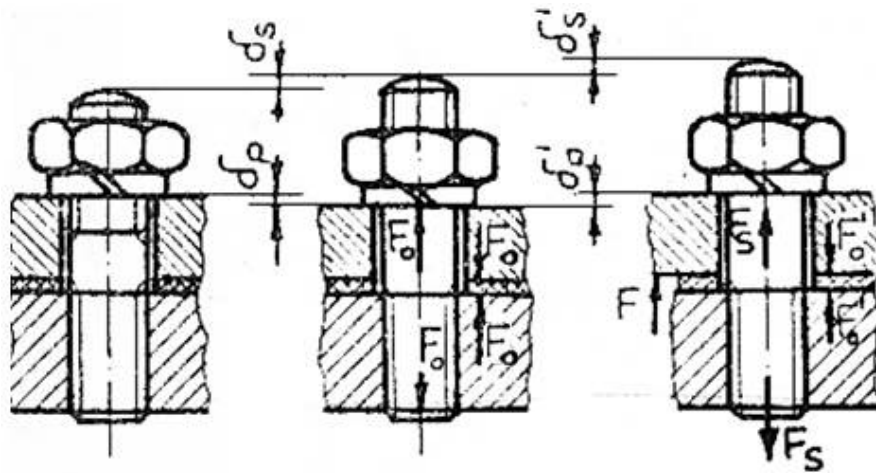
the case of using standardized wrenches ($L \approx 12 d$), it results that $F \approx 80 F_m$, even when the wrench is operated with a force $F_m = 150 \dots 350 \text{ N}$ (average manual force) of screws with small diameters, there is a risk of the screw breaking and it is recommended to use torque wrenches or limiting ones, thus controlling the tightening torque and implicitly the force F that requires assembly.

AFS-T.8.1.2 Model for calculating forces in prestressed threaded assemblies

Tab. AFS-T.8.1.2.1 *Calculation model for assemblies with axially loaded prestressed bolts*



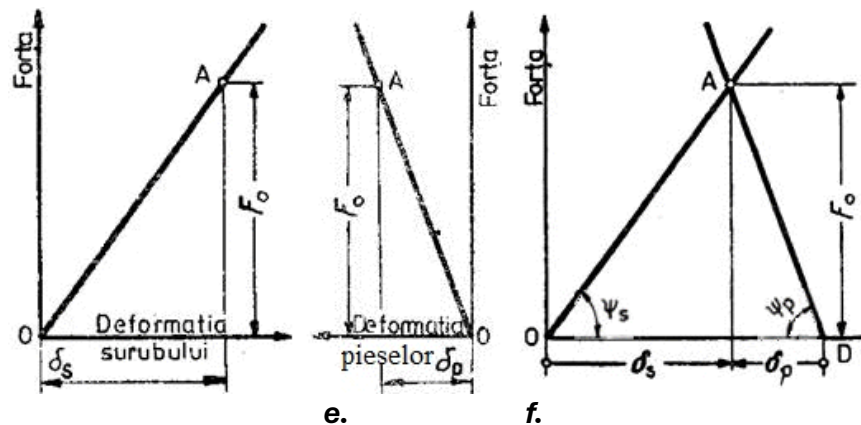
a



b.

c.

d.



e.

f.

Calculation assumptions:

- the screws securing the container lid are identical in size and load;
- the external operating force is considered static, but can also be pulsating;
- the initial tightening of the assembly must ensure that the tightness is maintained even during operation with pressure inside the container;
- considering the deformations of the assembly elements in the elastic domain, the stiffnesses of the screw and the tightened parts are determined with the relations:

$$k_s = \frac{F_0}{\delta_s} = \tan \psi_s, \quad k_p = \frac{F_0}{\delta_p} = \tan \psi_p, \quad (\text{AFS-T.8.1.2.1})$$

where, ψ_s and ψ_p are the angles of the elastic characteristics associated with the screw and the tightened parts, respectively; these quantities are determined by calculation or experimentally.

Installation and operation

- pre-assembly phase (fig. AFS-T.1.2.1,b), the screws and nuts are assembled with small forces until the clearances are eliminated and deformations do not occur during assembly;
- the actual assembly phase (fig. AFS-T.1.2.1,c,e,f), the nuts are tightened with the wrench and, as a result, the pre-tightening force, F_0 , appears in the assembly elements, which stresses the screws in tension and elongates them with the arrow δ_s , and compresses the assembled parts with the arrow δ_p , different from δ_s because the rigidities are different;
- the exploitation phase (fig. AFS-T.1.2.1,d,g) the external action occurs with the force, F_e , consequence of the introduction of a fluid with pressure p into the container, and as a result the screws are additionally elongated by the value δ'_s and the parts are decompressed by the value $\delta'_p = \delta'_s$..

Force calculation relationships:

The additional force that loads the screw, F_{sl} , respectively, the operating (residual) clamping force of the parts, F'_0 , a consequence of the action of the external force (operating, generated by the fluid under pressure) is determined based on the relationships:

$$F_{sl} = \delta'_s \tan \psi_s = \delta'_s k_s, \quad (\text{AFS-T.8.1.2.2})$$

$$F'_0 = F_0 - \delta'_p \tan \psi_p = F_0 - \delta'_p k_p$$

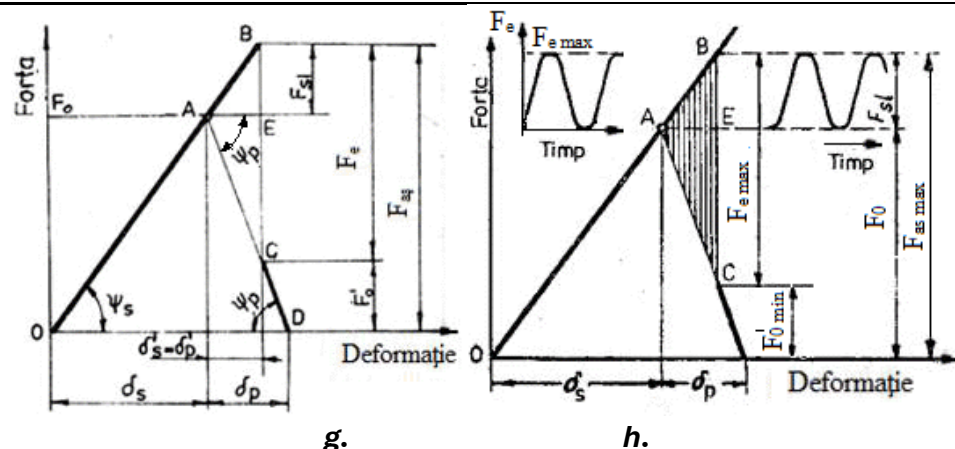


Fig. AFS-T.1.2.1 Calculation schemes: **a** – structure and loading; **b** – initial state; **c** – prestressed state; **d** – operating state (with external action); **e** – elastic characteristics of the bolt and nut; **f** – elastic characteristic of the prestressed threaded assembly; **g** – elastic characteristic of the prestressed threaded assembly externally loaded with static load; **h** – elastic characteristic of the prestressed threaded assembly externally loaded with variable load

and

$$F_e = F_{sl} + (F_0 - F'_0),, \quad (\text{AFS-T.8.1.2.3})$$

$$F_e = \delta'_s (k_s + k_p)$$

from which, taking into account the relationships (AFS-T.8.1.2.2), it results,

$$F_{sl} = \frac{k_s}{k_s + k_p} F_e = \chi F_e, \quad (\text{AFS-T.8.1.2.4})$$

where, denoted by $\chi = k_s / (k_s + k_p)$, the stiffness of the assembly.

From Fig. AFS-T.1.2.1,g the axial force loading the bolt and the axial force compressing the parts after the application of the operating force F_e , we obtain the total axial force in the bolt,

$$F_{as} = F_0 + \chi F_e, \quad (\text{AFS-T.8.1.2.5})$$

and the axial clamping force (remaining) in service,

$$F'_0 = F_0 - (1 - \chi) F_e \quad (\text{AFS-T.8.1.2.6})$$

Condition for ensuring tightness

To ensure tightness in the area of the assembled parts it is necessary that,

$F'_0 \geq 0$ and therefore, taking into account the relationship (AFS-T.8.1.2.6) the inequality takes the form,

$$F_0 \geq (1 - \chi) F_e, \quad (\text{AFS-T.8.1.2.7})$$

If the operating force is variable, as is frequently the case, the pulsating variation with the maximum force $F_{e \max}$ (fig. AFS-T.1.2.1,h), the maximum total axial force in the screw and the minimum tightening (remaining) force in operation are:

$$F_{as \max} = F_0 + \chi F_{e \max}, \quad (\text{AFS-T.8.1.2.8})$$

$$F'_{0 \min} = F_0 - (1 - \chi) F_{e \max} \quad (\text{AFS-T.8.1.2.9})$$

In this case, the condition for ensuring tightness (AFS-T.8.1.2.7) becomes,

$$F_0 \geq (1 - \chi) F_{e \max} \quad (\text{AFS-T.8.1.2.10})$$

Calculation of stiffnesses:

For bolts with constant section, from the expression for tensile elongation,

$$\delta_s = \frac{F_0 l_s}{A_s E_s}, \quad (\text{AFS-T.8.1.2.8})$$

the screw stiffness relation is obtained,

$$k_s = \frac{F_0}{\delta_s} = \frac{A_s E_s}{l_s}, \quad (\text{AFS-T.8.1.2.9})$$

where, A_s represents the cross-sectional area of the bolt, E_s – the modulus of elasticity of the bolt material, and l_s – the active length of the bolt measured from half the height of the bolt head to half the height of the nut.

In the case of bolts with n different sections, the stiffness is determined by considering the bolt with n sections with constant section *in series* with the relation,

$$k_s = \frac{1}{\sum_{i=1}^n \frac{1}{k_{si}}}, \quad (\text{AFS-T.8.1.2.10})$$

where, $k_{si} = A_{si} E_s / l_{si}$ – are the stiffnesses of sections 1, 2...n with A_{si} and l_{si} – the cross-sectional areas and, respectively, the lengths of the screw sections.

The rigidity of the assembled parts is determined by considering that the volume of material that participates in the load transmission is delimited on the outside by two frustoconical surfaces with the generator inclined at an angle γ (frequently, 45°), and on the inside by the surface of the screw passage with diameter d_0 .

Similarly, considering the assembled parts as cylindrical sections (equivalent to frustoconical sections) *in series*, the stiffness of the assembled parts is determined by the relationship

$$k_p = \frac{1}{\frac{1}{k_{p1}} + \frac{1}{k_g}}, \quad (\text{AFS-T.8.1.2.11})$$

where, $k_{p1} = A_{p1} E_{p1} / l_{p1}$ is the stiffness of the compressed section

associated with the cover, $A_{p1} = \frac{\pi}{4} (D_e^2 - d_0^2) = \frac{\pi}{4} ((S + l_{p1})^2 - d_0^2)$

, the cross-sectional area of the cylindrical section of equivalent diameter $D_{\text{associated}}$ with the cover, E_{p1} – the modulus of elasticity of the cover material,

l_{p1} – the thickness of the cover flange; $k_g = A_g E_g / l_g$ is the stiffness of the

section associated with the gasket with $A_g = \frac{\pi}{4} (D_o^2 - d_0^2)$, E_{p1} – the modulus of elasticity of the gasket material, l_g – the thickness of the gasket.

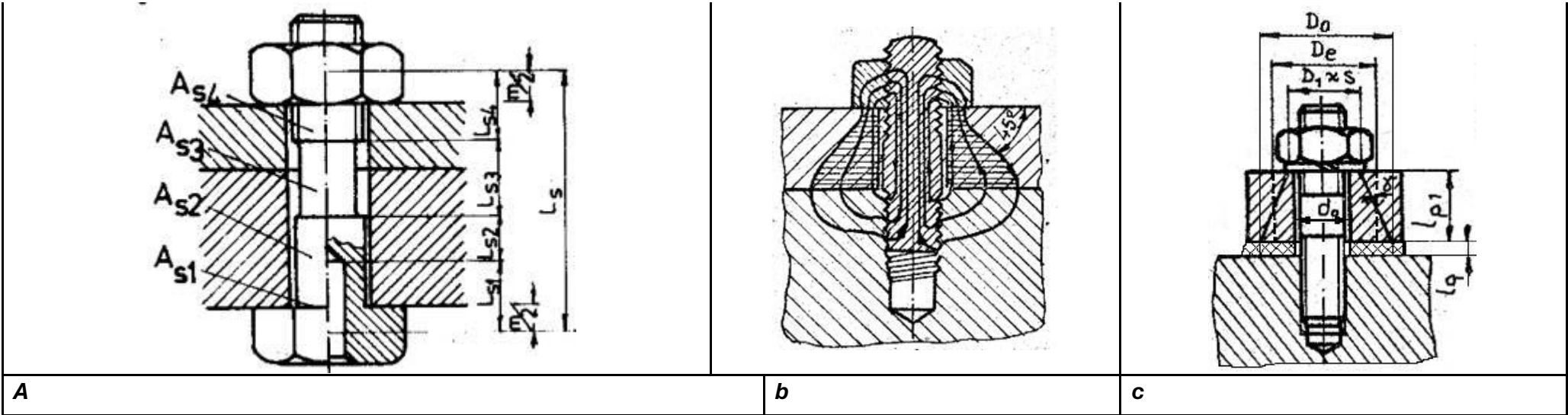


Fig. AFS-T.1.2.2 Calculation schemes for the stiffness of assembled parts: **a** – bolt with nut; **b** – stud with nut; **b** – stud with nut and sealing gasket

AFS-T.8.2 THREADED ASSEMBLY STRENGTH CALCULATION MODEL

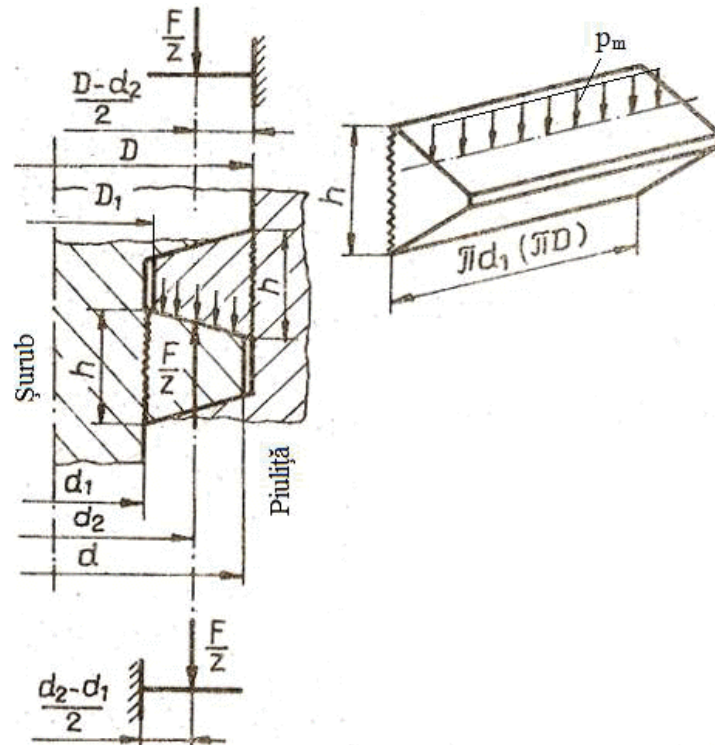


Fig. AFS-T.2.1.1 Calculation schemes for threads and threaded assemblies

Calculation assumptions:

- the axial load F is distributed uniformly over the z turns of the nut in contact with the turns of the nut;
- the load F/z acting on a coil is considered uniformly distributed on the contact surface with the pressure p_m in the direction parallel to the thread axis.

Crush resistance condition of the threaded assembly:

$$p_m = \frac{\frac{F}{z}}{\frac{\pi}{4}(d^2 - D_1^2)} = \frac{F}{\frac{\pi}{4}(d^2 - D_1^2)z} \leq \sigma_{as}, \quad (\text{AFS-T.8.2.1})$$

where, $\sigma_{as} = \min(\sigma_{as}^s, \sigma_{as}^p)$, where σ_{as}^s and σ_{as}^p are the allowable crushing stresses of the bolt and nut materials, respectively; for helical couplings $\sigma_{as} = p_a$, the allowable pressure of the lubricant film

Shear resistance conditions of the screw and nut thread:

$$\tau_f^s = \frac{\frac{F}{z}}{\pi d_1 h} = \frac{F}{\pi d_1 h z} \leq \tau_{af}^s, \quad (\text{AFS-T.8.2.2})$$

$$\tau_f^p = \frac{\frac{F}{z}}{\pi D h} = \frac{F}{\pi D h z} \leq \tau_{af}^p, \quad (\text{AFS-T.8.2.3})$$

where, τ_{af}^s and τ_{af}^p are the allowable shear stresses of the bolt and nut materials, respectively; h - the thickness of the thread at the base.

Conditions for bending strength of the screw and nut thread:

$$\sigma_i^s = \frac{M_i}{W_z} = \frac{\frac{F}{z} \frac{d_2 - d_1}{2}}{\frac{\pi d_1 h^2}{6}} = \frac{3 F (d_2 - d_1)}{\pi d_1 h^2 z} \leq \sigma_{ai}^s, \quad (\text{AFS-T.8.2.1.5})$$

$$\sigma_i^p = \frac{M_i}{W_z} = \frac{\frac{F}{z} \frac{D - d_2}{2}}{\frac{\pi D h^2}{6}} = \frac{3 F (D - d_2)}{\pi D h^2 z} \leq \sigma_{ai}^p, \quad (\text{AFS-T.8.2.1.6})$$

where, σ_{ai}^s and σ_{ai}^p are the allowable bending stresses of the bolt and nut materials, respectively.

Note: The verification of the threaded assembly and the threads of the screw and nut threads is done only in the case of using non-standard threads; for standardized threads, the above calculations are not necessary because the height of the nut was determined by imposing the condition of equal resistance of the screw rod and the threaded threads, resulting for the nut o with a height $m=0.8d$.

AFS-T.8.3 STRENGTH CALCULATION MODELS FOR BOLTED JOINTS

AFS-T.8.3.1 Strength calculation models for axially loaded bolted assemblies

Table AFS-T.8.3.1.1 Strength calculation model of bolted assembly under centric axial load

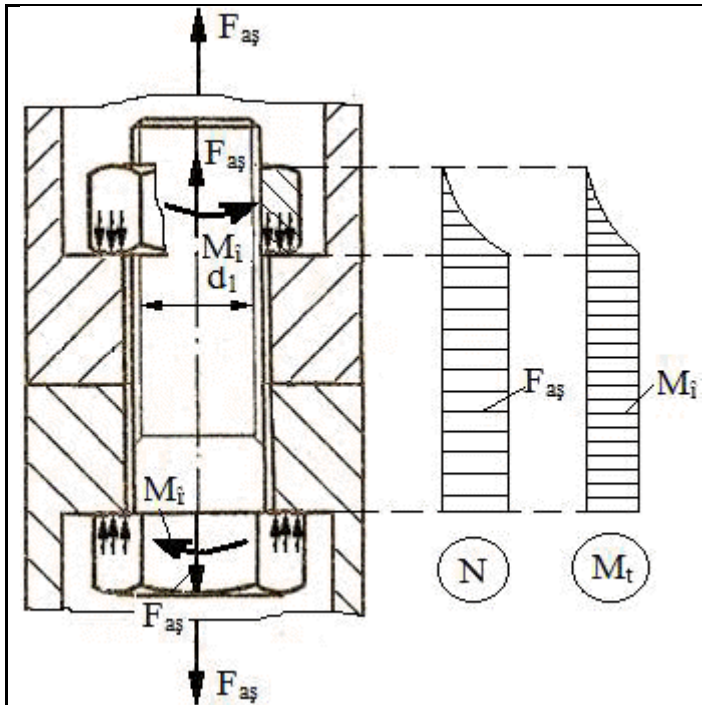


Fig. AFS-T.8.3.1 Calculation scheme of the assembly with a centric axially loaded screw

Calculation assumptions:

- the screw rod, in the worst case (tightening under load), is stressed in tension by the force F_{as}

$$M_i = F_{as} \frac{d_2}{2} \operatorname{tg}(\varphi' + \beta_m)$$

and in torsion by the screwing moment, (see rel. (AFS-T.8.1.1.5));

- the critical section, circular with a diameter equal to the internal diameter of the thread d_1 , corresponds to the area of the first turn of the nut.

Condition of resistance to compound stresses (tension and torsion) of the screw rod:

$$\sigma_e = \sqrt{\sigma_t^2 + 4\tau_t^2} = \sqrt{\left(\frac{4 F_{as}}{\pi d_1^2}\right)^2 + 4\left(\frac{16 M_i}{\pi d_1^3}\right)^2} \leq \sigma_{at}, \quad (\text{AFS-T.8.3.1.1.1})$$

where, $\sigma_{at} = \sigma_{02} / c$ is the allowable tensile stress with σ_{02} being the yield stress of the material and c – the safety factor (commonly, $c = 2.5$); for standard screw rods σ_{02} is determined from the group of mechanical characteristics (see subchapter AFS-T.5).

Note.

For design, the contribution of torsional stress to the equivalent stress is approximated by the factor $\beta = 1 \dots 1.3$ [Jula, 1986] and the AFS-T.8.3.1.1.1 relation becomes,

$$\sigma_e = \beta \frac{4 F_{as}}{\pi d_1^2} \leq \sigma_{at}, \quad (\text{AFS-T.8.3.1.1.2})$$

simplified verification relationship from which the diameter d_1 can be determined, for sizing

Table AFS-T.8.3.1.2 Calculation model for strength of bolted assembly under centric axial load

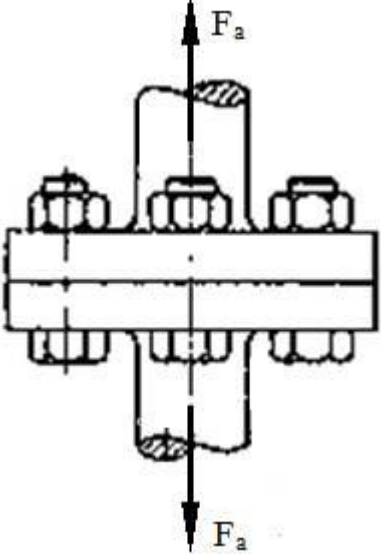
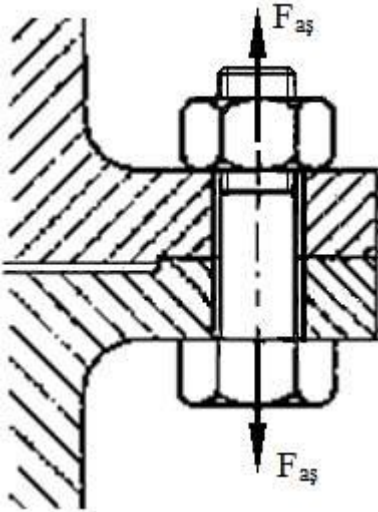
		<i>Calculation assumptions:</i> - the external axial force (static or variable) acts in the center of the screw axes which are considered to have parallel axes; - the screws are installed with shock and are tightened identically; - the external force is divided equally between the assembly bolts (which can also be pre-tensioned) and the axial force in a bolt, $F_{as} = F_a / z$; - the calculation of the resistance of the bolted assembly subjected to axial centric loading is reduced to the calculation of the bolted assembly. <i>The calculation relations of the assembly with z bolts loaded axially centrically are those in Tab. AFS-T.8.3.1.1 in which it is considered, $F_{as} = F_a / z$, keeping the assumptions adopted as valid</i>
A	b	

Fig. AFS-T.8.3.2.1 Calculation scheme for the strength of a bolted assembly subjected to axial centric loading: **a** – external loading scheme; **b** – loading scheme of a bolt

Table AFS-T.8.3.1.3 *Calculation model for the strength of the bolted assembly subjected to axial eccentric loading*

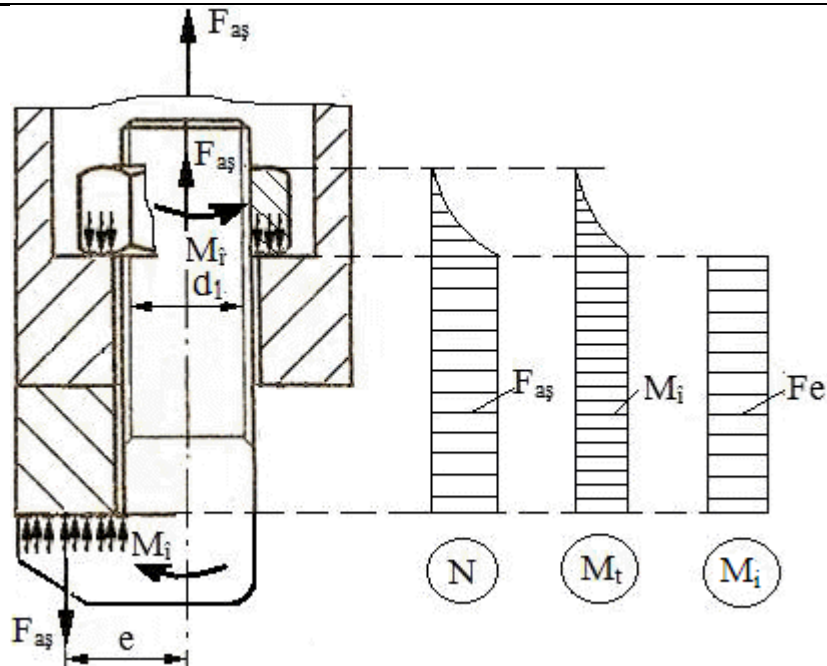


Fig. AFS- T.8.3.1.2.2 Calculation scheme for the resistance of the eccentric axially loaded screw rod

Calculation assumptions:

- the screw rod, in the worst case (tightening under load) is stressed in tension by the force F_{as} , in bending by the moment $M_i = F_{as} e$ and in torsion

$$M_i = F_{as} \frac{d_2}{2} \operatorname{tg}(\varphi' + \beta_m)$$

by the screwing moment, (AFS-T.8.1);

- the critical section, circular with a diameter equal to the internal diameter of the thread d_1 , corresponds to the area of the first turn of the nut.

The condition of resistance to compound stresses (tension, bending and torsion) of the screw rod:

$$\sigma_e = \sqrt{(\sigma_t + \sigma_i)^2 + 4\tau_t^2} = \sqrt{\left(\frac{4F_{as}}{\pi d_1^2} + \frac{32F_{as}e}{\pi d_1^3}\right)^2 + 4\left(\frac{16M_i}{\pi d_1^3}\right)^2} \leq \sigma_{at}$$

, (AFS-T.8.3.1.3.1)

where: $\sigma_{at} = \sigma_{02}/c$ represents the allowable tensile stress with σ_{02} being the yield stress of the material and c – the safety coefficient

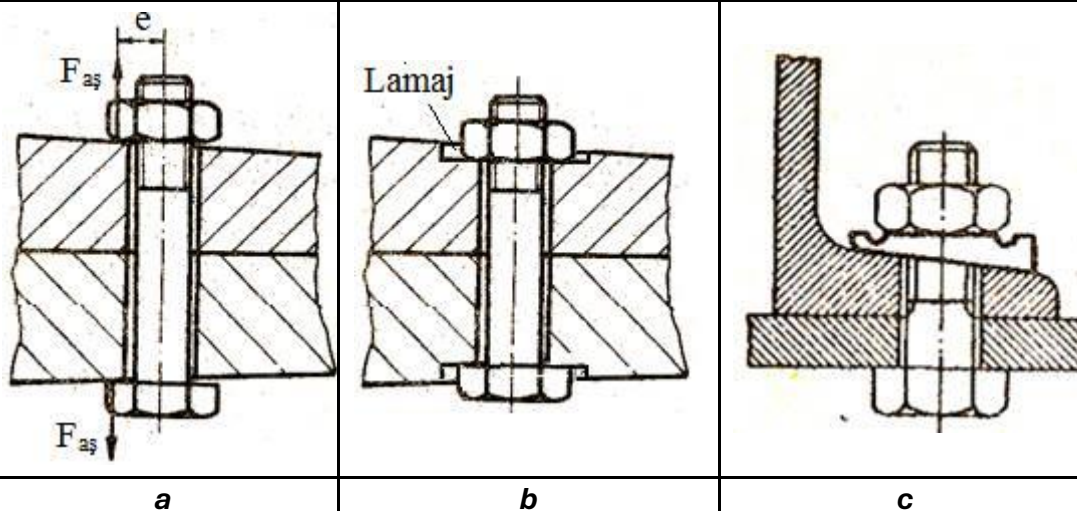


Fig. AFS-T.8.3.1.2.3 Avoiding bending stress on the screw shaft: **a** – the effect of non-perpendicularity of the bearing surfaces; **b** – machining of the bearing surface; **c** – use of special washers for U-profiles

Note.

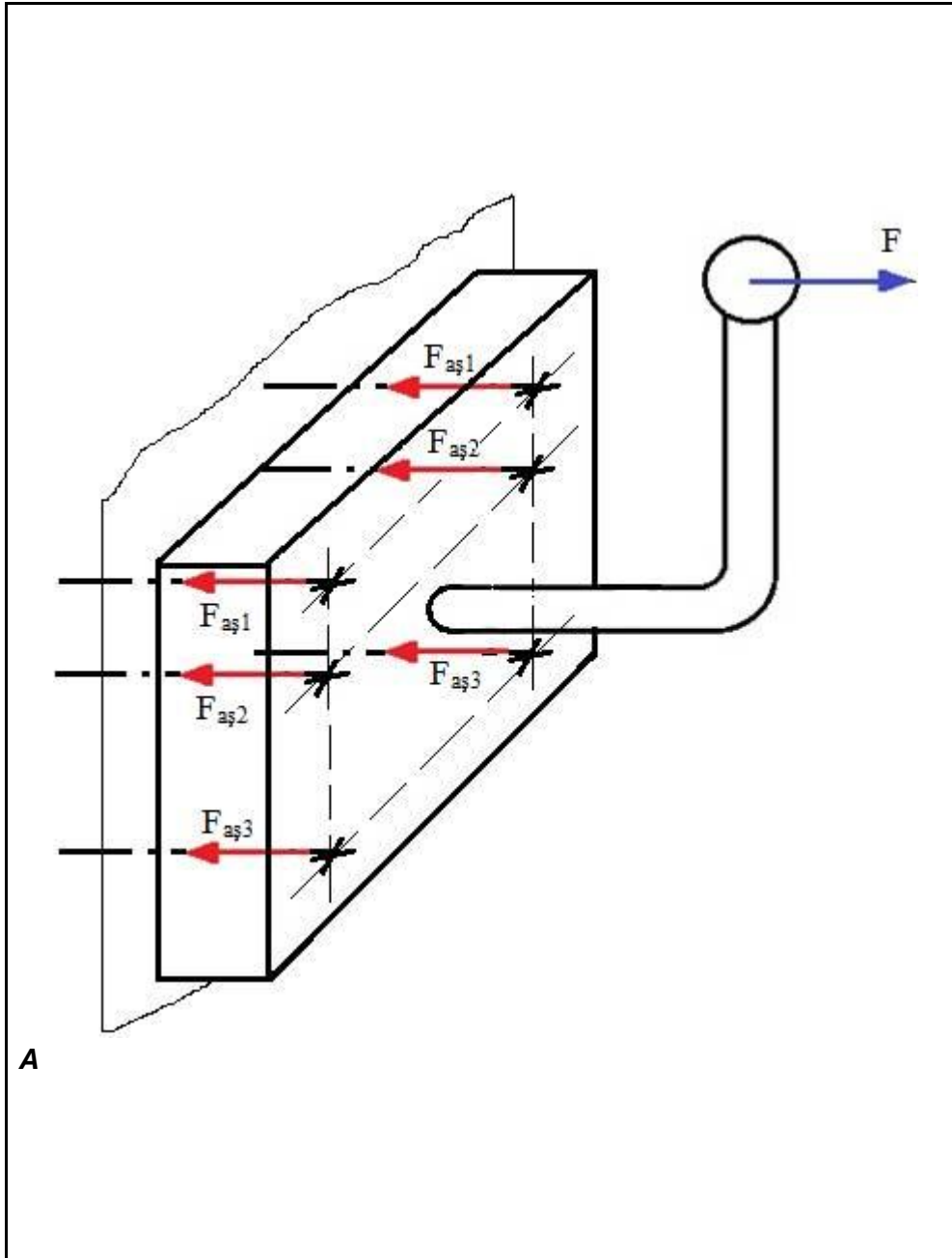
a. If the torsional stress is neglected, the equivalent stress relation (AFS-T.8.3.1.3.1) becomes,

$$\sigma_{tot} = \sigma_t + \sigma_i = \frac{4F_{as}}{\pi d_1^2} + \frac{32F_{as}e}{\pi d_1^3} = \sigma_t \left(1 + 8 \frac{e}{d_1}\right) \leq \sigma_{at}$$

(AFS-T.8.3.1.3.2)

from which, it can be deduced that for $e = d_1$, $\sigma_{tot} = 9 \sigma_t$ and, therefore, the bending stress has a very high weight and it is necessary to avoid, as much as possible, these types of assemblies;
b. if the bearing surfaces of the nut and/or bolt are not perpendicular to the hole axis (fig. AFS-T.8.3.1.3.3) the effect of rod bending is significant and to reduce it, measures are taken to process the bearing surfaces (AFS-T.8.3.1.2.3,b) or in the case of rolled U or I profiles, the use of special washers (AFS-T.8.3.1.2.3,c)

Table AFS-T.8.3.1.4 Calculation model for the strength of the bolted assembly under eccentric axial loading

 A	Calculation assumptions: - the screws being <u>identical in geometry and material</u> have the same rigidity, k_{and}; - the action of the external force (fig. AFS-T.8.3.1.2.4,a) is reduced to a load with force F in the center of the screw axes which produces in each of the 6 screws an axial force $F_{\text{s}} = F/6$ (fig. AFS-T.8.3.1.2.4,b) and, in addition, to a load with moment M_e which produces additional axial forces $F_{\text{s}} (i = 1,2,3)$; - considering the elastic deformations of the bolts, the flange (considered with infinite rigidity) <u>rotates with the angle, θ, around the support line passing through point O</u> (fig. AFS-T.8.3.1.2.4,c); in this case the elongations of the bolts in each row, δ_1, δ_2 and δ_3 are, on the one hand, proportional to the distances, L_1, L_2 and, respectively, L_3, according to the relations: $\delta_1/L_1 = \delta_2/L_2 = \delta_3/L_3 = \text{tg}\theta$, and, on the other hand, proportional to the forces in the bolts $F_{\text{s}1}, F_{\text{s}2}$ and, respectively, $F_{\text{s}3}$, according to the relations: $F_{\text{s}1}/\delta_1 = F_{\text{s}2}/\delta_2 = F_{\text{s}3}/\delta_3 = k_{\text{s}}$, from which it results, $\frac{F_{\text{s}1}}{L_1} = \frac{F_{\text{s}2}}{L_2} = \frac{F_{\text{s}3}}{L_3} = k_{\text{s}} \text{tg}\theta ; \quad (\text{AFS-T.8.3.1.2.3})$ - bending stresses in the bolt rods are neglected. Calculation relationships for additional forces from bolts: From the moment equilibrium equation at the support point O, considering two bolts in a row, $F L = 2' (F_{\text{s}1} L_1 + F_{\text{s}2} L_2 + F_{\text{s}3} L_3), \quad (\text{AFS-T.8.3.1.3.4})$ and taking into account the relationship (AFS-T.8.3.1.2.3) we obtain, $F_{\text{s}1} = \frac{F L}{2} \frac{L_1}{L_1^2 + L_2^2 + L_3^2} = \frac{M_e}{2} \frac{L_1}{L_1^2 + L_2^2 + L_3^2}, \quad (\text{AFS-T.8.3.1.3.5})$ $F_{\text{s}2} = \frac{F L}{2} \frac{L_2}{L_1^2 + L_2^2 + L_3^2} = \frac{M_e}{2} \frac{L_2}{L_1^2 + L_2^2 + L_3^2}, \quad (\text{AFS-T.8.3.1.3.5})$ $F_{\text{s}3} = \frac{F L}{2} \frac{L_3}{L_1^2 + L_2^2 + L_3^2} = \frac{M_e}{2} \frac{L_3}{L_1^2 + L_2^2 + L_3^2}, \quad (\text{AFS-T.8.3.1.3.7})$ where $F_{\text{s}1}, F_{\text{s}2}$ and $F_{\text{s}3}$ are the <u>additional forces generated by the moment M_e in the bolts on rows 1, 2 and 3, respectively.</u>
--	---

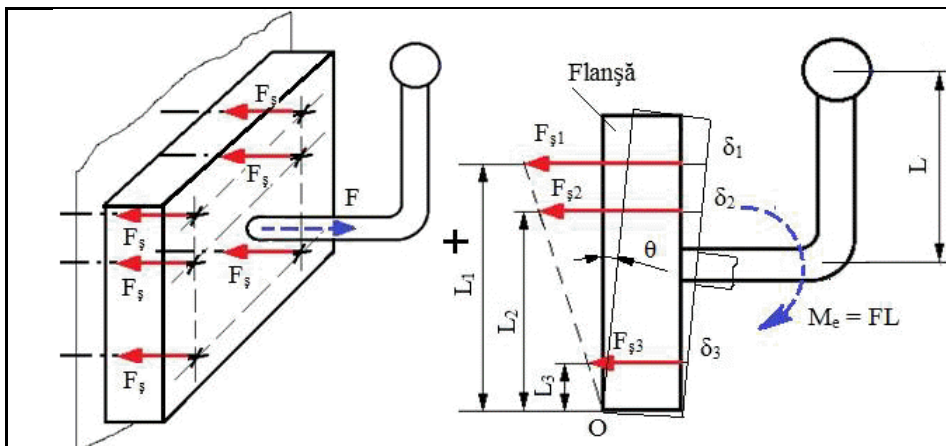
**b**

Fig. AFS- T.8.3.1.2.4 Bolt loading diagrams of an eccentrically loaded assembly: **a** – general; **b** – with the components (force and moment) of the reduction torque

The resulting axial forces from bolts $i = 1, 2, 3$ are

$$F_{asi} = F_{si} + F_{si} = F_{si} + F/6. \quad (\text{AFS-T.8.3.1.3.8})$$

In the general case of an assembly with n rows of bolts and m bolts per row ($p = mn$), the resultant axial forces from the bolts in the $i = 1, 2, 3 \dots n$ rows are determined from the relationship,

$$F_{asi} = \frac{FL}{n} \frac{L_i}{\sum_i^m L_i^2} + \frac{F}{mn} = \frac{M_e}{n} \frac{L_i}{\sum_i^m L_i^2} + \frac{F}{mn} \quad (\text{AFS-T.8.3.1.3.8})$$

The strength calculation of the axially eccentrically loaded bolt assembly is reduced to the calculation of the most stressed bolt loaded with the axial force, $F_{as} = \max (F_{as1}, F_{as2} \dots F_{asm})$, according to the model in Table AFS-T.8.3.1.1

AFS-T.8.3.2 Calculation models for the strength of transversely loaded bolted joints

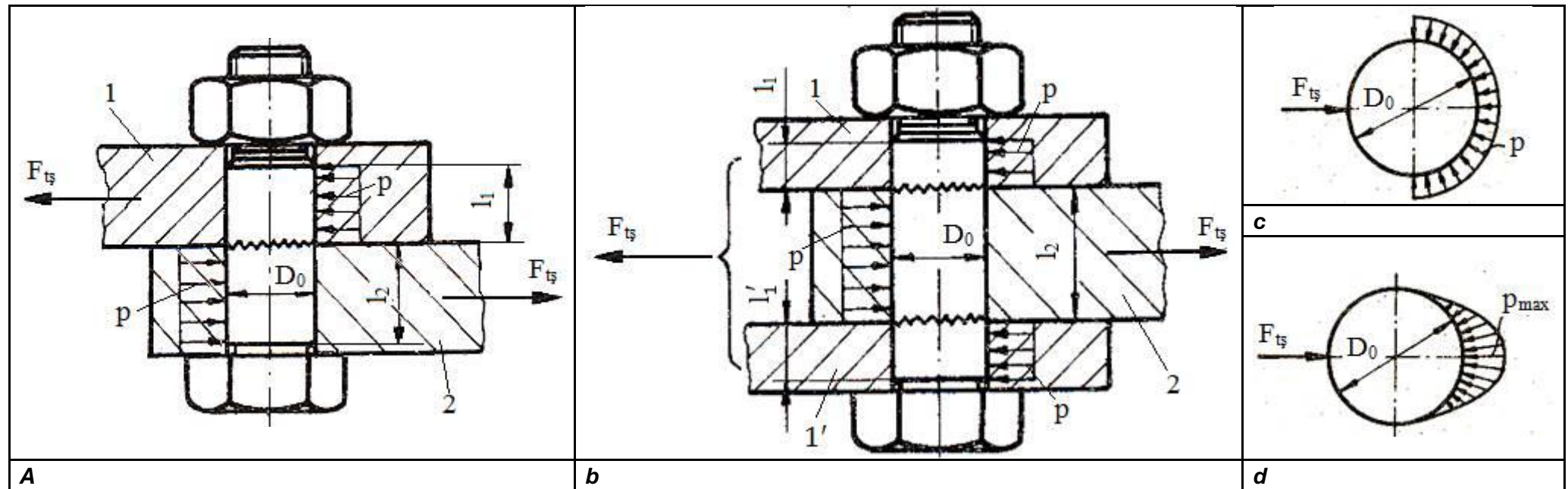
Tab. AFS-T.8.3.2.1 Calculation model for the strength of a bolted assembly installed without clearance and loaded transversely

Fig. AFS-T.8.3.2.1 Calculation schemes for bolted assembly without clearance: **a** – non-centered transverse load; **b** – centered transverse load; **c** – uniform pressure distribution; **d** – cosine pressure distribution

Calculation assumptions:

- the direction of the external loading force, F_{ts} , is perpendicular to the screw axis;
- the screw is mounted without play (stepped) forming an intermediate bore (cosinusoidal pressure distribution, fig. AFS-T.8.3.2.1,c) or with reduced tightening (uniform pressure distribution, fig. AFS-T.8.3.2.1,d);
- the external load F_{ts} is transmitted from plate 1 to plate 2 (fig. AFS-T.8.3.2.1,a) or from plates 1 and 1' to plate 2 (fig. AFS-T.8.3.2.1,b) through direct (semi-cylindrical) contacts, according to the flows: 1 -> screw rod -> 2, respectively, 1 and 1' -> screw rod -> 2.

Assembly strength conditions:

- condition of crushing resistance (contact) screw-assembled parts:

$$p = \frac{F_{ts}}{D_0 l_{\min}} \leq \sigma_{as} \text{ or } p_{\max} = \frac{F_{ts}}{0,8 D_0 l_{\min}} \leq \sigma_{as}, \quad (\text{AFS-T.8.3.2.1.1})$$

for uniform pressure distribution (fig. AFS-T.8.3.2.1,c), or respectively, cosine (fig. AFS-T.8.3.2.1,d), where: $l_{\min} = \min(l_1, l_2)$, for the case in fig. AFS-T.8.3.2.1,a or,

- the shear resistance condition of the screw rod:

$$\tau_f = \frac{F_{ts}}{i \frac{\pi}{4} D_0^2} = \frac{4 F_{ts}}{i \pi D_0^2} \leq \tau_{af}, \quad (\text{AFS-T.8.3.2.1.2})$$

where, τ_{af} is the allowable shear stress of the bolt material $\tau_{af} = (0.6 \dots 0.8) \sigma_{at}$, the maximum values are for variable loads; $i = 1$ or 2 , the number of shear zones for the case in Fig. AFS-T.8.3.2.1,a or Fig. AFS-T.8.3.2.1,b respectively.

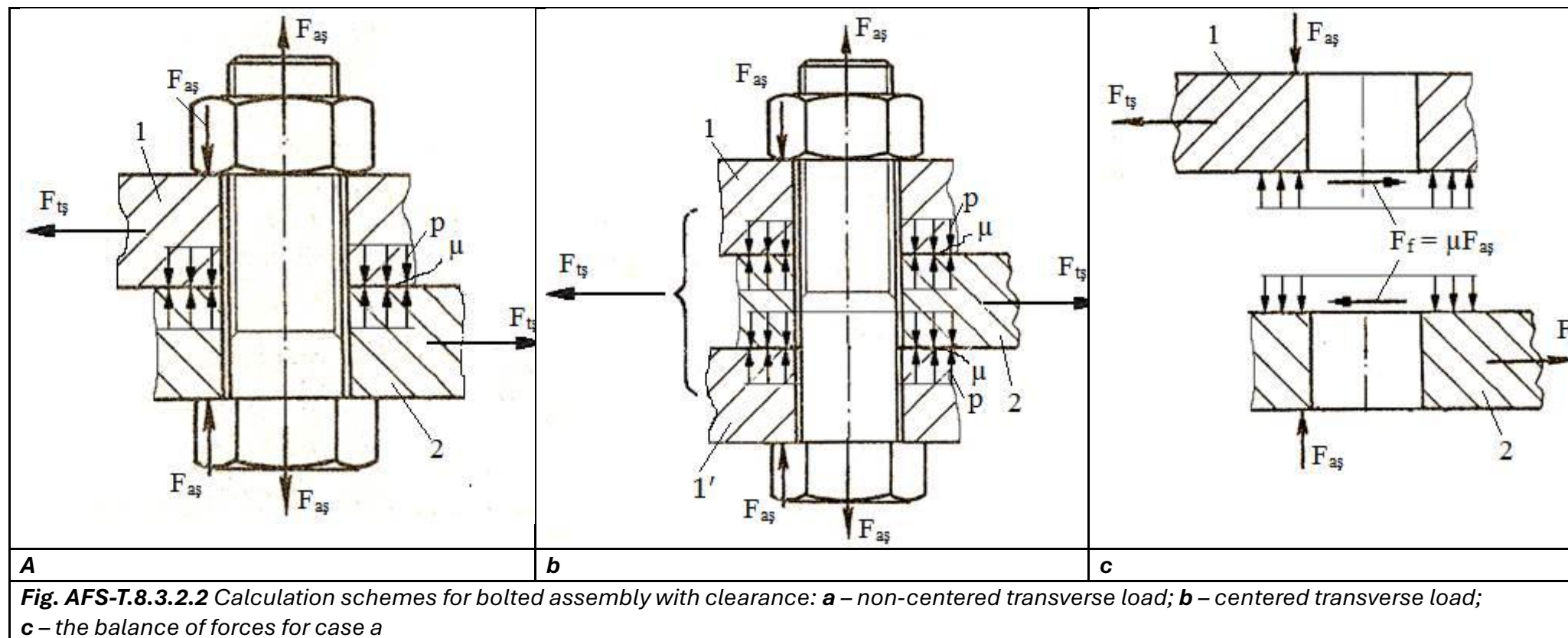
Note.

Compared to assemblies with screws mounted with clearance (subchapter AFS-T.8.3.2.2), in the case of these assemblies, smaller screws are obtained, but the technology is more demanding (the screw shaft is ground and the holes are reamed) and the

$l_{\min} = \min(l_1 + l'_1, l_2)$, for the case in fig. AFS-T.8.3.2.1,b; D_0 – nominal diameter of the bolt shank; $\sigma_{as} = \min(\sigma_{as1}, \sigma_{as2})$ – minimum crushing stresses of the materials of sheets 1 (respectively, 1 and 1'), 2 and the bolt ($\sigma_{as} = (0.2...0.4)\sigma_{02}$)

required precision is much higher and, therefore, much higher costs.

Tab AFS-T.8.3.2.2 Strength calculation model for a bolted assembly with a clearance loaded transversely



Calculation assumptions:

- the direction of the external loading force, $F_{t\varnothing}$, is perpendicular to the screw axis;
- the screw is mounted with a large clearance (there is no contact between the sheet metal and the screw rod);
- the transverse force, $F_{t\varnothing}$, is transmitted from plate 1 to plate 2 (fig. AFS-T.8.3.2.2,a) or from plates 1 and 1' to plate 2 (fig. AFS-T.8.3.2.2,b) by friction, a consequence of the tightening of the plates with the axial force from the screw $F_{a\varnothing}$, without relative movement and contact with the screw rod occurring.

The condition for force transmission through friction:

$$F_f = \mu F_{a\varnothing} i \geq \beta_a F_{t\varnothing} \quad (\text{AFS-T.8.3.2.2.1})$$

where, i is the number of friction surfaces, $i = 1$, for the case in fig. AFS-T.8.3.2.2,a and $i = 2$, for the case in fig. AFS-T.8.3.2.2,b; μ - friction coefficient; β_a – safety factor for sliding of the sheets.

Bolt strength calculation

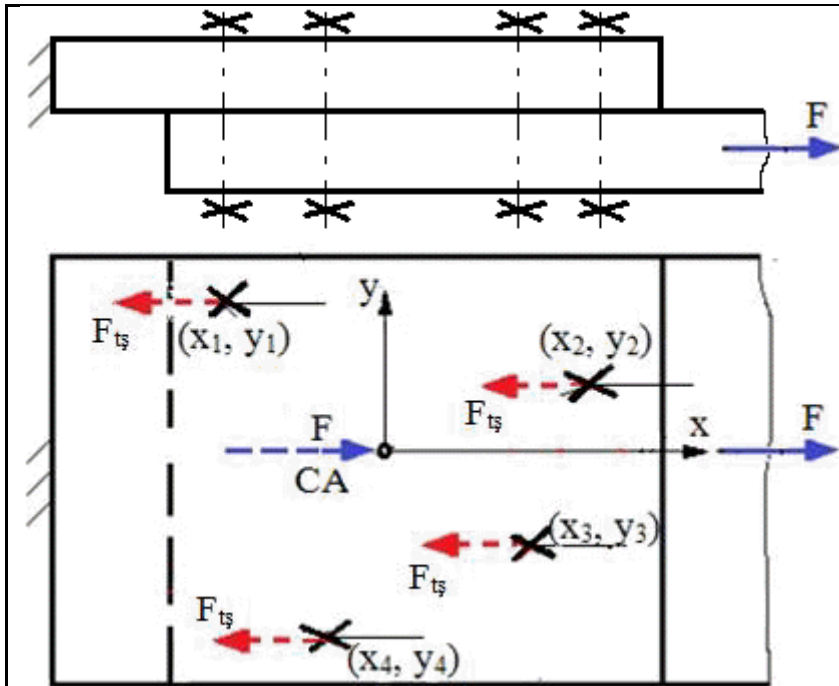
If the condition of force transmission by friction is met, the bolts are stressed in tension and during assembly and in torsion by the screwing moment; the axial force in a bolt is determined from the relation (AFS-T.8.3.2.2.1),

$$F_{a\varnothing} = \frac{\beta_a F_{t\varnothing}}{\mu i}, \quad (\text{AFS-T.8.3.2.2.2})$$

The calculation of the bolted assembly with clearance loaded transversely is reduced to the calculation of the bolted assembly with axially loaded centrically according to the model in Table AFS-T.8.3.1.1 considering the load from the relation (AFS-T.8.3.2.2.2).

Note.

Since the clamping force is greater than the force to be transmitted, large diameter screws result, which is a disadvantage; the advantage of these assemblies is that they require low machining precision of the screws and the bores in which they are mounted.

Tab AFS-T.8.3.2.3 Strength calculation model for bolted assemblies (mounted with or without clearance) subjected to centric transverse loading

Calculation assumptions:

- the screws have axes parallel to an axis containing their center, CA, where and with x_i and y_i are the coordinates of the points associated with the assembly screws;

$$\sum_{i=1}^4 x_i = 0 \quad \sum_{i=1}^4 y_i = 0$$

- the bolts, as a consequence of the loading with the external force F with the direction passing through point CA, are loaded with equal transverse forces.

$$F_{ts} = \frac{F}{4} \quad (\text{AFS-T.8.3.2.3.1})$$

or, for the general case with z screws,

$$F_{ts} = \frac{F}{z} \quad (\text{AFS-T.8.3.2.3.2})$$

- the calculation of bolted assemblies mounted with clearance and loaded transversely centrically, finally, is reduced to the calculation of the bolted assembly (mounted without clearance or with clearance) according to the models in tab. AFS-T.8.3.2.1 or, respectively, tab. AFS-T.8.3.2.2 loaded with the transverse force, F_{ts} .

Fig. AFS-T.8.3.2.2.1 Calculation scheme for bolted assemblies subjected to centric transverse loads

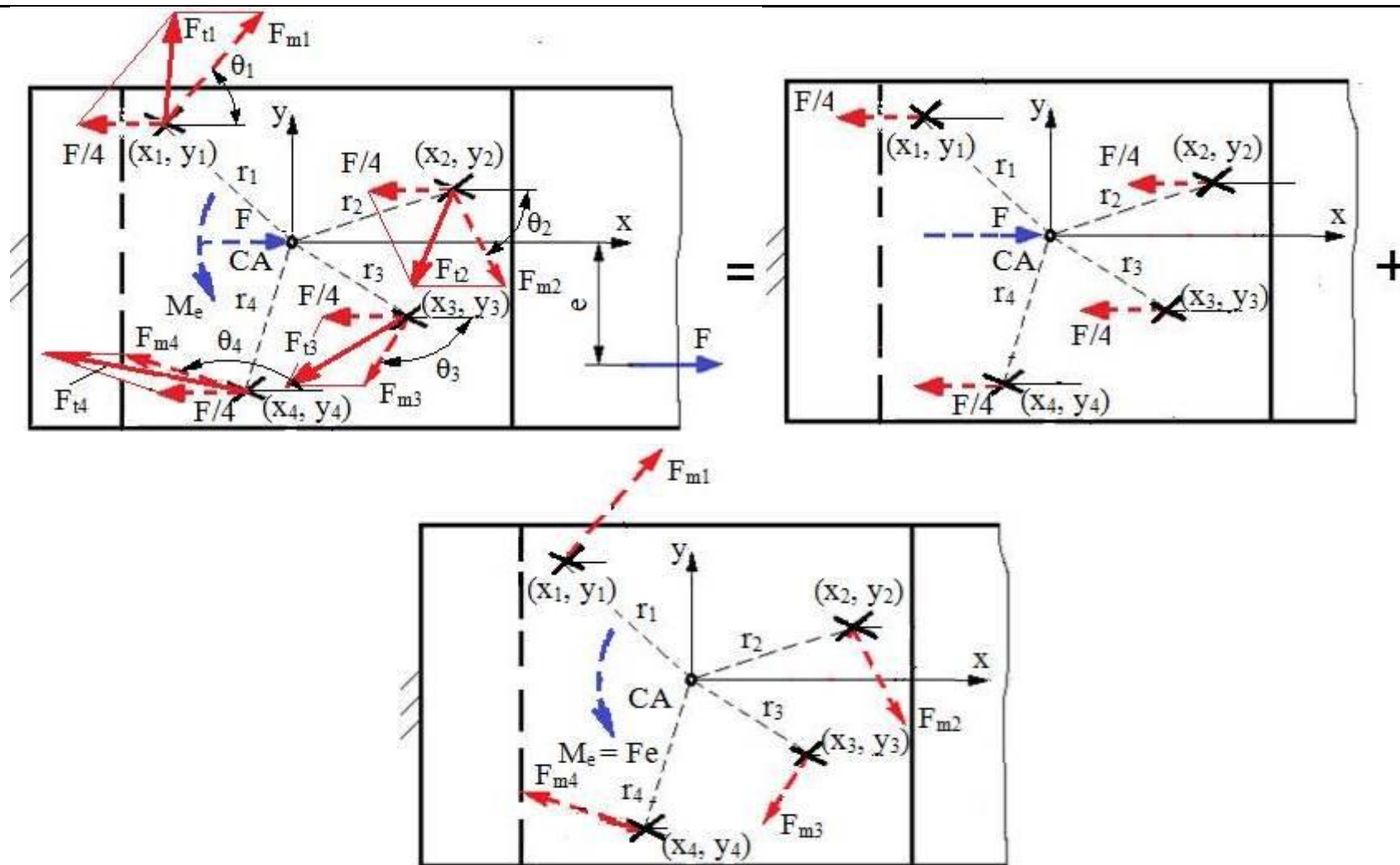
Table AFS-T.8.3.2.4 Strength calculation model for bolted assemblies subjected to transverse eccentric loading**A****b****c**

Fig. AFS-T.8.3.2.4.1 Calculation scheme of the assembly with eccentrically loaded transverse bolts: **a** – general scheme; **b** – reduced force loading scheme;
b – reduced moment loading scheme

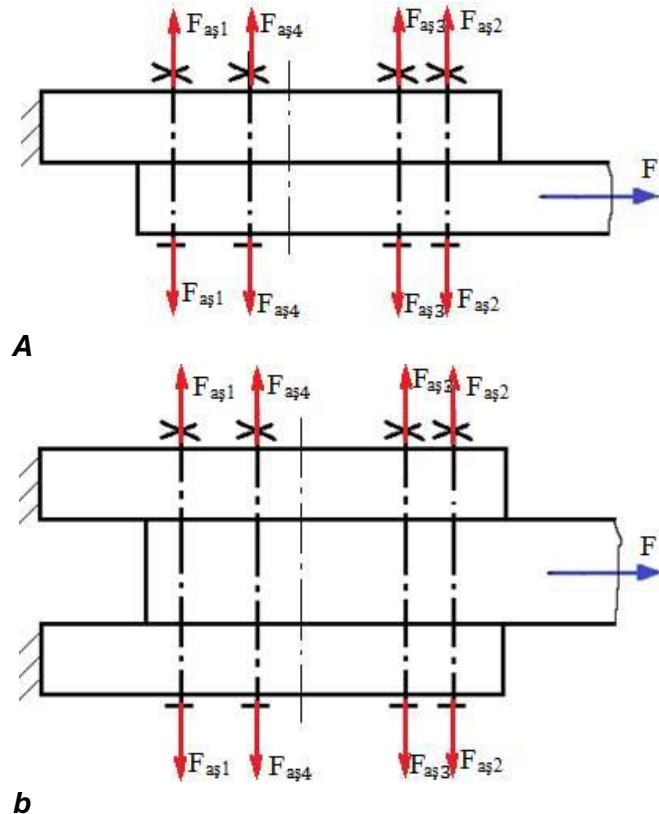


Fig. AFS-T.8.3.2.4.2 Axial loading diagrams of bolts installed with clearance in the assembly subjected to transverse eccentric loading: **a** – non-centered structure; **b** – centered structure

Calculation assumptions:

- the screws have axes parallel to an axis containing their center, CA, where and with x_i and y_i are the coordinates of the points associated with the axes of the assembly screws; $\sum_1^4 x_i = 0$
 $\sum_1^4 y_i = 0$

- the external transverse force, F, acting eccentrically on the assembly is reduced at the level of the axis passing through point CA to a force equal to F and to the moment,

$$M_e = F_e; \quad (\text{AFS-T.8.3.2.4.1})$$

- the force F is transmitted in equal proportions by F/4 by each bolt (fig. AFS-T.8.3.2.3.1,b) and the moment M_e is transmitted by the forces F_{m1} , F_{m2} , F_{m3} and F_{m4} whose moments respect the relationship,

$$F_{m1}r_1 + F_{m2}r_2 + F_{m3}r_3 + F_{m4}r_4 = M_e, \quad (\text{AFS-T.8.3.2.4.2})$$

where, $r_i = \sqrt{x_i^2 + y_i^2}$ ($i = 1, 2, 3, 4$) are the distances of the application points to the center of CA (fig. AFS-T.8.3.2.3.1,c);

- the forces F_{m1} , F_{m2} , F_{m3} and F_{m4} are proportional to the distances (arms), r_1 , r_2 , r_3 and, respectively, r_4 , according to the relationship,

$$\frac{F_{m1}}{r_1} = \frac{F_{m2}}{r_2} = \frac{F_{m3}}{r_3} = \frac{F_{m4}}{r_4} = K, \quad (\text{AFS-T.8.3.2.4.3})$$

where K is some constant;

- the bolts, as a consequence of the loading from point CA, with force F and moment M_e , are loaded with transverse forces F_{t1} , F_{t2} , F_{t3} and F_{t4} which are obtained by vectorial summation of the forces associated with the dimension F/4 with the forces F_{m1} , F_{m2} , F_{m3} and, respectively, F_{m4} (fig. AFS-T.8.3.2.4.1,a).

Calculation relationships for transverse bolt loading forces:

From the relations (AFS-T.8.3.2.4.1), (AFS-T.8.3.2.4.2) and (AFS-T.8.3.2.4.3) it results that the forces that generate the components of the moment M_e ,

$$F_{mi} = M_e \frac{r_i}{r_1^2 + r_2^2 + r_3^2 + r_4^2} = F_e \frac{r_i}{r_1^2 + r_2^2 + r_3^2 + r_4^2}, \quad (\text{AFS-T.8.3.2.4.4})$$

where, $i = 1, 2, 3, 4$.

By summing, according to the “parallelogram rule”, the resultant bolt loading forces are determined with the relationship (fig. AFS-T.8.3.2.4.1),

$$F_{ti} = \sqrt{\left(\frac{F}{4}\right)^2 + F_{mi}^2} + \frac{F}{4} F_{mi} \cos(\pi - \theta_i) = \sqrt{\left(\frac{F}{4}\right)^2 + F_{mi}^2} - \frac{F}{4} F_{mi} \frac{x_i}{r_i} \quad (\text{AFS-T.8.3.2.4.5})$$

where, $i = 1, 2, 3, 4$ and the forces F_{mi} are according to the relation (AFS-T.8.3.2.4.4); θ_i - the angles of the directions of the forces F_{mi} (fig. AFS-T.8.3.2.4.1,a)

Calculation relationship for axial load forces of bolts, $i = 1, 2, 3, 4$, mounted with clearance (fig. AFS-T.8.3.2.4.2):

$$F_{a\varphi i} = \frac{\beta_a F_{ti}}{\mu j z}, \quad (\text{AFS-T.8.3.2.4.6})$$

where, $j = 1$ (fig. AFS-T.8.3.2.4.2,a) or $j = 2$ (fig. AFS-T.8.3.2.4.2,b) represents the number of friction surfaces, z – the number of bolts; μ – the friction coefficient; β_a – safety factor for sliding of the sheets, F_{ti} – the transverse loading force of bolt i .

The strength calculation of assemblies with bolts mounted with clearance loaded transversely eccentrically, finally, is reduced to the calculation of the bolt assembly (mounted without clearance or with clearance) according to the models in tab. AFS-T.8.3.2.1 or, respectively, tab. AFS-T.8.3.2.2 loaded with the transverse force, $F_{t\varphi} = \max(F_{t1}, F_{t2}, F_{t3}, F_{t4})$, for the assembly with bolts mounted without clearance, or, respectively, the maximum axial force, $F_{a\varphi} = \max(F_{a\varphi 1}, F_{a\varphi 2}, F_{a\varphi 3}, F_{a\varphi 4})$, for the assembly with bolts mounted with clearance

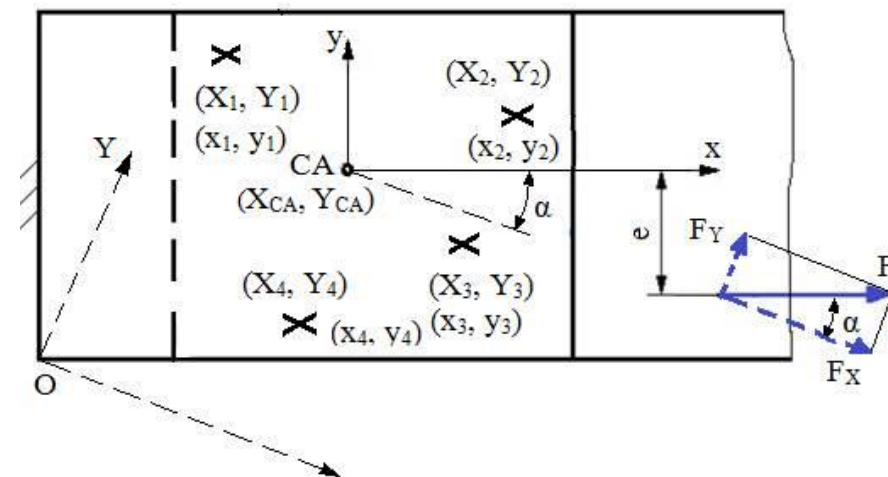


Fig. AFS-T.8.3.2.4.3 Calculation scheme for the center of parallel axes associated with bolts and generalization regarding the direction of the external force

Note.

1. The coordinates of the center of the bolt axes, CA, can be defined as the center of the parallel axes associated with the assembly bolts:

a. for the case of simple figures (triangles, square, rhombus, parallelogram) the properties of elementary geometry are used;

b. for the general case (fig. AFS-T.8.3.2.4.2) when the coordinates of the points associated with the screw axes (X_i, Y_i) are known in relation to the XOY coordinate system, the coordinates of the center of the axes are determined,

$$X_{CA} = \frac{\sum_1^4 X_i}{4}, \quad Y_{CA} = \frac{\sum_1^4 Y_i}{4}; \quad (\text{AFS-T.8.3.2.4.6})$$

and the coordinates of the centers of the axes relative to the coordinate system xCAy rotated by angle α with respect to the coordinate system XOY,

$$x_i = (X_i - X_{CA}) \cos \alpha - (Y_i - Y_{CA}) \sin \alpha, \quad (\text{AFS-T.8.3.2.4.7})$$

$$y_i = (X_i - X_{CA}) \sin \alpha - (Y_i - Y_{CA}) \cos \alpha, \quad (\text{AFS-T.8.3.2.4.8})$$

2. For the general case of loading the assembly with orthogonal forces F_x and F_y relative to the XOY coordinate system, the resultant force is determined, and the

$$F = \sqrt{F_x^2 + F_y^2} \quad \alpha = \arctg(F_y/F_x)$$

AFS-T.9 CONSTRUCTION STRUCTURES

AFS-T.9.1 CONSTRUCTION FORMS OF BOLTS, NUTS AND FLAT WASHERS

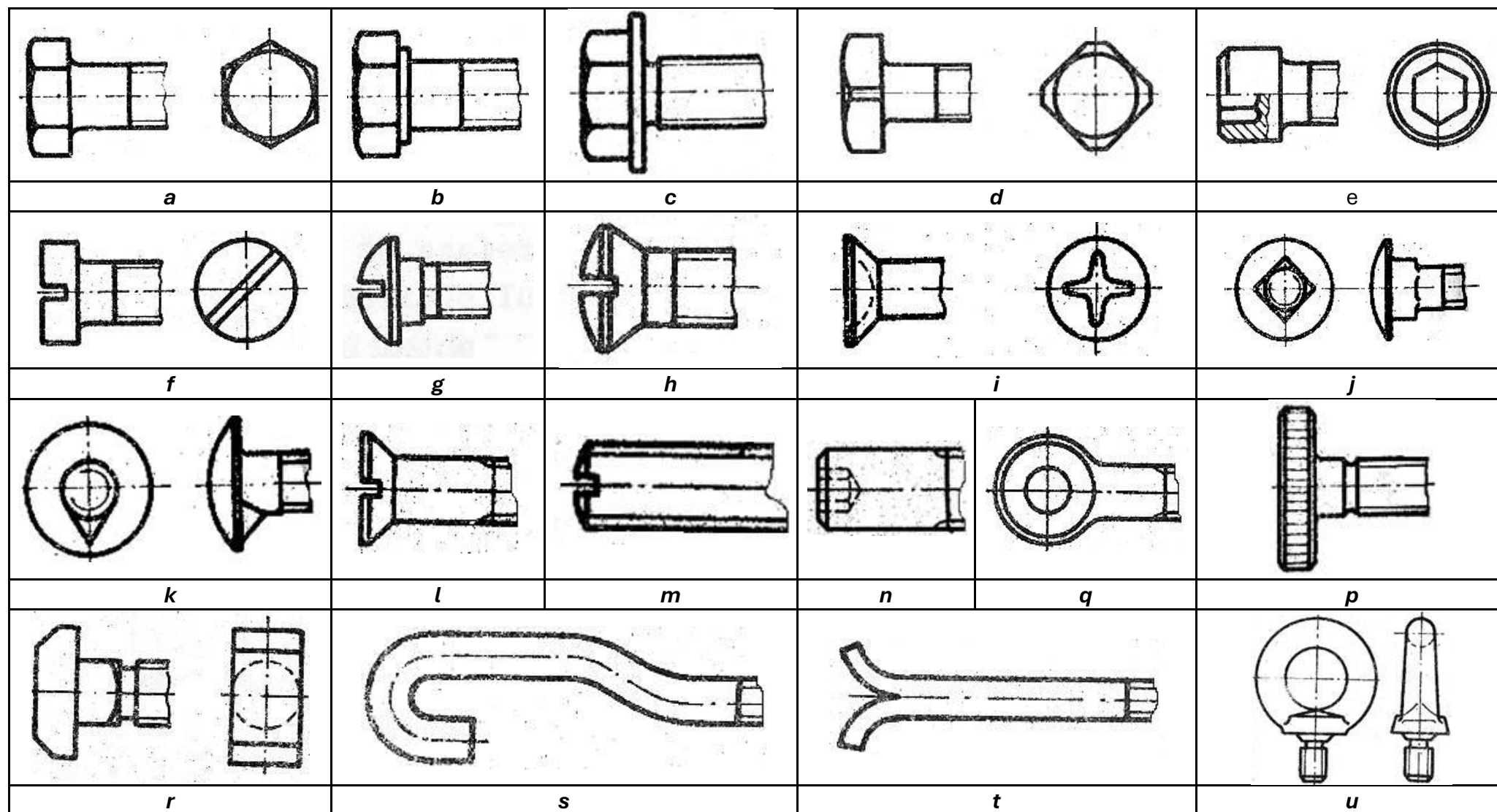


Fig. AFS-T.9.1.1 Constructional shapes of the screw head: **a** – hexagonal; **b** – hexagonal with intermediate threshold; **c** – hexagonal with collar; **d** – square; **e** – cylindrical with internal hexagonal recess; **f** – cylindrical head with notch; **g** – semi-round with notch; **h** – semi-countersunk with notch; **i** – countersunk with cross recess; **j** – convex and square anti-rotation; **k** – convex and anti-rotation nose; **l** – countersunk with notch; **m** – headless with notch; **n** – headless internal hexagonal recess; **o** – ring (eye); **p** – ribbed cylindrical; **r** – hammer (T-shaped); **s** – foundation hook; **t** – foundation split; **u** – annular

Note.

- the hexagonal head (fig. AFS-T.9.1,a,b,c) ensures the smallest maneuvering space when assembling/disassembling with the open-end wrench;

- the square head (fig. AFS-T.9.1,d) provides a larger contact surface with the mounting/dismounting key and has high durability during repeated mounting/dismounting;
- the hexagonal head with intermediate threshold (fig. AFS-T.9.1,b) ensures a reduced stress concentrator;
- the hexagonal head with collar (fig. AFS-T.9.1,c) is used for assembling parts made of soft materials (wood, aluminum, etc.) to reduce the contact pressure on the seating surface;
- the head with internal hexagonal recess (fig. AFS-T.9.1,e) is used for high clamping forces and the head with cross recess or notch (fig. AFS-T.9.1,f,g,h,i,l,m,n) for the screwdriver are used for low clamping forces;
- the square head or anti-rotation nose deforms the material (usually wood) of the parts and prevents rotation when the nut is tightened;
- the shapes in Fig. AFS-T.9.1,o,p,r are used in the construction of technological devices;
- the shapes in Fig. AFS-T.9.1,s,t are used for fixing to the foundation by burying in concrete;
- the knurled cylindrical head (fig. AFS-T.9.1,p) is used for tightening with reduced hand forces;

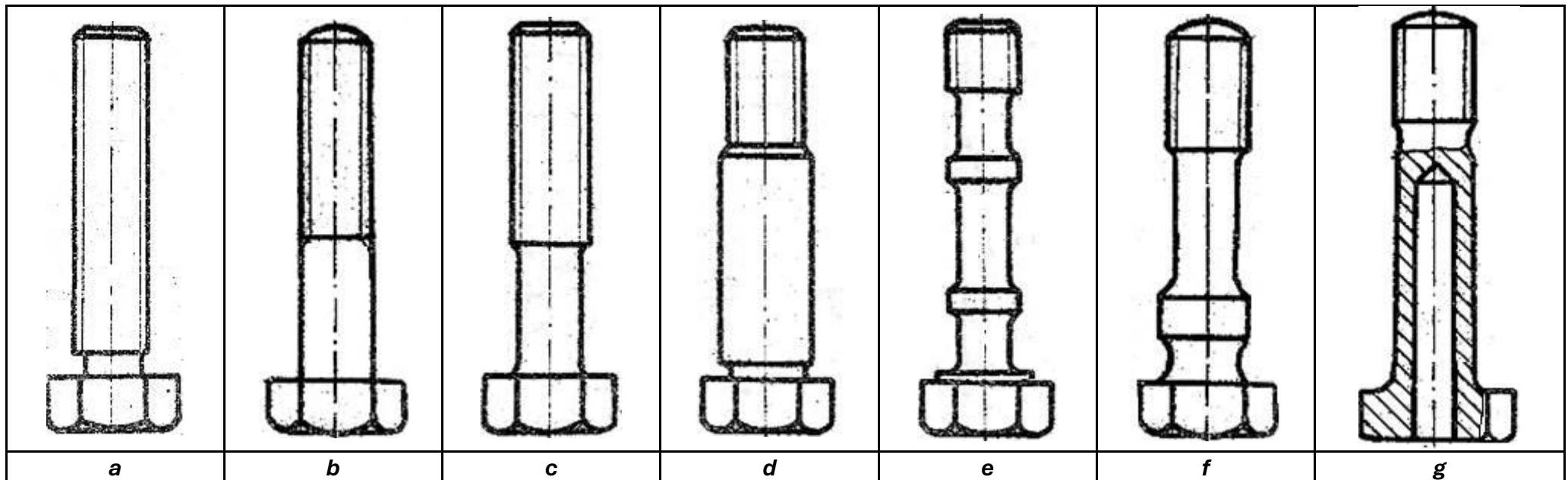


Fig. AFS-T.9.1.2 Construction forms of the screw shaft: **a** – threaded along the entire length; **b,c** – threaded only on one portion; **d** – with a stepping portion; **e, f** – thinned with guiding portions; **g** – thinned with a bore

Note.

- the diameter of the threaded portion can be smaller (fig. AFS-T.9.2,d) when the stress in this area is lower and thus the stress concentrator is reduced;
- the diameter of the unthreaded portion may be larger (fig. AFS-T.9.2,d) than the threaded one in order to ensure mating areas with direct contact with the assembled parts;

- to increase the elasticity of screw rods subjected to variable stresses, the diameter of the unthreaded rod is reduced less in the guiding areas in the bore in the assembled parts (fig. AFS-T.9.2,e,f) or an axial hole is applied (fig. AFS-T.9.2,g);
- the connecting portions from the threaded rod to the unthreaded one with increased radii (fig. AFS-T.9.2,e,f) or with connection (fig. AFS-T.9.2,g) ensure an increase in fatigue resistance;

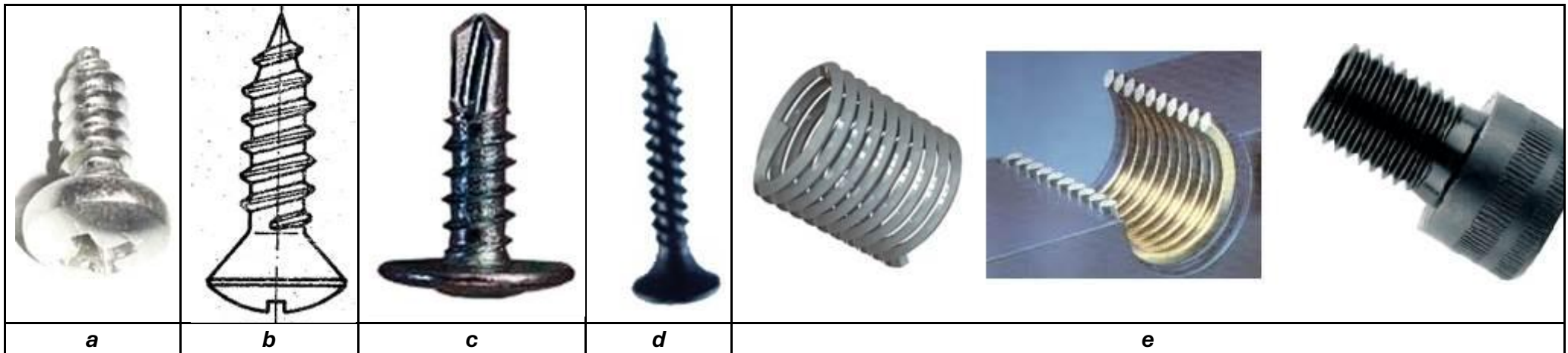


Fig. AFS-T.9.1.3 Special screws: **a** – for assembling sheets; **b** – self-tapping; **c** – self-drilling; **d** – hollow screw; **e** – with helical gaskets (Heli Coil)

Note.

- for assembling thin sheets with thicknesses comparable to the thread pitch, a through hole will be drilled in the sheet under the screw head and a hole with a diameter equal to the internal diameter of the thread in the second sheet with a thickness smaller than the screw pitch (fig. AFS-T.9.1.3,a);
- to reduce assembly costs in the case of elements made of soft materials (including steel), the screw thread is configured and hardened by heat treatment so that it can, when screwed in, perform the thread, and therefore also the tightening (fig. AFS-T.9.1.3,b) or, in addition, the screw may have a drill tip for machining the bore (fig. AFS-T.9.1.3,c);
- to increase the efficiency of threaded assemblies made of solid parts made of soft materials, a helical gasket with a rhombic section is inserted between the screw and the nut (fig. AFS-T.9.1.3,d) which has the role of uniforming the load distribution on the contacting threads.

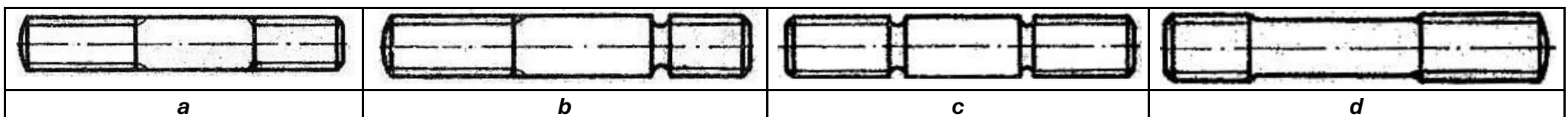


Fig. AFS-T.9.1.4 Bolt constructions: **a** – the unthreaded rod has the same diameter as the threaded rod and no connections; **b** – the unthreaded rod has the same diameter as the threaded rod and a through connection; **c** – the unthreaded rod has the same diameter as the threaded rod and two connections; **d** – the unthreaded rod has the smaller diameter

Note.

- *pins* are cylindrical rods threaded at both ends that are used when the material of a part does not ensure sufficient durability of the thread, during frequent assembly and disassembly, or when nuts cannot be used for constructive reasons;
- the length of the threaded portion in the base piece is determined depending on the characteristics of its materials (aluminum, cast iron, steel, etc.)

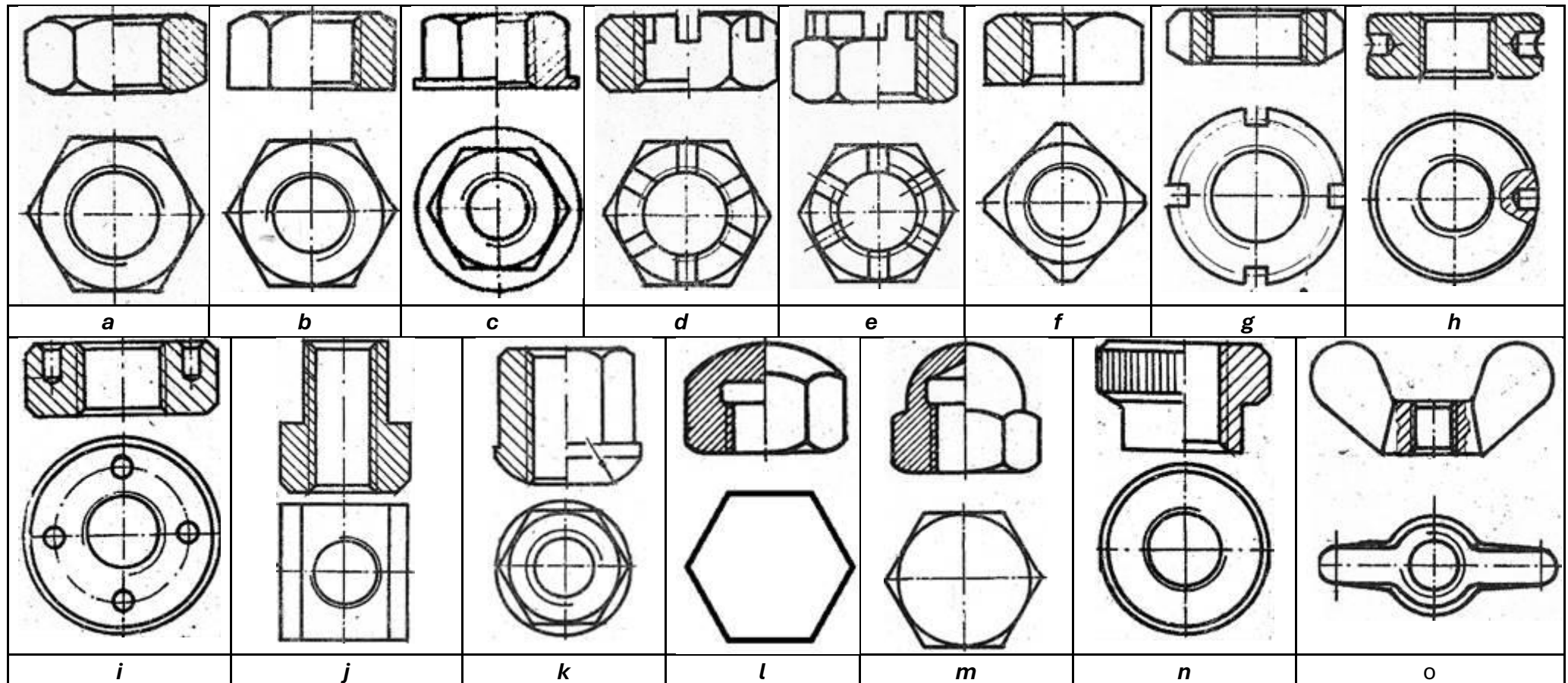


Fig. AFS-T.9.1.5 Constructional shapes of nuts: **a** – regular hexagonal (shape A); **b** – regular hexagonal (shape B); **c** – hexagonal with collar; **d,e** – crenellated; **f** – square; **g** – round with splines; **h** – round with radial holes; **i** – round with radial holes **j** – for T-splines; **k** – hexagonal with spherical seating surface; **l, m** – countersunk; **n** – knurled; **o** – butterfly

Note.

- the hexagonal nut with the spherical seating surface (fig. AFS-T.9.5,k) allows the nut to be centered on the bolt axis and is used when mounting vehicle wheels;
- the round nut with splines (fig. AFS-T.9.5,g) is used for axial fixation of the inner rings of the bearings

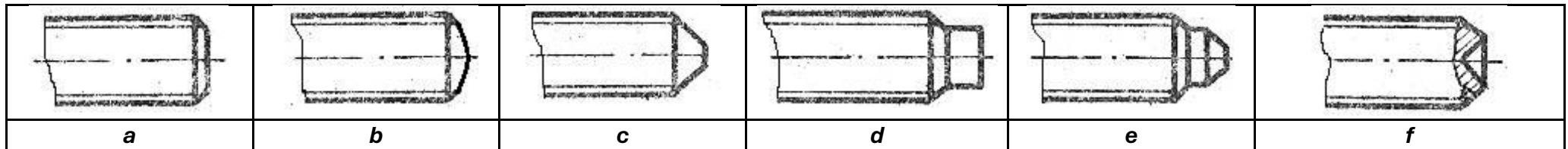


Fig. AFS-T.9.1.6 Construction forms of the threaded rod in the head area: **a** – flat; **b** – convex; **c** – with conical tip; **d** – with flat cylindrical pin; **e** – with conical cylindrical pin; **f** – with cylindrical and cone **g** – with inner cone

Note: They are used as threaded pins to prevent relative movement of parts through contacts between the head surfaces and the associated recesses (flattening, tapered hole, cylindrical hole)

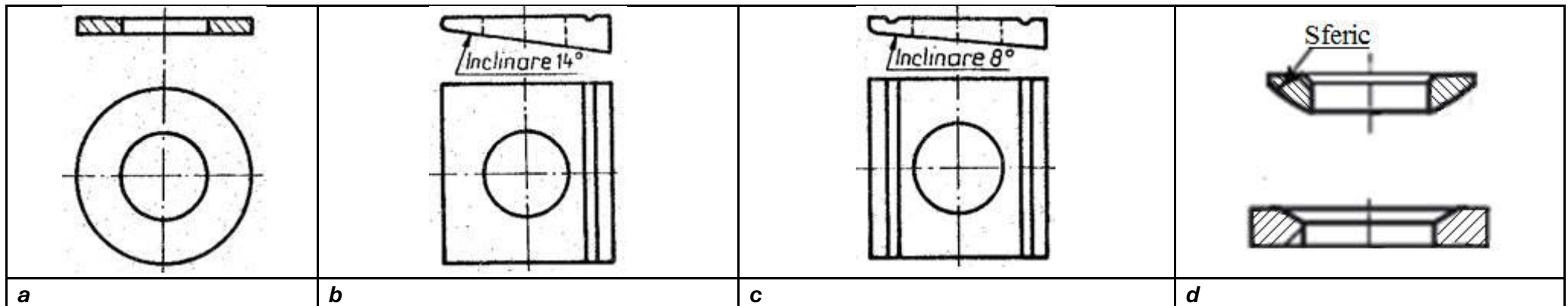


Fig. AFS-T.9.1.7 Washer construction forms: **a** – circular flat; **b** – square compensation for I-profile; **c** – square compensation for U-profile; **d** – spherical compensation

Note: Washers are used to reduce the pressure on the nut support surface (fig. AFS-T.9.7,a) or for the correct placement of the nut when the support surface is rough or inclined (fig. AFS-T.8.3.1.2.2; fig. AFS-T.9.7,b – for assembling laminated I profiles, AFS-T.9.7,c – for U profiles)

AFS-T.9.2 SOLUTIONS FOR CONSTRUCTIVE OPTIMIZATION OF THREADED AND BOLTED ASSEMBLIES

Purpose: to ensure adequate shapes and minimum dimensions to eliminate most cases of breakage, especially through fatigue, at low cost.

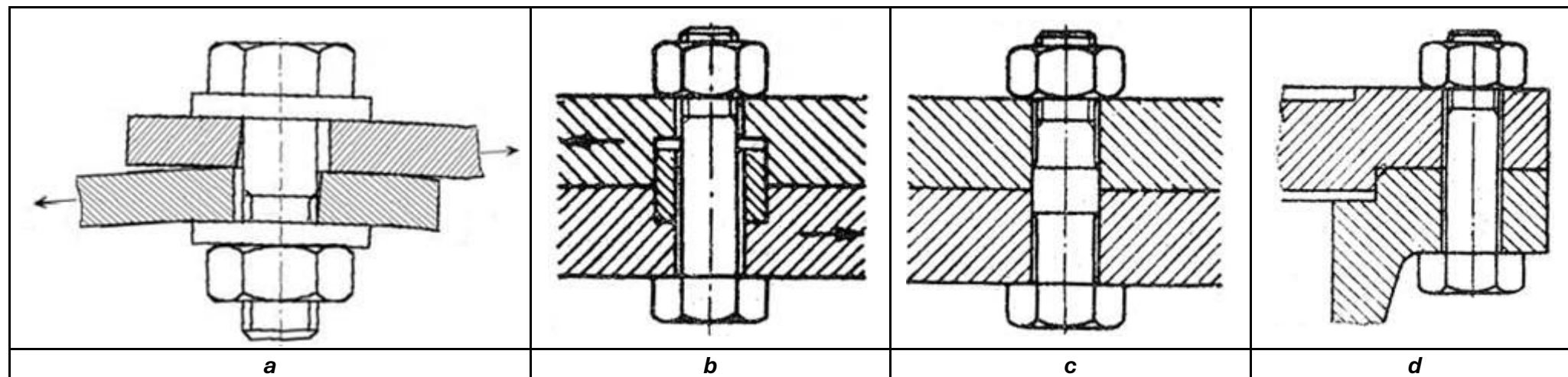


Fig. AFS-T.9.2.1 Constructive solutions for safety and centering of transversely loaded clearance assemblies: **a** – improper functional condition; **b** – centering rings; **c** – with centering section; **d** – with centering threshold

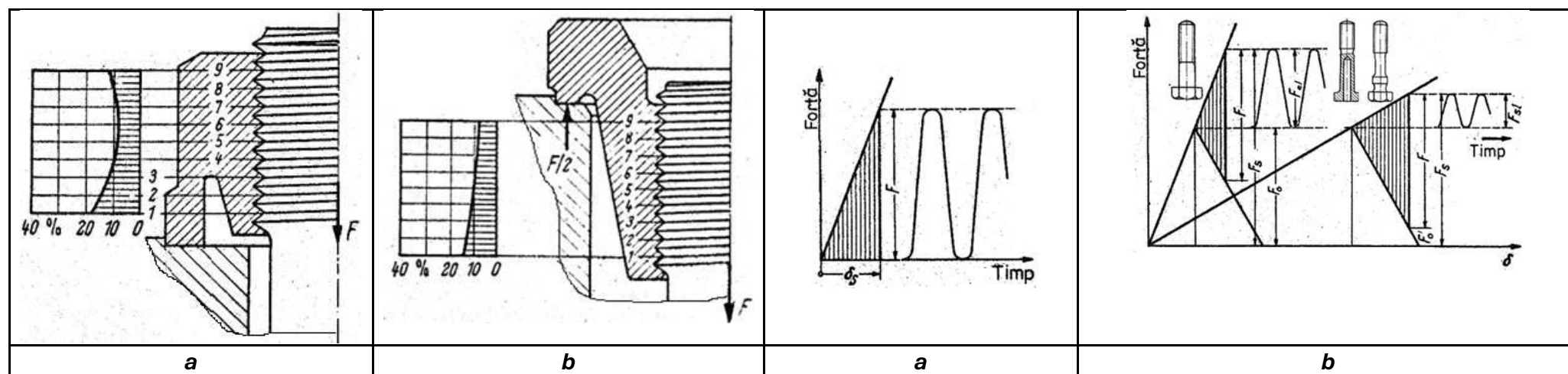


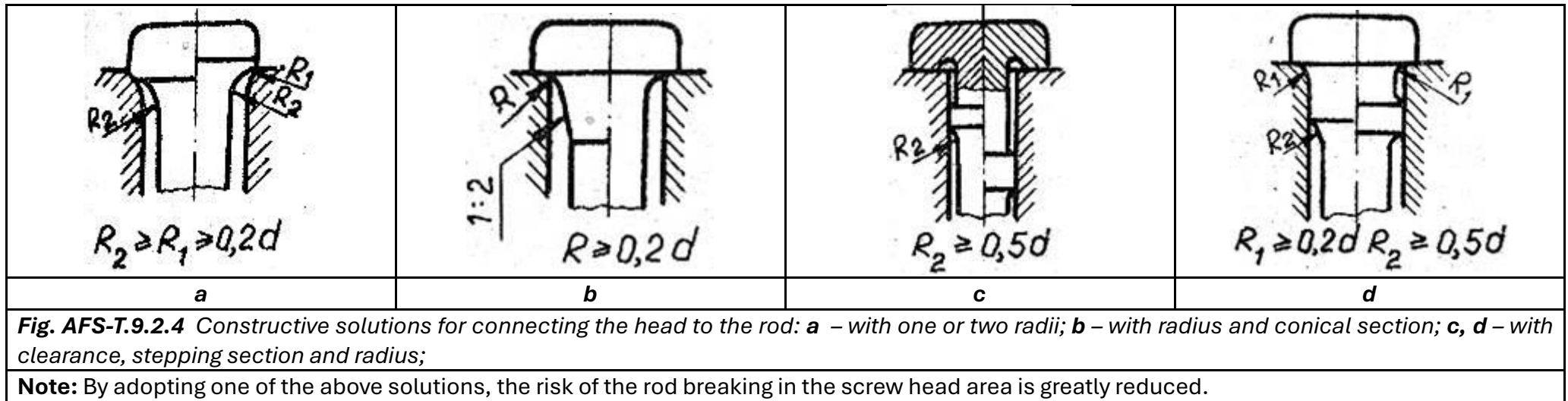
Fig. AFS-T.9.2.2 Constructive solutions for uniforming the distribution of external force on the coils in contact: **a** – compressed elastic nut; **b** – stretched elastic nut

Fig. AFS-T.9.2.3 Effects of reducing the stiffness of bolts on the behavior of assemblies with axially loaded prestressed bolts: **a** – characteristic and pulsating variation of the external force; **b** – characteristic of the assembly

Note.

a. in the case of constructive variants of elasticization of the nut body to uniformize the distribution of the external force on the coils in contact, loads of the first coil of up to 20% are obtained (fig. AFS-T.9.2.2) unlike the case of distribution for the rigid nut where the value of the load of the first coil is 30...40% (fig. AFS-T.6.1);

b. in the case of axially loaded prestressed assemblies, the use of bolts with increased elasticity (AFS-T.9.1.2, e,f,g) leads to a decrease in the additional force, F_{sl} (including its amplitude) and the total force F_s (fig. AFS-T.9.2.3,b) that loads the bolt compared to the use of bolts with constant section (AFS-T.9.1.2,b) loaded by the same external force F and with the same variation mode (pulsating).



AFS-T.9.3 SECURING THREADED ASSEMBLIES AGAINST SELF-LOOSENING

Purpose: avoiding self-loosening of threaded assemblies which, although meeting the self-locking condition (rel. AFS-T.8.1.1.6), may loosen due to the actions of **variable and shock loads, vibrations and temperature differences** which have the effect of reducing friction in the assembly.

The principles underlying constructive insurance solutions are:

- **increasing friction** forces between assembly elements;
- preventing the relative rotation of the nut with respect to the screw by means of **additional elements**;
- **local plastic deformation** of the bolt and/or nut material or application of additional material (usually spot welding)

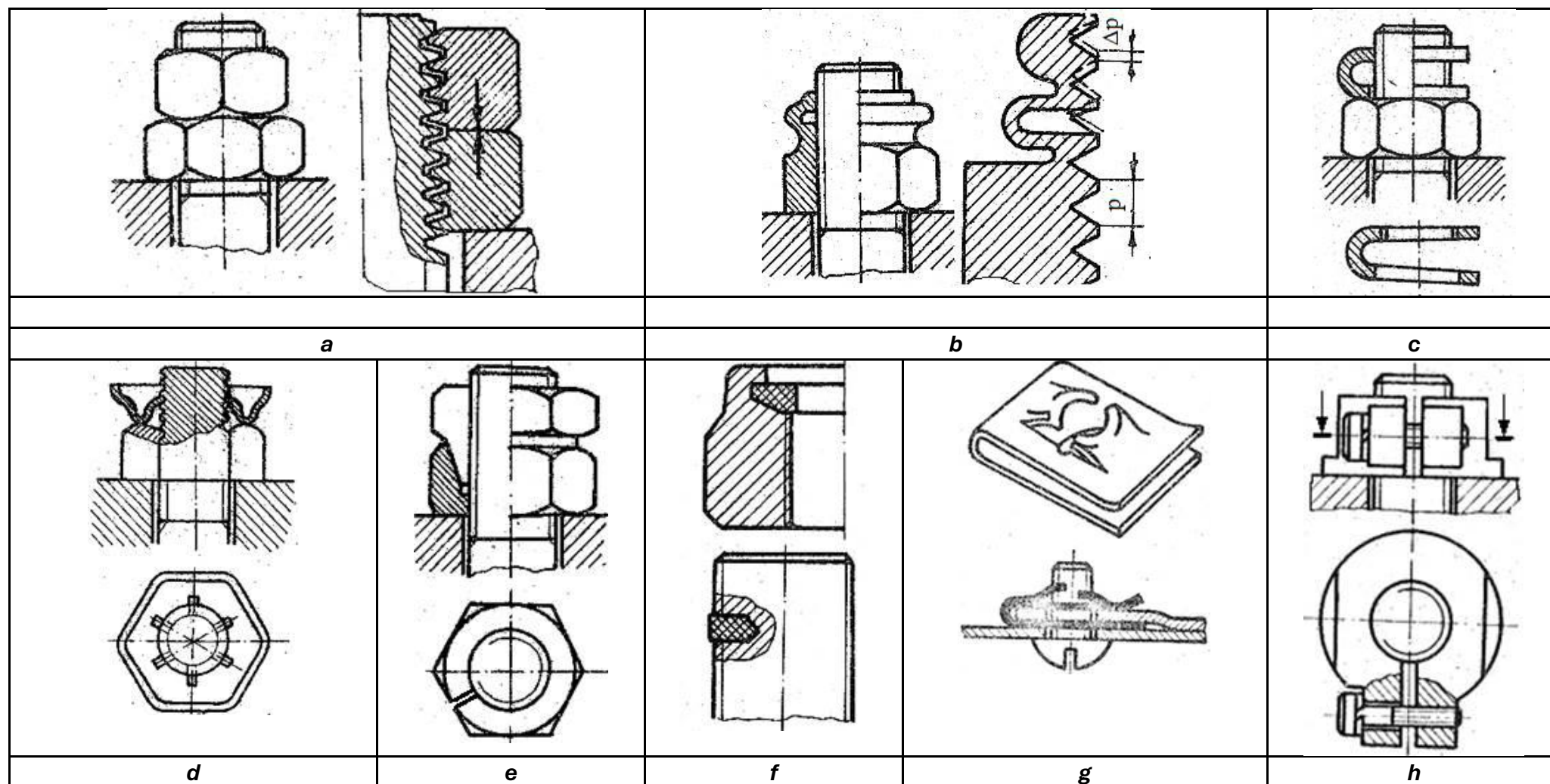


Fig. AFS-T.9.3.1 Constructive solutions for securing against self-loosening by increasing friction forces: **a** – with lock nut; **b** – with solid elastic nut; **c** – with lamellar elastic lock nut; **d** – with stamped elastic lock nut; **e** – with conical elastic lock nut; **f** – with elastic inserts; **d** – sheet metal elastic nut; **g** – sectioned nut

Description:

Fig. AFS-T.9.3.1,a: by tightening the locknut, additional pressures are created between the threads of the nut and the bolt, which are maintained even if the external force is canceled due to variable loading, vibrations and/or temperature variations.

Fig. AFS-T.9.3.1,b: the elastic locknut, after tightening with the wrench, provides, due to the difference in pitch of the outer portion, a loading with a local axial force that generates a friction force independent of the external load.

Fig. AFS-T.9.3.1,c,d: by tightening with the wrench the upper threaded part deforms in relation to the lower part and an elastic force is produced which locally stresses the screw and generates friction force independent of the external load.

Fig. AFS-T.9.3.1,e: The split conical lock nut creates additional axial forces and strong radial pressure by tightening in the thread of the lower nut.

Fig. AFS-T.9.3.1,f: the nut or bolt contains inserts, usually made of nylon, which, as a result of deformation during tightening, provide increased frictional forces.

Fig. AFS-T.9.3.1,g: the self-tightening sheet metal nut, used to assemble vehicle body elements, has two teeth that act as a thread and when tightened, it compresses and presses on the screw thread, generating friction that opposes disassembly.

Fig. AFS-T.9.3.1,h: By tightening the tangential screw, increased thread pressure forces are generated in the thread of the radially sectioned nut.

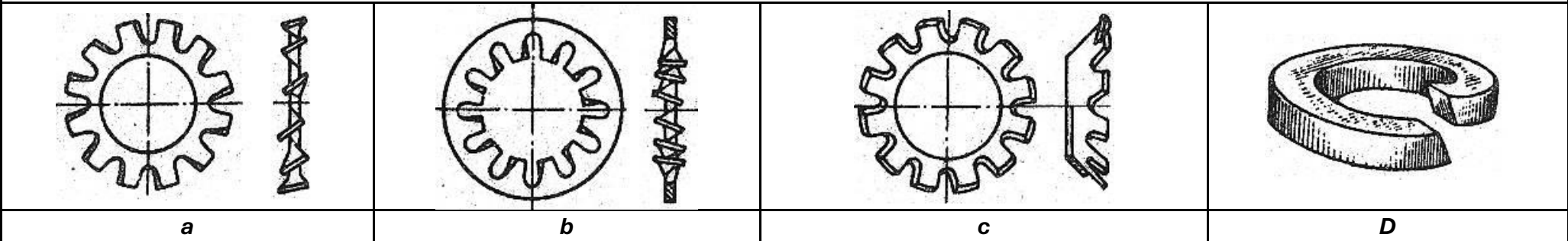


Fig. AFS-T.9.3.2 Constructive solutions for securing against self-loosening by increasing friction forces with elastic washers: a – flat toothed on the outside; b – flat toothed on the inside; c – conical with external teeth; d – Grower

Fig. AFS-T.9.3.2: the edges of the washer teeth and the flared ends of the Grower washers contribute to preventing the nut from rotating by locally deforming the surfaces with which they come into contact.

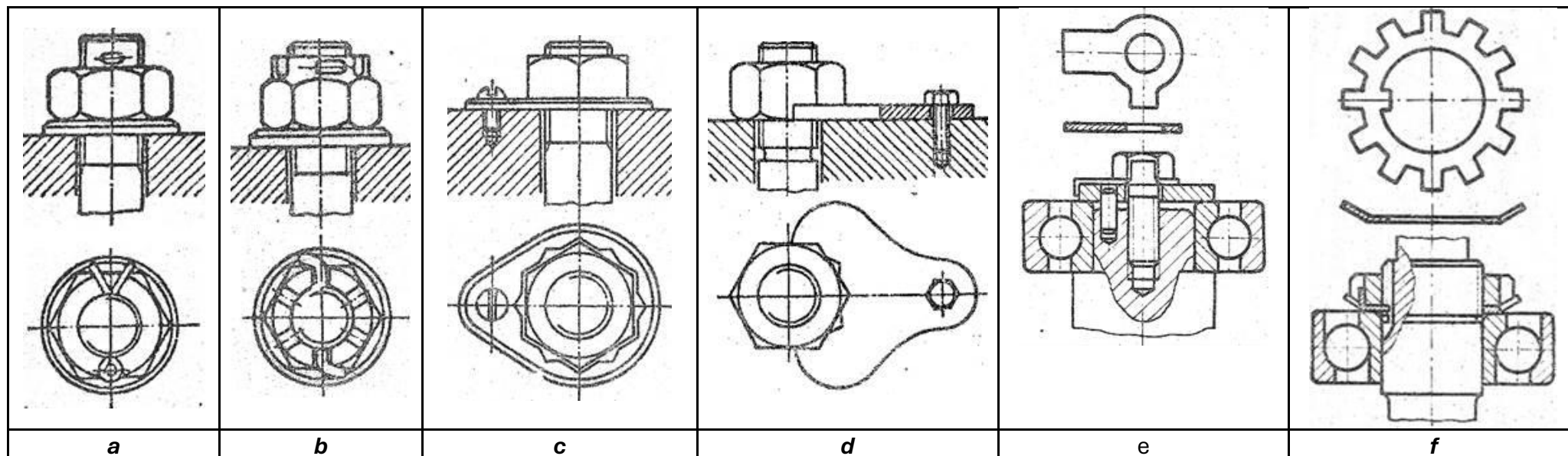


Fig. AFS-T.9.3.3 Constructive solutions for securing against self-loosening by additional elements: **a** – by split nail (splint); **b** – deformed pin in crenellated nut; **c** – profiled plate on the entire contour; **d** – profiled plate on the semi-contour; **e** – washer with two wings; **f** – safety washer for splined nuts of bearings

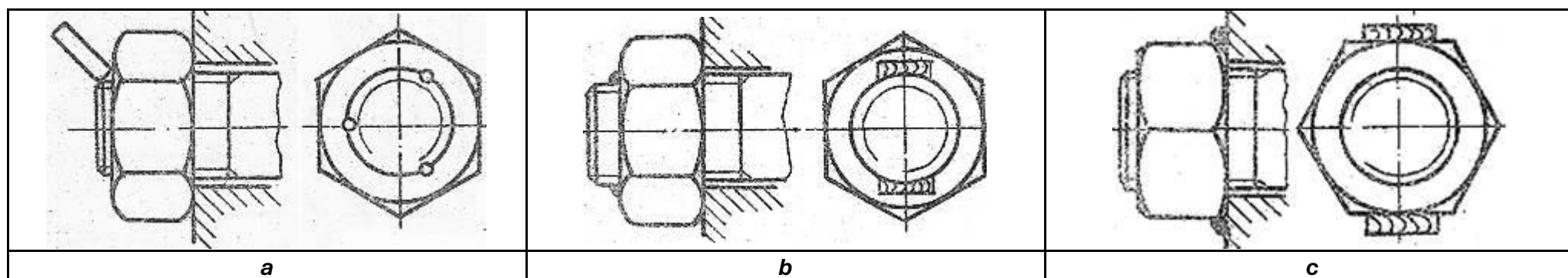


Fig. AFS-T.9.3.4 Constructive solutions for securing against self-loosening through plastic deformation or welding: **a** – plastic deformation; **b** – nut-bolt welding; **c** – nut weld in relation to the tightened part

AAB. SHAFT-HUB ASSEMBLIES

AAB-T. CONSTRUCTION AND MODELING OF SHAFT-HUB ASSEMBLIES

CONTENTS

AAB-T.1 FORM ASSEMBLIES	6-5
AAB-T.1.1 PARALLEL KEY ASSEMBLIES	6-5
AAB-T.1.1.1 DEFINITION	6-5
AAB-T.1.1.2 CONSTRUCTION STRUCTURE	6-5
AAB-T.1.1.3 AREAS OF USE, ADVANTAGES AND DISADVANTAGES	6-6
AAB-T.1.1.4 CLASSIFICATION	6-6
AAB-T.1.1.5 MATERIALS AND TECHNOLOGIES	6-7
AAB-T.1.1.6 FORMS AND CAUSES OF DISPOSAL OR INCOMPATIBLE BEHAVIOR	6-7
AAB-T.1.1.7 FUNCTIONAL AND CONSTRUCTIVE PARAMETERS	6-8
AAB-T.1.1.8 CALCULATION MODELS	6-8
AAB-T.1.2 ASSEMBLIES THROUGH DISC KEYS	6-11
AAB-T.1.2.1 DEFINITION	6-11
AAB-T.1.2.2 CONSTRUCTION STRUCTURE	6-11
AAB-T.1.2.3 AREAS OF USE, ADVANTAGES AND DISADVANTAGES	6-11
AAB-T.1.2.4 CLASSIFICATION	6-12
AAB-T.1.2.5 MATERIALS AND TECHNOLOGIES	6-12
AAB-T.1.2.6 FORMS AND CAUSES OF DISPOSAL OR INCOMPATIBLE BEHAVIOR	6-12
AAB-T.1.2.7 FUNCTIONAL AND CONSTRUCTIVE PARAMETERS	6-13
AAB-T.1.2.8 CALCULATION MODELS	6-13
AAB-T.1.3 THE SPLINE ASSEMBLY	6-15
AAB-T.1.3.1 DEFINITION	6-15
AAB-T.1.3.2 CONSTRUCTION STRUCTURE	6-16
AAB-T.1.3.3 AREAS OF USE, ADVANTAGES AND DISADVANTAGES	6-16
AAB-T.1.3.4 CLASSIFICATION	6-17
AAB-T.1.3.5 MATERIALS AND TECHNOLOGIES	6-17
AAB-T.1.3.6 FORMS AND CAUSES OF DISPOSAL OR INCOMPATIBLE BEHAVIOR	6-18
AAB-T.1.3.7 FUNCTIONAL AND CONSTRUCTIVE PARAMETERS	6-18
AAB-T.1.3.8 CALCULATION MODELS	6-18
AAB-T.1.4 PROFILED ASSEMBLIES (ON POLYGONAL CONTOUR)	6-21
AAB-T.1.4.1 DEFINITION	6-21
AAB-T.1.4.2 CONSTRUCTION STRUCTURE	6-22
AAB-T.1.4.3 AREAS OF USE, ADVANTAGES AND DISADVANTAGES	6-22
AAB-T.1.4.4 CLASSIFICATION	6-23
AAB-T.1.4.5 MATERIALS AND TECHNOLOGIES	6-23
AAB-T.1.4.6 FORMS AND CAUSES OF DISPOSAL OR INCOMPATIBLE BEHAVIOR	6-24
AAB-T.1.4.7 FUNCTIONAL AND CONSTRUCTIVE PARAMETERS	6-25
AAB-T.1.4.8 CALCULATION MODELS	6-25
AAB-T.1.5 ASSEMBLIES USING ELASTIC RINGS	6-27

AAB-T.1.5.1 DEFINITION	6-27
AAB-T.1.5.2 CONSTRUCTION STRUCTURE	6-27
AAB-T.1.5.3 AREAS OF USE, ADVANTAGES AND DISADVANTAGES	6-27
AAB-T.1.5.4 CLASSIFICATION	6-27
AAB-T.1.5.5 MATERIALS AND TECHNOLOGIES.....	6-29
AAB-T.1.5.7 FUNCTIONAL AND CONSTRUCTIVE PARAMETERS	6-29
AAB-T.1.5.8 CALCULATION MODELS.....	6-30
AAB-T.1.5.9 INSTALLATION AND OPERATION.....	6-30
AAB-T.2 FRICTION JOINTS.....	6-32
AAB-T.2.1 SELF-CLEANING (PRESSED) ASSEMBLIES.....	6-32
AAB-T.2.1.1 DEFINITION.....	6-32
AAB-T.2.1.2 CONSTRUCTION STRUCTURE	6-32
AAB-T.2.1.3 AREAS OF USE, ADVANTAGES AND DISADVANTAGES	6-32
AAB-T.2.1.4 CLASSIFICATION	6-33
AAB-T.2.1.5 MATERIALS AND TECHNOLOGIES.....	6-33
AAB-T.2.1.6 FORMS AND CAUSES OF DISPOSAL OR INCOMPATIBLE BEHAVIOR	6-33
AAB-T.2.1.7 FUNCTIONAL AND CONSTRUCTIVE PARAMETERS	6-34
AAB-T.2.1.8 CALCULATION MODELS.....	6-34
AAB-T.2.1.9 INSTALLATION AND OPERATION.....	6-39
AAB-T.2.2 CYLINDER TIGHTENING ASSEMBLIES (BRACELET)	6-39
AAB-T.2.2.1 DEFINITION.....	6-39
AAB-T.2.2.2 CONSTRUCTION STRUCTURE	6-40
AAB-T.2.2.3 AREAS OF USE, ADVANTAGES AND DISADVANTAGES	6-40
AAB-T.2.2.4 CLASSIFICATION	6-40
AAB-T.2.2.5 MATERIALS AND TECHNOLOGIES.....	6-41
AAB-T.2.2.6 FORMS AND CAUSES OF DISPOSAL OR INCOMPATIBLE BEHAVIOR	6-41
AAB-T.2.2.7 FUNCTIONAL AND CONSTRUCTIVE PARAMETERS	6-41
AAB-T.2.2.8 CALCULATION MODELS.....	6-41
AAB-T.2.3 ASSEMBLIES BY TIGHTENING ON THE CONE	6-43
AAB-T.2.3.1 DEFINITION.....	6-43
AAB-T.2.3.2 CONSTRUCTION STRUCTURE	6-43
AAB-T.2.3.3 AREAS OF USE, ADVANTAGES AND DISADVANTAGES	6-43
AAB-T.2.3.4 CLASSIFICATION	6-44
AAB-T.2.3.5 MATERIALS AND TECHNOLOGIES.....	6-45
AAB-T.2.3.6 FORMS AND CAUSES OF DISPOSAL OR INCOMPATIBLE BEHAVIOR	6-45
AAB-T.2.3.7 FUNCTIONAL AND CONSTRUCTION PARAMETERS	6-45
AAB-T.2.3.8 CALCULATION MODELS.....	6-46
.....	6-47
AAB-T.2.4 ASSEMBLIES WITH TRUNCATED CONICAL RING	6-47
AAB-T.2.4.1 DEFINITION.....	6-47
AAB-T.2.4.2 CONSTRUCTION STRUCTURE	6-48

AAB-T.2.4.3 AREAS OF USE, ADVANTAGES AND DISADVANTAGES	6-49
AAB-T.2.4.4 CLASSIFICATION	6-49
AAB-T.2.4.5 MATERIALS AND TECHNOLOGIES.....	6-50
AAB-T.2.4.6 FORMS AND CAUSES OF DISPOSAL OR INCOMPATIBLE BEHAVIOR	6-50
AAB-T.2.4.7 FUNCTIONAL AND CONSTRUCTIVE PARAMETERS	6-50
AAB-T.2.4.8 CALCULATION MODELS.....	6-51

AAB-T.1 FORM ASSEMBLIES

AAB-T.1.1 PARALLEL KEY ASSEMBLIES

AAB-T.1.1.1 DEFINITION

The parallel key assembly is a removable assembly of the shaft-hub type, which transmits the torsional moment by means of form contact with the shaft and hub on the side faces of a prismatic intermediate element (fig. AAB-T.1.1.2.1).

AAB-T.1.1.2 CONSTRUCTION STRUCTURE

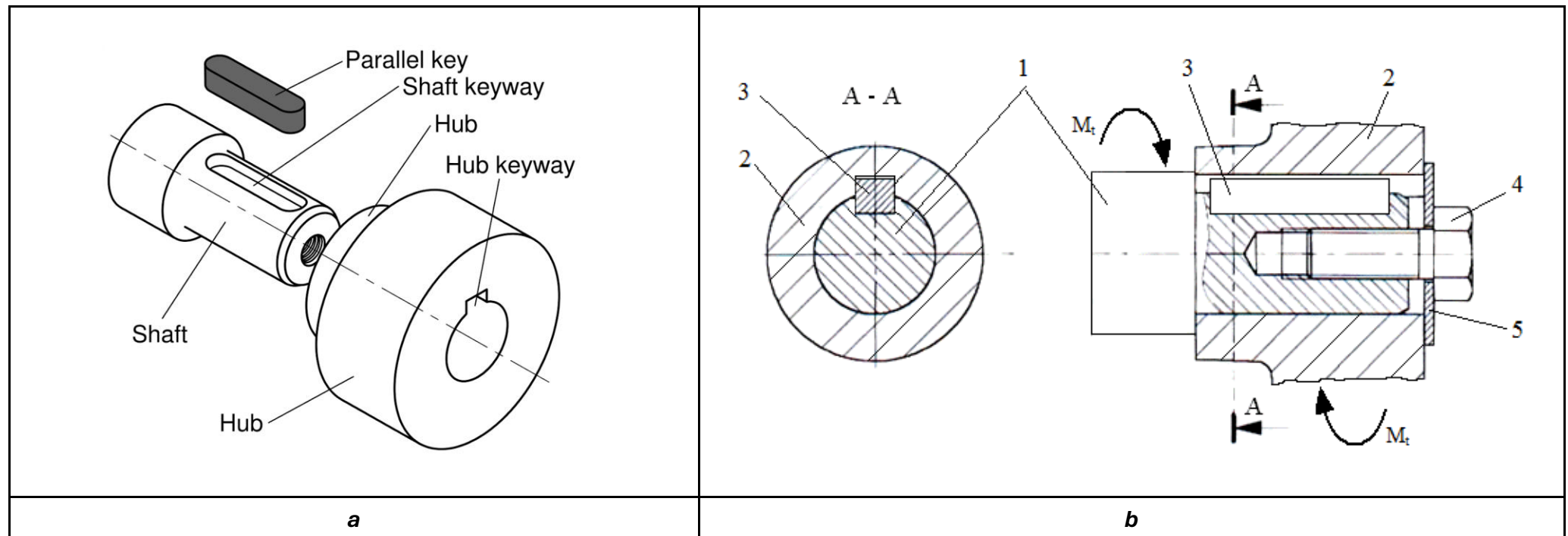


Fig. AAB-T.1.1.2.1 Constructive structure of the parallel key assembly: **a** – exploded axonometric view of the elements of a parallel key assembly; **b** – axial and transverse section

Note. The torque is transmitted through the form, according to the flow: 1 -> 3 -> 2; 1 - shaft, 2 - hub, 3 - parallel key; the axial fixation of the hub 2 is done with the screw 5 and the washer 4.

AAB-T.1.1.3 AREAS OF USE, ADVANTAGES AND DISADVANTAGES

Main areas of use: all transmissions involving the attachment of gear wheels, chain or belt wheels, half-couplings, etc. to shafts.

Tab. AAB-T.1.1.3.1 Advantages and disadvantages of parallel key joints

Advantages *	Disadvantages *
- easy assembly and disassembly. - international standardization. - high durability. - the shaft and hub can be reused in case of damage to the quill	- the keyway in the shaft represents a strong stress concentrator (fig. AAB-T.1.1.8.1); - do not take up axial forces
* compared to other types of assemblies (by shape, by friction forces, non-dismountable, etc.)	

AAB-T.1.1.4 CLASSIFICATION

Tab. AAB-T.1.1.4. 1 Classification of parallel key joints [Jula, 1986; Rădulescu, 1981]

criterion	Assembly type	Scheme
Shape of the key ends (only for parallel key assembly)	Parallel key with round ends (form A)	Fig. AAB-T.1.1.4.1,a
	Parallel key with straight ends (form B)	Fig. AAB-T.1.1.4.1,b
	Parallel key with one round head and the other straight (shape C) – only usable on shaft ends	Fig. AAB-T.1.1.4.1,c
Functional role	Fixed assemblies	Fig. AAB-T.1.1.4.1,a,b,c
	Mobile assemblies	Fig. AAB-T.1.1.4.1,d,e

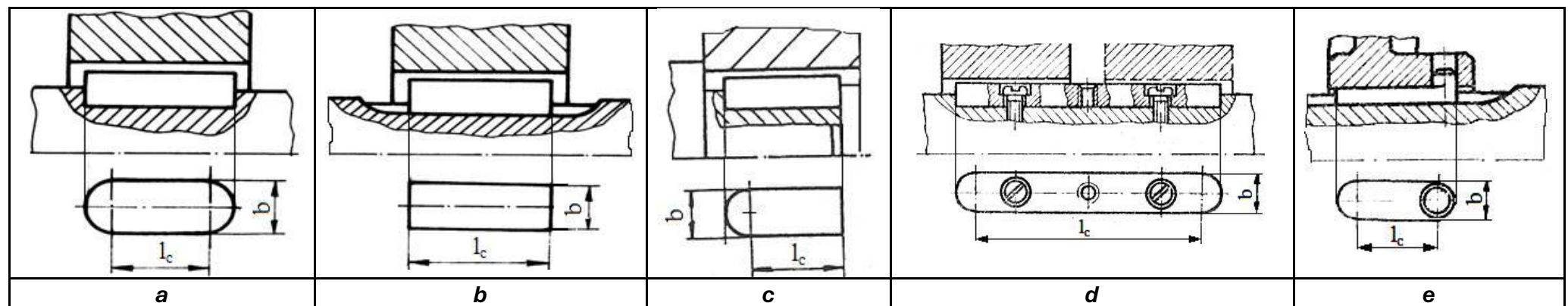
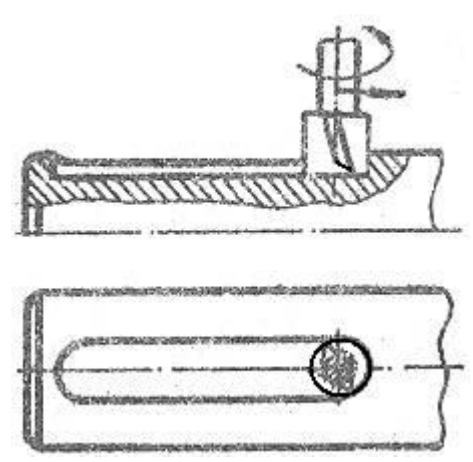
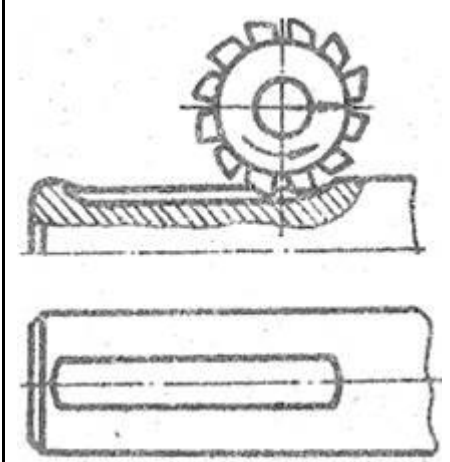
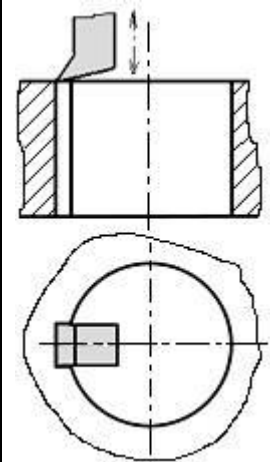


Fig. AAB-T.1.1.4.1 Shapes and types of parallel keys: **a** – A (with rounded ends); **b** – B (as straight ends); **c** – C (one end rounded and the other straight); **d** – A, fixed to the shaft by screw assemblies (for mobile assemblies); **e** – A, fixed to the hub by a spigot (for mobile assemblies)

AAB-T.1.1.5 MATERIALS AND TECHNOLOGIES

The keys are usually made of steels with a tensile strength of $\sigma_t = 500...600 \text{ MPa}$, steels for mechanical construction: E295 (OL50), E335 (OL60) according to SR EN 10025-2 (STAS 500/2) or carbon steels for improvement: C35 (OLC35), C45 (OLC45) according to SR EN 10083 (STAS 880).

Tab. AAB-T.1.1.5. 1 Key channel machining processes

Key channel with rounded ends (for keys type A or C)	Key channel with straight ends (for type B keys)	Keyway in hub
Milling with a finger mill	Milling with a disc cutter	Mortising (or broaching)
		
Note: The parallel key is processed by milling and grinding.		

AAB-T.1.1.6 FORMS AND CAUSES OF DISPOSAL OR INCOMPATIBLE BEHAVIOR

Tab. AAB-T.1.1.6.1 Forms and causes of disuse or improper behavior

Shapes	Result	Appearance	Cause	Manifestation	Avoidance
Key shearing	Disposal	At overloads Hardened shafts and logs and soft material keys	Exceeding permissible strength the shear	Longitudinal sectioning of the key	Limitation by calculation of the shear stress that occurs in the key, to permissible values
Damage to the key flanks and/or channels by crushing (plastic deformation)	Locking assembly and preventing disassembly	For fixed assemblies For heavily loaded springs, shafts and	Exceeding permissible crushing stresses	Damage to the active flanks of the parallel key in the lateral areas of	Limitation of crushing stresses by calculating to permissible values

		wheels made of soft materials		contact with the shaft and/or hub	
Expulsion of the lubricant film	Difficult axial movement (only for mobile assemblies)	In axially moving assemblies, lubricated	Exceeding the permissible pressure to prevent the lubricant film from being expelled	Dry friction, wear	Limiting by calculation the non-expulsion pressure of the lubricant to permissible values

AAB-T.1.1.7 FUNCTIONAL AND CONSTRUCTIVE PARAMETERS

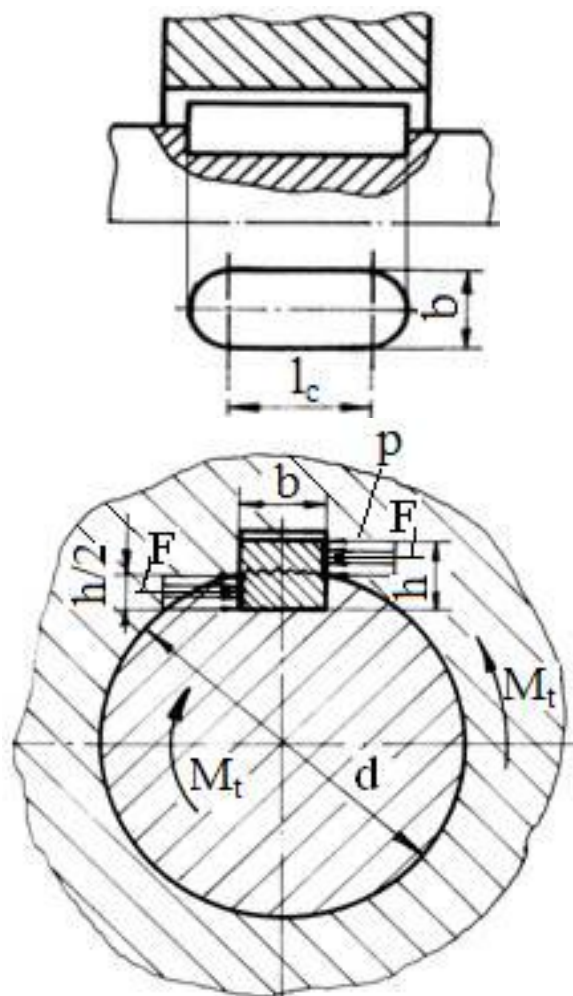
Tab. AAB-T.1.1.7.1 Construction parameters of parallel keys [STAS-1005]

The image shows three types of parallel keys: A (rounded ends), B (straight ends), and C (one rounded end, one straight). Each type is shown in side and end views. Dimensions include width b , height h , length l , and head radius $R=b/2$. A detail view shows a 45-degree bevel with height f and radius r .

Standard construction forms:
A – with both ends rounded,
B – with both ends straight,
C – with one end round and the other straight

Geometric parameters:
 b [mm] – width of the key,
 h [mm] – height of the key,
 l [mm] – length of the key,
 R [mm] – radius of the key head,
 f [mm] – bevel height,
 r [mm] – contour radius

AAB-T.1.1.8 CALCULATION MODELS

Tab. AAB-T.1.1.8.1 Resistance calculation model [Jula, 1986; Rădulescu, 1981; Florea, 2007]**Fig. AAB-T.1.1.8.1** Strength calculation model**Calculation assumptions and requirements:**

- the contact pressures are uniformly distributed on the active faces of the key;
- it is considered that the key is inserted, half of its height in the channel made in the shaft, and the other half in the channel made in the hub.
- the resultant force arm of the contact pressures (with the shaft, respectively with the hub), is approximated to the value $d/2$ and for the calculation,

$$F = \frac{2 M_t}{d} ; \text{ (APP-T.8.1.1.8.1)}$$

- for the calculation of the crushing of the assembly and the shear of the key, the rounded portions of the head are not taken into account.

The torsional resistance condition of the shaft in the keyway area:

$$\frac{\beta_k M_t}{\frac{\pi d^3}{16}} \leq \tau_{at}, \quad \text{ (APP-T.8.1.1.8.2)}$$

where, $\tau_{at} = 15...45$ MPa is the conventional allowable torsional stress (also considering the existence of bending stress) and $\beta_k = 2...3$, the stress concentrator factor (for the calculation of variable, fatigue stresses); in the case of quasi-static stresses, $\beta_k = 1...2$ is considered.

The diameter of the shaft in the key area, from this condition (rel. APP-T.8.1.1.8.2), becomes,

$$d \geq \sqrt[3]{\frac{16 \beta_k M_t}{\pi \tau_{at}}} . \quad \text{ (APP-T.8.1.1.8.3)}$$

Crush resistance condition of the assembly

The contact pressure on the contact surfaces of the key with the shaft or hub,

$$p = \frac{F}{A_s} = \frac{\frac{2 M_t}{d}}{\frac{h}{2} l_c} = \frac{4 M_t}{d h l_c} \leq \sigma_{as}(p_a), \quad \text{ (APP-T.8.1.1.8.4)}$$

where, M_t [Nmm] is the torsional moment that loads the assembly, d [mm] – shaft diameter, h [mm] – key height (adopted from STAS 1004, depending on the shaft diameter, d), l_c [mm] – calculated key length; σ_{as} [MPa] – allowable crushing stress (for fixed assemblies, fer. APP-M.1); p_a [MPa] – the admissible pressure, not to expel the lubricant film (for mobile assemblies, fer. APP-M.1); from the last part of this relation the dimensioning relations are obtained,

$$l_c = \frac{4 M_t}{d h \sigma_{as}}, \text{ for fixed assemblies and } l_c = \frac{4 M_t}{d h p_a}, \text{ for mobile assemblies.}$$

Shear resistance condition of the key

The shear stress of the key,

$$\tau_f = \frac{F_f}{A_f} = \frac{\frac{2 M_t}{d}}{b l_c} = \frac{2 M_t}{d b l_c} \leq \tau_{af}, \quad (\text{APP-T.8.1.1.8.5})$$

where,

b [mm] is the width of the key (adopted from STAS 1004, depending on the diameter, d, of the shaft) and

τ_{af} – the allowable shear stress [MPa] (fer. [APP-M.1](#))

Note.

- to reduce the stress concentrator effect (increasing fatigue resistance), the keys are made with beveled edges and, consequently, the key channels are made with connections (fig. AAB-T.1.1.8.1);
- verification of standard keys is usually no longer done in design practice, because the shear strength condition was used to determine the width b , considering the key material indicated in the standard.

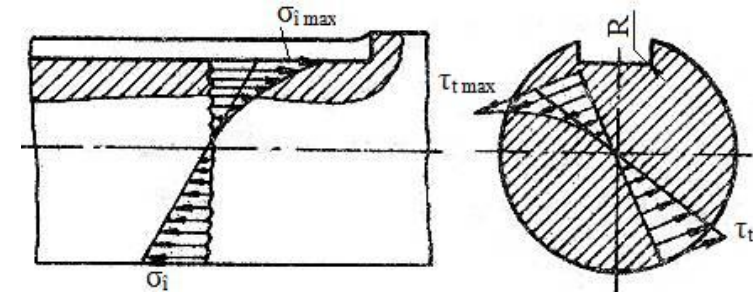


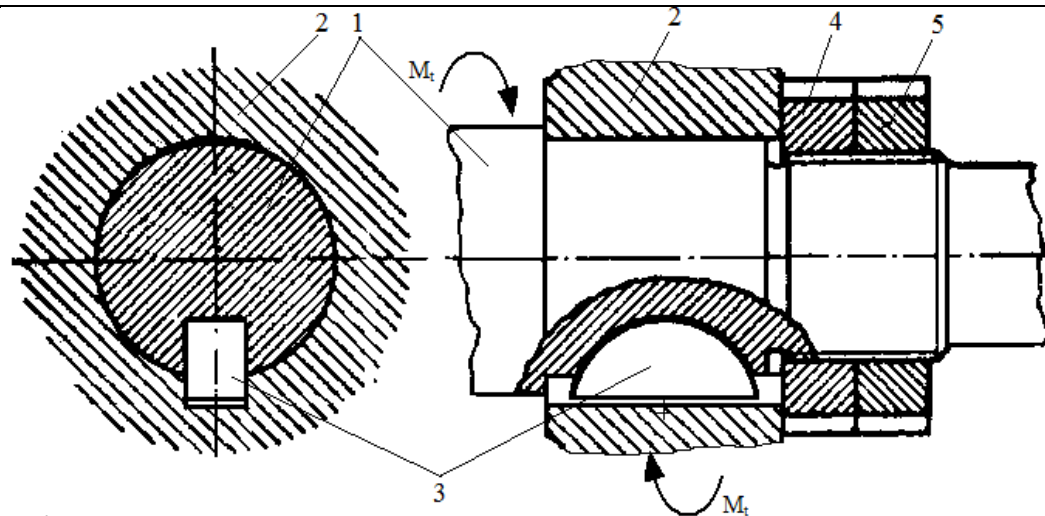
Fig. AAB-T.1.1.8.1 Distributions of bending and torsional stresses in the case of movable loads

AAB-T.1.2 ASSEMBLIES THROUGH DISC KEYS

AAB-T.1.2.1 DEFINITION

The *disc key assembly* is a removable assembly of the shaft-hub type, which transmits the torque by means of form contacts with the shaft and hub on the side faces of an intermediate element in the form of a disc segment (fig. AAB-T.1.2.2.1).

AAB-T.1.2.2 CONSTRUCTION STRUCTURE



Note. The torque is transmitted through the form, according to the flow: 1 -> 3 -> 2; 1 - shaft, 2 - hub, 3 - disc key; the axial fixation of the hub 2 is done with the nut 4 secured against unscrewing with the locknut 5.

Fig. AAB-T.1.1.2.1 Structural structure of the disc key assembly in axial and transverse section

AAB-T.1.2.3 AREAS OF USE, ADVANTAGES AND DISADVANTAGES

Main areas of use: in the construction of machine tools, vehicles, agricultural machinery, etc., in particular, for assembling wheels on shaft ends

Tab. AAB-T.1.2.3 Advantages and disadvantages of disc key assemblies

Advantages *	Disadvantages *
- easy assembly and disassembly. - international standardization. - high durability. - the shaft and hub can be reused in case of damage to the quill	- the keyway in the shaft represents a strong stress concentrator (greater than in the case of parallel key assemblies); - do not take up axial forces; - unsuitable for small diameter shafts
* Compared to other types of assemblies (by shape, by friction forces, non-dismountable, etc.)	

AAB-T.1.2.4 CLASSIFICATION

AAB-T.1.2.5 MATERIALS AND TECHNOLOGIES

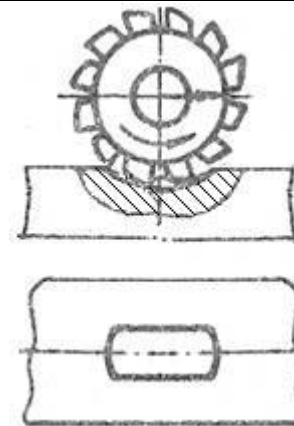
Materials

The keys are usually made of steels with a tensile strength of $s_r = 500...600$ MPa, steels for mechanical construction: E295 (OL50), E335 (OL60) according to SR EN 10025-2 (STAS 500/2) or carbon steels for improvement: C35 (OLC35), C45 (OLC45) according to SR EN 10083 (STAS 880).

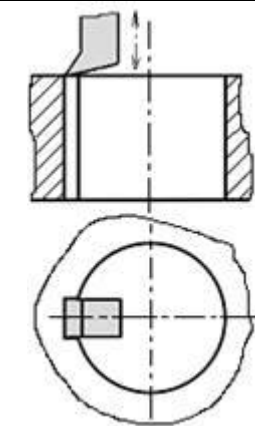
TECHNOLOGY

Channel processing:

- by cutting with a disk cutter (fig. AAB-T.1.2.5.1,a), in the shaft
- by cutting: **mortising** (fig. AAB-T.1.2.5.1,b) or **broaching** for large production runs



a



b

Fig. AAB-T.1.2.5.1 Machining of splines for the disc key: **a** – in the shaft; **b** – in the hub

AAB-T.1.2.6 FORMS AND CAUSES OF DISPOSAL OR INCOMPATIBLE BEHAVIOR

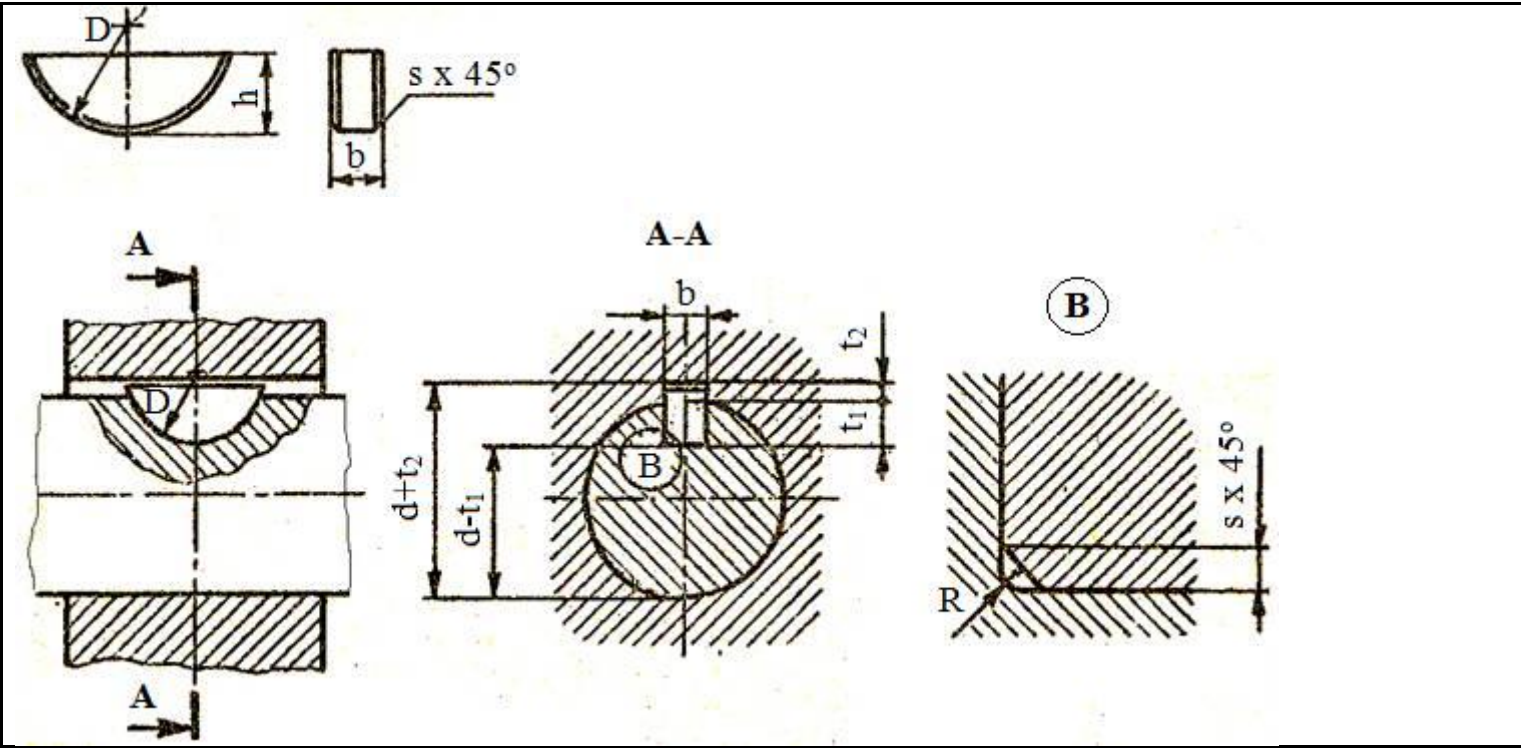
Tab. AAB-T.1.1.6.1 Forms and causes of disuse or improper behavior

shapes	result	Appearance	CAUSE	Manifestation	Avoidance
Key shearing	Disposal	At overloads	Exceeding the permissible shear strength	Longitudinal sectioning of the key	Limitation by calculation of the shear stress that occurs in the

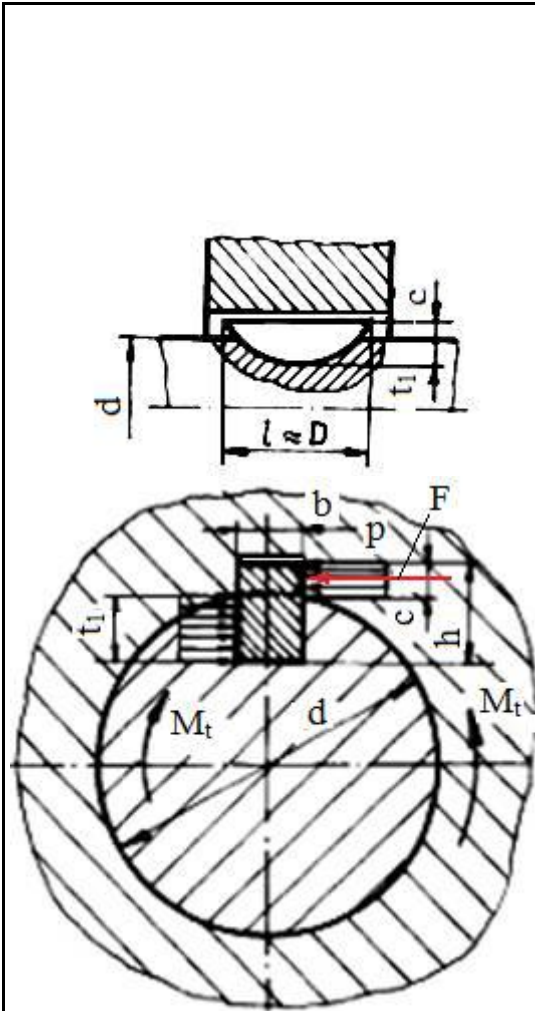
		Hardened shafts and logs and soft material keys			key, to permissible values
Damage to the key flanks and/or channels by crushing (plastic deformation)	Locking assembly and preventing disassembly	For fixed assemblies For heavily loaded springs, shafts and wheels made of soft materials	Exceeding permissible crushing stresses	Damage to the active flanks of the parallel key in the lateral areas of contact with the shaft and/or hub	Limitation of crushing stresses by calculation to permissible values

AAB-T.1.2.7 FUNCTIONAL AND CONSTRUCTIVE PARAMETERS

Tab. AAB-T.1.2.7.1 Construction parameters of disc keys and shaft and hub splines [STAS-1012]

	Geometric parameters of the keys and channels: b [mm] – width of the key, h [mm] – height of the key, D [mm] – sector diameter, R [mm] – connection radius at the bottom of the channel, t_1 [mm] – height of the channel in the shaft, t_2 [mm] – height of the channel in the hub, s [mm] – thickness of the key bevel,
--	---

AAB-T.1.2.8 CALCULATION MODELS

Tab. AAB-T.1.2.8.1 Resistance calculation model [Jula, 1986; Rădulescu, 1981; Florea, 2007]**Fig. AAB-T.1.2.8.1** Strength calculation model

Calculation assumptions and requirements:

- the contact stresses are uniformly distributed on the active faces of the key;
- the resultant force arm of the contact pressures (with the shaft, respectively with the hub), is approximated to the value d/2 and for calculation,

$$F = \frac{2 M_t}{d} \quad (\text{APP-T.8.1.2.8.1})$$

- the crushing calculation of the assembly is made for the most stressed contact, key-disc, for which the contact surface is considered to be rectangular with length $l \approx D$ and width $c = h t_1 \approx$
- the shear section of the key is considered rectangular with length $l \approx D$ and width $b \approx$

The torsional resistance condition of the shaft in the keyway area:

$$\frac{\beta_k M_t}{\frac{\pi d^3}{16}} \leq \tau_{at}, \quad (\text{APP-T.8.1.2.8.2})$$

where, $\tau_{at} = 15 \dots 45$ MPa represents the conventional allowable stress at torsion (also takes into account the existence of bending stress) and $\beta_k = 2 \dots 4$, the stress concentrator factor (for the calculation of variable, fatigue stresses); in the case of quasi-static stresses, $\beta_k = 1 \dots 2$ is considered

The diameter of the shaft in the key area, from this condition (rel. APP-T.8.1.2.8.2), becomes,

$$d \geq \sqrt[3]{\frac{16 \beta_k M_t}{\pi \tau_{at}}} \quad (\text{APP-T.8.1.2.8.3})$$

Crush resistance condition of the assembly

The contact pressure on the contact surface of the key with the hub,

$$p = \frac{F}{A_s} = \frac{\frac{2 M_t}{d}}{D c} \leq \sigma_{as}, \quad (\text{APP-T.8.1.2.8.4})$$

where, M_t [Nmm] represents the torsional moment that loads the assembly, d [mm] – shaft diameter, h [mm] – key height (adopted from S tas 1012, depending on the shaft diameter, d), D [mm] – calculated length of the key; c [mm] – width of the contact area with the hub; σ_{as} [MPa] – allowable crushing stress of the assembly (fer. APP-M.1).

Shear resistance condition of the key

The shear stress of the key,

$$\tau_f = \frac{F_f}{A_f} = \frac{\frac{2 M_t}{d}}{b D} \leq \tau_{af}, \quad (\text{APP-T.8.1.2.8.5})$$

where, b [mm] represents the width of the key (adopted from Stas 1012, depending on the diameter, d [mm] of the shaft) and τ_{af} [MPa] – the allowable shear stress (fer. [APP-M.1](#)).

AAB-T.1.3 THE SPLINE ASSEMBLY

AAB-T.1.3.1 DEFINITION

The spline assembly is a removable assembly of the shaft-hub type, which can be considered as an assembly with multiple parallel keys, integral with the shaft and equiangularly positioned on its periphery (fig. AAB-T.1.3.2.1,c).

AAB-T.1.3.2 CONSTRUCTION STRUCTURE

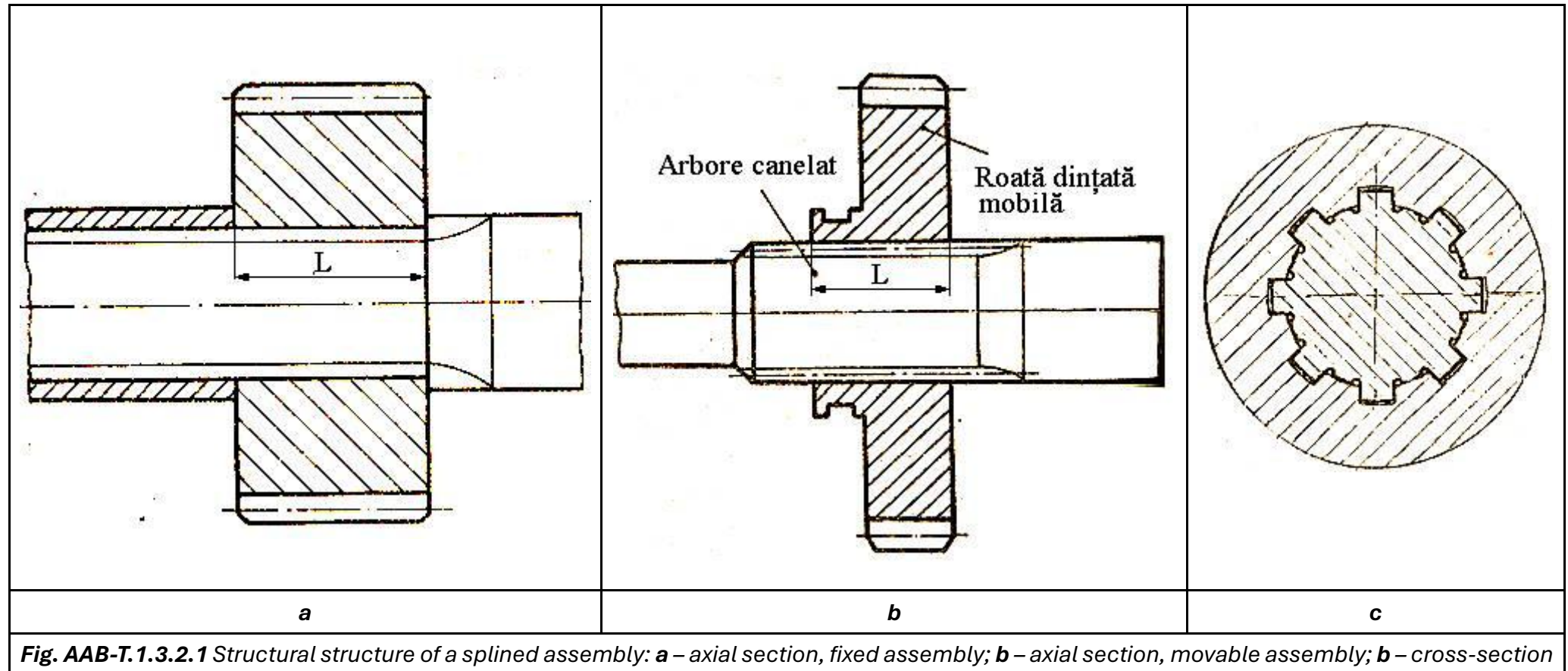


Fig. AAB-T.1.3.2.1 Structural structure of a splined assembly: **a** – axial section, fixed assembly; **b** – axial section, movable assembly; **c** – cross-section

AAB-T.1.3.3 AREAS OF USE, ADVANTAGES AND DISADVANTAGES

Main areas of use: all transmissions involving the attachment of gear wheels, chain or belt wheels to shafts, half-couplings that transmit higher torques than assemblies using two or even three parallel keys.

Tab. AAB-T.1.3.4.1 Advantages and disadvantages of spline assemblies

Advantages *	Disadvantages *
- higher load capacity; - greater fatigue resistance; - better centering and guidance	- more demanding execution technologies, and therefore higher costs
* related to key assemblies	

AAB-T.1.3.4 CLASSIFICATION

Tab. AAB-T.1.3.4.1 Classification of spline assemblies [Jula, 1986; Rădulescu, 1981]

critereon	Assembly type	Scheme
Spline profile shape	With rectangular profile	Fig. AAB-T.1.3.4.1,a,b,c
	With involute profile	Fig. AAB-T.1.3.4.2
	With triangular profile	Fig. AAB-T.1.3.4.3
Functional role	Fixed assemblies	Fig. AAB-T.1.3.2.1,a,b
	Mobile assemblies	Fig. AAB-T.1.3.2.1,b
Cross-sectional dimensions (only for rectangular profile splines)	Light series - the torque transmitted by the assembly is lower than that transmitted by the shaft	
	Medium series - the torque transmitted by the assembly is equal to that transmitted by the shaft	
	Heavy series - the torque transmitted by the assembly is with large oscillations and shocks.	

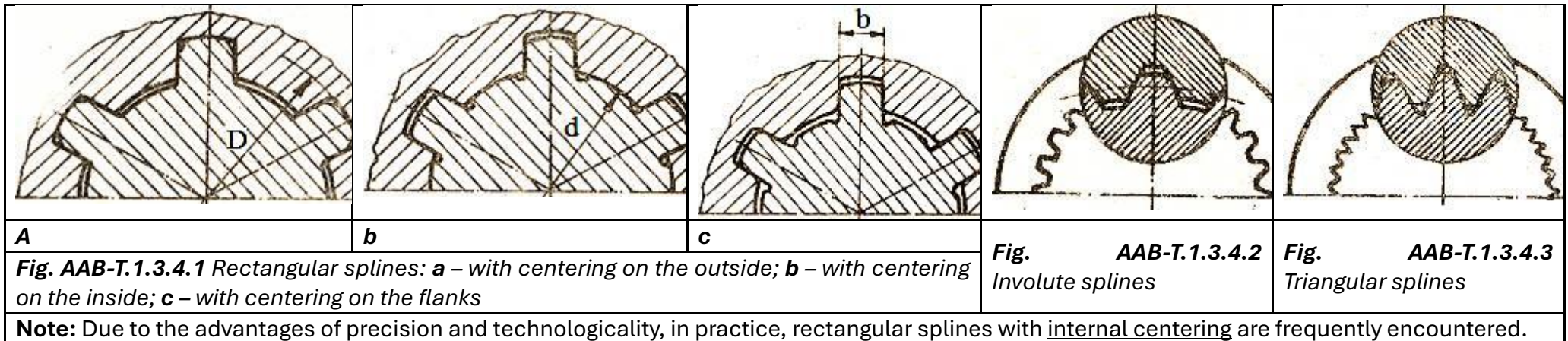


Fig. AAB-T.1.3.4.1 Rectangular splines: **a** – with centering on the outside; **b** – with centering on the inside; **c** – with centering on the flanks

Fig. AAB-T.1.3.4.2
Involute splines

Fig. AAB-T.1.3.4.3
Triangular splines

Note: Due to the advantages of precision and technologicality, in practice, rectangular splines with internal centering are frequently encountered.

AAB-T.1.3.5 MATERIALS AND TECHNOLOGIES

Since in this case there is no intermediate element between the assembled elements (shaft and hub), the materials of the two elements are taken into account for the calculation.

Tab. ACA-T.5.1 Spline machining processes

The splines in the shaft	The splines in the hub
Milling with a disk or worm cutter and grinding (optional)	Mortising or broaching

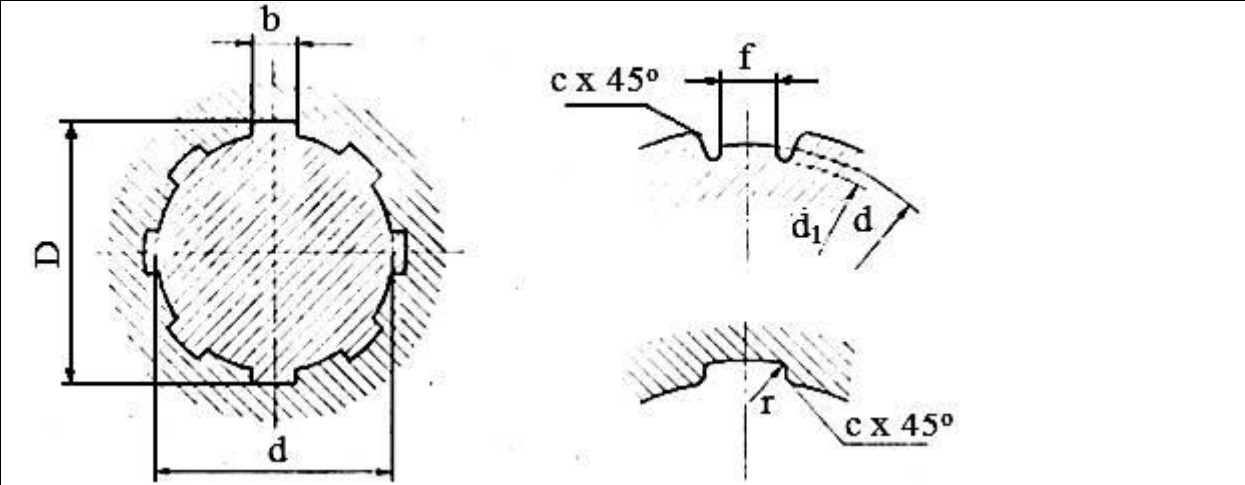
AAB-T.1.3.6 FORMS AND CAUSES OF DISPOSAL OR INCOMPATIBLE BEHAVIOR

Tab. AAB-T.1.3.6.1 *Forms and causes of disuse or improper behavior*

Shapes	Result	Appearance	Cause	Manifestation	Avoidance
Damage to the flanks of the splines in the shaft and hub by crushing (plastic deformation)	Locking assembly and preventing disassembly	On shafts and hubs made of heavily loaded soft materials	Overcoming crushing stresses of materials	Damage to the active surfaces of the flanks in the lateral areas of contact with the shaft and, possibly, the hub	Limiting crushing stresses to permissible values by calculation
Expulsion of the lubricant film	Axial movement hindered	In axially moving assemblies, lubricated	Exceeding the non-expulsion pressure of the lubricant film	Dry friction, wear	Limiting by calculation the non-expulsion pressure of the lubricant to permissible values

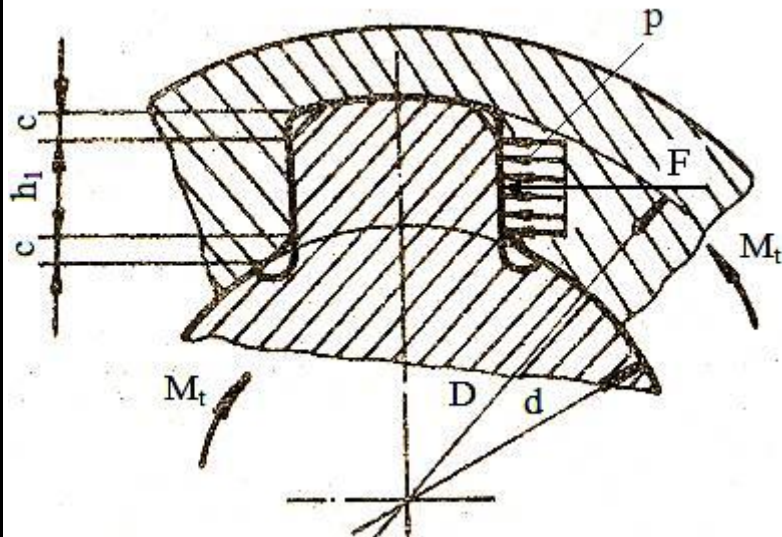
AAB-T.1.3.7 FUNCTIONAL AND CONSTRUCTIVE PARAMETERS

Tab. AAB-T.1.3.7. 1 *Construction parameters of rectangular splines [STAS-1768, STAS-1769, STAS-1770]*

	<i>Standard functional-constructive forms:</i> - light series, - middle series, - heavy series <i>Geometric parameters:</i> D [mm] – outer diameter, d [mm] – inner diameter, b [mm] – spline width, d_1 [mm] – clearance diameter, f [mm] – width of the centering surface, c [mm] – bevel height, r [mm] – connection radius
--	--

AAB-T.1.3.8 CALCULATION MODELS

AAB-T.1.3.8.1 Calculation model of rectangular spline joints

Tab. AAB-T.1.3.8.1.1 Strength calculation model [Jula, 1986; Rădulescu, 1981; Gafițanu, 1999]**Fig. ACA-T.8.1** Strength calculation model

Calculation assumptions and requirements:

- contact pressures are uniformly distributed on the active faces;
- due to inaccuracies in execution and assembly, the load distribution is uneven on the z splines in contact and for the calculation a correction factor is introduced, usually with the value 0.75, considering that approximately 75% of the splines participate in the transmission of the torsional moment;
- the resultant force arm of the contact pressures (with the shaft, respectively with the hub), is approximated by the value of the average radius of arrangement of the z splines and

$$F = \frac{2 M_t}{z D_m} = \frac{4 M_t}{z(D+d)} \quad (\text{AAB-T.1.3.8.1.1})$$

The condition of torsional resistance of the shaft in the spline area

The diameter of the shaft in the spline area, from the condition of resistance to torsional stress,

$$\frac{\beta_k M_t}{\frac{\pi d^3}{16}} \leq \tau_{at} \quad (\text{AAB-T.8.1.3.8.2})$$

must respect the relationship,

$$d \geq \sqrt[3]{\frac{16 \beta_k M_t}{\pi \tau_{at}}} \quad (\text{AAB-T.8.1.3.8.4})$$

where, $\tau_{at} = 15...45$ MPa is the allowable stress at conventional torsion (also takes into account the existence of bending stress) and $\beta_k = 3...5$, the stress concentrator factor (for the calculation at variable, fatigue stresses).

Crush resistance condition of the assembly

$$p = \frac{\frac{4 M_t}{z(D+d)}}{0,75 \left(\frac{D-d}{2} - 2c \right) L} = \frac{4 M_t}{0,75 z(D+d) \left(\frac{D-d}{2} - 2c \right) L} \leq \sigma_{as} (p_a) \quad (\text{AAB-T.8.1.3.8.5})$$

where, M_t [Nmm] is the torsional moment loading the assembly, z – number of splines, d [mm] – inner diameter, D [mm] – outer diameter, c [mm] – chamfer height, L [mm] – spline length; σ_{as} [MPa] – allowable crushing stress (for fixed assembly); p_a [MPa], allowable non-expulsion pressure of the lubricant film (for mobile assembly).

For design calculations, the required length of the splined hub (Fig. AAB-T.1.3.2.1) is obtained from the relationship (APP-T.8.1.3.8.5),

$$L \geq \frac{S'}{s'} = \frac{\frac{4 M_t}{(D+d)} \frac{1}{\sigma_{as}}}{0,75 z \left(\frac{D-d}{2} - 2c \right)} \quad \text{or} \quad L \geq \frac{S'}{s'} = \frac{\frac{4 M_t}{(D+d)} \frac{1}{p_a}}{0,75 z \left(\frac{D-d}{2} - 2c \right)}, \quad \text{AAB-T.8.1.3.8.6)}$$

for fixed and mobile assembly, respectively, where S' represents the required bearing surface and s' - the effective bearing surface corresponding to the unit of contact length between the shaft and the hub

AAB-T.1.3.8.2 Calculation model of involute spline assemblies

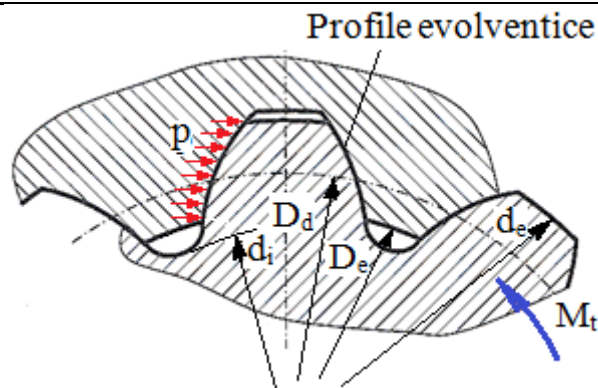


Fig. ACA-T.8.2 Strength calculation model

Calculation assumptions:

- the contact pressures are uniformly distributed on the involute active flanks;
- due to inaccuracies in execution and assembly, the load distribution is uneven on the z splines in contact and for the calculation a correction factor is introduced, usually with the value 0.75, considering that approximately 75% of the splines participate in the transmission of the torsional moment;
- the resultant force arm of the contact pressures p (with the shaft, respectively with the hub), is approximated by the value of the division radius at which the thickness of the solid is equal to that of the hollow) and

$$F = \frac{2 M_t}{z D_d} \quad (\text{AAB-T.1.3.8.2.1})$$

where, $D_d = mz$ is the pitch diameter with z - the number of splines and m - the tooth modulus with standard values (usually, $m = 1, 1.5, 2, 3, 4, 5$ or 8 mm).

The condition of torsional resistance of the shaft in the spline area

The diameter of the shaft in the spline area, from the condition of resistance to torsional stress,

$$\frac{\beta_k M_t}{\frac{\pi d_i^3}{16}} \leq \tau_{at}, \quad (\text{APP-T.8.1.3.8.2.2})$$

must respect the relationship,

$$d_i \geq \sqrt[3]{\frac{16 \beta_k M_t}{\pi \tau_{at}}} \quad (\text{APP-T.8.1.3.8.2.3})$$

where, $\tau_{at} = 15...45$ MPa is the allowable stress at conventional torsion (also taking into account the existence of bending stress) and $\beta_k = 1.5...3$, the stress concentrator factor (for fatigue calculation).

Crush resistance condition of the assembly:

$$p = \frac{\frac{2 M_t}{z D_d}}{0,75 \frac{D_e - d_e}{2} L} = \frac{4 M_t}{0,75 z D_d (D_e - d_e) L} \leq \sigma_{as} (p_a) \quad , (AAB-T.1.3.8.2.4)$$

where, M_t [Nmm] represents the torque, d_e [mm] - the outer diameter of the shaft spline, D_e [mm] - the outer diameter of the hub spline, L [mm] - the length of the spline; σ_{as} [MPa] - the allowable crushing stress (for fixed assembly); p_a [MPa], the allowable non-expulsion pressure of the lubricant film (for mobile assembly).

AAB-T.1.4 PROFILED ASSEMBLIES (ON POLYGONAL CONTOUR)

AAB-T.1.4.1 DEFINITION

The profiled assembly (on a polygonal contour) is a removable assembly, of the shaft-hub type, which transmits the torsional moment through the shape through contacts on three, four or six faces made equiangularly on the shaft and hub (fig. AAB-T.1.1.4.1).

AAB-T.1.4.2 CONSTRUCTION STRUCTURE

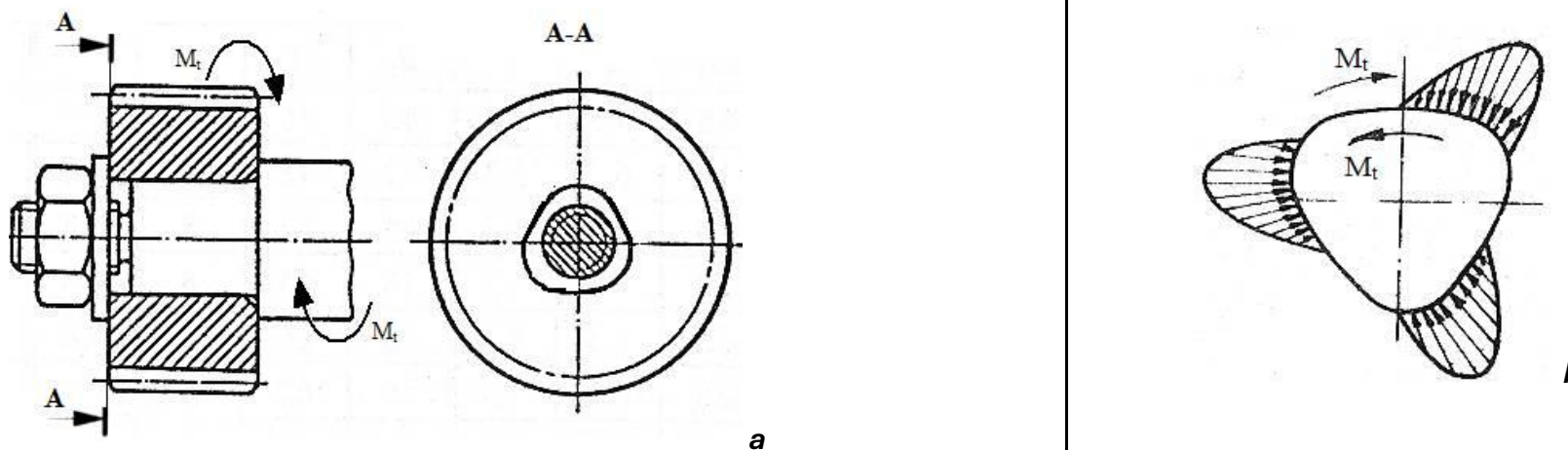


Fig. AAB-T.1.3.2.1 Structural structure of a profiled assembly: **a** – axial section, fixed assembly; **b** – front view and shaft loading

AAB-T.1.4.3 AREAS OF USE, ADVANTAGES AND DISADVANTAGES

Main areas of use: for fixing gears, ratchet wheels, levers on shafts, etc.

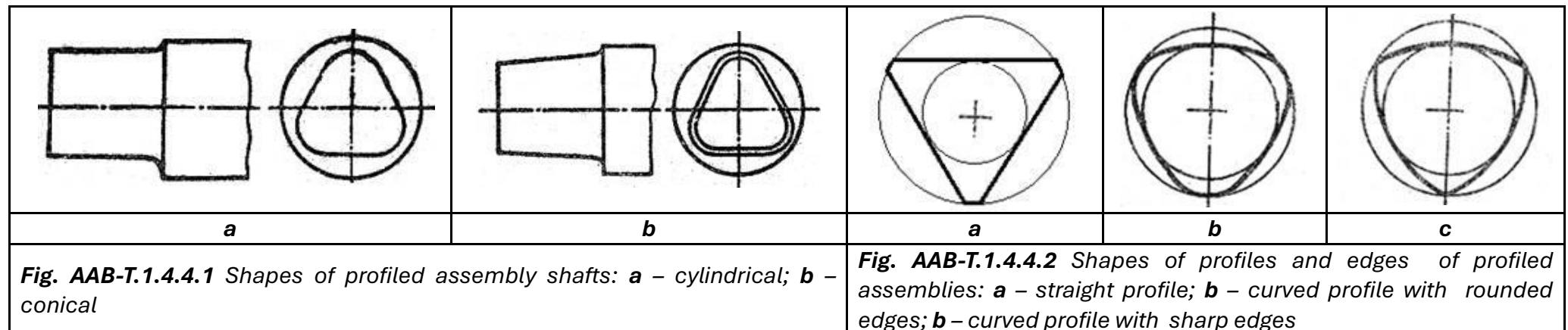
Tab. AAB-T.1.4.3.1 Advantages and disadvantages of profiled assemblies

Advantages *	Disadvantages *
- higher load capacity; - greater fatigue resistance; - better centering and guidance	- high execution precision, and therefore higher costs
* related to parallel keyway or spline assemblies	

AAB-T.1.4.4 CLASSIFICATION

Tab. AAB-T.1.4.4.1 Classification of profiled assemblies [Jula, 1986; Rădulescu, 1981]

Criterion	Assembly type	Scheme
Surface shape	Straight (flat surfaces)	Fig. AAB-T.1.4.4.1,a
	Curved surfaces	Fig. AAB-T.1.4.4.1,b
Number of active surfaces	With triangular profile (three sides)	Fig. AAB-T.1.4.4.2
	With square profile (four sides)	
	With hexagonal profile (six sides)	
The shape of the connecting edges of the active faces of the shaft	With sharp edges	Fig. AAB-T.1.4.4.2,c
	With rounded edges	Fig. AAB-T.1.4.4.2,b
The shape of the profiled shaft section	Cylindric	Fig. AAB-T.1.4.4.1,a
	Conical	Fig. AAB-T.1.4.4.1,b
Possibility of axial displacement of the hub	fixed	Fig. AAB-T.1.4.4.1,b
	Mobile	Fig. AAB-T.1.4.4.2,c



AAB-T.1.4.5 MATERIALS AND TECHNOLOGIES

Since in this case there is no intermediate element between the assembled elements (shaft and hub), the materials of the two elements are considered for the calculation.

Tab. AAB-T.1.4.5.1 Profile processing processes

Shaft profiles	The profiles in the hub
Copy turning; grinding	Drilling, turning, broaching

AAB-T.1.4.6 FORMS AND CAUSES OF DISPOSAL OR INCOMPATIBLE BEHAVIOR

Tab. AAB-T.1.4.6.1 *Forms and causes of disuse or improper behavior*

Shapes	Result	Appearance	Cause	Manifestation	Avoidance
Damage to the flanks of the shaft and hub profiles by crushing (plastic deformation)	Locking assembly and preventing disassembly	On shafts and hubs made of heavily loaded soft materials	Overcoming crushing stresses of materials	Damage to the active surfaces of the flanks in the lateral areas of contact with the shaft and, possibly, the hub	Limiting crushing stresses to permissible values by calculation
Expulsion of the lubricant film	Axial movement hindered	In axially moving assemblies, lubricated	Exceeding the non-expulsion pressure of the lubricant film	Dry friction, wear	Limiting by calculation the non-expulsion pressure of the lubricant to permissible values

AAB-T.1.4.7 FUNCTIONAL AND CONSTRUCTIVE PARAMETERS

Tab. AAB-T.1.4.7.1 Constructive parameters of the profiled assembly [Rădulescu, 1981]

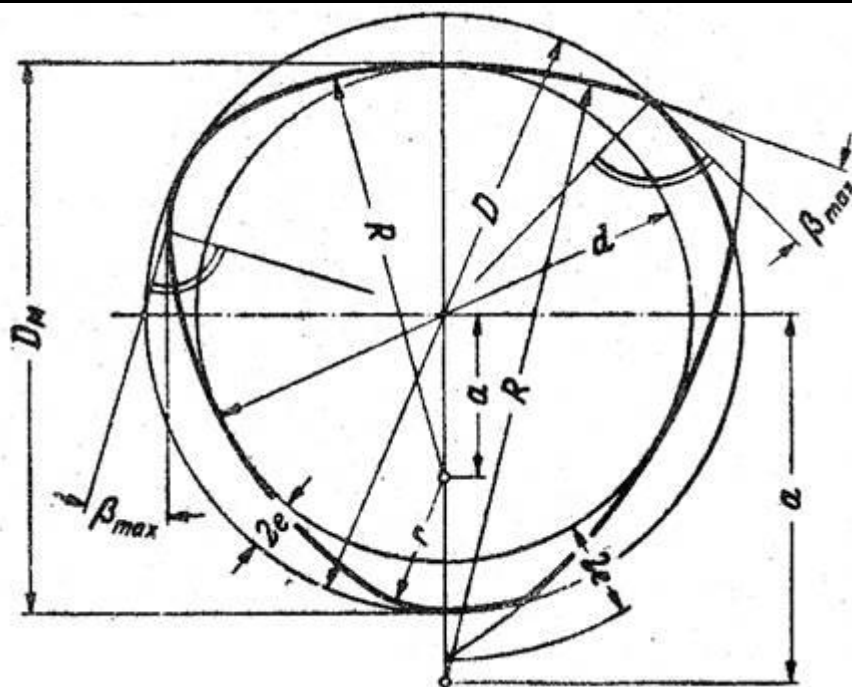
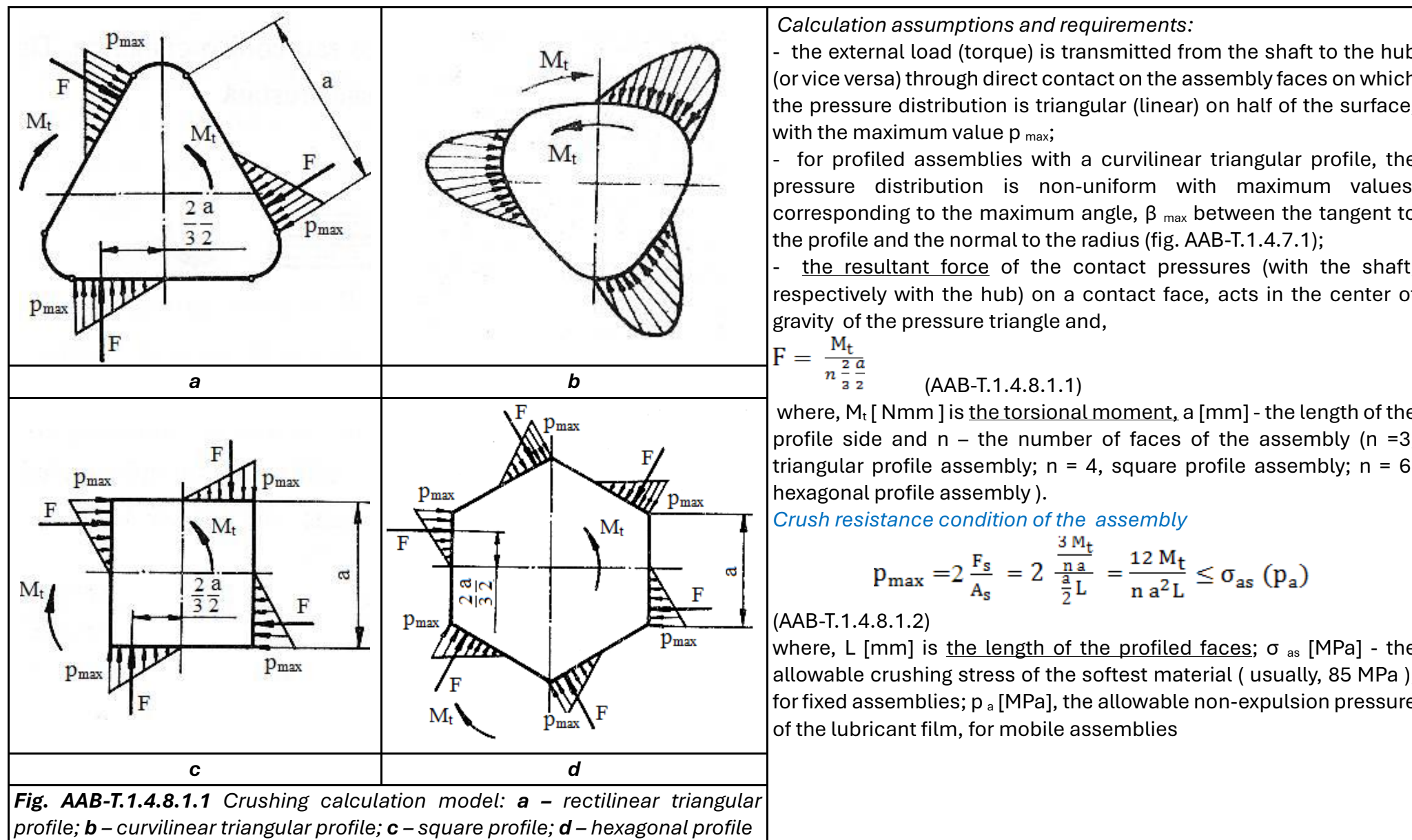


Fig. AAB-T.1.4.7.1 Profile geometry

Profiles closer to a circle (fig. AAB-T.1.4.7.1) are recommended for fixed assemblies, and those closer to a triangular shape can be used for axially mobile assemblies.

Geometric parameters:

AAB-T.1.4.8 CALCULATION MODELS

Tab. AAB-T.1.4.8.1.1 Strength calculation model [Jula, 1986; Rădulescu, 1981; Gafițanu, 1999]

AAB-T.1.5 ASSEMBLIES USING ELASTIC RINGS

AAB-T.1.5.1 DEFINITION

The *elastic ring assembly* is a removable, support - type connection that prevents axial displacement by taking over (transmitting) the axial force through the shape of the elastic ring mounted on the shaft or in the bore (fig. AAB-T.1.5.2.1).

AAB-T.1.5.2 CONSTRUCTION STRUCTURE

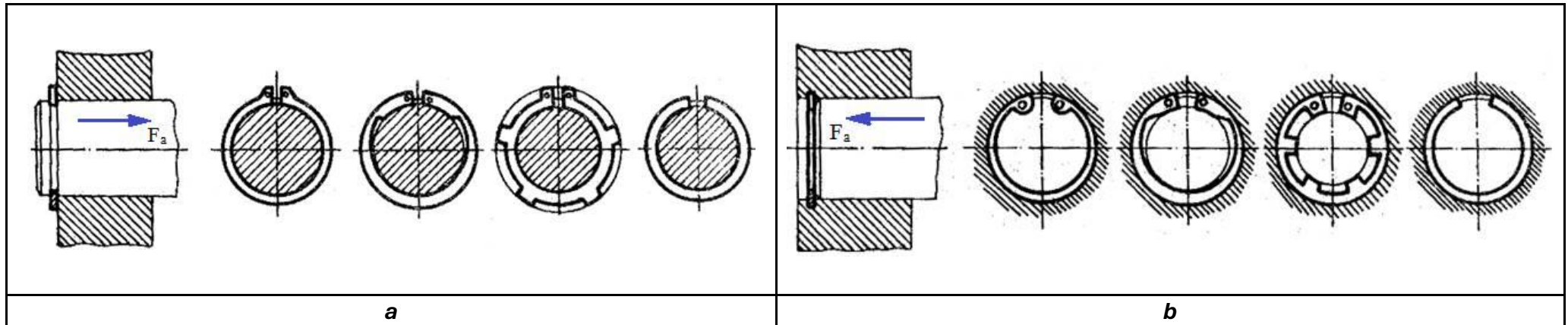


Fig. AAB-T.1.5.2.1 Snap rings: **a** – shaft type; **b** – bore type

AAB-T.1.5.3 AREAS OF USE, ADVANTAGES AND DISADVANTAGES

Main areas of use: for fixing gears, bearings, spacers, etc.

Tab. AAB-T.1.5.3.1 Advantages and disadvantages of elastic ring assemblies

Advantages *	Disadvantages *
- simple from a constructive point of view; - easy assembly and disassembly	- take up low axial forces; - inability to adjust the axial position

AAB-T.1.5.4 CLASSIFICATION

Tab. AAB-T.1.5.4.1 Classification of assemblies by elastic rings [Jula, 1986; Rădulescu, 1981]

Criterion	Assembly type	Scheme
The element on which it is mounted	For the shaft	Fig. AAB-T.1.5.4.1, AAB-T.1.5.4.3, AAB-T.1.5.4.5
	For bore	Fig. AAB-T.1.5.4.2, AAB-T.1.5.4.4, AAB-T.1.5.4.4

Technological process	turned	Fig. AAB-T.1.5.4.1, AAB-T.1.4.4.2
	die	Fig. AAB-T.1.5.4.5
	Plastic deformation	Fig. AAB-T.1.5.4.3, AAB-T.1.5.4.4, AAB-T.1.5.4.6
Mounting method	Axial	Fig. AAB-T.1.5.4.1, AAB-T.1.5.4.2, AAB-T.1.5.4.3, AAB-T.1.5.4.4
	Radial	Fig. AAB-T.1.5.4.5, AAB-T.1.5.4.6

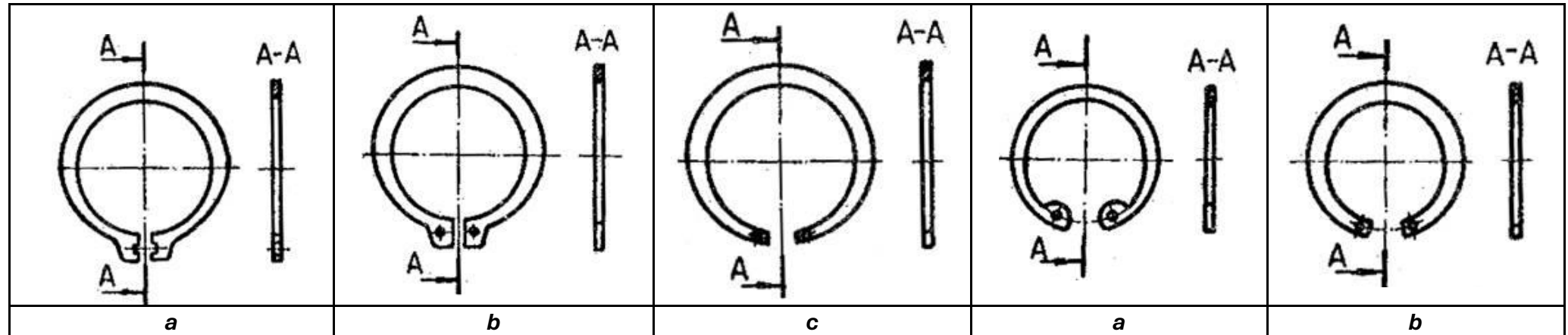


Fig. AAB-T.1.5.4.1 Eccentric snap rings with ears: a – for shafts with $d \leq 10 \text{ mm}$; b – for shafts with $10 < d \leq 165 \text{ mm}$; c – for shafts with $d > 165 \text{ mm}$

Fig. AAB-T.1.4.4.2 Eccentric elastic rings with ears: a – for bores with $D \leq 165 \text{ mm}$; b – for bores with $D > 165 \text{ mm}$

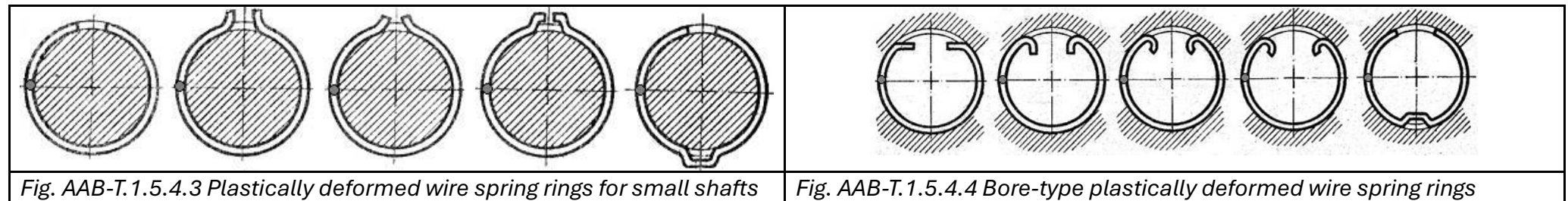


Fig. AAB-T.1.5.4.3 Plastically deformed wire spring rings for small shafts

Fig. AAB-T.1.5.4.4 Bore-type plastically deformed wire spring rings

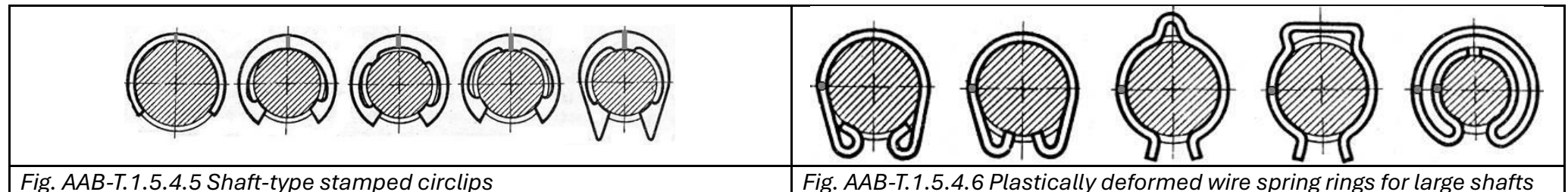


Fig. AAB-T.1.5.4.5 Shaft-type stamped circlips

Fig. AAB-T.1.5.4.6 Plastically deformed wire spring rings for large shafts

AAB-T.1.5.5 MATERIALS AND TECHNOLOGIES

To ensure the elastic deformation capacity required during assembly, the following groups of materials are used:

- spring steel (C55) with a hardness of 40-45 HRC.
- steels alloyed with chromium, silicon and vanadium (e.g. 50VCr11) for rings that must withstand high temperatures.
- beryllium bronzes for rings subjected to corrosive agents.

Tab. AAB-T.1.4.5.1 Processing processes for elastic rings and splines

Procedures for elastic rings	Processes for channels in shaft or bore
Turning, stamping, plastic deformation, grinding	Turning, rarely milling

AAB-T.1.5.6 FORMS AND CAUSES OF DISPOSAL OR INCOMPATIBLE BEHAVIOR

Tab. AAB-T.1.5.6.1 Forms and causes of disuse or improper behavior

Shapes	Result	Appearance	Cause	Manifestation	Avoidance
Damage to the flanks of the splines in the shaft and hub by crushing (plastic deformation)	Locking assembly and preventing disassembly	On shafts and hubs made of heavily loaded soft materials	Overcoming crushing stresses of materials	Damage to the active surfaces of the flanks in the lateral areas of contact with the shaft and, possibly, the hub	Limiting crushing stresses to permissible values by calculation

AAB-T.1.5.7 FUNCTIONAL AND CONSTRUCTIVE PARAMETERS

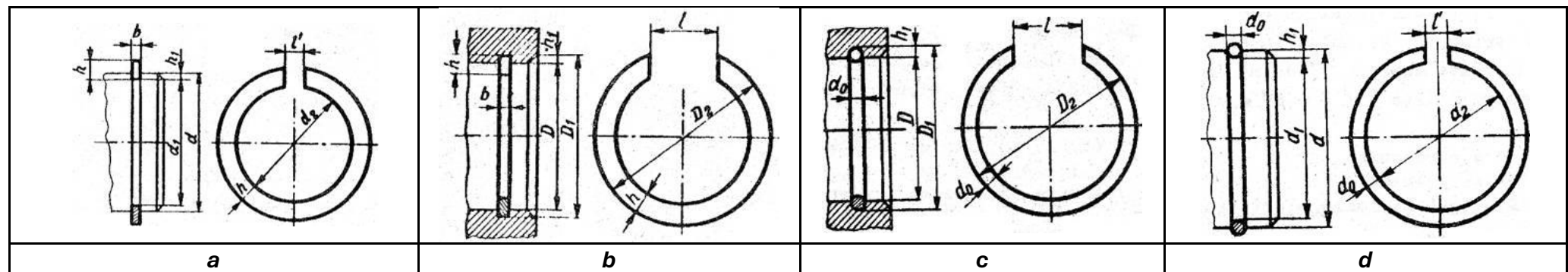


Fig. AAB-T.1.5.7.1 Geometry of elastic rings and channels in the shaft and bore [Rădulescu, 1981]

AAB-T.1.5.8 CALCULATION MODELS

		<i>Calculation assumptions and requirements:</i> - the axial force F_a is transmitted through the shape through the contact between the elastic ring and the lateral face of the channel on which the pressure p appears, which is considered uniformly distributed and a crushing calculation is required; - the bending stress of the ring is neglected. <i>Crush resistance condition of the snap ring assembly mounted in the shaft or hub</i> $p = \frac{F_s}{A_s} = \frac{F_a}{\pi d h_d} \leq \sigma_{as} \quad \text{or,} \quad p = \frac{F_s}{A_s} = \frac{F_a}{\pi D h_D} \leq \sigma_{as}$ (AAB-T.1.5.8.1.1) where, F_a [N] is the axial force; d [mm] – shaft diameter; D [mm] – bore diameter; h_d [mm] – shaft spline height; h_D [mm] – hub spline height; σ_{as} [MPa] – allowable crushing stress of the softest material (usually, 100 MPa). <i>Shear resistance condition of the snap ring mounted in the shaft or hub</i> $\tau_f = \frac{F_f}{A_f} = \frac{F_a}{\pi d b} \leq \tau_{af} \quad \text{or, respectively,} \quad \tau_f = \frac{F_f}{A_f} = \frac{F_a}{\pi D b} \leq \tau_{af}$ (AAB-T.1.5.8.1.2) where, F_a [N] is the axial force; b [mm] is the ring thickness; τ_{af} [MPa] – allowable shear stress of the ring material (usually, 80...100 MPa)
a	b	

Fig. AAB-T.1.5.8.1.1 Strength calculation model: a – ring mounted in the shaft; b – ring mounted in the hub

AAB-T.1.5.9 INSTALLATION AND OPERATION

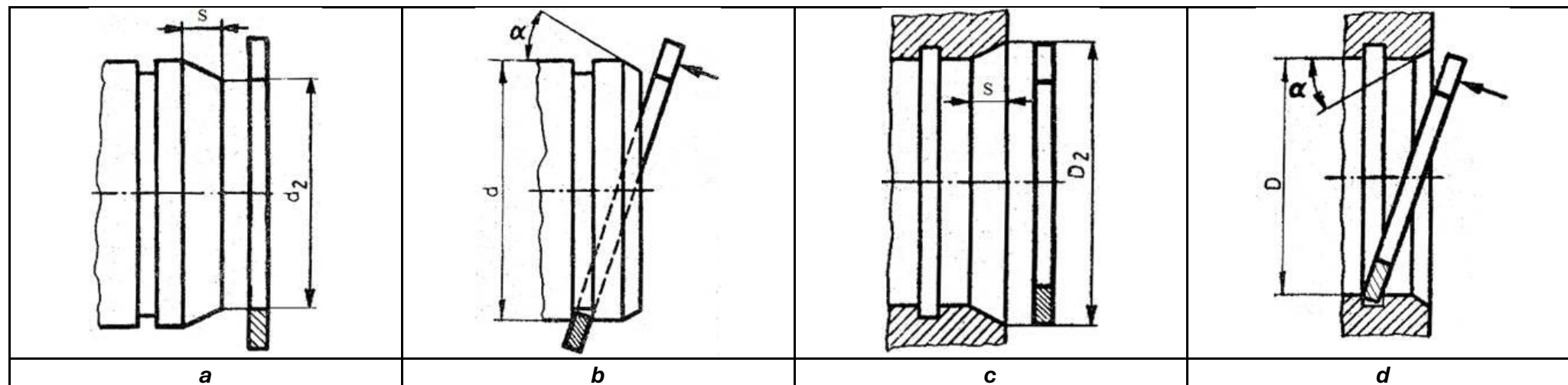


Fig. AAB-T.1.5.9.1 Axial mounting of shaft-type snap rings: a – radial extension and axial displacement; b – radial extension and tilting

Fig. AAB-T.1.5.9.2 Axial mounting of bore-type elastic rings: a – radial compression and axial displacement; b – radial compression and tilting

Note: To mount the rings in the shaft or hub spline, it is recommended to chamfer the surfaces, with an angle $\alpha = 30^\circ$ on the portions $s = 0.1d$ (D)

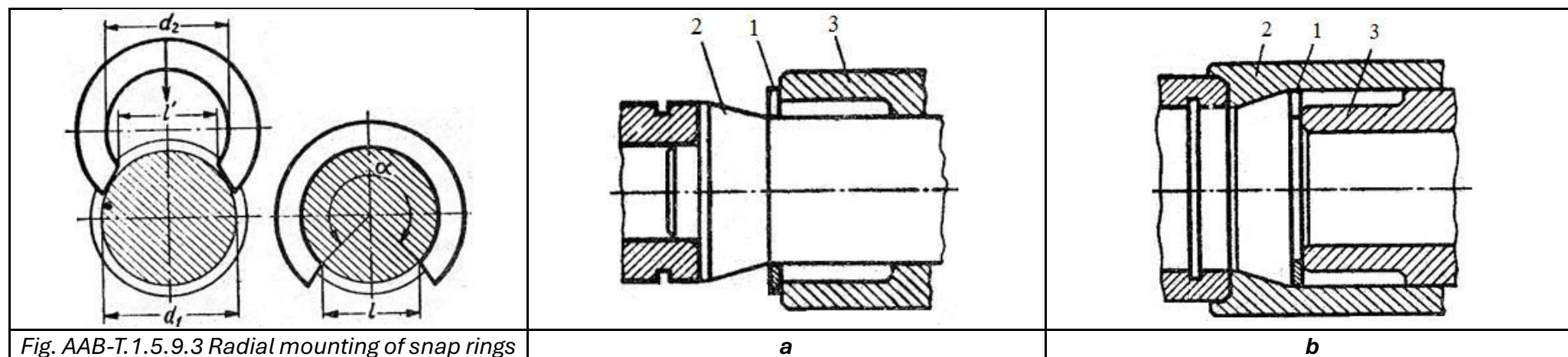


Fig. AAB-T.1.5.9.3 Radial mounting of snap rings

Fig. AAB-T.1.5.9.4 Guide devices for axial mounting of snap rings: a – shaft type; b – bore type

Note: When using axial mounting devices (fig. AAB-T.1.5.9.4), the ring 1 is expanded (compressed) by the conical rod (bush) 2, as a result of the axial thrust by the punch 3.

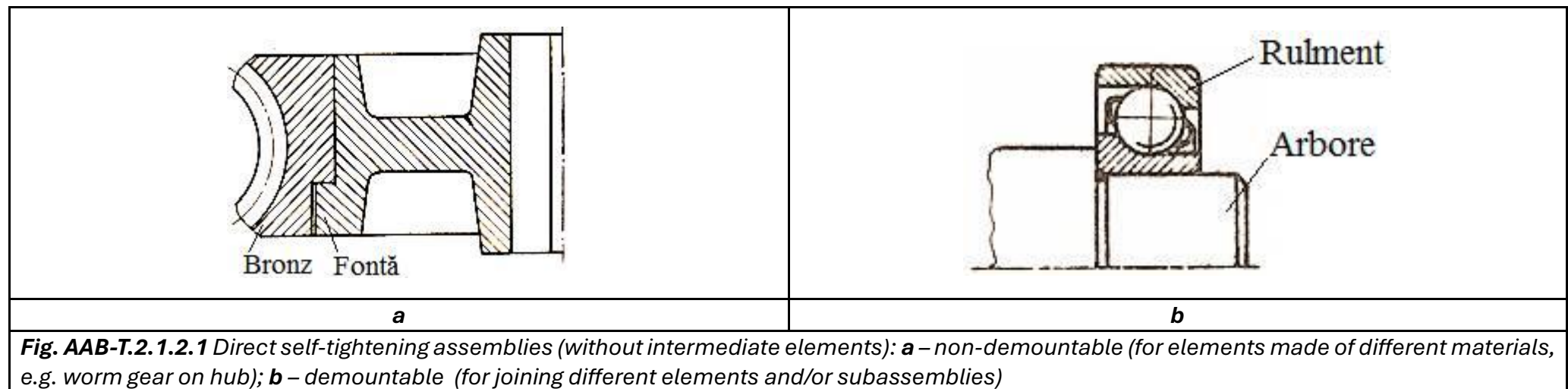
AAB-T.2 FRICTION JOINTS

AAB-T.2.1 SELF-CLEANING (PRESSED) ASSEMBLIES

AAB-T.2.1.1 DEFINITION

Self-tightening (pressed) assembly is a shaft-hub type connection, non-detachable or, rarely, detachable, which transmits loads through friction that arises on the contact surface between the shaft and the hub under the action of contact pressures that appear after assembly due to mutual tightening generated by elastic deformations (fig. AAB-T.2.1.1.1).

AAB-T.2.1.2 CONSTRUCTION STRUCTURE



AAB-T.2.1.3 AREAS OF USE, ADVANTAGES AND DISADVANTAGES

Main areas of use: mounting flywheels, bearing bushings in housings; tires mounted on railway vehicle wheels; crankshafts assembled by pressing; joining elements of chain links; assembling gear rings on the wheel body; assembling bearings on shaft spindles.

Tab. ACA-T.3.1 Advantages and disadvantages of self-tightening assemblies

Advantages *	Disadvantages *
- capacity to transmit large loads; - good performance under variable loads; - good centering of the assembled parts;	- high execution precision of the active elements, imposed by the operating and assembly conditions; - damage to functional surfaces, during repeated disassembly and assembly;

- | | |
|--|--|
| - make it possible to make parts from several parts in order to save expensive and/or scarce materials or to facilitate manufacturing technology | - the need to sort the parts to rigorously ensure the required tightenings |
|--|--|

AAB-T.2.1.4 CLASSIFICATION

Tab. AAB-T.2.1.4.1 Classification of assemblies by self- tightening [Horovitz, 1969; Jula, 1986; Rădulescu, 1981]

Criterion	Assembly type
The constructive solution	Without clamping elements
	With clamping elements
Possibility of dismantling	demountable
	of permanent
Assembly technology	Pressed (forced axial insertion of the shaft into the hub, usually at ambient temperature)
	Fretted (heating the hub and/or cooling the shaft to obtain higher tightenings than in the case of cold pressed ones)
Note: The removable or non-removable nature is determined by the size of the tightening and the assembly technology used.	

AAB-T.2.1.5 MATERIALS AND TECHNOLOGIES

In the case of self-tightening assemblies on cylindrical, pressed or crimped surfaces, there is no intermediate element between the assembled elements (shaft and hub), the materials of the two elements are taken into account for the calculation.

The cold or hot pressing operation is performed with manual presses (small tightenings) or hydraulic presses (large tightenings); to reduce the axial pressing force, the contact surfaces can be lubricated (the load transmission capacity decreases).

The shrink fit operation involves heating the hub (over an open flame, in an oven or in an oil bath), usually below 600 ° C, and/or cooling the shaft (in liquid oxygen or carbon dioxide) in order to reduce or even eliminate the axial mounting force.

AAB-T.2.1.6 FORMS AND CAUSES OF DISPOSAL OR INCOMPATIBLE BEHAVIOR

Tab. AAB-T.2.1.6.1 Forms and causes of disuse or improper behavior

Shapes	Result	Appearance	Cause	Manifestation	Avoidance
Damage to the contact surface of the shaft or hub (plastic deformation)	Locking assembly and preventing disassembly	On shafts and hubs made of heavily loaded soft materials	Overcoming crushing stresses of materials	Damage to the active contact surfaces of the shaft and/or hub	Limiting crushing stresses to permissible values by calculation

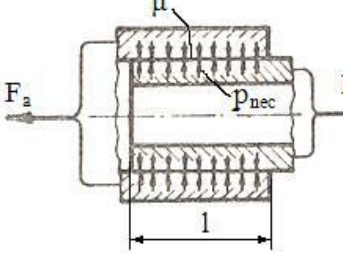
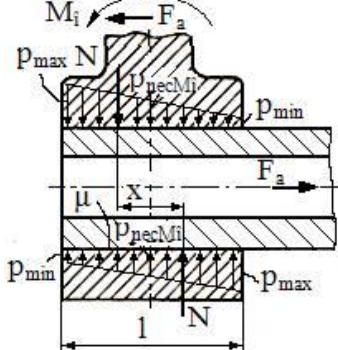
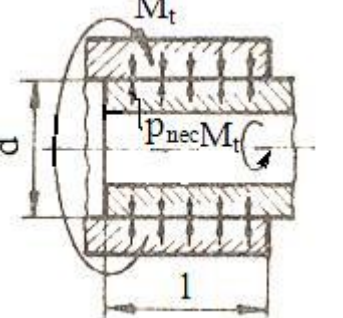
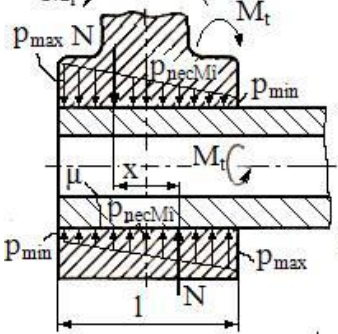
Increased radial elastic deformations	Reducing clearances in adjacent elements (bearings, sliding bearings)	When assembling bearing elements	Exceeding permissible radial deformations	Shaft compression and/or radial hub extension	Limitation of radial deformations by calculation
---------------------------------------	---	----------------------------------	---	---	--

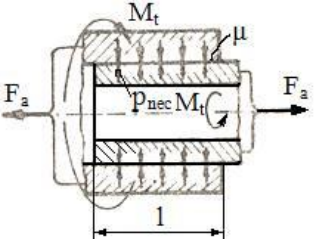
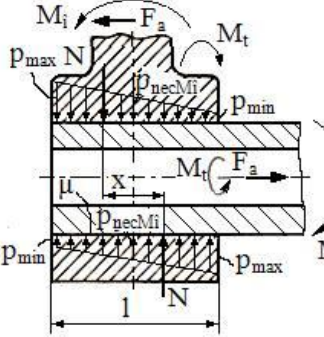
AAB-T.2.1.7 FUNCTIONAL AND CONSTRUCTIVE PARAMETERS

AAB-T.2.1.8 CALCULATION MODELS

AAB-T.2.1.8.1 Load calculation models

Tab. AAB-T.2.1.8.1.1 Loading situations and associated conditions of load transmission by friction

Cas e	Type of tasks	Loading scheme	Condition of load transmission by friction	Cas e	Type of tasks	Loading scheme (see note 1)	Conditions for load transmission through friction and shape (see note 1)
1	Axial force		$F_a \leq \mu \pi d l p_{nec}$ or $p_{nec} \geq \frac{F_a}{\mu \pi d l}$	4	Bending moment and axial force		$F_a \leq \mu \pi d l p_{nec} F_a,$ $M_i \leq d l^2 \frac{p_{nec} M_i}{3}$ or $p_{nec} = \max(p_{nec}, p_{nec} M_i) = \frac{F_a}{\max(\mu \pi d l, \frac{3 M_i}{d l^2})}$
2	Torque		$M_t \leq \mu \pi d l p_{nec} \frac{d}{2}$ or $p_{nec} \geq \frac{2 M_t}{\mu \pi d^2 l}$	5	Bending moment and torsional moment		$M_t \leq \mu \pi d l p_{nec} \frac{d}{2},$ $M_i \leq d l^2 \frac{p_{nec} M_i}{3}$ or $p_{nec} = \max(p_{nec}, p_{nec} M_i) = \frac{2 M_t}{\max(\mu \pi d l, \frac{3 M_i}{d l^2})}$

3	Axial force and torque		$\sqrt{\left(\frac{2 M_t}{d}\right)^2 + F_a^2} \leq \mu \pi d l p_{nec}$ or $p_{nec} \geq \frac{\sqrt{\left(\frac{2 M_t}{d}\right)^2 + F_a^2}}{\mu \pi d l}$	6	Bending moment, axial force and torsional moment	 $\sqrt{\left(\frac{2 M_t}{d}\right)^2 + F_a^2} \leq \mu \pi d l p_{nec}$, $M_i \leq d l^2 \frac{p_{nec} M_i}{3}$ or $p_{nec} = \max(p_{nec}, p_{nec M_i})$ max($\frac{\sqrt{\left(\frac{2 M_t}{d}\right)^2 + F_a^2}}{\mu \pi d l}, \frac{3 M_i}{d l^2}$)
Note.	a. In cases 1, 2 and 3, due to axial loads, the pressure distribution on the contact surface is uniform; the transmission of external loads is entirely through friction. b. In cases 3,4,5, due to the loading with the bending moment, the pressure distribution on the contact surface is non-uniform, with the average pressure $p_{nec M_i} = (p_{min} + p_{max})/2$. For the calculation, the limiting case $p_{min} = 0$ ($p_{nec M_i} = p_{max}/2$), when the action of the bending moment in the diametrical plane is equivalent to a couple by the relationship, $M_i = N x = d l p_{nec M_i} x = d l p_{nec M_i} l/3 = d l p_{max} l/3 = d l^2 p_{max} l/6$, in which $x=l/3$ was considered and, considering the pressure distribution in the frontal plane constant, the force $N = d l p_{nec M_i}$. Since, in these cases, the transmission of the torsional moment is done by shape and the axial loads by friction, the conditions for transmitting the axial loads are independent and, consequently, for the calculation, it is necessary to adopt the maximum values corresponding to the two situations (cases 1, 2 and 3 are particular situations of cases 4, 5 and 6 respectively for which it is considered, $M_i = 0$).					

AAB-T.2.1.8.2 Calculation model of the required elastic radial stresses (deformations)

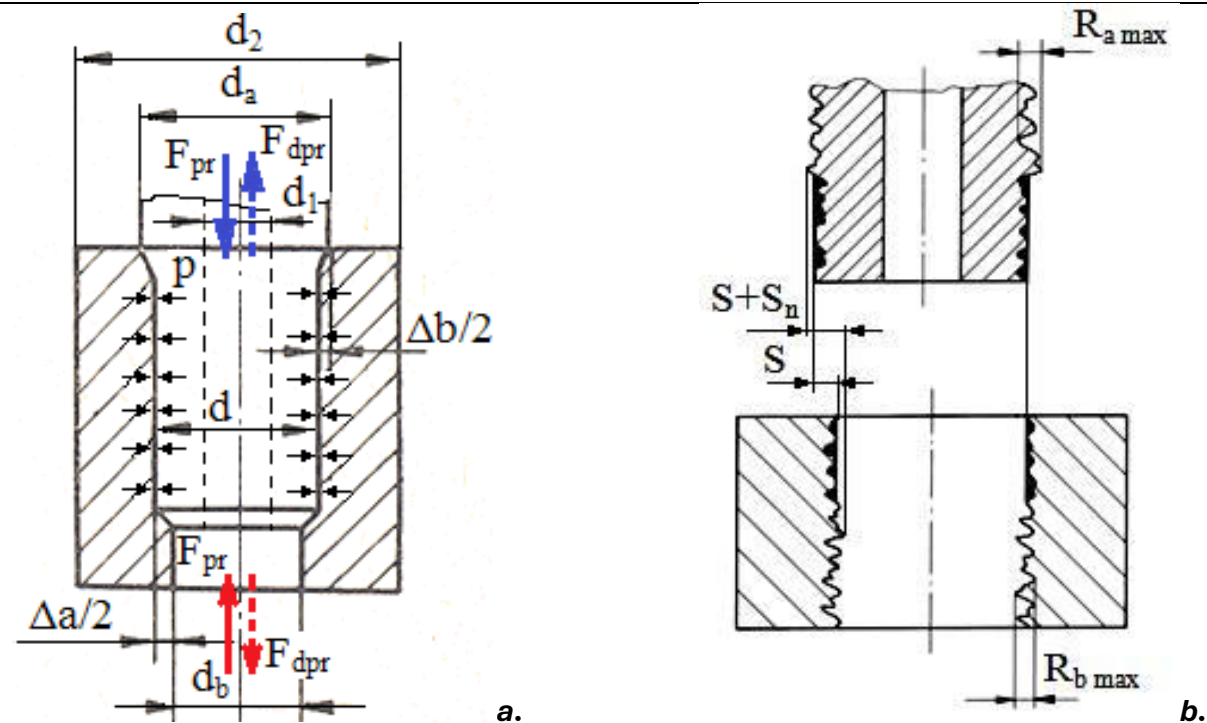


Fig. AAB-T.2.1.8.1.2.1 Deformation calculation schemes: **a** – geometric parameters; **b** - influence of surface irregularities on tightening [Roloff, 2008]

Calculation assumptions:

- the shaft and hub materials deform in the elastic range.
- the contact pressure on the common surface is considered constant, uniformly distributed.
- external loads are transmitted through frictional forces that appear on the contact surfaces due to tightening.

Calculation of tightening

Theoretical tightening required (according to the theory of elasticity of thick tubes, Lame's relation):

$$S = d_a - d_b = \Delta a + \Delta b = p \left(\frac{k_a}{E_a} + \frac{k_b}{E_b} \right) d \cdot 10^3 \text{ } [\mu\text{m}], \quad (\text{AAB-T.2.1.8.2.1})$$

where: p [MPa] represents the pressure between the surfaces in contact, $p = p_{\text{nec}}$; d [mm] – diameter of the contact surface; d_a and d_b initial diameters of the shaft and, respectively, the hub; Δa and Δb radial deformation of the shaft (expansion) and, respectively, the hub (contractions); E_a [MPa] and E_b [MPa] - modulus of elasticity of the materials of the shaft and, respectively, the hub; of the shaft and, respectively, the hub; k_a and k_b - coefficients defined by the relations:

$$k_a = \frac{d^2 + d_1^2}{d^2 - d_1^2} - \nu_a, \quad k_b = \frac{d_2^2 + d^2}{d_2^2 - d^2} + \nu_b$$

where, v_a and v_b represent the transverse contraction coefficients (Poisson)

The corrected tightening required takes into account the actual execution and assembly conditions, according to the relationship,

$$S_c = S + S_n + S_t$$

where: S_n represents the correction due to the machining microirregularities of the surfaces that come into contact according to the expression,

$$S_n = 1.2 (R_{a \max} + R_{b \max}) [\mu m],$$

with $R_{a,b \max}$ the maximum heights of the irregularities, initially existing on the surface of the shaft, respectively the hub (see fer. ASP-P.1);

S_t – correction of the effect of thermal expansion of the assembled parts in operation, according to the relationship

$S_t = [\alpha_b (t_b - t_0) - \alpha_a (t_a - t_0)] \cdot d \cdot 10^{-3} [\mu m]$, where $\alpha_{a,b}$ are the thermal expansion coefficients of the shaft or hub material; t_0 – ambient temperature at which the assembly was performed; $t_{a,b}$ – operating temperatures of the shaft or hub, respectively, in °C.

The axial forces required for pressing or depressurizing are determined with the relationships,

$$F_{pr} = \mu_1 \pi d l p \text{ or, respectively, } F_{dpr} = \mu_2 \pi d l p,$$

where, μ_1 represents the coefficient of friction when pressing; μ_2 – the coefficient of friction when releasing ($\mu_2 > \mu_1$); to reduce the pressing force, the contact surfaces can be lubricated with oil.

Note.

For design calculations, $p = p_{nec}$ can be considered, corresponding to the loading scheme (subchapter AAB - T.2.1.8.1)

AAB-T.2.1.8.3 Strength calculation model for self-tightening pressed or crimped assemblies [Jula, 1986; Rădulescu, 1981; Catrina, 2009]

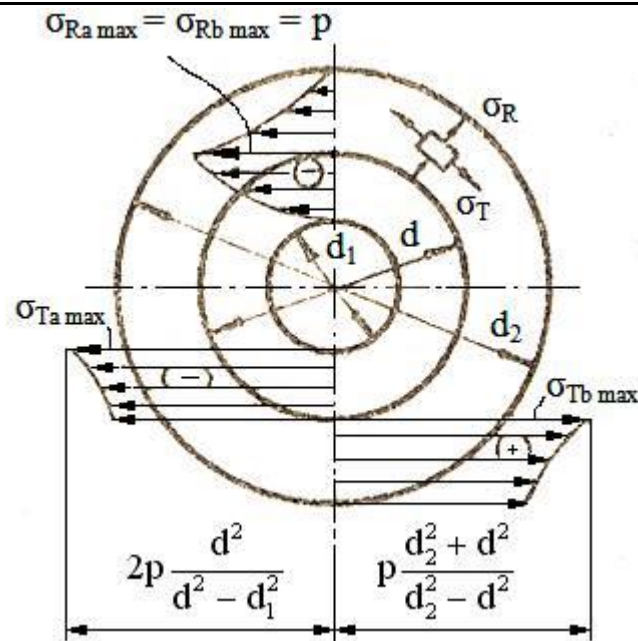


Fig. AAB-T.2.1.8.1.3.1 Stress calculation scheme

Simplifying assumptions:

- the stresses in the assembled parts determined by the contact pressures are calculated based on the relationships established in the study of thick-walled tubes in the elastic domain;
- radial stresses, σ_R , appear in the shaft and hub, with zero values on the inside of the shaft and the outside of the hub, and tangential stresses, σ_T , of traction in the hub and compression in the shaft;
- in order for the assembled parts not to deform plastically, according to the theory of tangential stresses, the following condition must be met:

$$\tau_{max} = \frac{\sigma_T - \sigma_R}{2} < \tau_{02} \cong \frac{\sigma_{02}}{2}, \quad (\text{AAB-T.2.1.8.3.1})$$

The conditions for avoiding plastic deformations (ensuring elastic tightening) for the shaft and hub are determined with the relations,

$$\tau_{b \max} = \frac{\sigma_{Tb \max} - \sigma_{Rb \max}}{2} = \frac{p \frac{d_2^2 + d^2}{d_2^2 - d^2} - (-p)}{2} = p \frac{d_2^2}{d_2^2 - d^2} \leq \tau_{02b} \approx \frac{\sigma_{02b}}{2}, \quad (\text{AAB-T.2.1.8.3.2})$$

and respectively,

$$\tau_{a \max} = \frac{\sigma_{Ta \max} - \sigma_{Ra \min}}{2} = \frac{-2p \frac{d^2}{d^2 - d_1^2} - 0}{2} = -p \frac{d^2}{d^2 - d_1^2} \leq \tau_{02a} \approx \frac{\sigma_{02a}}{2}, \quad (\text{AAB-T.2.1.8.3.3})$$

where, $\sigma_{Ta,b}$ and $\sigma_{Ra,b}$ are the tangential and, respectively, radial stresses in the shaft and, respectively, the hub; $\tau_{02a,b}$ and $\sigma_{02a,b}$ – the yield strengths of the shaft and, respectively, the hub materials (in the case of brittle materials, e.g. cast iron, the breaking strengths τ_r and σ_r are taken into account).

The maximum values (critical, allowed at the limit) that the contact pressures can reach, at the yield limits of the shaft and hub materials (σ_{02a} and, respectively, σ_{02b}), can be established from the relationships (AAB-T.2.1.8.3.2 and AAB-T.2.1.8.3.3):

$$p_{ca} = \frac{1}{2} \sigma_{02a} \frac{d^2 - d_1^2}{d^2} \text{ and, respectively, } p_{cb} = \frac{1}{2} \sigma_{02b} \frac{d_2^2 - d^2}{d_2^2} \quad (\text{AAB-T.2.1.8.3.4})$$

AAB-T.2.1.8.4 Deformation calculation model for self-tightening pressed or crimped assemblies [Jula, 1986; Rădulescu, 1981; Catrina, 2009]

Simplifying assumptions:

- the shaft and hub materials deform in the elastic range;
- the contact pressure on the common surface is considered constant;
- the diametrical elastic deformations $\Delta d_{1,2}$ of the parts (shaft and hub) assembled at the level of diameters $d_{1,2}$ are calculated from Hooke's relation,

$$\sigma_{R1} = E_a \varepsilon_1 = E_a \frac{\Delta d_1}{d_1} \text{ and } \sigma_{R2} = E_a \varepsilon_1 = E_a \frac{\Delta d_1}{d_1} \quad (\text{AAB-T.2.1.8.4.1})$$

Conditions for avoiding limit radial displacements

The diametral elastic displacements, according to Hooke's relation from the theory of elasticity, of decreasing the shaft diameter d_1 and increasing the hub diameter d_2 are:

$$\Delta d_1 = \sigma_{R1} \frac{d_1}{E_a} = 2p \frac{d^2}{d^2 - d_1^2} \frac{d_1}{E_a} 10^3 [\mu\text{m}] \text{ and, respectively, } \Delta d_2 = \sigma_{R2} \frac{d_2}{E_b} = 2p \frac{d^2}{d_2^2 - d^2} \frac{d_2}{E_b} 10^3 [\mu\text{m}] \quad (\text{AAB-T.2.1.8.4.2})$$

AAB-T.2.1.8.5 Thermal calculation model of cylindrical assemblies with self-tightening by welding [Jula, 1986; Rădulescu, 1981; Catrina, 2009]

Note: In the case of a fretted assembly, the thermal expansion equations in the diametrical direction of the shaft or hub,

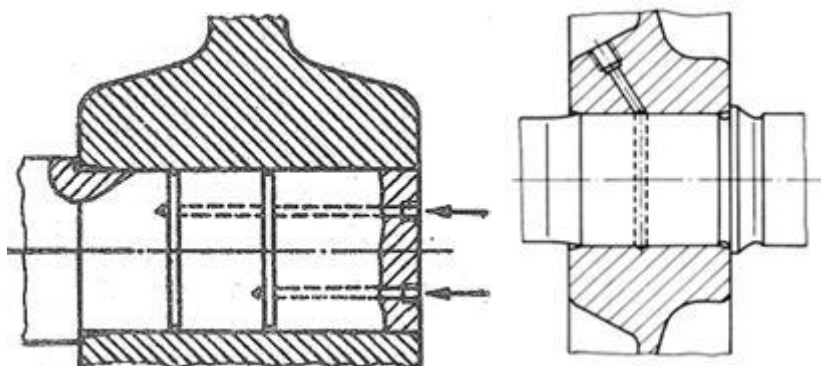
$$S_{\max} + j_m = \alpha_a (t_0 - t_a) d \cdot 10^3 \text{ and respectively, } S_{\max} + j_m = \alpha_b (t_b - t_0) d \cdot 10^3 \quad (\text{AAB-T.2.1.8.1.5.1})$$

the shaft cooling or hub heating temperatures are obtained,

$$t_a = t_0 - \frac{S_{\max} + j_m}{\alpha_a d} 10^{-3} [\text{°C}] \text{ and/or respectively, } t_b = t_0 + \frac{S_{\max} + j_m}{\alpha_b d} 10^{-3} [\text{°C}] \quad (\text{AAB-T.2.1.8.5.2})$$

where, t_0 [° C] is the temperature of the mounting environment; $S_{\max}$ [μm] – the maximum tightening to be achieved according to the adopted adjustment; $\alpha_{a,b}$ [$1/^\circ\text{C}$] – the unidirectional thermal expansion coefficients of the shaft and hub materials, respectively; j_m [μm] – the additional mounting clearance required for ease of mounting.

AAB-T.2.1.9 INSTALLATION AND OPERATION

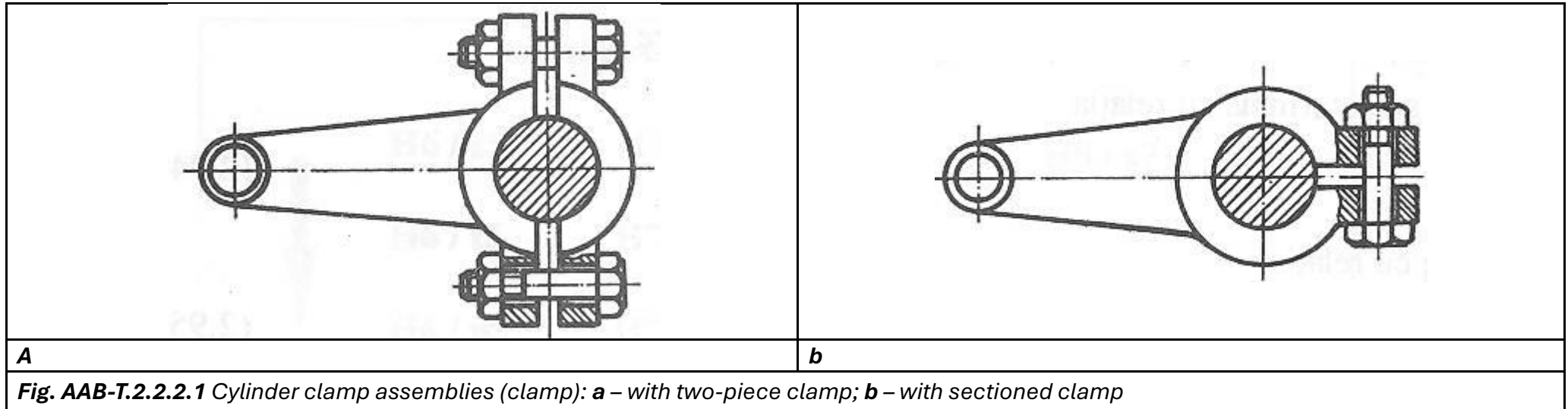


AAB-T.2.2 CYLINDER TIGHTENING ASSEMBLIES (BRACELET)

AAB-T.2.2.1 DEFINITION

The cylinder (clamp) clamping assembly is a shaft-hub type, removable assembly that transmits loads through the friction that occurs on the cylindrical contact surface between the shaft and a clamp pressed by means of screw assemblies (fig. AAB-T.2.2.1.1).

AAB-T.2.2.2 CONSTRUCTION STRUCTURE



AAB-T.2.2.3 AREAS OF USE, ADVANTAGES AND DISADVANTAGES

Main areas of use: machine construction for fixing various cranks, arms, supports on bars, rods, columns, axles or shafts; fixing actuating or control levers

Tab. AAB-T.2.2.3.1 Advantages and disadvantages of cylinder tightening assemblies

Advantages *	Disadvantages *
-different axial and angular positionings can be made;	- relatively low loads; - it is necessary to adjust the tightness; - unsuitable for frequent assembly and disassembly

AAB-T.2.2.4 CLASSIFICATION

Tab. AAB-T.2.2.4.1 Classification of assemblies by tightening on the cylinder (clamp)

Criterion	Assembly type	Scheme
Bracelet shape	With two-piece bracelet	Fig. AAB-T.2.2.2.1,a
	With sectioned bracelet	Fig. AAB-T.2.2.2.1,b
Threaded assembly	With nut	
	With thread in the bracelet body	

AAB-T.2.2.5 MATERIALS AND TECHNOLOGIES

The bracelets are made of rolled steel or sheet metal, and it is recommended that the roughness of the contact surfaces be as high as possible.

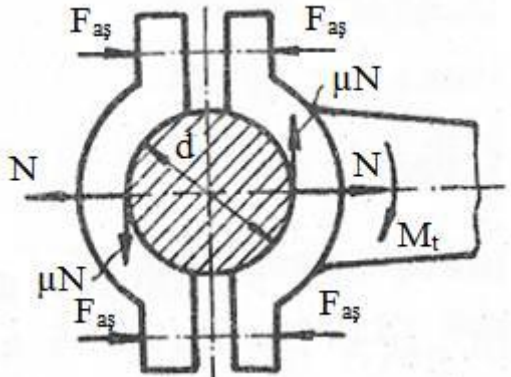
AAB-T.2.2.6 FORMS AND CAUSES OF DISPOSAL OR INCOMPATIBLE BEHAVIOR

Tab. AAB-T.2.1.6.1 Forms and causes of disuse or improper behavior

shapes	result	Appearance	CAUSE	Manifestation	Avoidance
Damage to the contact surface (plastic deformation)	Interruption of the transmission of the load	For shafts and bracelets made of heavily loaded soft materials	Overcoming crushing stresses of materials	Damage to the active contact surfaces of the shaft and/or the collar	Limiting crushing stresses to permissible values by calculation
Damage to threaded assembly	Reducing clearances in adjacent elements (bearings, sliding bearings)	When assembling bearing elements	Exceeding permissible radial deformations	Shaft compression and/or radial hub extension	Limitation of radial deformations by calculation

AAB-T.2.2.7 FUNCTIONAL AND CONSTRUCTIVE PARAMETERS

AAB-T.2.2.8 CALCULATION MODELS

	<i>Calculation assumptions and requirements:</i> - the external load is transmitted entirely through friction; - the pressures on the contact surfaces are uniformly distributed; - the clamp tightening screws are loaded identically and are subjected to tensile stress. <i>Shaft torsional resistance condition in the assembly area</i> The minimum diameter of the shaft in the assembly area, from the condition of resistance to torsional stress, $\frac{\beta_k M_t}{\frac{\pi d^3}{16}} \leq \tau_{at}, \quad (\text{APP -T.2.1.8.1.1})$
--	--

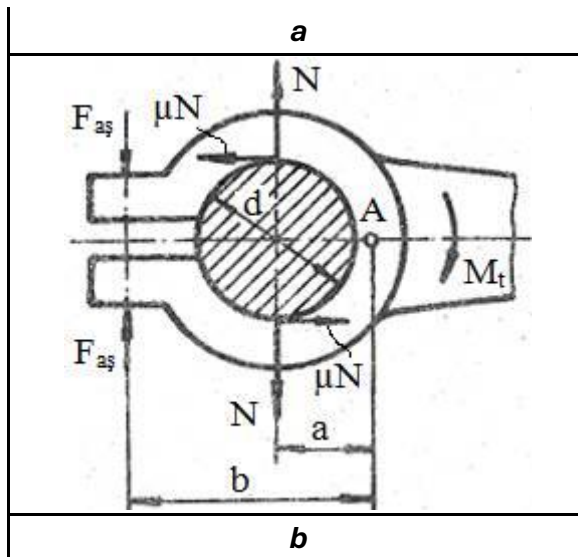


Fig. AAB-T.2.1.8.1 Calculation scheme of the cylinder clamping assembly: **a** – with two-piece clamp: **b** – with sectioned clamp

must respect the relationship,

$$d \geq \sqrt{\frac{z \cdot 16 \beta_k M_t}{\pi \tau_{at}}} \quad (\text{APP-T.2.1.8.1.2})$$

where, $\tau_{at} = 15...45$ MPa is the allowable stress at conventional torsion (also takes into account the existence of bending stress, maximum values are taken when bending is reduced or even zero (at the shaft end); the material is steel) and $\beta_k = 1...2$, the stress concentrator factor (for fatigue calculation).

The condition for load transmission by friction is determined from the moment equilibrium equation,

$$\mu N d \geq \beta_a M_t, \quad (\text{AAB-T.2.1.8.1.3})$$

where, M_t [Nmm] is the transmitted torque, d [mm] – shaft diameter, μ - friction coefficient (for steel parts $\mu = 0.15...0.2$), N [N] - normal pressing force of the clamps; $\beta_a = 1.2...1.8$ – safety factor against sliding (skating).

From the equilibrium equation of the forces acting on a bracelet (variant in Fig. AAB-T.2.1.8.,a) or a half-bracelet articulated at point A (variant in Fig. AAB-T.2.1.8.1,b), we obtain,

$$N = z F_{as} \quad \text{or, respectively,} \quad N a = z F_{as} b, \quad (\text{AAB-T.2.1.8.1.4})$$

where, z is the number of bolts tightening the brackets; a and b - lengths adopted constructively.

The condition of crushing resistance of the assembly, taking into account the relationship (AAB-T.2.1.8.1.3) is,

$$p = \frac{F_s}{A_s} = \frac{N}{d l} = \frac{\frac{\beta_a M_t}{\mu d}}{d l} = \frac{\beta_a M_t}{\mu d^2 l} \leq \sigma_{as}, \quad (\text{AAB-T.2.1.8.1.5})$$

where, l is the length of the bracelets, $\sigma_{as} = \min(\sigma_{as a}, \sigma_{as b})$ – the allowable crushing stress ($\sigma_{as a, b}$ – the allowable crushing stresses of the shaft and bracelet materials, respectively).

Bolt assembly calculation

From the relation (AAB-T.2.1.8.1.3), taking into account the relation (AAB-T.2.1.8.1.4), the axial tightening force of a screw results,

$$F_{as} = \frac{\beta_a M_t}{\mu z d} \quad \text{sau} \quad F_{as} = \frac{\beta_a a M_t}{\mu z d b}, \quad (\text{AAB-T.2.1.8.1.6})$$

for the assembly with the two-piece bracket or, respectively, for the one with the articulated bracket; the values obtained with these relations are used for the calculation of the axially loaded single-bolt assembly.

AAB-T.2.3 ASSEMBLIES BY TIGHTENING ON THE CONE

AAB-T.2.3.1 DEFINITION

The *taper fit* is a shaft-hub type, removable connection that transmits loads through friction that occurs on the conical contact surface between the shaft and a hub that is pressed by axial displacement (fig. AAB-T.2.3.2.1).

AAB-T.2.3.2 CONSTRUCTION STRUCTURE

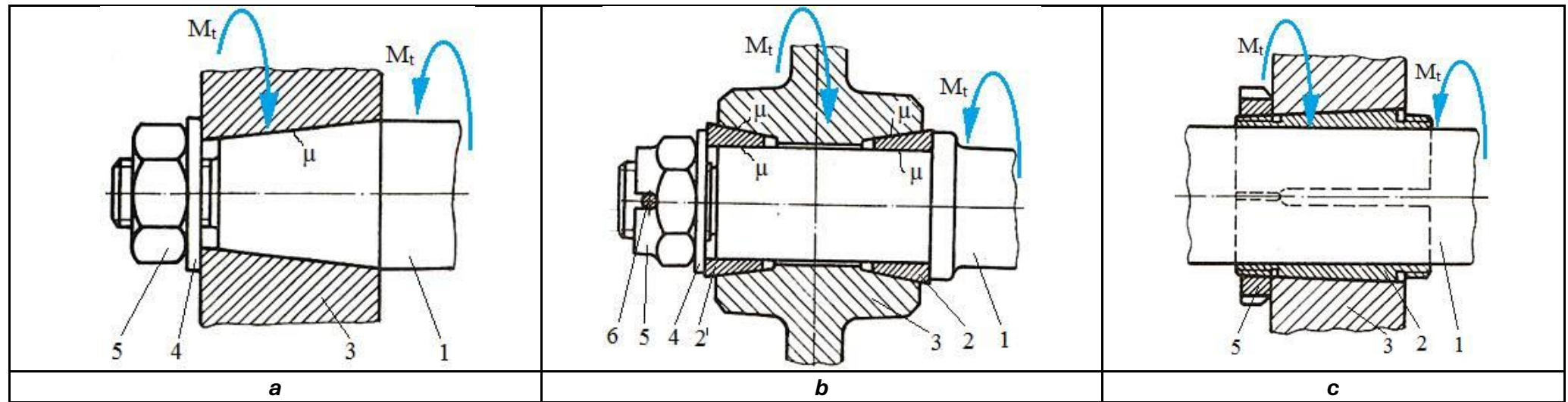


Fig. AAB-T.2.3.2.1 Cone-tightening assemblies: **a** – direct, without intermediate element; **b** – with elastic sleeves; **c** – with notched elastic sleeve

Description of operation and construction:

- fig. AAB-T.2.3.2.1,a – the torque, M_t , is transmitted by friction from shaft 1 to hub 3 when the axial force generated by tightening nut 5 presses the hub onto the shaft through washer 4;
- fig. AAB-T.2.3.2.1,b – the torque, M_t , is transmitted by friction from the shaft 1 to the elastic bushings 2, 2' and from them by friction to the hub 3 when the axial force generated by tightening the crenelated nut 5 keys the bushings 2, 2' between the shaft and the hub through the washer 4;
- fig. AAB-T.2.3.2.1,c – the torque, M_t , is transmitted by friction from the shaft 1 to the elastic bushing 2 and from them by friction to the hub 3 when the axial force generated by tightening the splined nut 5 keys the bushing 2 between the shaft and the hub;

AAB-T.2.3.3 AREAS OF USE, ADVANTAGES AND DISADVANTAGES

Main areas of use: machine construction for fixing transmission wheels on shaft ends; for fixing cutting tools; fixing bearings on shafts

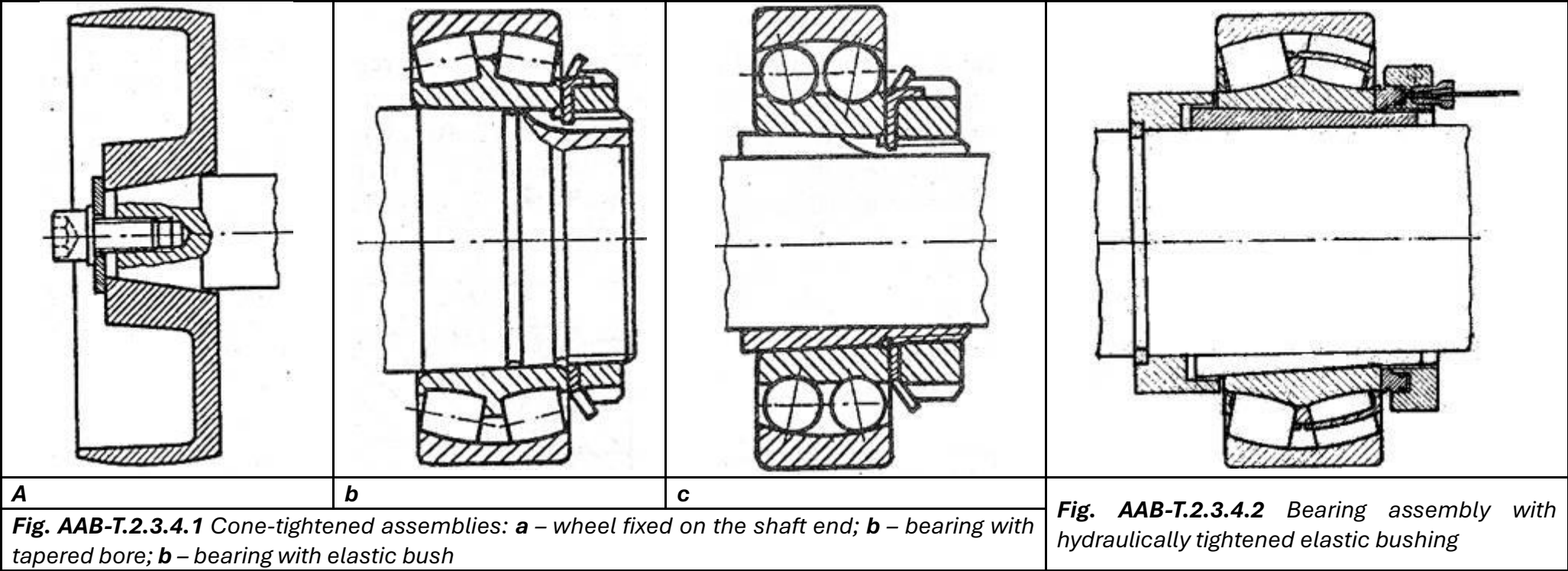
Tab. AAB-T.2.3.3.1 Advantages and disadvantages of cone tightening assemblies (relative to other friction assemblies)

Advantages *	Disadvantages *
- axial and angular positioning of the hub in relation to the shaft can be made; - do not introduce voltage concentrators	- transmit relatively low moments; - tightening adjustment is required

AAB-T.2.3.4 CLASSIFICATION

Tab. AAB-T.2.2.4.1 Classification of cone tightening assemblies

Criterion	Assembly type	Scheme
Number of contact surfaces	Direct (without intermediate element, with a single contact surface)	Fig. AAB-T.2.3.4.1,a,b
	With elastic bushings (with two or more contact surfaces)	Fig. AAB-T.2.3.4.1,c; AAB-T.2.3.4.2
Tightening method	To assemble with screws	Fig. AAB-T.2.3.4.1,b,c
	Hydraulic	Fig. AAB-T.2.3.4.2
Possibility of adjustment when mounting in axial direction	On tapered shaft (axially fixed)	Fig. AAB-T.2.3.4.1,a,b
	On cylindrical shaft	Fig. AAB-T.2.3.4.1,c; AAB-T.2.3.4.2



AAB-T.2.3.5 MATERIALS AND TECHNOLOGIES

The bracelets are made of rolled steel or sheet metal, and it is recommended that the roughness of the contact surfaces be as high as possible.

AAB-T.2.3.6 FORMS AND CAUSES OF DISPOSAL OR INCOMPATIBLE BEHAVIOR

Tab. AAB-T.2.1.6.1 Forms and causes of disuse or improper behavior

Shapes	Result	Appearance	Cause	Manifestation	Avoidance
Damage to the contact surface (plastic deformation)	Interruption of the transmission of the load	On shafts and hubs made of heavily loaded soft materials	Overcoming crushing stresses of materials	Damage to the active contact surfaces of the shaft and/or hub	Limiting crushing stresses to permissible values by calculation

AAB-T.2.3.7 FUNCTIONAL AND CONSTRUCTION PARAMETERS

The angle of inclination of the cone is 1:5 for easily removable machine parts: pulleys, pivots, couplings, valves, bearings, etc.; 1:10 for adjustable bearing bushes, shaft ends, etc.; 1:20 for tools and shafts for machine tools

AAB-T.2.3.8 CALCULATION MODELS

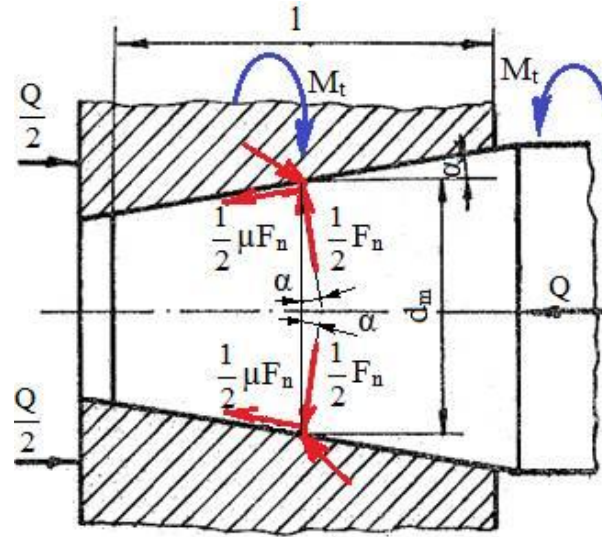


Fig. AAB-T.2.1.8.1 Calculation scheme of the cone tightening assembly

Calculation assumptions and requirements:

- the external load is transmitted entirely through friction;
- the pressures on the contact surfaces are uniformly distributed;
- for the calculation of the assembly on the conical surface, the loading on diametrically opposite semi-cones is considered

Shaft torsional resistance condition in the assembly area

The average diameter of the shaft in the assembly area, from the condition of resistance to torsional stress,

$$\frac{16 \beta_k M_t}{\pi d_m^3} \leq \tau_{at} \quad (\text{APP-T.2.3.8.1})$$

is determined by the relationship,

$$d_m = \sqrt[3]{\frac{16 \beta_k M_t}{\pi \tau_{at}}} \quad (\text{APP-T.2.3.8.2})$$

where, $\tau_{at} = 15 \dots 45$ MPa is the allowable stress at conventional torsion (also takes into account the existence of bending stress, maximum values are taken when bending is reduced or even zero (at the shaft end); the material is steel) and $\beta_k = 1 \dots 2$, the stress concentrator factor (for fatigue calculation).

The condition for transmitting the load through friction is determined from the equilibrium equation of axial moments,

$$\frac{1}{2} \mu F_n d_m \geq \beta_a M_t \quad (\text{AAB-T.2.3.8.3})$$

where, M_t [Nmm] is the transmitted torque, d_m [mm] – average diameter of the assembly, μ – coefficient of friction (for steel parts, $\mu = 0.15 \dots 0.3$), F_n [N] – normal pressing force; $\beta_a = 1.1 \dots 1.3$ – safety factor against sliding (skidding).

From the equilibrium equation of the forces acting in the axial direction, the clamping force is obtained,

$$Q = F_n \sin \alpha + \mu F_n \cos \alpha = F_n \sin \alpha + \operatorname{tg} \varphi F_n \cos \alpha = F_n \frac{\sin(\alpha + \varphi)}{\cos \varphi}, \quad (\text{AAB-T.2.3.8.4})$$

where, $\varphi = \arctan \mu$ - the friction angle.

The condition for the crushing resistance of the assembly, taking into account the relationships (AAB-T.2.3.8.3) and (AAB-T.2.3.8.4) is,

$$p = \frac{F_n}{A_s} = \frac{\frac{2 \beta_a M_t}{\mu d_m}}{\pi d_m l} = \frac{2 \beta_a M_t}{\pi \mu d_m^2 l} \leq \sigma_{as} \quad (\text{AAB-T.2.3.8.5})$$

where, l is the length, $\sigma_{as} = \min(\sigma_{as a}, \sigma_{as b})$ – the allowable crushing stress ($\sigma_{as a, b}$ – the allowable crushing stresses of the shaft and hub materials, respectively).

Calculation of threaded assembly or clamping device

From the relationship (AAB-T.2.3.8.4), taking into account the relationship (AAB-T.2.3.8.3), the clamping force results,

$$Q = \frac{2 \beta_a M_t}{\mu d_m} \frac{\sin(\alpha + \varphi)}{\cos \varphi} \quad (\text{AAB-T.2.3.8.6})$$

The value obtained with this relationship is used to calculate the axially loaded bolt assembly and/or its tightening torque.

AAB-T.2.4 ASSEMBLIES WITH TRUNCATED CONICAL RING

AAB-T.2.4.1 DEFINITION

The assembly with truncated conical rings is a shaft-hub type connection, removable, which transmits loads through the friction that arises on the conical contact surfaces between the shaft and the hub and a pair or more truncated conical rings which, when pressed axially, deform elastically radially and generate normal pressures (fig. AAB-T.2.4.2.1).

AAB-T.2.4.2 CONSTRUCTION STRUCTURE

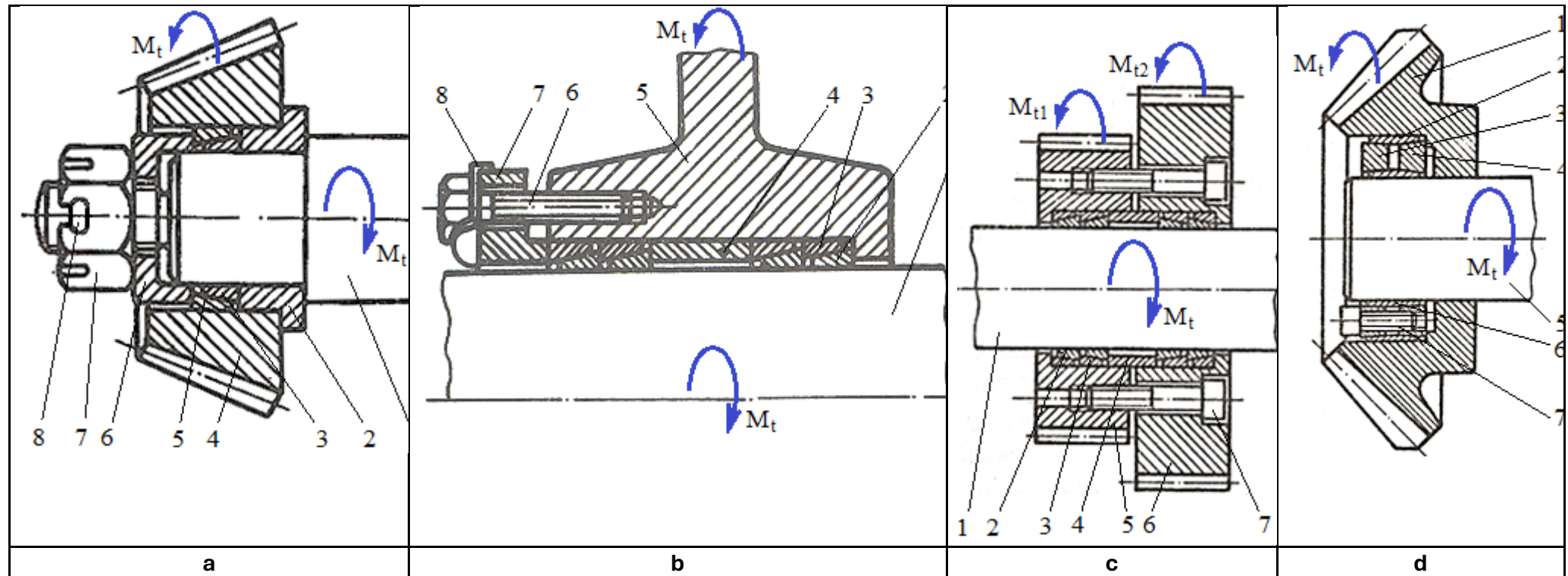


Fig. AAB-T.2.4.2.1 Tightening assemblies with conical rings: **a** – with a pair of conical rings and closing the axial force flow through the shaft; **b** – with four pairs of frustoconical rings and closing the axial force flow through the hub; **c** – with four pairs of frustoconical rings and closing the axial force flow through the hub; **d** – with a pair of bi-conical rings and closing the axial force flow inside

Description, operation and construction:

- fig. AAB-T.2.4.2.1,a – the torque M_t is transmitted by friction from the shaft 1 to the inner ring 3, to the outer ring 5 and to the bevel pinion 4 when the axial force generated by tightening the nut 7 pushes the outer ring 5 axially through the washer 6, which as a result of the contact after the generators with the inner ring 3, they stretch, respectively compress radially, and after the disappearance of the clearances adhere to the hub, respectively to the shaft; the nut 7 is secured against unscrewing by the split pin 8; the flow of the axial tightening force is closed through the shaft and the bushing 2.
- fig. AAB-T.2.4.2.1,b – the torque M_t is transmitted by friction from the shaft 1, through four parallel paths, to the inner rings 2, the outer rings 3 and the hub 5 when the axial force generated by tightening the screws 8 pushes/pulls axially the outer/inner rings 3/2 which, as a result of the contact after the generators with the inner/outer rings 2/3, stretch/compress, respectively compress/stretch radially, and after the clearances disappear, adhere to the hub/shaft, respectively to the shaft/hub; the spacer bushing 4 has the role of closing the axial tightening flow from the second inner ring on the left to the first outer ring on the right.

- fig. AAB-T.2.4.2.1,c – the torque M_t is transmitted by friction from the shaft 1, on four parallel paths (two for driving the wheel 5 and two for driving the wheel 6), to the inner rings 2, to the outer rings 3 and to the gear wheels 5 and 6 when the axial force generated by tightening the screws 7 pushes/pulls axially the outer rings 3 which, as a result of the contact after the generators with the inner rings 2, stretch, respectively compress radially, and after the disappearance of the clearances, adhere to the hub, respectively to the shaft; the spacer bushing 4 has the role of closing the axial tightening flow between the central inner rings.
- fig. AAB-T.2.4.2.1,d – the torque M_t is transmitted by friction from the shaft 5 to the inner bitronconical ring 6, to the intermediate washers 3 and 4, to the outer bitronconical ring 2 and to the bevel gear 1 when the axial force generated by tightening the screws 7 pushes and pulls axially the washers 3 and 4 which, as a result of the contact after the generators with the inner bitronconical rings 6 and outer 2, are compressed, respectively stretched radially, and after the disappearance of the clearances they adhere to the hub, respectively to the shaft.

AAB-T.2.4.3 AREAS OF USE, ADVANTAGES AND DISADVANTAGES

Main areas of use: machine construction for fixing transmission wheels on shafts.

Tab. AAB-T.2.3.3.1 *Advantages and disadvantages of truncated cone ring assemblies (compared to other shaft-hub assemblies)*

Advantages *	Disadvantages *
- ensures good centering of the assembled parts; - allow repeated assembly and disassembly without damaging the contact surfaces; - does not require channel processing that would weaken the resistance and introduce stress concentrators; - role of protection against overloads of assembled parts as a result of the appearance of skidding; - allow adjustment of the axial position of the hub relative to the shaft	- the need for additional tightening elements; - high execution precision; - in the case of overloads with the appearance of slipping, high temperatures occur; - requires precise execution due to small radial elastic deformations of the rings

AAB-T.2.4.4 CLASSIFICATION

Tab. AAB-T.2.2.4.1 *Classification of assemblies with truncated conical rings*

Criterion	Assembly type	Scheme
Number of inner/outer ring pairs	With a pair	Fig. AAB-T.2.4.2.1,a
	With two or more pairs	Fig. AAB-T.2.4.2.1,b,c
Ring shape	Troncon	Fig. AAB-T.2.4.2.1,a,b,c
	Bitronconical	Fig. AAB-T.2.4.2.1,d
Axial clamping force flow	With shaft closure	Fig. AAB-T.2.4.2.1,a
	With lockable handle	Fig. AAB-T.2.4.2.1,b,c
	With internal locking (in the clamping subsystem)	Fig. AAB-T.2.4.2.1,d

AAB-T.2.4.5 MATERIALS AND TECHNOLOGIES

The truncated conical rings are made of spring steels: **38Si7; 46Si7; 56Si7; 60Cr3; 51CrV4**

AAB-T.2.4.6 FORMS AND CAUSES OF DISPOSAL OR INCOMPATIBLE BEHAVIOR

Tab. AAB-T.2.4.6.1 Forms and causes of decommissioning or improper behavior

Shapes	Result	Appearance	Cause	Manifestation	Avoidance
Damage to the contact surface (plastic deformation)	Interruption of the transmission of the load	For shafts and wheels made of heavily loaded soft materials	Overcoming crushing stresses of materials	Damage to the active contact surfaces of the shaft and/or the collar	Limiting crushing stresses to permissible values by calculation

AAB-T.2.4.7 FUNCTIONAL AND CONSTRUCTIVE PARAMETERS

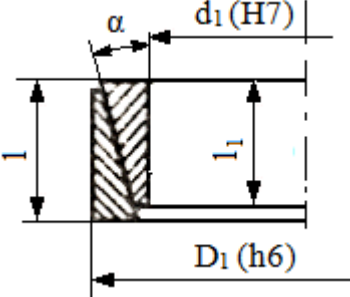
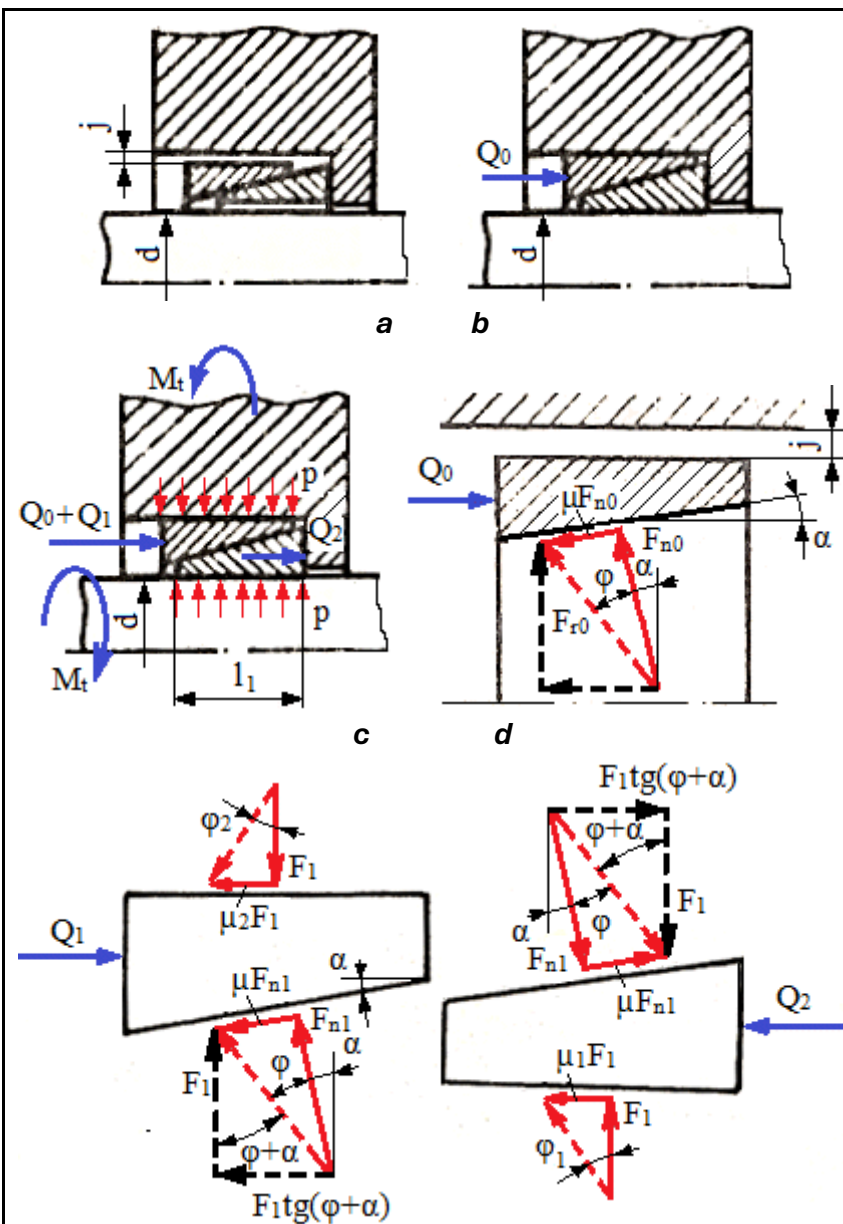
	Construction parameters: Tilt angle, $\alpha = 12 \dots 17^\circ$ Outer diameter, $D_1 = \{15, 18, 20, 22, 25, 26, 30, 32, 36, 44, 52, 56, 70, 78 \dots 212\}$ Inner diameter, $d_1 = \{12, 14, 16, 18, 20, 22, 25, 28, 32, 36, 40, 45, 50, 63, 71 \dots 200\}$ Length of the pair of rings, $l = \{4.5; 4.5; 4.5; 4.5; 6.3; 6.3; 6.3; 6.3; 7; 8; 8; 8; 9; 10 \dots 16\}$ Contact length (inside/outside), $l_1 = \{3.5; 3.5; 3.5; 3.5; 5.3; 5.3; 5.3; 5.3; 6; 6.8; 6.8; 7.6; 8.4 \dots 12.4\}$
---	--

Fig. AAB-T.2.4.7.1 Geometric parameters of a pair of conical rings

AAB-T.2.4.8 CALCULATION MODELS

AAB-T.2.4.8.1 Calculation model of assemblies with a pair of rings



Calculation assumptions:

- initially, as a consequence of the radial clearances provided between the rings and the assembled parts, the ring pack is mounted freely by hand (fig. AAB-T.2.4.8.1,a);
- as a result of tightening with force Q_0 , the rings are elastically deformed radially (the outer ring stretches and the inner one compresses) until the radial clearances are eliminated (fig. AAB-T.2.4.8.1,b);
- increasing the clamping force by the value Q_1 leads to the generation on the cylindrical surfaces (external, on the shaft and internal, in the hub) of the contact pressure, p , necessary for transmitting the torque M_t through friction (fig. AAB-T.2.4.8.1,c);
- the pressures on the contact surfaces are uniformly distributed.

Calculation relations:

Radial elastic displacement (stretching) of the outer ring with the clearance value,

$$\Delta l = j = \frac{N l}{E A} = \frac{F_{r0} d_m}{2 E A_i} = \frac{\frac{Q_0}{\tan(\alpha+\phi)} \frac{D_1+d_1}{2}}{2 E A_i} = \frac{Q_0 (D_1+d_1)}{4 E A_i \tan(\alpha+\phi)}, \quad (\text{APP-T.2.4.8.1})$$

where: j represents the radial clearance during assembly; $d_m = (D_1 + d_1)/2$ - the average diameter of the pair of rings (fig. AAB-T.2.4.7.1); F_{r0} - the radial force generated by the resultant force of the normal force F_{n0} and the friction force μF_{n0} (fig. AAB-T.2.4.8.1,a,d); A_i - the cross-sectional area of a ring; E - the longitudinal modulus of elasticity of the ring material; $\phi = \arctan \mu$ - the friction angle with μ , the coefficient of friction between the conical surfaces of the rings.

The **clamping force required** to cancel the games, from the relationship above (APP-T.2.4.8.1),

$$Q_0 = \frac{4 E A_i \tan(\alpha+\phi) j}{D_1+d_1}. \quad (\text{APP-T.2.4.8.2})$$

The **additional axial clamping force** (after eliminating the clearance) Q_1 for creating the contact pressure on the active cylindrical surfaces (of the shaft and hub), necessary for transmitting the torque through friction, is determined from the axial balance relationship of the outer ring (fig. AAB-T.2.4.8.1,e),

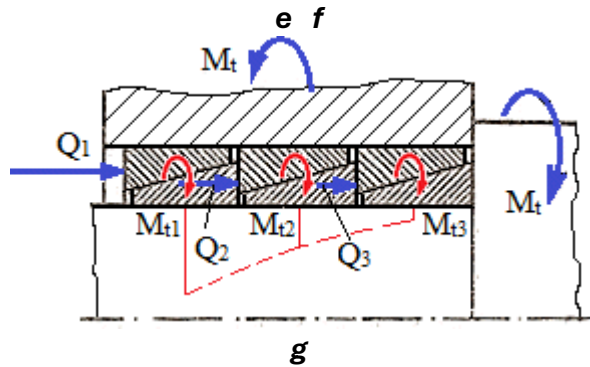


Fig. AAB-T.2.4.8.1 Calculation schemes for the assembly with truncated conical rings: **a** – initial assembly; **b** – tightening to eliminate clearances; **c** – tightening to generate the necessary pressure; **d** – tightening forces to eliminate clearances; **e** – outer ring forces; **f** – inner ring forces; **g** – moment distribution in the case of assemblies with several pairs of truncated conical rings

$$Q_1 = F_1 [\operatorname{tg}(\alpha + \varphi) + \mu_2], \quad (\text{APP-T.2.4.8.3})$$

where: F_1 represents the radial force that generates the contact pressure, μ_2 - the coefficient of friction between the cylindrical surfaces of the outer ring and the hub.

The *minimum frictional moment* transmitted through the contact surface between the shaft and the inner ring, considering F_1 from the relationship (APP-T.2.4.8.3),

$$M_{fmin} = \mu_1 F_1 \frac{d}{2} = \mu_1 \frac{Q_1}{\operatorname{tg}(\alpha + \varphi) + \mu_2} \frac{d}{2} = \frac{\mu_1 Q_1 d}{2 [\operatorname{tg}(\alpha + \varphi) + \mu_2]}, \quad (\text{APP-T.2.4.8.3})$$

where: d represents the shaft diameter; μ_1 represents the friction coefficient between the cylindrical surfaces of the inner ring and the shaft.

The condition for the transmission of the torque through friction, $M_{fmin} \geq \beta_a M_t$, taking into account the relationship (APP-T.2.4.8.3), becomes,

$$\frac{\mu_1 F_1 d}{2} \geq \beta_a M_t \text{ sau } \frac{\mu_1 Q_1 d}{2 [\operatorname{tg}(\alpha + \varphi) + \mu_2]} \geq \beta_a M_t. \quad (\text{APP-T.2.4.8.4})$$

where: $\beta_a = 1.5 \dots 3$ represents the coefficient of safety against slipping; M_t - transmitted torque.

The *total axial pressing (tightening) force*, taking into account the relationships (APP- T.2.4.8.2) and (APP- T.2.4.8.4),

$$Q = Q_0 + Q_1 = \frac{4 E A_i \operatorname{tg}(\alpha + \varphi) j}{D_1 + d_1} + \frac{2 \beta M_t [\operatorname{tg}(\alpha + \varphi) + \mu_2]}{\mu_1 d}. \quad (\text{APP-T.2.4.8.5})$$

The *condition of crushing resistance*, taking into account the relationship (APP- T.2.4.8.3),

$$p = \frac{F}{A} = \frac{F_1}{\pi d l_1} = \frac{Q_1}{\pi d l_1 [\operatorname{tg}(\alpha + \varphi) + \mu_2]} \leq \sigma_{as}, \quad (\text{APP-T.2.4.8.7})$$

where: l_1 represents the active length of the rings, σ_{as} - the allowable crushing stress corresponding to the softest material (inner ring or shaft).

The *axial reaction force of the hub on the inner ring*,

$$Q_2 = F_1 [\operatorname{tg}(\alpha + \varphi) - \mu_1]. \quad (\text{APP-T.2.4.8.6})$$

Note: Given that in the case of assemblies with multiple pairs of rings (fig. AAB-T.2.4.8.1,g) the pressure force on the last pair is much reduced, $Q_3 < Q_2 < Q_1$ (see rel. (APP- T.2.4.8.3) and (APP- T.2.4.8.6)), the transmitted torsional moments decrease progressively, $M_{t3} < M_{t2} < M_{t1}$ and, therefore, it is not recommended to increase the number of pairs of rings (there can be a maximum of 3 pairs); in the case of assembly with two or three pairs of rings, considering that the first pair is the most stressed (the transmitted moment is maximum, fig. AAB-T.2.4.8.1,a,d) only the first pair will be dimensioned and/or checked during the design.

ASB. ASSEMBLIES BY PINS AND BOLTS

ASB-T. CONSTRUCTION AND MODELING OF ASSEMBLIES USING PINS AND BOLTS

CONTENTS

ASB-T.1 DEFINITION 7-3

ASB-T.2 CONSTRUCTION STRUCTURES..... 7-3

ASB-T.3 AREAS OF USE, ADVANTAGES AND DISADVANTAGES 7-4

ASB-T.4 CLASSIFICATION 7-5

ASB-T.5 MATERIALS AND TECHNOLOGIES 7-6

ASB-T.6 FORMS AND CAUSES OF DISPOSAL OR INAPPROPRIATE BEHAVIOR..... 7-6

ASB-T.7 FUNCTIONAL AND CONSTRUCTIVE PARAMETERS.....7-7

ASB-T.8 CALCULATION MODELS 7-8

 ASB-T.8.1 Calculation model for asymmetrically transversely loaded pin assemblies 7-8

 ASB-T.8.2 Calculation model for shaft-hub assemblies with symmetrical transversely loaded pin 7-9

 ASB-T.8.3 Calculation model of shaft-hub assemblies with longitudinal cylindrical pins7-10

 ASB-T.8.4 Bolted joint calculation model7-11

ASB-T.9 CONSTRUCTIVE SOLUTIONS7-12

ASB-T.1 DEFINITION

Pins are removable assembly elements, with a cylindrical or conical shape that relatively position two or more elements as well as transmit reduced loads through their shape.

Bolts are larger cylindrical pins used as connecting elements in joints (low-speed rotating couplings).

ASB-T.2 CONSTRUCTION STRUCTURES

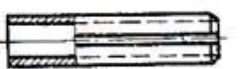
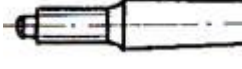
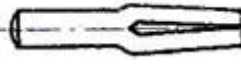
					
<i>a</i>	<i>b</i>	<i>c</i>	<i>d</i>	<i>e</i>	<i>f</i>
					
<i>g</i>	<i>h</i>	<i>i</i>	<i>j</i>	<i>k</i>	<i>l</i>

Fig. ASB-T.2.1 Types of pins: *a* – smooth solid cylindrical; *b* – tubular cylindrical; *c* – notched cylindrical along the entire length; *d,e* – notched cylindrical on portions; *f* – ribbed cylindrical; *g* – smooth conical; *h* – conical with threaded pin for disassembly; *i* – conical with a split end; *j* – notched conical along the entire length; *k* – combined and notched on the conical portion; *l* – combined and notched on the conical portions at the ends

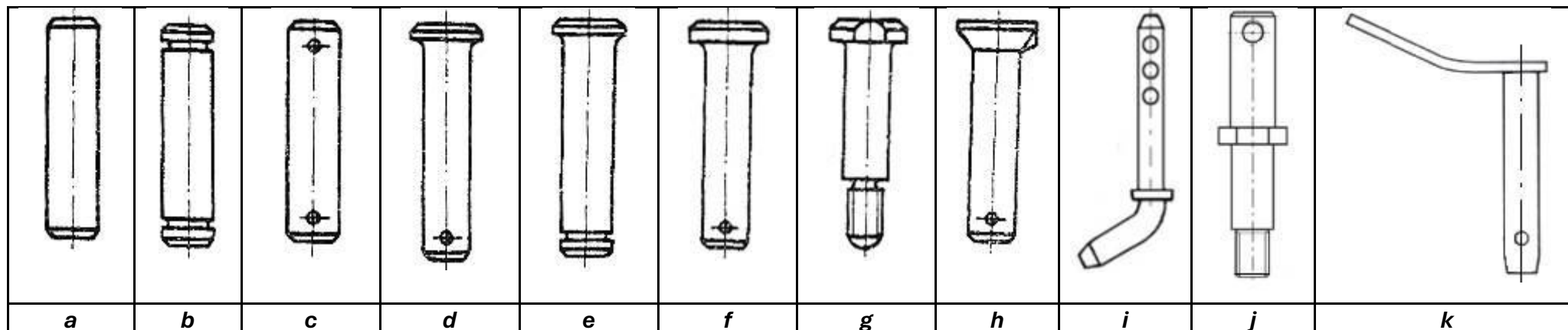


Fig. ASB-T.2.2 Types of bolts: **a** – headless and without cotter pin hole; **b** – with slots for snap rings; **c** – headless and cotter pin holes; **d** – with small head and cotter pin hole; **e** – with small head and slot for snap ring; **f** – with large head and cotter pin hole; **g** – with head and threaded pin; **h** – with nose head and cotter pin hole; **i** – with bent head; **j** – double with threaded pin; **k** – with handle

ASB-T.3 AREAS OF USE, ADVANTAGES AND DISADVANTAGES

Main areas of use: machine construction, technological and industrial equipment, etc.

Tab. ASB-T.3.1 Advantages and disadvantages

Advantages *	Disadvantages *
- simplicity of implementation; - precision of the joint; - low cost; - speed of assembly and disassembly; - small size	- introduce stress concentrators in the channel area - transmit low loads

ASB-T.4 CLASSIFICATION

Tab. ASB-T.4.1 Classification of pins

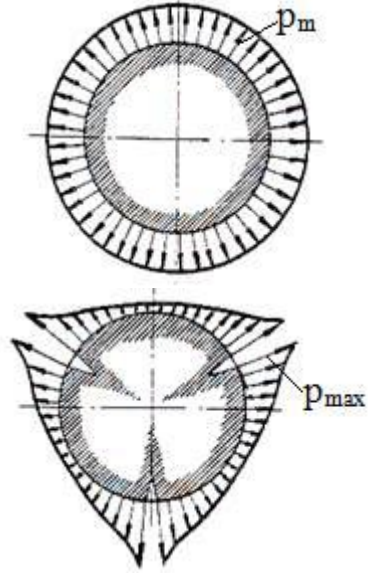
Criterion	Type	Scheme	Note.
Destination	Positioning (centering)	Fig. ASB-T.9.1,a,f,g,i	Ensuring the relative position of two parts
	Fixing (assembly)	Fig. ASB-T.9.1,b,c,f,g,i	Transmission of relatively small loads
	Safety	Fig. ASB-T.9.1,j	Protection of mechanical transmission components against overloads
By rod shape	Cylindrical with solid section	Fig. ASB-T.2.1,a,c...l	 a b
	Cylindrical with tubular section	Fig. AFT-T.2.2,c	
	tapered	Fig. ASB-T.2.1,g,h,i,j	
	Combined sections	Fig. ASB-T.2.1,k,l	
According to the shape of the outer surface	With smooth surface	Fig. ASB-T.2.1,a,b	
	With notched surface (lengthwise, locally)	Fig. ASB-T.2.1,c,d,e,j,k,l	
	With a striated surface	Fig. ASB-T.2.1,f	
According to the shape of the head	With threaded head	Fig. ASB-T.2.1,h	
	Notched head	Fig. ASB-T.2.1,i	

Fig. ASB-T.4.1 Distribution of pressures in the circumferential plane: **a** – uniform for the smooth pin; **a** – non-uniform with peaks p_{max} that ensure against disassembly

Tab. ASB-T.4.1 Bolt classification

Criterion	Type	Scheme
The shape of the final portions	Headless	Fig. ASB-T.2.2,a,b,c,j
	With head	Fig. ASB-T.2.2,d...i

	With spline for snap ring	Fig. ASB-T.2.2,b,e
	With pin hole	Fig. ASB-T.2.2,c,d,f,h,i,j,k
	With threaded pin	Fig. ASB-T.2.2,g,j
	With anti-rotation nose	Fig. ASB-T.2.2,h

ASB-T.5 MATERIALS AND TECHNOLOGIES

E285, C45, OSC 8 steels for bolts and pins are used depending on the purpose of the assembly.

The elastic tubular pins are made of spring steel with high tensile strength.

Bolts sometimes require heat treatments to harden the surface.

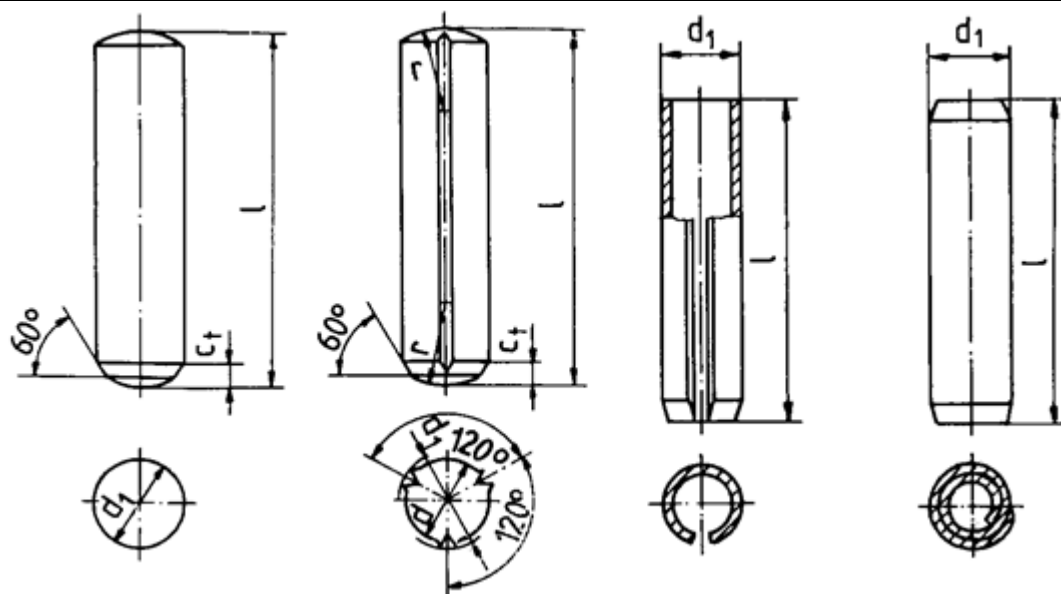
The execution technology of pins and bolts depends on the production series: in the case of unique pieces and small series, turning followed by grinding is recommended; in the case of large production series, they can be made by rolling (plastic deformation).

ASB-T.6 FORMS AND CAUSES OF DISPOSAL OR INAPPROPRIATE BEHAVIOR

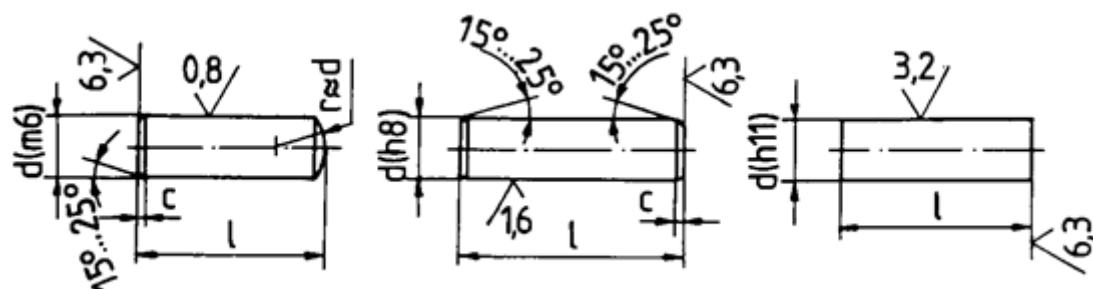
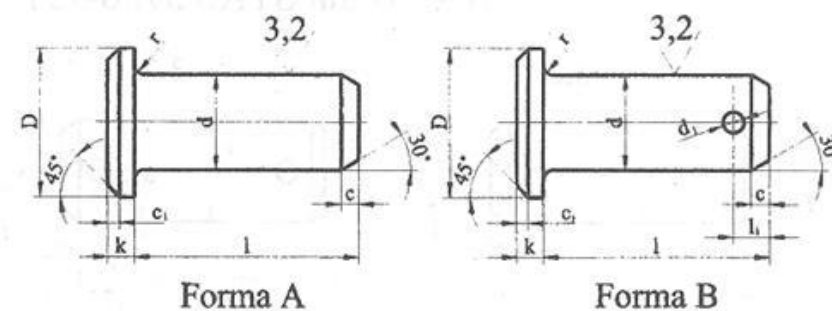
Tab. ASB-T.6.1 *Forms and causes of inappropriate behavior or removal from use*

Shapes	Result	Appearance	Cause	Manifestation	Avoidance
Rod breakage due to overload	Disposal	The rupture occurs in the shear section area.	Accidental and/or shock overloads	Interruption of the transmission of the load	Avoiding overloads
Damage to active surfaces	Difficult disassembly and impossibility of reassembly	In contact areas with part surfaces	Exceeding the crushing strength of the material	Plastic deformation of contacting surfaces	Limitation by calculation of crushing stresses in contact areas to permissible values

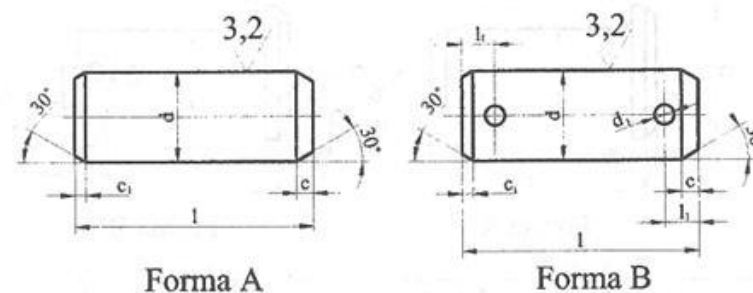
ASB-T.7 FUNCTIONAL AND CONSTRUCTIVE PARAMETERS



BOLȚ CU CAP MIC (Extras din STAS 5754/2-80)

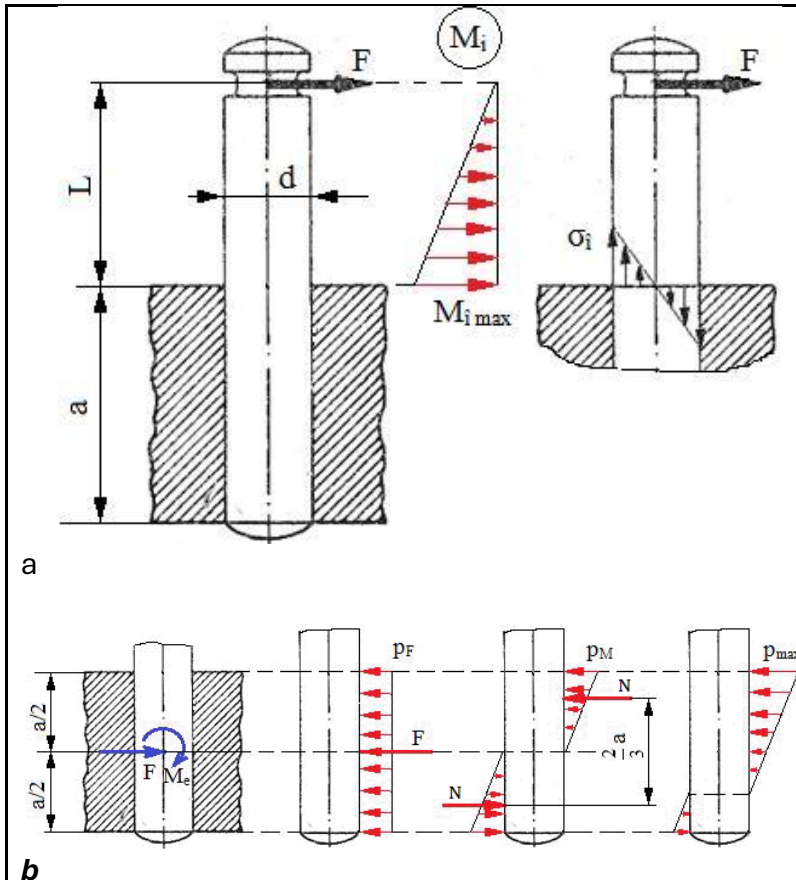


BOLȚURI FĂRĂ CAP (Extras din STAS 5756-73)



ASB-T.8 CALCULATION MODELS

ASB-T.8.1 CALCULATION MODEL FOR ASYMMETRICALLY TRANSVERSELY LOADED PIN ASSEMBLIES



Calculation assumptions and requirements:

- the calculation model is considered as a fixed bar subjected to bending stress
- the pin is pressed into the body of the base element and the transmission of force F is done through the cylindrical contact surface on which crushing pressures appear uniformly distributed on the circumference;
- the external force F is reduced at the center point of the contact area to the force F and the reduced moment, $M_e = F(L + a/2)$;
- the force F in the center of the contact area generates on the opposite semi-cylindrical surface pressures, p_F , uniformly distributed among the generators;
- the reduced moment M_e generates on two semi-cylindrical surfaces pressures distributed linearly after the generators with the maximum, p_M ;
- the shear stress is neglected.

Calculation relations:

The condition of the assembly's resistance to crushing,

$$p_{\max} = p_F + p_M = \frac{F}{a d} + \frac{2 N}{d \frac{a}{2}} = \frac{F}{a d} + \frac{2 \frac{M_e}{\frac{2}{3} a}}{d \frac{a}{2}} = \frac{F}{a d} + \frac{2 \frac{F(L + \frac{a}{2})}{\frac{2}{3} a}}{d \frac{a}{2}} = \frac{F}{a d} \left(1 + 6 \frac{L + \frac{a}{2}}{a} \right) = 2 \frac{F}{a d} \left(2 + 3 \frac{L}{a} \right) \leq \sigma_{as} \quad (\text{ASB-T.8.1.1})$$

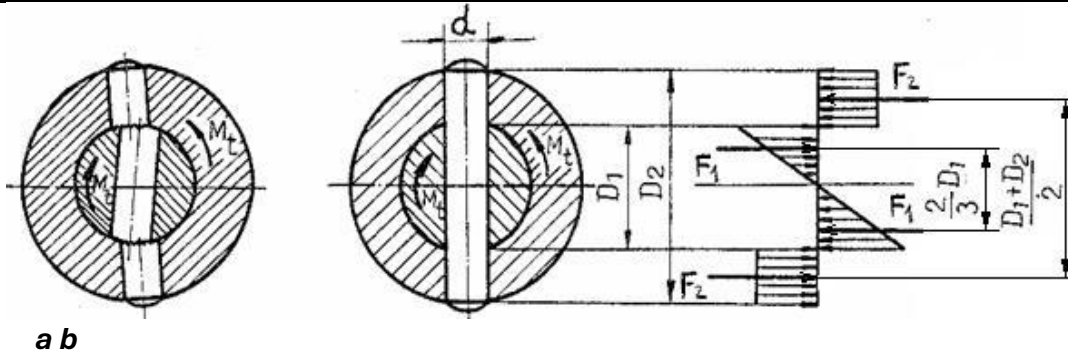
where $\sigma_{as} = \min(\sigma_{as}^s, \sigma_{as}^p)$ with σ_{as}^s is the allowable crushing stress of the pin material and σ_{as}^p – the allowable crushing stress of the material of the part in which the pin is mounted.

The condition of bending strength,

$$\sigma_i = \frac{M_{i \max}}{W_z} = \frac{F L}{\frac{\pi d^3}{32}} = \frac{32 F L}{\pi d^3} \leq \sigma_{ai} \quad , (\text{ASB-T.8.1.2})$$

where, σ_{ai} is the allowable bending stress of the pin material.

ASB-T.8.2 CALCULATION MODEL FOR SHAFT-HUB ASSEMBLIES WITH SYMMETRICAL TRANSVERSELY LOADED PIN



a b
Fig. ASB-T.8.2.1 Calculation schemes: **a** – shear; **b** – crushing

Calculation assumptions and requirements:

- the pin is mounted tightly in the hub and the pressure distribution in the contact areas of the pin with the hub is considered to be uniform;
- the pin is mounted with clearance in the shaft and the pressure distribution in the contact areas of the pin with the shaft is linear along the generator and uniform along the circumference;
- the transmission of forces through assembly is done through the shape;
- stresses: crushing between pin and shaft, crushing between pin and hub, pin shear

Calculation relations:

The condition of crushing resistance in the pin-hub contact areas,

$$p_{\max}^b = \frac{F}{A_s} = \frac{F_2}{A_s} = \frac{\frac{2 M_t}{D_1 + D_2}}{d \frac{D_2 - D_1}{2}} = \frac{4 M_t}{d (D_2^2 - D_1^2)} \leq \sigma_{as} \quad , \quad (\text{ASB-T.8.2.1})$$

where, $\sigma_{as} = \min(\sigma_{as}^s, \sigma_{as}^b)$ with σ_{as}^s is the allowable crushing stress of the pin material and σ_{as}^b – the allowable crushing stress of the hub material.

The condition of crushing resistance in the pin-shaft contact areas,

$$p_{\max}^a = \frac{F}{A_s} = 2 \frac{F_1}{A_s} = 2 \frac{\frac{2 M_t}{D_1}}{d \frac{D_1}{2}} = \frac{6 M_t}{d D_1^2} \leq \sigma_{as} \quad (\text{ASB-T.8.2.2})$$

where, $\sigma_{as} = \min(\sigma_{as}^s, \sigma_{as}^a)$ with σ_{as}^s is the allowable crushing stress of the pin material and σ_{as}^a – the allowable crushing stress of the shaft material.

The shear strength condition of the pin,

$$\tau_f = \frac{T}{A_f} = \frac{\frac{M_t}{D_1}}{\frac{\pi d^2}{4}} = \frac{4 M_t}{\pi D_1 d^2} \leq \tau_{af} \quad , \quad (\text{ASB-T.8.2.3})$$

where, τ_{af} is the allowable shear stress of the pin material

Note: for notched pins, the allowable crushing stress is adopted 30% lower than for smooth pins.

ASB-T.8.3 CALCULATION MODEL OF SHAFT-HUB ASSEMBLIES WITH LONGITUDINAL CYLINDRICAL PINS

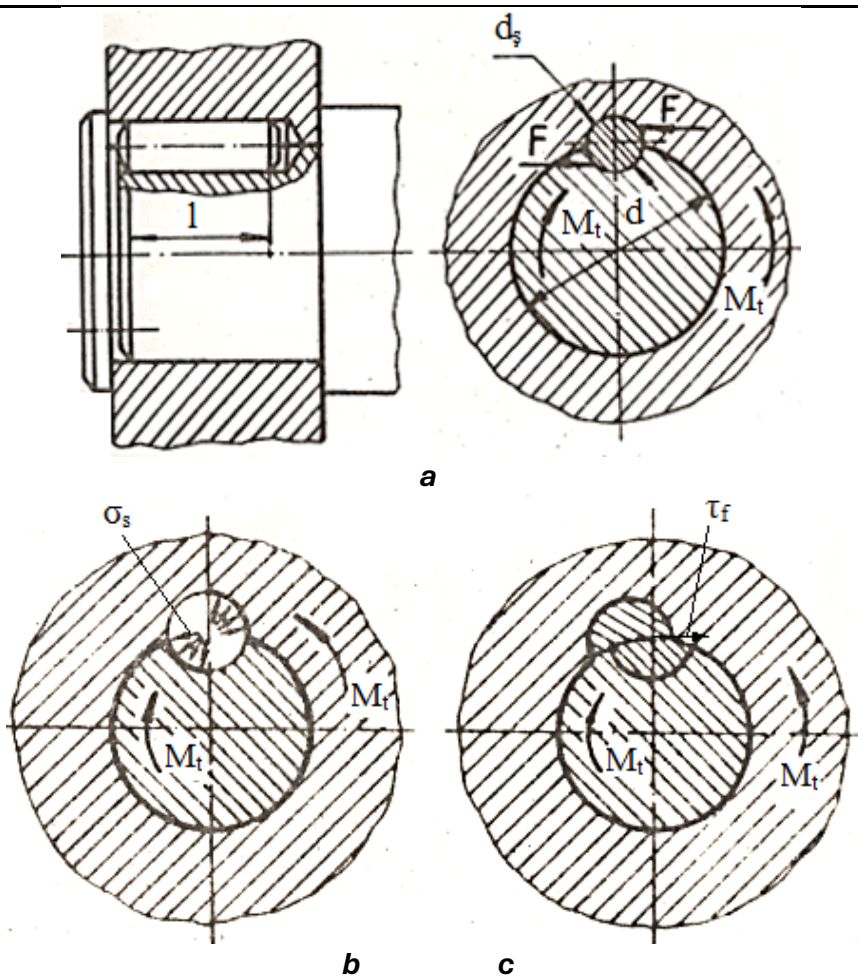


Fig. ASB-T.8.4.1 Calculation schemes: **a** – axial and transverse sections; **b** – crushing stress; **c** – shear stress

Calculation assumptions and requirements:

- the pin is pressed into a cylindrical bore, half in the shaft and half in the hub, made in the state of the shaft assembled in the hub;
- the internal force that requires the assembly is considered to act in the cylindrical area and is determined with the relationship, $F = 2 M_t / d$;
- the internal force F generates uniformly distributed crushing stresses σ_s on the quarter-cylindrical contact surface of the pin with the shaft/hub;
- as a result of the action of the internal force F , the pin is also subjected to shear in the diametrical plane with the shear stress τ_f .

Calculation relations:

The condition of the assembly's resistance to crushing,

$$\sigma_s = \frac{F}{A_s} = \frac{\frac{2 M_t}{d}}{z \frac{d_s}{2} l} = \frac{4 M_t}{z d d_s l} \leq \sigma_{as} \quad (\text{ASB-T.8.3.1})$$

where: σ_{as} represents the allowable crushing stress of the softer material, z – number of pins, d_s – pin diameter, d – shaft diameter, M_t – transmitted torque.

The shear strength condition of the assembly,

$$\tau_f = \frac{F}{A_f} = \frac{\frac{2 M_t}{d}}{z d_s l} = \frac{2 M_t}{z d d_s l} \leq \tau_{af} \quad (\text{ASB-T.8.3.2})$$

where, τ_{af} represents the allowable shear stress of the pin material

ASB-T.8.4 BOLTED JOINT CALCULATION MODEL

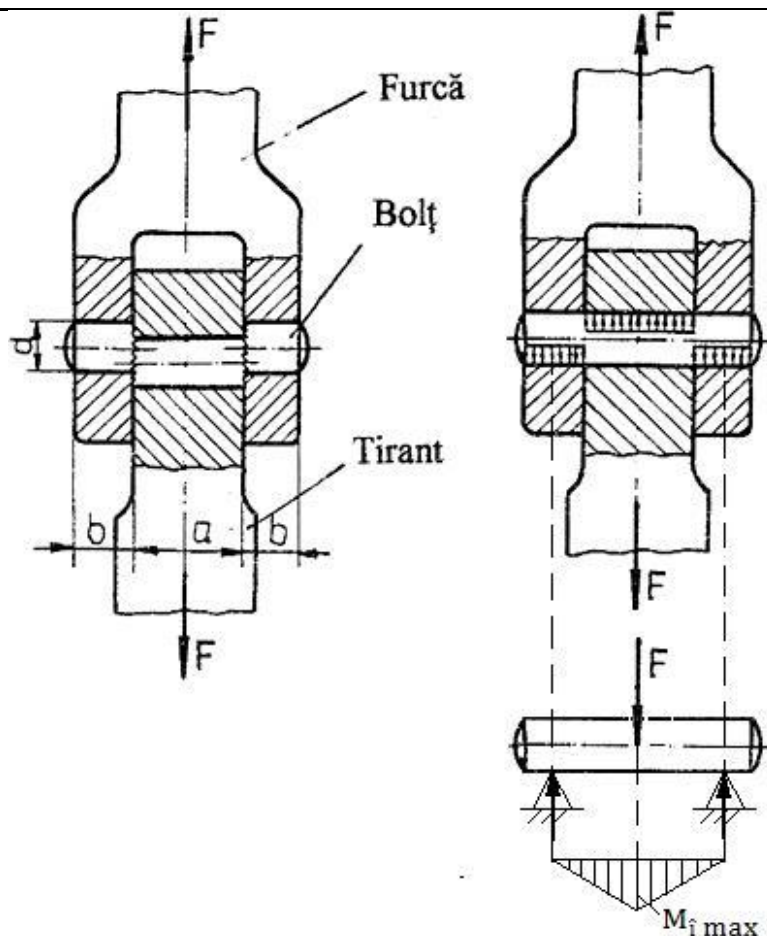


Fig. ASB-T.8.3.1 Calculation schemes: **a** – shear; **b** – crushing and bending

Calculation assumptions and requirements:

- the bolt is mounted pressed into the fork body and with play in the tie rod;
- pressures are considered to be uniformly distributed on the contact surfaces;
- stresses: crushing between bolt and fork, crushing between bolt and tie rod, bolt shear, bolt bending.

Calculation relations:

The condition of crushing resistance in the pin-fork contact areas,

$$p_m^f = \frac{F_s}{A_s} = \frac{\frac{F}{2}}{b d} \leq \sigma_{as}, \quad (\text{ASB-T.8.4.1})$$

where, $\sigma_{as} = \min(\sigma_{as}^b, \sigma_{as}^f)$ with σ_{as}^b is the allowable crushing stress of the bolt material and σ_{as}^f – the permissible crushing stress of the fork material.

The condition of crushing resistance in the bolt-tie contact area,

$$p_m^t = \frac{F_s}{A_s} = \frac{F}{a d} \leq \sigma_{as} \quad \text{or} \quad \frac{F}{a d} \leq p_a, \quad (\text{ASB-T.8.4.2})$$

for assembly without lubrication, or respectively for assembly with lubrication, where: $\sigma_{as} = \min(\sigma_{as}^b, \sigma_{as}^t)$ with σ_{as}^b , the allowable crushing stress of the bolt material and σ_{as}^t , the allowable crushing stress of the tie rod material; p_a - the allowable (non-expulsion) pressure of the lubricant.

The shear strength condition of the bolt,

$$\tau_f = \frac{F_f}{A_f} = \frac{\frac{F}{2}}{\frac{\pi d^2}{4}} \leq \tau_{af}, \quad (\text{ASB-T.8.4.3})$$

where, τ_{af} represents the allowable shear stress of the bolt material.

The condition of the bending strength of the bolt

$$\sigma_i = \frac{M_{i \max}}{W_z} = \frac{\frac{F}{2} \frac{a+b}{2}}{\frac{\pi d^3}{32}} = \frac{8 F (a+b)}{\pi d^3} \leq \sigma_{ai}, \quad (\text{ASB-T.8.4.4})$$

where, σ_{ai} is the allowable bending stress of the bolt material

Note: Sometimes, in the case of relative movement between the fork and the tie rod, it is necessary to lubricate the assembly with consistent grease in the area of the tie rod-bolt clearance adjustment and in the ASB-T.8.4.2 relation, instead of σ as, p_a is introduced, the contact pressure that ensures the non-expulsion of the lubricant film.

ASB-T.9 CONSTRUCTIVE SOLUTIONS

Tab. ASB-T.9.1 Constructive solutions for pin assemblies

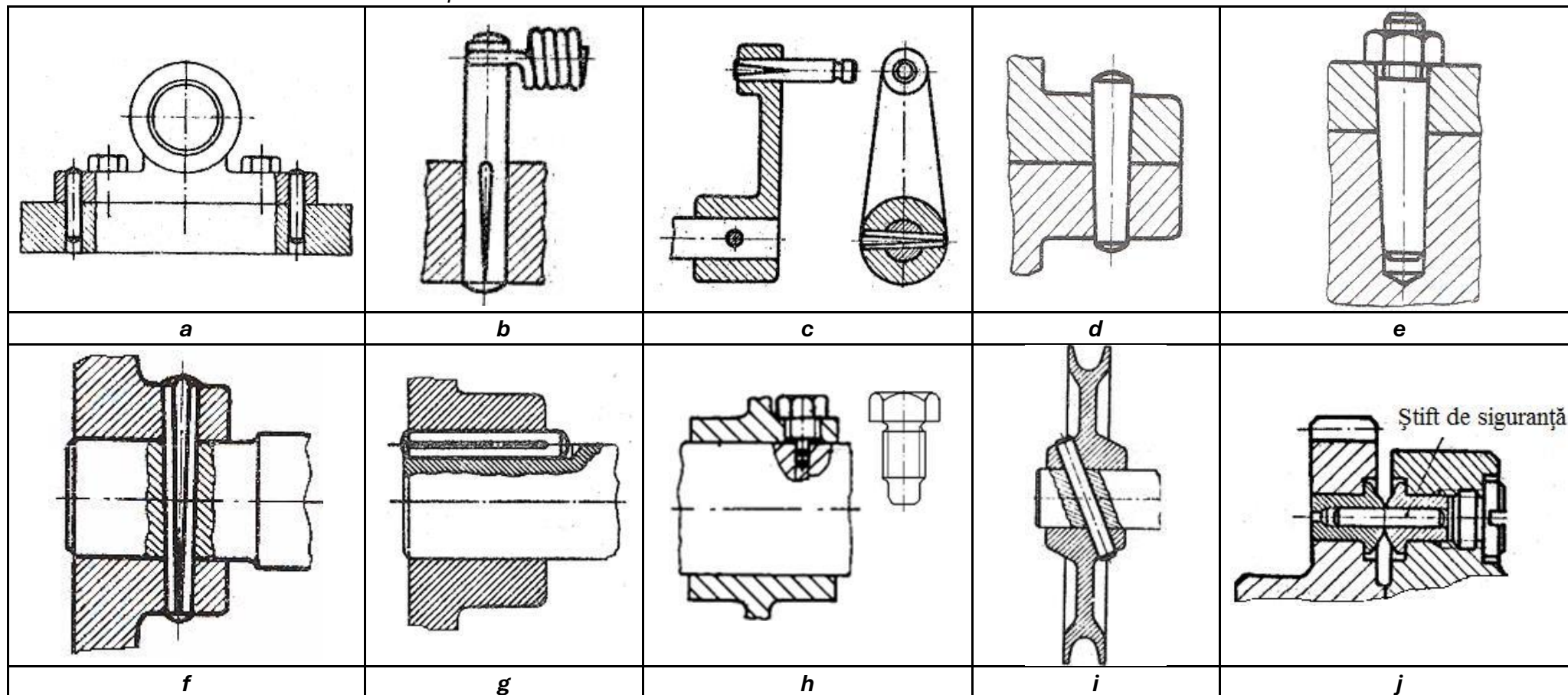
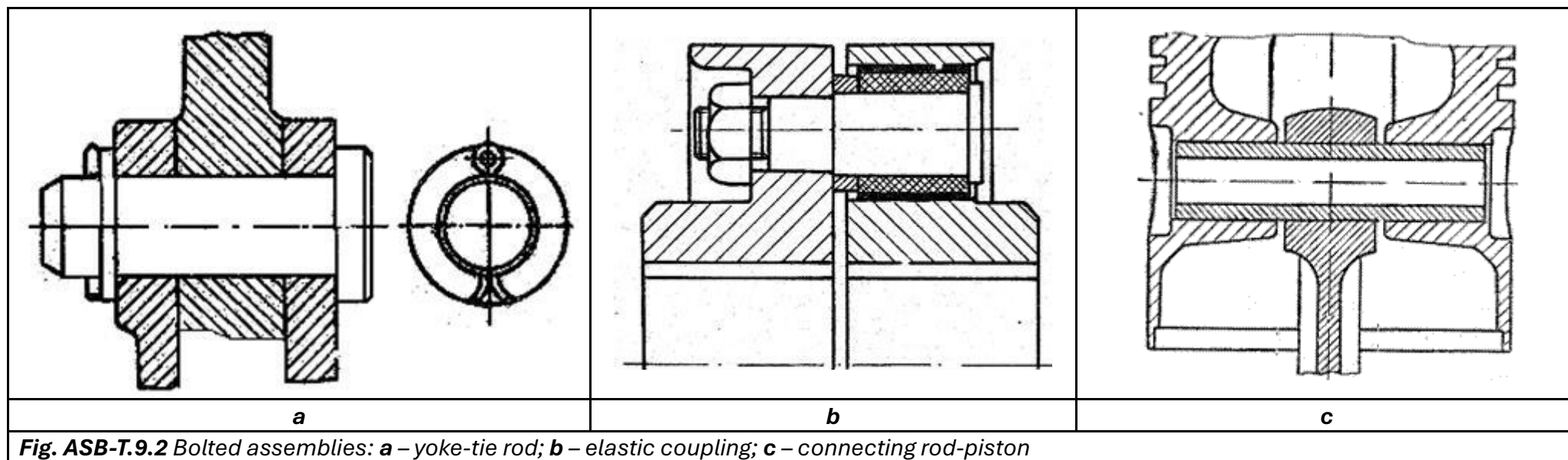


Fig. ASB-T.9.1 Pin assemblies: **a** – for positioning a bearing body; **b** – for hanging tension springs; **c** – for assembling cranks; **d** – flange positioning cone; **e** – positioning cone and with threaded pin for disassembly; **f** – cone with notch for transverse shaft-hub assembly; **g** – cylindrical with notch for longitudinal shaft-hub assembly; **h** – threaded with pin for fixing the hub to the shaft; **i** – cylindrical with for inclined shaft-hub assembly; **j** – safety cylindrical as part of a coupling

Tab. ASB-T.9. *Constructive solutions for bolted assemblies*

ARC. ELASTIC ELEMENTS (SPRINGS)

ARC-T. CONSTRUCTION AND MODELING OF ELASTIC ELEMENTS (SPRINGS)

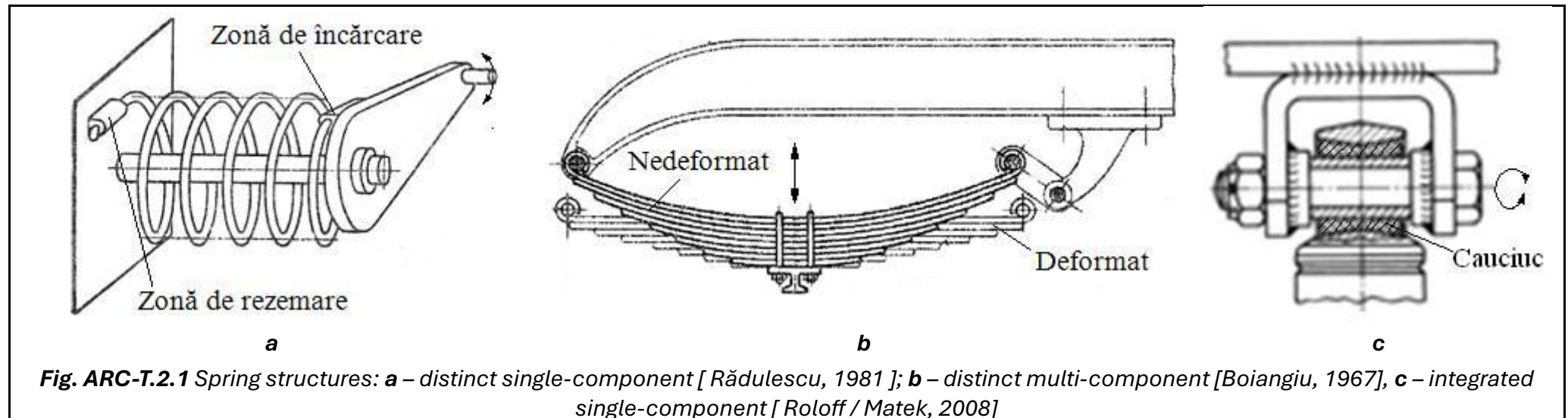
CONTENTS

ARC-T.1 DEFINITION	8-3
ARC-T.2 CONSTRUCTION STRUCTURE	8-3
ARC-T.3 FUNCTIONS AND AREAS OF USE	8-3
ARC-T.4 CLASSIFICATION	8-5
ARC-T.5 MATERIALS AND TECHNOLOGIES.....	8-8
ARC-T.6 FORMS AND CAUSES OF DISPOSAL OR INAPPROPRIATE BEHAVIOR	8-8
ARC-T.7 FUNCTIONAL AND CONSTRUCTIVE PARAMETERS	8-9
ARC-T.8 CALCULATION MODELS.....	8-14
ARC-T.8.1 Calculation models of cylindrical compression/tension helical springs	8-14
ARC-T.8.2 Calculation models for cylindrical torsion coil springs	8-18
ARC-T.8.3 Calculation models for leaf and lamellar springs	8-21
ARC-T.8.3.1. Straight single leaf springs	8-21
ARC-T.8.3. 2 Leaf springs (multi-leaf)	8-22
ARC-T.8.4 Torsion bar spring calculation models	8-26
ARC-T.8.5 Rubber Spring calculation models	8-28
ARC-T.8.5. 1. Rubber compression springs	8-29
ARC-T.8.5. 2. Rubber shear springs	8-29
ARC-T.8.5. 3. Rubber torsion springs	8-30

ARC-T.1 DEFINITION

Springs are machine elements that, due to the shape and elastic properties of the materials, store the mechanical work of external forces during deformation, and return it, quasi-totally or partially, during the period of return to the initial shape [Jula, 1986].

ARC-T.2 CONSTRUCTION STRUCTURE



ARC-T.3 FUNCTIONS AND AREAS OF USE

Tab. ARC-T.3.1 Functions and areas of use of springs

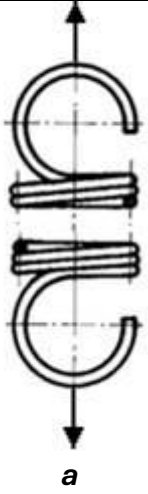
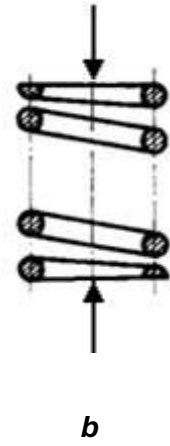
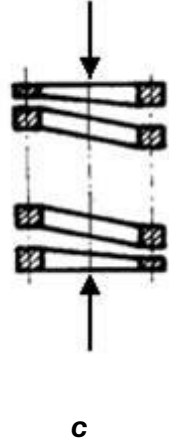
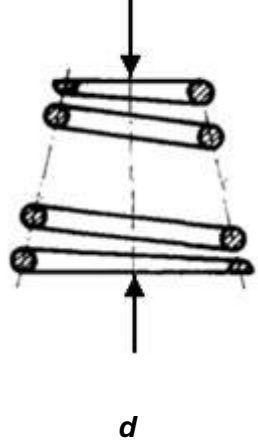
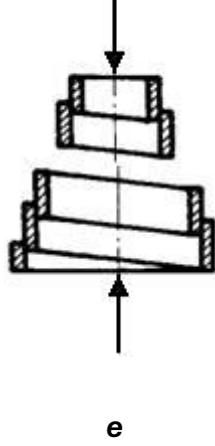
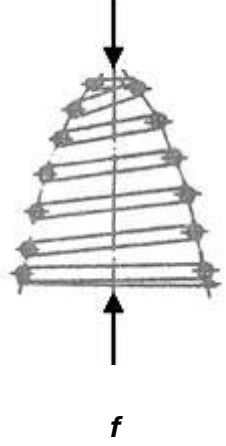
Functions	Areas of use
Exercising permanent <u>forces</u>	Safety valves, elastic safety couplings,
Shock and vibration <u>damping</u> (energy dissipation)	Vehicle suspensions, buffers, elastic couplings, crane hooks, presses, machine foundations, connection systems, etc.

Accumulation of mechanical energy to be released instantly or gradually	Clocks, mechanically operated mechanisms, hydraulic or pneumatic systems
Limiting maximum force values	Stamps and dies, safety couplings
Adjusting the functional parameters of some devices	Control valves,
Measuring forces and moments	Torque wrenches, scales, measuring instruments, test stands
Changing the natural frequencies of mechanical elements or systems	Suspensions, elastic couplings

ARC-T.4 CLASSIFICATION

Tab. ARC-T.4.1 *Classification of springs [Boiangiu, 1967; Demian, 1980]*

Criterion	Type
Construction form	Helical (fig. ARC-T.4.1,a,b,c,d,e,f,g,j), torsion bar (fig. ARC-T.4.1,i), annular (fig. ARC-T.4.1,p), disc (fig. ARC-T.4.1,h), sheet (fig. ARC-T.4.1,l), flat spirals (fig. ARC-T.4.1,k), block (fig. ARC-T.4.1,q)
Main material requirement	Tension-compression: ring (fig. ARC-T.4.1,g); disc (fig. ARC-T.4.1,h)
	Bending: lamellar, in sheets (fig. ARC-T.4.1,l; fig. ARC-T.1,b), helical loaded with torsional moments (fig. ARC-T.4.1,j; fig. ARC-T.1,a), flat spiral springs (fig. ARC-T.4.1,k)
	Torsion (twisting): torsion bar (fig. ARC-T.4.1,i), helical (fig. ARC-T.4.1,j)
Mode of action of the external load	Loaded with tensile loads: helical (fig. ARC-T.4.1,a)
	Loaded with compression loads: helical (fig. ARC-T.4.1,b,c,d,e,f,g), annular springs (fig. ARC-T.4.1,g), disc springs (fig. ARC-T.4.1,h), rubber springs (fig. ARC-T.4.1,q)
	Loaded with bending loads: lamellar, in sheets (fig. ARC-T.4.1,l)
	Loaded with torsional loads: torsion bar (fig. ARC-T.4.1,i), torsional helical (fig. ARC-T.4.1,j)
rigidity	Constant (fig. ARC-T.4.1,a,b,c)
	Progressively variable (fig. ARC-T.4.1,d,e)
	Regressive variable (fig. ARC-T.4.1,g)
Nature of the material	Metallic: ferrous (fig. ARC-T.4.1,a,b,c,d,e,g,h,i,j,k,l,m,n,o,p), non-ferrous
	Non-metallic: rubber (fig. ARC-T.2,q,r); plastics

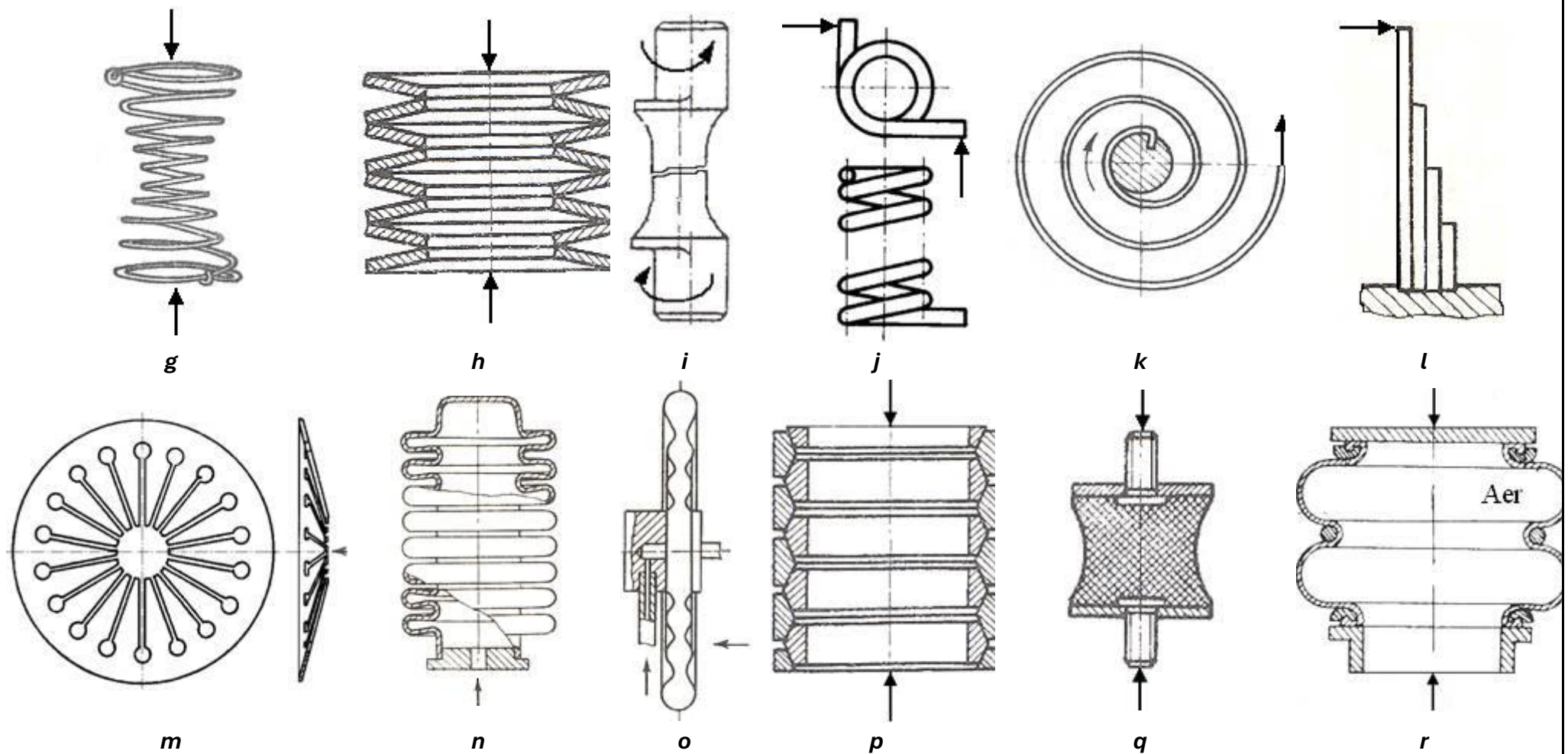


Fig. ARC-T.4.1 Types of springs: **a** – cylindrical tension coil; **b, c** – cylindrical compression coil; **d, e** – conical compression coil; **f, g** – parabolic compression coil; **h** – disc; **i** – torsion bar; **j** – torsion coil; **k** – flat spiral; **l** – sheet; **m** – diaphragm; **n** – corrugated tube; **o** – corrugated membrane; **p** – annular; **q** – rubber block; **r** – air cushion

ARC-T.5 MATERIALS AND TECHNOLOGIES

Tab. ARC-T.5.1 Material groups and technological flows

Material group	Common and semi-finished brands	Technological flows	Spring type
Spring steels	High-quality carbon steels with high C content (0.4...1.25): C55A, C65A, C75A, C85A (see fer. ARC-M.1); patented circular wire (hardening to increase strength), grades RR, RM, RS (STAS 893).	Cold rolling → stress relief (return)	Screw
	Carbon alloy steels (SR EN 10089): 51Si17A, 50VCr11A, 56Si15A, 56Si17A; drawn wire (STAS 892), wide steel for leaf springs (SR EN 10092-2), peeled steel bars for valve springs (STAS 8371-80), cold rolled steel strips (DIN 17222)	Quenching and tempering → coiling (d < 8...10 mm) or cold forming → stress relief; winding (d > 8...10 mm) or hot forming → quenching-tempering	Helical, lamellar, sheet, disc, annular Lamella, flat spiral
Non-ferrous materials	Brass (Cu70, Cu 72, Ni6, Al1.5), bronzes (Cr0.6-0.85; Be2, Cu97.5); wire, strips	Forming → heat treatment	Springs with anti-corrosion properties
Non-metallic materials	Rubber (natural or synthetic), cork, plastics	Vulcanization (for rubber)	Buffers, shock absorbers
Note: To increase the fatigue resistance of steel springs, shot blasting (cast iron, steel) can be done, which leads to hardening of the surface layer; anti-corrosion protection can be done by browning, galvanic coating or painting.			

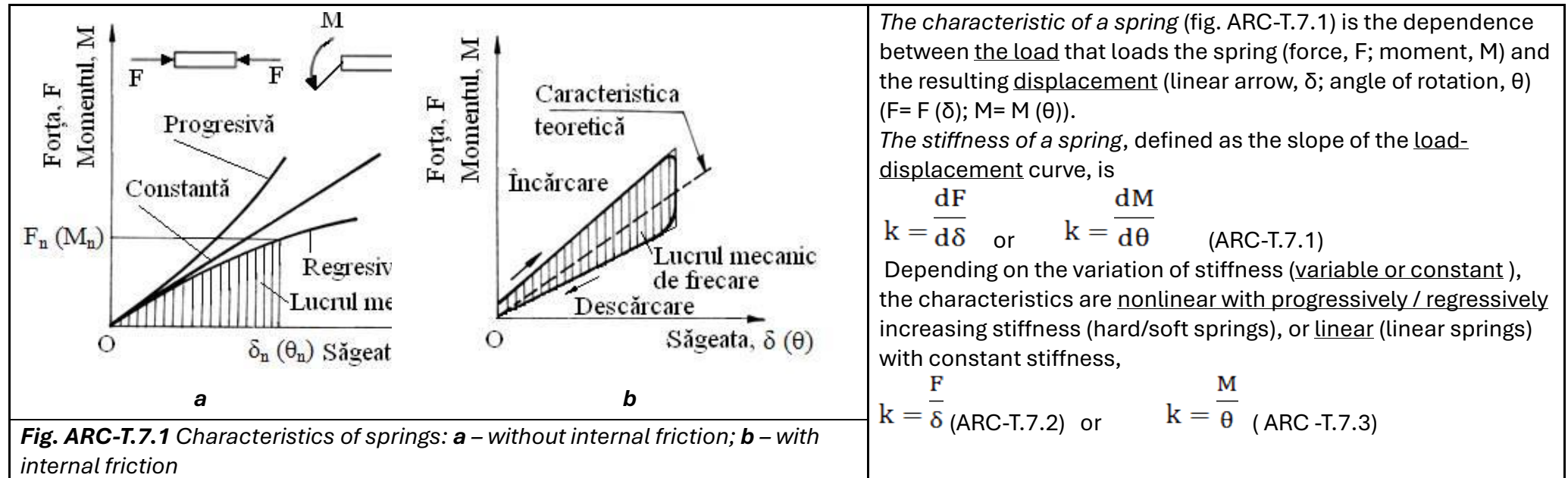
ARC-T.6 FORMS AND CAUSES OF DISPOSAL OR INAPPROPRIATE BEHAVIOR

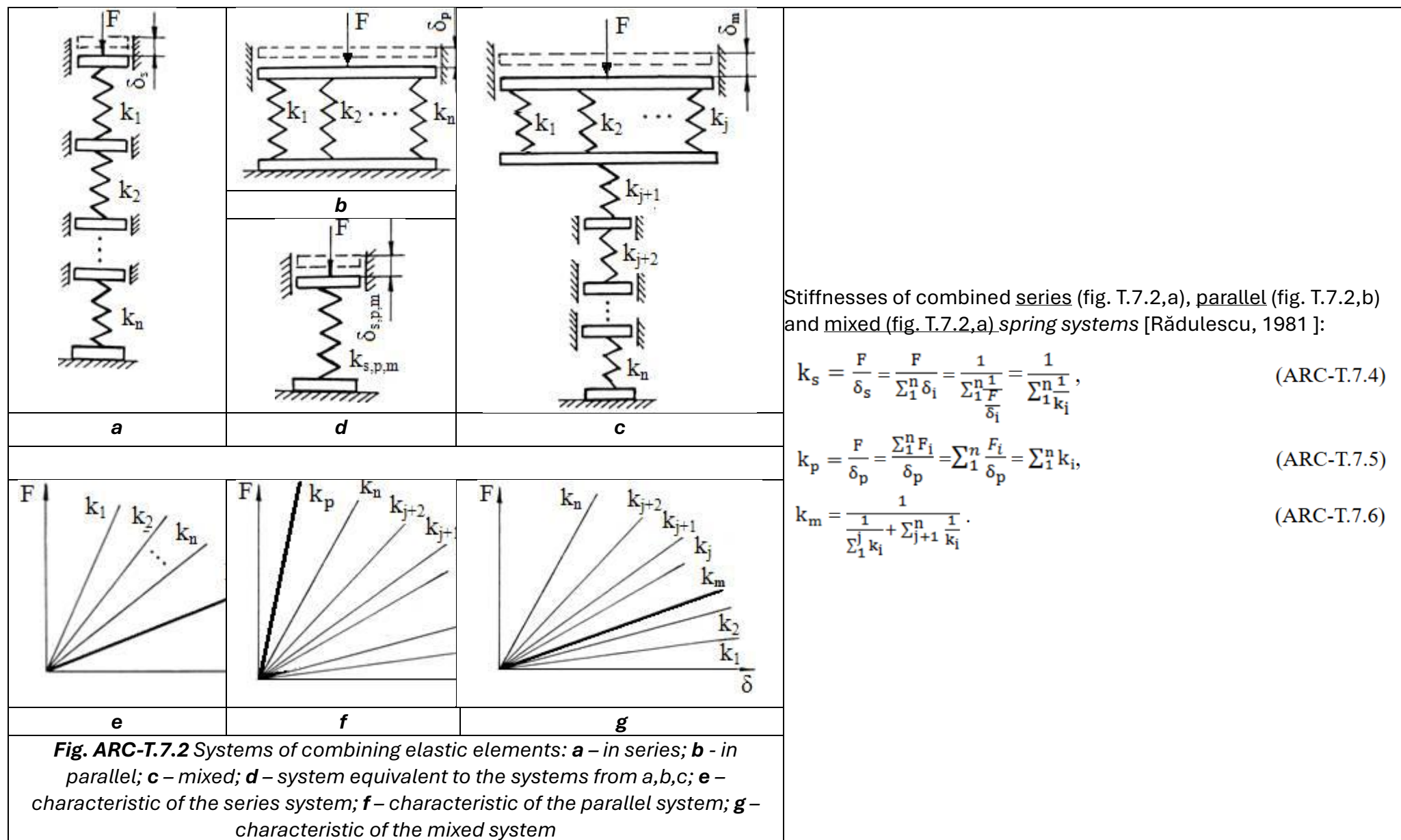
Tab. ARC-T.6.1 Forms and causes of inappropriate behavior or removal from use

Shapes	Appearance	Cause	Result	Manifestation	Avoidance
Fracture	Especially for springs loaded with variable loads	Exceeding the static or fatigue strength of the material	Disposal	Interruption of the connection in the system in which it is mounted	Limiting maximum static or fatigue stresses to permissible values; avoiding operating overloads

Behavior without energy accumulation	Especially for rubber springs	Loss of elastic properties of the material over time	Inappropriate behavior	Movements with non-uniformities of the system integrating the spring	Limiting operating time
--------------------------------------	-------------------------------	--	------------------------	--	-------------------------

ARC-T.7 FUNCTIONAL AND CONSTRUCTIVE PARAMETERS





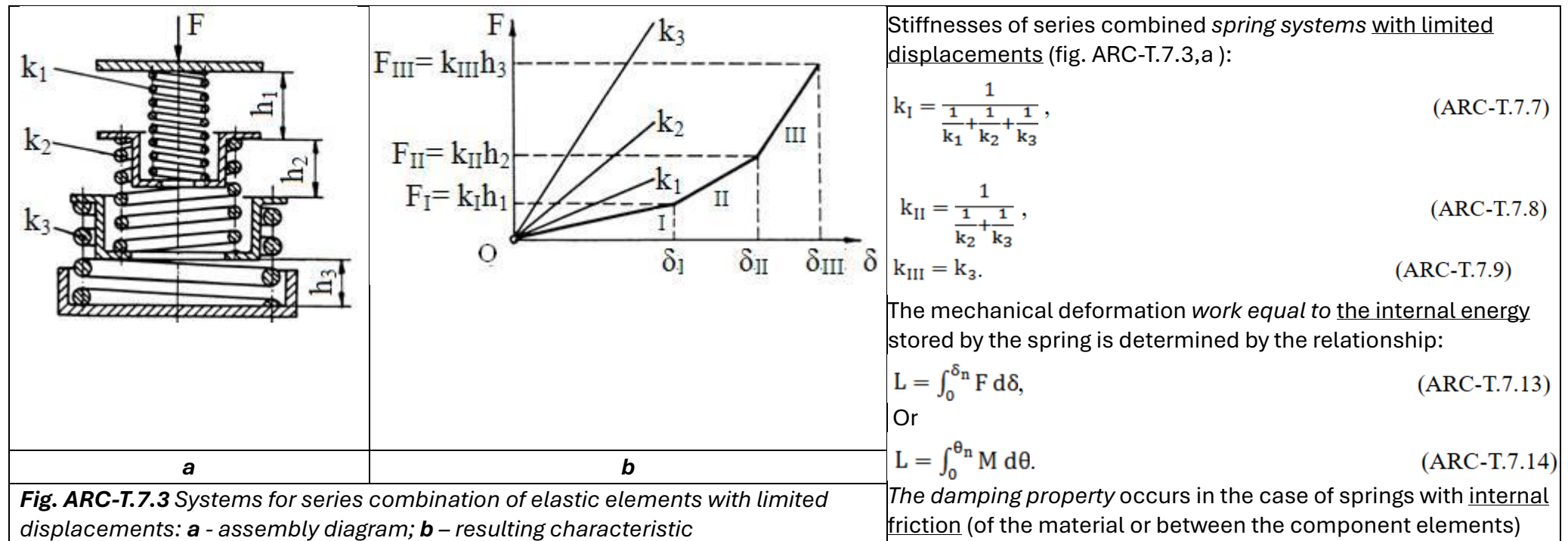
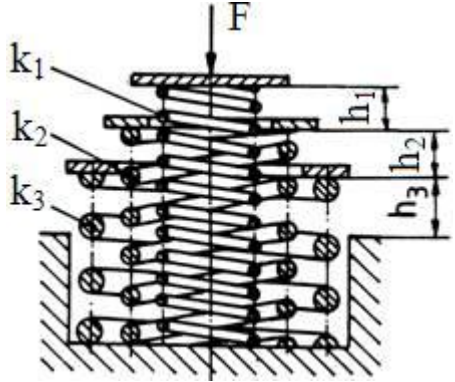
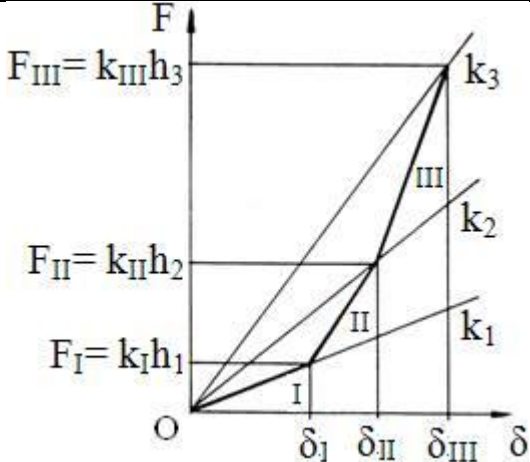


Fig. ARC-T.7.3 Systems for series combination of elastic elements with limited displacements: **a** - assembly diagram; **b** – resulting characteristic

		that have the <u>load characteristic</u> above the theoretical characteristic (without friction) and the return characteristic below it (fig. ARC-T.7.1). The mechanical work of deformation, L_i, is greater than the mechanical work returned upon relaxation, L_d, with the friction losses (hysteresis) being transformed into heat. <u>Stiffnesses of spring systems combined in parallel with limited displacements</u> (fig. ARC-T.7.4,b): $k_I = k_1, \quad (\text{ARC-T.7.10})$ $k_{II} = k_1 + k_2, \quad (\text{ARC-T.7.11})$ $k_{III} = k_1 + k_2 + k_3. \quad (\text{ARC-T.7.12})$
Fig. ARC-T.7.4 Parallel combination systems of elastic elements with limited displacements: a - assembly diagram; b – resulting characteristic		The quantitative evaluation of friction losses can be done through <u>the efficiency</u> $\eta = \frac{L_d}{L_i}, \quad (\text{ARC-T.7.15})$ or by <u>the depreciation coefficient</u> $c = \frac{L_i - L_d}{L_i + L_d}. \quad (\text{ARC-T.7.16})$
Observations 1. The characteristic in Fig. ARC-T.7.1,b is specific to springs with internal frictions that occur in the spring material or between the spring elements (e.g. leaf springs) when the loading and unloading curves are different. The loading characteristic is above the theoretical characteristic, the force at zero displacement is necessary to overcome static friction, when this force is exceeded, elastic deformation begins. During the return, in the first phase the decrease in the external force does not produce a decrease in the deflection as a result of the reaction of static friction forces to the change of direction. The mechanical work required to deform the spring (highlighted by the surface under the loading characteristic) is greater than the mechanical work to relax the spring (highlighted by the surface under the unloading characteristic) with the mechanical work of internal frictions (highlighted by the hatched surface). The accumulation of energy by the arc can be a harmful or favorable phenomenon, the assessment can only be made according to the intended purpose. 2. In general, in the calculation of linear springs, the expressions for the mechanical work of deformation (ARC-T.7.13/ ARC-T.7.14) can be arranged in the following forms: $L = \eta_a \frac{\sigma_{\max}^2}{E} V, \quad L = \eta_a \frac{\tau_{\max}^2}{G} V,$		

where, $\sigma_{\max}$ and $\tau_{\max}$ represent the maximum normal and tangential stresses, respectively, that occur in the spring material; E and G - the longitudinal and transverse elastic moduli, respectively; V - the volume of the spring material; η_a - the specific utilization coefficient depending on the distribution of stresses in the section, the shape and loading mode of the spring. In some situations, the volumetric utilization coefficient is used,

$$\eta_V = \frac{L}{V} = \eta_a \frac{\sigma_{\max}^2}{E}, \quad \eta_V = \frac{L}{V} = \eta_a \frac{\tau_{\max}^2}{G},$$

which indicates the overall degree of use of the material, from the point of view of energy accumulation; it is obvious that springs with high values of this coefficient are preferable from an energetic point of view

ARC-T.8 CALCULATION MODELS

ARC-T.8.1 CALCULATION MODELS OF CYLINDRICAL COMPRESSION/TENSION HELICAL SPRINGS

Tab. ARC-T.8.1.1 Calculation parameters of cylindrical compression/tension helical springs [Rădulescu, 1981]

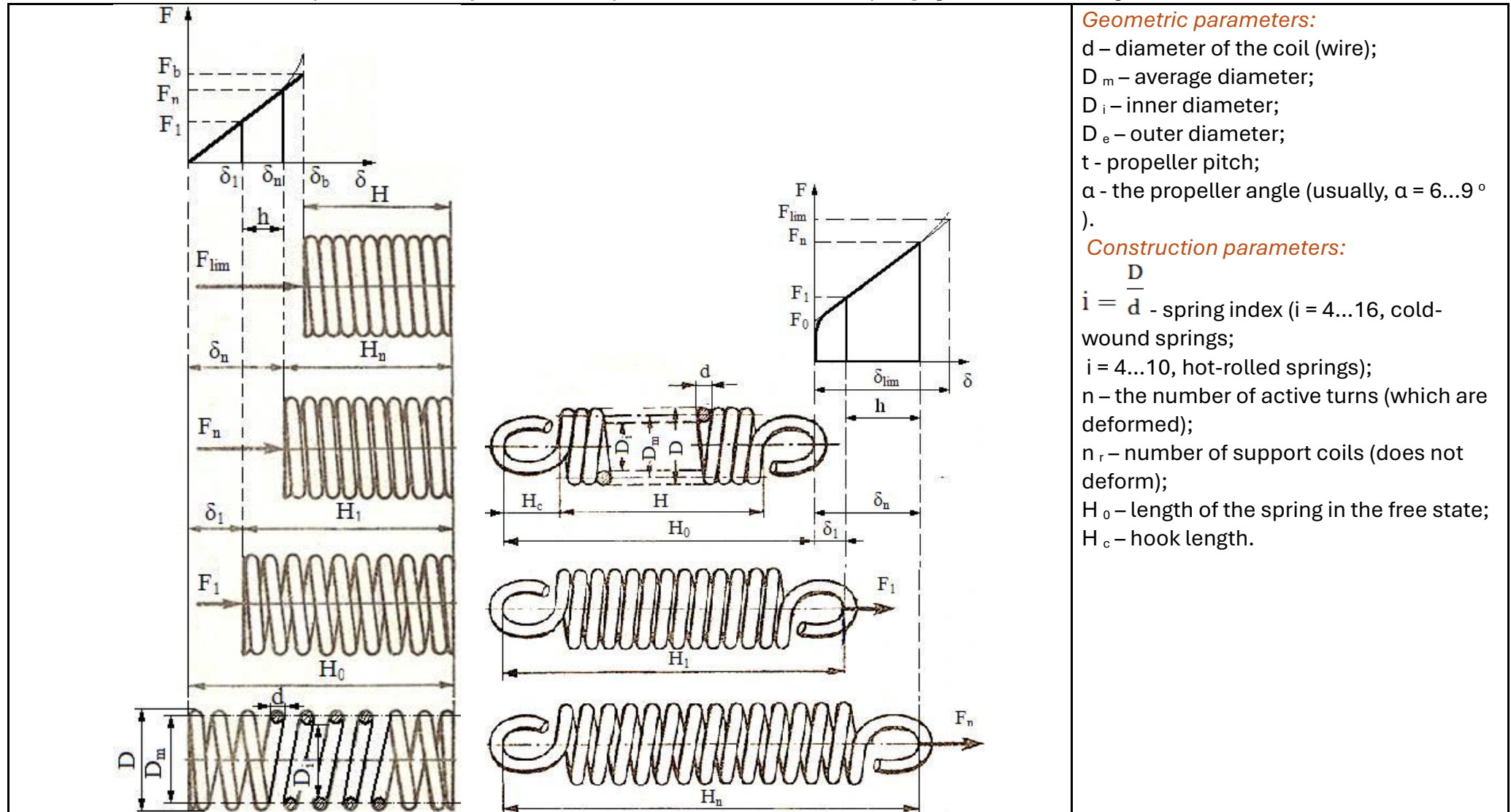


Fig. ARC-T.8.1.1 Calculation parameters of the cylindrical helical spring: **a** – compression; **b** – tension

Functional parameters:

δ_1, H_1 – the deflection, respectively the length of the mounted spring prestressed with the force F_1 ;

δ_n, H_n – the deflection, respectively the length of the spring at maximum load F_n ;

δ_b, H_b – the deflection, respectively the length of the spring when locked with the load F_b ;

δ_{lim} – maximum deflection of the spring (corresponding to the yield strength of the material, for the tension spring) at the limit load F_{lim} ;

h – spring stroke.

Tab. ARC-T.8.1.2 Calculation models for compression/tension helical springs

Stress and strain calculation model

Efforts and stresses in the spring coil		
effort	Calculation relationship	request
Torque	$M_t = F \cos \alpha \frac{D_m}{2}$ (ARC-T.8.1.1)	Torsion
Bending moment	$M_i = F \sin \alpha \frac{D_m}{2}$ (ARC-T.8.1.2)	Bending
Cutting force	$T = F \cos \alpha$ (ARC-T.8.1.3)	Shear
Normal force	$N = F \sin \alpha$ (ARC-T.8.1.4)	Compression / tension

Fig. ARC-T.8.1.2 Strength calculation model: **a** – spring loading; **b** – stresses in the spring coil

Simplifying assumptions: the propeller angle being small, $\alpha = 6...9^\circ$, it will be considered, $\sin \alpha \approx 0$, $\cos \alpha \approx 1$; shear, compression/traction and bending stresses are neglected; the main stress is torsion.

The torsional stress, considering the arc segment as a straight bar, is determined with the relationship,

$$\tau_t = \frac{M_t}{W_p} = \frac{F \frac{D_m}{2}}{\frac{\pi d^3}{16}} = \frac{8 F D_m}{\pi d^3}. \quad (\text{ARC-T.8.1.17})$$

Since the arc coil is a curved bar in space, the tangential tension τ_t is non-uniform and has a maximum of,

$$\tau_{t \max} = K \tau_t = K \frac{8 F D_m}{\pi d^3} = K \frac{8 F i}{\pi d^2} = K \frac{8 F i^3}{\pi D_m^2} \leq \tau_{at}, \quad (\text{ARC-T.8.1.18})$$

in the inner area of the coil; $K=1+1.6/i$ is the torsion stress concentration factor [Drăghici, 1982]

Model for calculating displacement (sag), stiffness and mechanical work

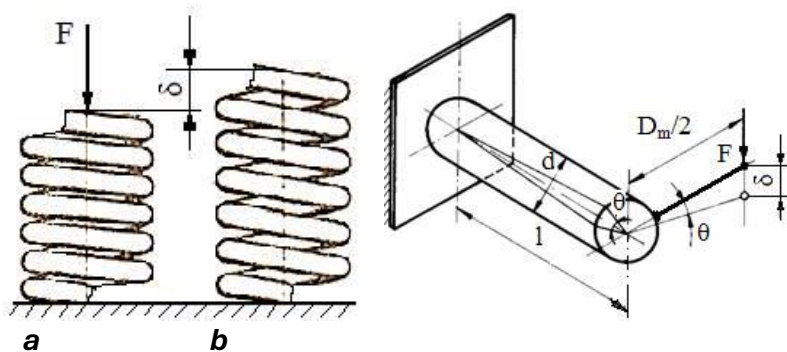


Fig. ARC-T.8.1.3 Strength calculation model: **a** – spring deformation; **b** – simplified model

Simplifying assumptions: the deflection δ , as a result of the deformation of the spring with the force F , is equal to the displacement of the force F as a result of the twisting by the angle θ of the straight bar of length $l = \pi D_m n$, which represents the unfolded spring; the characteristic of the spring is linear.

The arrow of the spring is determined by the relationship,

$$\delta = \frac{D_m}{2} \theta = \frac{D_m}{2} \frac{M_t l}{G I_p} = \frac{D_m}{2} \frac{\frac{F \pi D_m n}{G \frac{\pi d^4}{32}}}{\pi d^4} = \frac{8 F D_m^3 n}{G d^4} = \frac{8 F n i^3}{G d}, \quad (\text{ARC-T.8.1.19})$$

where G is the transverse modulus of elasticity, and I_p – the moment of inertia of the coil section.

Spring stiffness,

$$k = \frac{F}{\delta} = \frac{G d^4}{8 D_m^3 n} = \frac{G d}{8 n i^3}. \quad (\text{ARC-T.8.1.20})$$

The accumulated energy is equal to the mechanical work (linear characteristic, see tab. ARC-T.8.1.1, a),

$$L = \frac{F \delta}{2} = \frac{4 n D_m^3 F^2}{G d^4} = \frac{4 n i^3 F^2}{G d}. \quad (\text{ARC-T.8.1.21})$$

Note 1 The deflection of the tension spring prestressed with force F_0 (see fig. ARC-T1,b), loaded with force F ,

$$\delta = \frac{8 n D_m^3}{G d^4} (F - F_0) \quad (\text{ARC-T.8.1.22})$$

Note 3 Calculation relations for stress and deformation of helical springs with rectangular coil

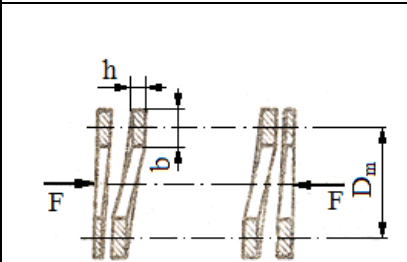


Fig. ARC-T.8.1.4 Parameters of the helical spring with rectangular helix

Maximum voltage,

$$\tau_{t \max} = \frac{F D_m}{k_1 b h^2}. \quad (\text{ARC-T.8.1.24})$$

Arrow,

$$\delta = \frac{D_m^3 n F}{k_2 G b h^3}. \quad (\text{ARC.8.1.25})$$

The factors k_1 and k_2 are dependent on the b/h ratio (fer. [ARC-F.1](#))

Note 2

The length of the wire (semi-finished product), necessary to make the spring,

$$L_s = \frac{\pi D_m n}{\cos \alpha} \quad (\text{ARC-T.8.1.23})$$

Note 4.

Stability (buckling) calculation of compression coil springs

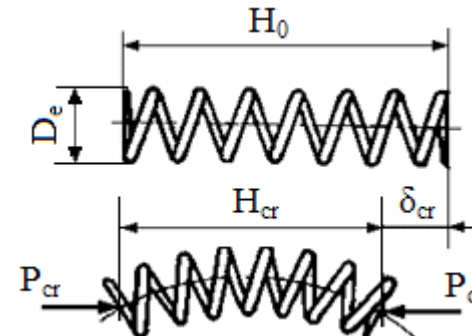


Fig. ARC-T.8.1.5 Buckling calculation parameters

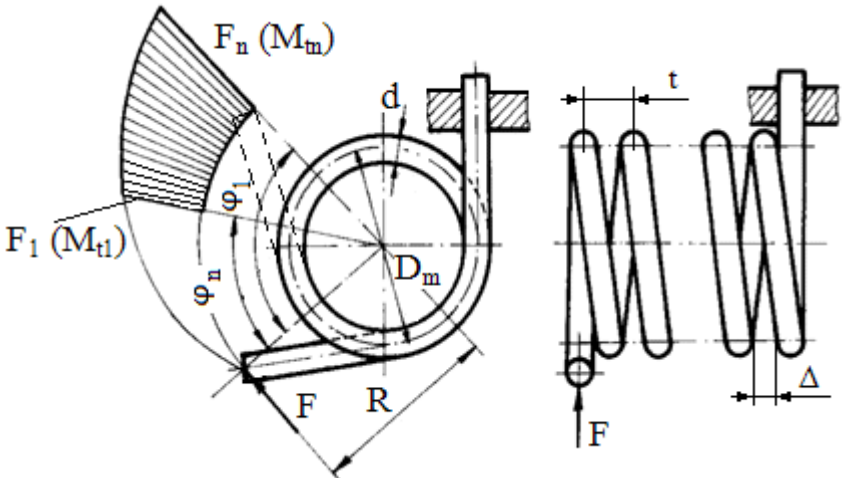
Simplifying hypothesis: the spring is assimilated to a straight bar stressed in compression (fig. ARC-T.8.1.5) in various support situations (fer. [ARC-P.1.1](#)).

Buckling calculation [Demian, 1980] involves determining the critical load, P_{cr} , dependent on the slenderness factor, H_0/D_m ; the springing factor, δ/H_0 ; the type of support (fer. [ARC-P.1.1](#)).

Note 5. Fatigue calculation, in the case of springs with variable loads (e.g. valve springs), involves determining safety factors with classical methods or with specific methods [Roloff / Matek, 2008]

ARC-T.8.2 CALCULATION MODELS FOR CYLINDRICAL TORSION COIL SPRINGS

Tab. ARC-T.8.2.1 Calculation parameters of cylindrical torsion helical springs

	Geometric parameters: d – diameter of the coil (wire); D_m – average diameter; t - propeller pitch; α - the propeller angle (usually, α = 6...9 °). Construction parameters: $i = D_m/d$, the arc index; n – the number of active turns (which are deformed); R – radius (length) of the load action arm. Functional parameters: F₁ (M_{t1}), φ₁ – the assembly force (torque), respectively the spring deflection during assembly; F_n (M_{tn}), φ_n – maximum force (torque), respectively maximum working deflection; φ_f – spring stroke (φ_f = φ_n – φ₁)
--	---

Tab. ARC-T.8.2.2 Calculation models for helical torsion springs

Stress and strain calculation model	
	Efforts and stresses in the spring coil

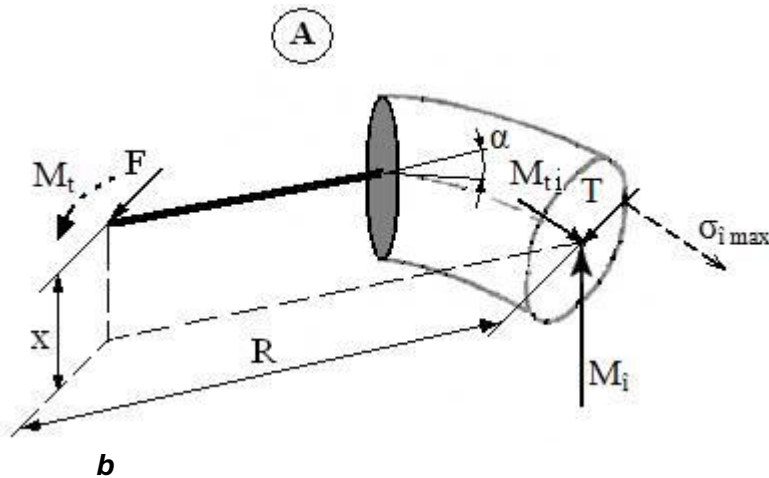
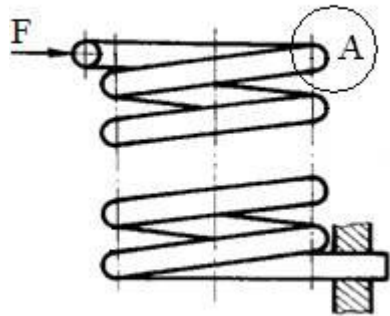


Fig. ARC-T.8.2.2 Strength calculation model: **a** – spring loading; **b** – stresses in the spring coil

Model for calculating displacement (sag), stiffness and mechanical work

effort	Calculation relationship	request
Bending moment	$M_i = F R$ (ARC-T.8.2.1)	Bending
Internal torsion moment	$M_{ti} = F x$ (ARC-T.8.2.2)	Torsion
Cutting force	$T = F$ (ARC-T.8.2.3)	Shear

Simplifying assumptions: shear stress is neglected; dimension x is reduced and, therefore, torsion is neglected; the main stress is bending with the maximum at the outside of the coil.

The bending stress, according to the straight bar bending model, is

$$\sigma_i = \frac{M_i}{W_z} = \frac{M_t}{\frac{\pi d^3}{32}} = \frac{32 M_t}{\pi d^3} = \frac{32 F R}{\pi d^3} \quad (\text{ARC-T.8.2.2})$$

Since the spring coil is a curved bar in space, the bending stress is non-uniform and, as a result of correcting the bending model of a straight bar, has a maximum

$$\sigma_{i \max} = K_i \sigma_i = K_i \frac{32 M_t}{\pi d^3} = K_i \frac{32 F R}{\pi d^3} \leq \sigma_{ai}, \quad (\text{ARC-T.8.2.3})$$

in the outer area of the coil; $K_i = (4i-1)/(4i-4)$ is the concentration factor of the maximum bending stresses on the outside of the coil ($i = D_m/d$) [Gafițanu, 1999]

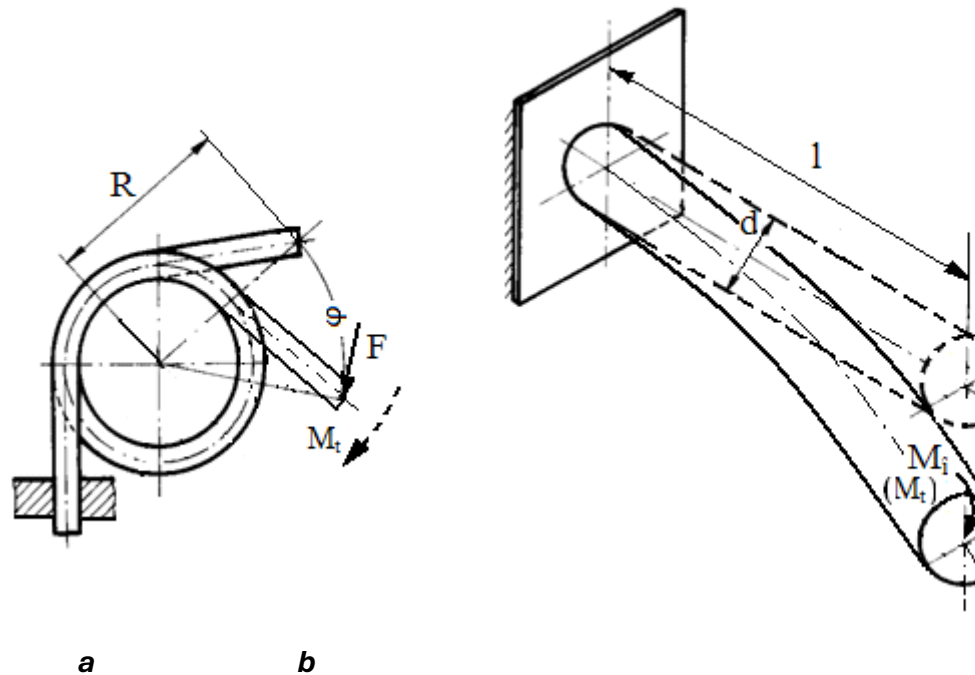


Fig. ARC-T.8.2.3 Arrow calculation model: **a** – spring deformation; **b** – simplified model

Simplifying assumptions: the rotation ϕ , as a result of the deformation of the spring under the action of the force F , is considered to be the angle by which a straight bar of length $l = \pi D_m n$, representing the unfolded spring, rotates; the characteristic of the spring is linear.

The arrow (rotation) of the arc is determined by the relationship,

$$\varphi = \frac{M_i l}{E I_z} = \frac{M_t \pi D_m n}{E \frac{\pi d^4}{64}} = \frac{64 M_t D_m n}{E d^4} = \frac{64 M_t n i}{E d^3} = \frac{64 F R n i}{E d^3}, \quad (\text{ARC})$$

where, E represents the longitudinal modulus of elasticity, and I_z – the axial moment of inertia of the coil section.

Spring stiffness,

$$k = \frac{M_i}{\varphi} = \frac{E d^4}{64 D_m n} = \frac{E d^3}{64 n i}. \quad (\text{ARC})$$

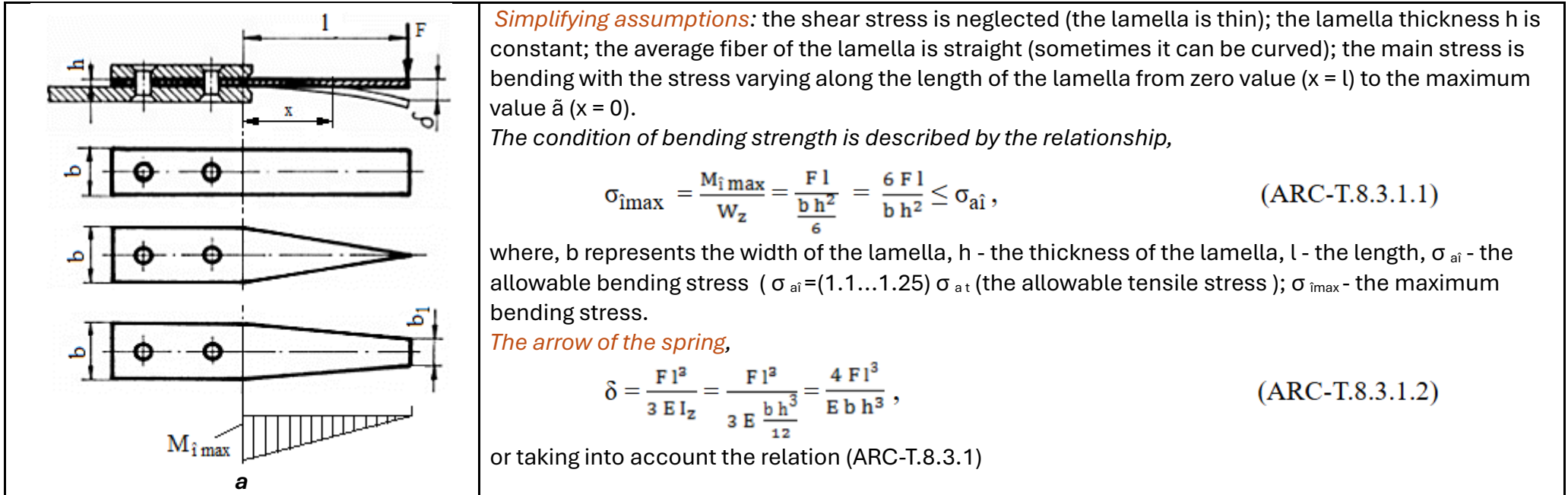
The **accumulated energy** is equal to the mechanical work (linear characteristic),

$$L = \frac{M_i \varphi}{2} = \frac{32 M_t^2 D_m n}{E d^4} = \frac{32 F^2 R^2 D_m n}{E d^4} = \frac{64 F^2 R^2 n i}{E d^3} \quad (\text{ARC})$$

ARC-T.8.3 CALCULATION MODELS FOR LEAF AND LAMELLAR SPRINGS

ARC-T.8.3.1. Straight single leaf springs

Tab. ARC-T.8.3.1 Calculation models for strength, stiffness and stored energy



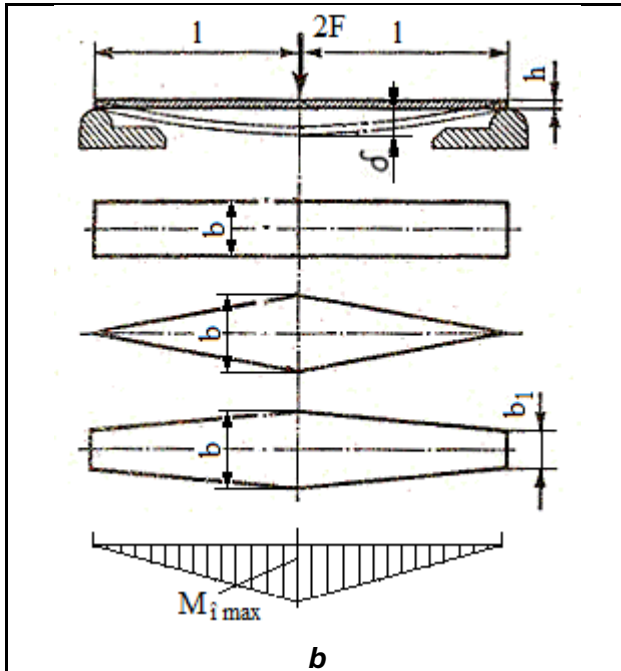


Fig. ARC-T.8.3.1 Calculation models of single leaf springs: **a** – embedded; **b** – doubly supported

$$\delta = \frac{2}{3} \frac{\sigma_{i\max} l^2}{E h},$$

(ARC-T.8.3.1.3)

where, E is the longitudinal modulus of elasticity of the material, I_z - axial moment of inertia of the spring section.

Spring stiffness,

$$k = \frac{F}{\delta} = \frac{F}{\frac{4Fl^3}{Ebh^3}} = \frac{Ebh^3}{4l^3}$$

(ARC-T.8.3.1.4)

is constant and therefore, the elastic characteristic is linear.

The accumulated energy is equal to the mechanical work (linear characteristic), also taking into account (ARC-T.8.3.1.1),

$$L = \frac{F\delta}{2} = \frac{F}{2} \frac{4Fl^3}{Ebh^3} = \frac{2F^2 l^3}{Ebh^3} = \frac{1}{18} \frac{\sigma_{i\max}^2}{E} V,$$

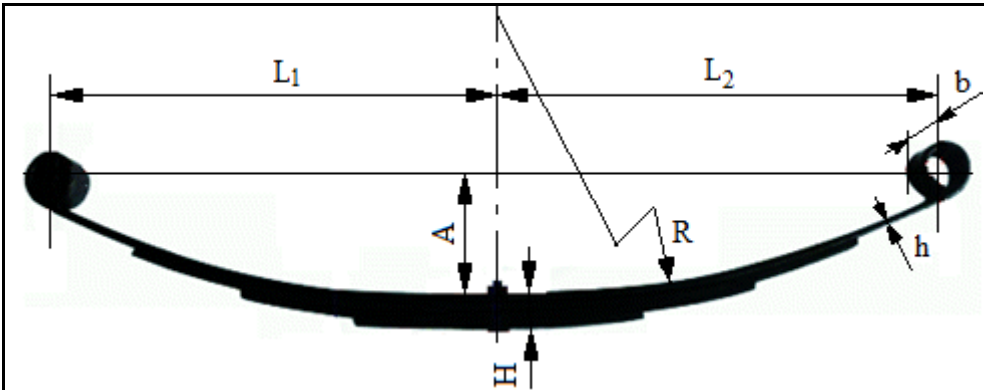
(ARC- T.8.3.1.5)

where, $V = bhl$ represents the volume of the spring; a low value of the specific utilization coefficient ($\eta_a = 1/18$) is highlighted, which shows low efficiency of energy accumulation.

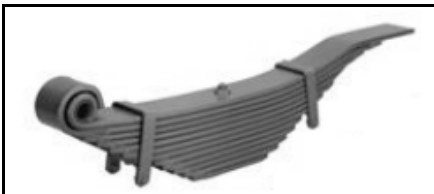
ARC-T.8.3. 2 Leaf springs (multi-leaf)

Leaf springs are elastic subsystems composed of multiple overlapping leaves that work (deform) simultaneously.

Tab. ARC-T.8.3.2.1 Construction, efficiency and geometry of leaf springs

	Description The first and longest leaf whose chord is equal to the span of the spring is called the <u>main leaf</u>. The other shorter leaves mounted below it are called <u>secondary leaves</u>. The leaves of the spring are arranged and sized so that the entire spring comes as close as possible to a <u>solid of equal bending resistance</u>. Constructive geometric parameters L_1 - length of the front/rear arm (left/right); L_2 - length of the rear/front arm (right/left); R - curvature; H - thickness; A - height; h - sheet thickness; sheet width
Fig. ARC-T.8. 3.2.1 Geometry of the leaf spring	Advantages: can also take up transverse forces; also have damping capacity; simple construction Disadvantages: increased weight (the ratio of specific potential energy, energy stored during deformation/volume, is lower than in the case of coil springs or torsion bars); reduced durability (service life).

Tab. ARC-T.8.3.2.1 Types of leaf springs

				
a	b	c	d	e
Fig. ARC-T.8. 3.2.2 Types of leaf springs: a – with sliding head; b – with bilinear stiffness (progressive); c – with variable stiffness; d – with prestressed (different initial curvatures); e – with parabolic shaped sheets				

Tab. ARC-T.8.3.2.1 Characteristics of leaf springs

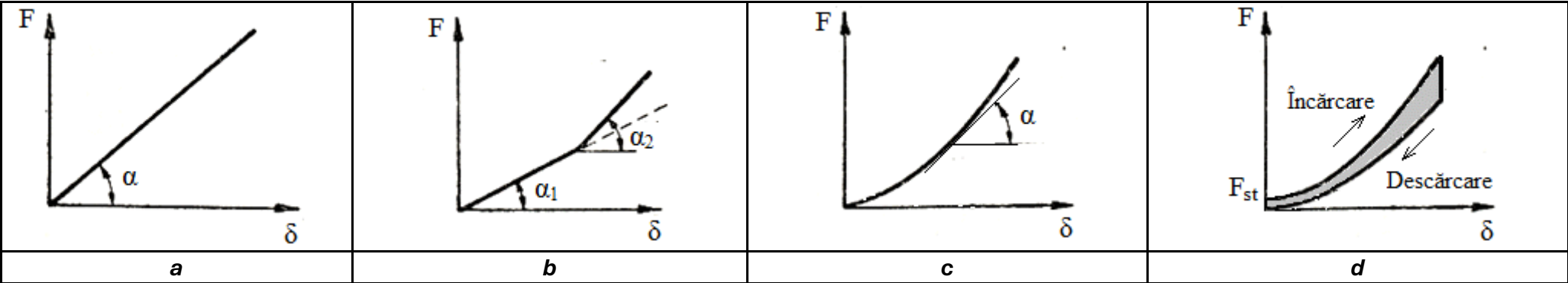


Fig. ARC-T.8. 3.2.3 Characteristics of leaf springs: **a** – linear (with constant stiffness); **b** – bilinear; **c** – nonlinear (with variable stiffness); **d** – nonlinear with hysteresis (damping)

Note.
Leaf springs they also have a damping capacity, generated by the internal friction between the blades. Friction in the spring occurs as a result of the relative sliding of the sheets; the spring begins to store internal energy only after the friction force is overcome, greater than F_{st} (fig. ARC-T.8. 3.2.3,d). Due to the friction between the leaves, the spring has the ability to attenuate shocks and vibrations. Shocks caused by external forces from below are transmitted directly, without damping, to the vehicle frame, worsening the comfort and stability of the ride; for this reason, leaf springs are not used for passenger vehicles.
To obtain the optimal (non-linear) characteristic, in the case of springs with linear characteristic, other springs with linear or non-linear characteristic (made of rubber) are used.
The nonlinear characteristic of leaf springs is also obtained by the variable (sliding) support of their ends.

Tab. ARC-T.8.3. 2 Resistance, stiffness and stored energy calculation models

a b	<i>Simplifying features:</i> shear stress is neglected (sheets are thin); the thicknesses of the lamellae h are constant; the average fibers of the lamellae can be straight or curved (ellipse, parabola); friction between the lamellae is neglected; the main stress is bending with variable tension along the lamellae. <i>The condition of bending strength,</i> $\sigma_{i \max} = \frac{M_{i \max}}{W_z} = \frac{F l}{n \frac{b h^2}{6}} = \frac{6 F l}{n b h^2} \leq \sigma_{ai}, \quad (\text{ARC-T.8.3.2.1})$ where, b represents the width of the sheets, h - thickness, l - length, n - number of sheets; σ_{ai} - allowable bending stress ($\sigma_{ai} = (1.1 \dots 1.25) \sigma_{at}$ (allowable tensile stress)). <i>The arrow of the spring,</i> $\delta = \frac{F l^3}{3 E I_z} = \frac{F l^3}{3 E n \frac{b h^3}{12}} = \frac{4 F l^3}{n E b h^3}, \quad (\text{ARC-T.8.3.2.2})$ or taking into account the relation (ARC-T.8.3.1) $\delta = \frac{2}{3} \frac{\sigma_{i \max} l^2}{E h}, \quad (\text{ARC-T.8.3.2.3})$ <i>Theoretical spring stiffness,</i> $k = \frac{F}{\delta} = \frac{F}{\frac{4 F l^3}{n E b h^3}} = \frac{n E b h^3}{4 l^3}, \quad (\text{ARC-T.8.3.2.4})$ is constant and therefore, the theoretical elastic characteristic is linear. <i>The accumulated energy is equal to the mechanical work (theoretical linear characteristic),</i> $L = \frac{F \delta}{2} = \frac{F}{2} \frac{4 F l^3}{n E b h^3} = \frac{2 F^2 l^3}{n E b h^3}, \quad (\text{ARC-T.8.3.2.5})$
Fig. ARC-T.8. 3.2 Calculation models of leaf springs: a – with a single arm; b – with two arms	

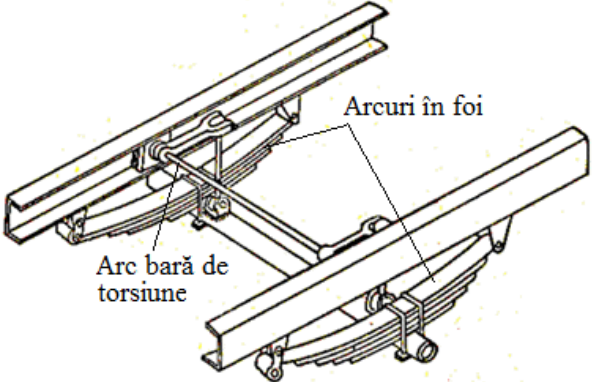
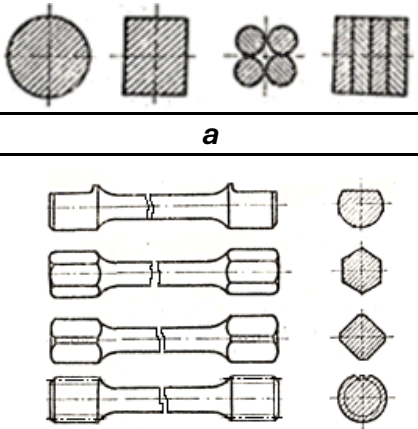
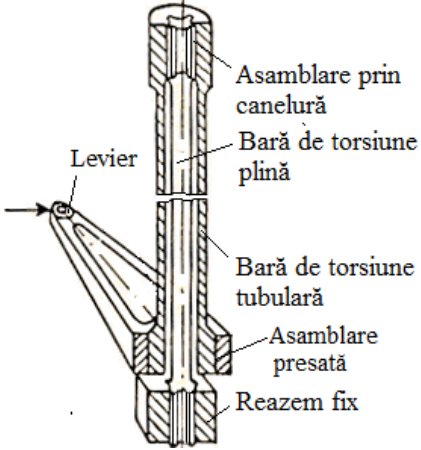
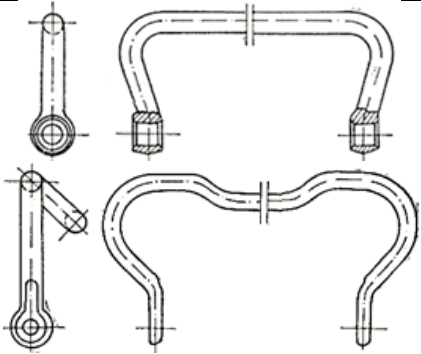
ARC-T.8.4 TORSION BAR SPRING CALCULATION MODELS

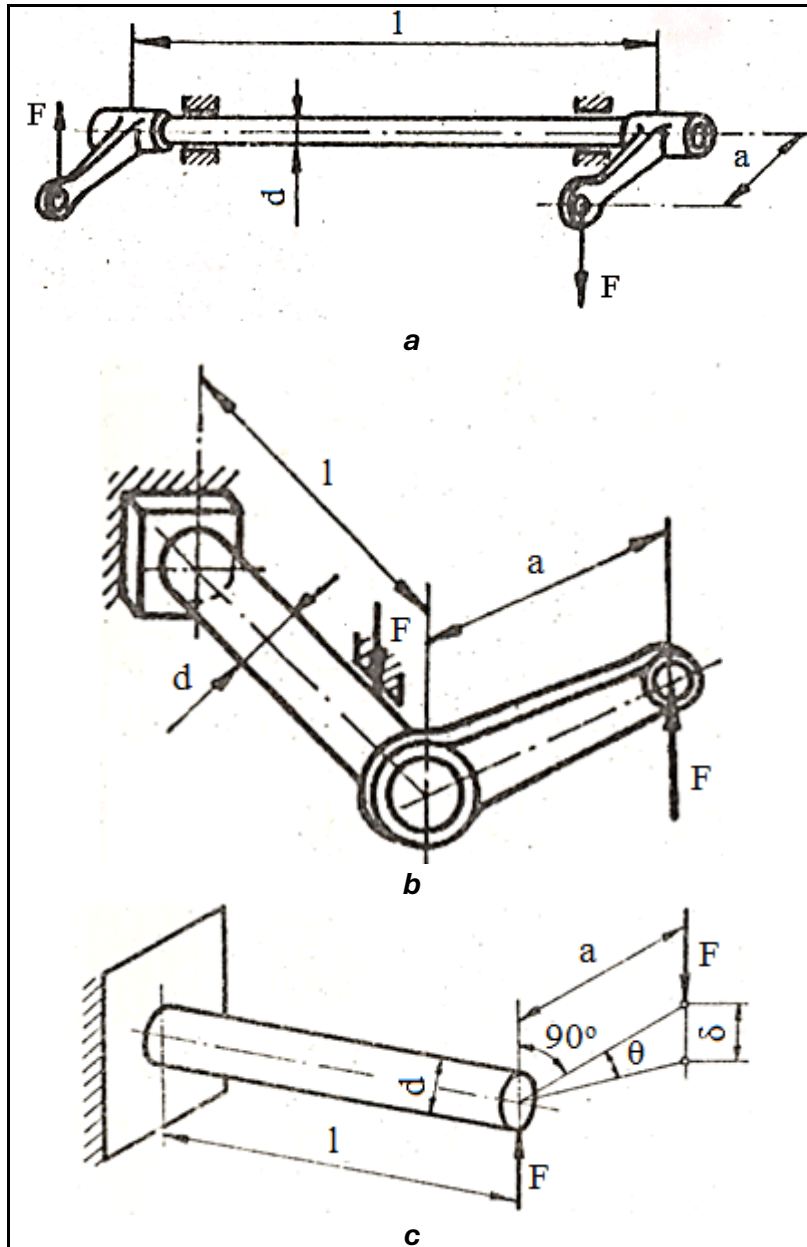
The torsion bar spring has the shape of a straight bar, usually of constant section along its entire length, actuated by forces applied to the ends of levers arranged at one (fig. ARC-T.8. 4. 3,b) or both ends (fig. ARC-T.8. 4. 1, fig. ARC-T.8. 4. 4,a). The sections of the springs can have simple or compound shapes (fig. ARC-T.8. 4. 2,a); simple sections can be round, square (fig. ARC-T.8. 4. 2,a), rectangular, hexagonal, annular, etc.; compound sections can be made up of several bars of round or rectangular section (fig. ARC-T.8. 4. 2,a). The shapes of the ends of the bar springs that are assembled with the levers can be: cylindrical with flattening, hexagonal, square, splined (fig. ARC-T.8. 4. 3,b).

Advantages: simple construction and technology; easy assembly and maintenance; no internal friction; relatively small radial dimensions; high load-bearing capacity (the energy stored per unit volume is approximately 4 times greater than a leaf spring).

The main disadvantage is the increased axial dimensions (for reduction, the combined variant can be adopted, fig. ARC-T.8. 4. 3).

In most cars, torsion bars are integrated as stabilizers (fig. ARC-T.8. 4. 4) with the role of reducing the inclinations of the car around the longitudinal axis; the stabilizers are arranged transversely to the longitudinal axis of the car and are attached to the guide arms of independent suspensions or to the axle in the case of dependent suspensions.

			
Fig. ARC-T.8. 4.1 Vehicle suspension with two leaf springs and a transversely mounted torsion bar spring [Boiangiu, 1958]	Fig. ARC-T.8. 4.2 Geometric shapes: a – of sections; b – of fixing ends	Fig. ARC-T.8. 4.3 Combined torsion bar spring (solid bar with tubular bar)	Fig. ARC-T.8. 4.4 U-shaped stabilizer bars

Tab. ARC-T.8.4. 1 Calculation models for strength, stiffness and stored energy

And simplifying moves:

- they do not take on longitudinal (axial) and/or transverse (shear and bending) forces as a consequence of mounting with support bearings (sliding bearings);
- deformations of the levers and the fixing head assemblies are not taken into account;
- the main stress is torsion.

Torsion resistance condition,

$$\tau_t = \frac{M_t}{W_p} = \frac{16 F a}{\pi d^3} \leq \tau_{at}, \quad (\text{ARC-T.8.4.1})$$

where, a represents the length of the lever, d - the diameter of the bar, F - the loading force, τ_{at} - the allowable torsional stress (600...800 MPa, for steel).

The arc deflection, taking into account the relationship (ARC-T.8.4.1), is determined with the relationships:

$$\delta \cong \theta a = \frac{M_t l}{G I_p} \quad a = \frac{M_t l}{G \frac{\pi d^4}{32}} \quad a = \frac{32 F a^2 l}{\pi d^4 G} = 2 \frac{l}{d} \frac{\tau_t}{G}, \quad (\text{ARC-T.8.4.2})$$

where, l represents the length of the bar; I_p - polar moment of inertia of the section; G - transverse modulus of elasticity; for the case $\tau_t = \tau_{at}$ at the maximum deflection is obtained from the torsional resistance condition.

The stiffness of the spring,

$$k = \frac{M_t}{\theta} = \frac{M_t l}{\frac{M_t l}{G I_p}} = \frac{G I_p}{l} = \frac{G \frac{\pi d^4}{32}}{l} = \frac{\pi d^4 G}{32 l}, \quad (\text{ARC-T.8.4.3})$$

The mechanical deformation work, taking into account the relationship (ARC-T.8.4.1), is determined with the relationships:

$$L = \frac{1}{2} M_t \theta = \frac{1}{2} M_t \frac{M_t l}{G I_p} = \frac{1}{2} F a \frac{F a l}{G \frac{\pi d^4}{32}} = \frac{16 a^2 l F^2}{\pi d^4 G} = \frac{1}{4} \frac{\tau_t^2}{G} V, \quad (\text{ARC-T.8.4.4})$$

where, $V = \pi d^2 l / 4$ represents the volume of the spring; a high value of the specific utilization coefficient ($\eta_a = 1/4$) is highlighted, indicating increased efficiency of energy accumulation

Fig. ARC-T.8. 4.5 Calculation models of torsion bar springs: **a** – with two arms; **b** – with a single arm

ARC-T.8.5 RUBBER SPRING CALCULATION MODELS

Description

Rubber springs are widely used in the construction of motor vehicles due to their special properties: high damping capacity (they take up to 40% of the mechanical deformation work), simple construction and technology, low price, safe and silent operation. These properties, on the one hand, are influenced by the ambient environment (temperature, humidity, chemical agents, etc.) and, on the other hand, diminish over time through the aging process.

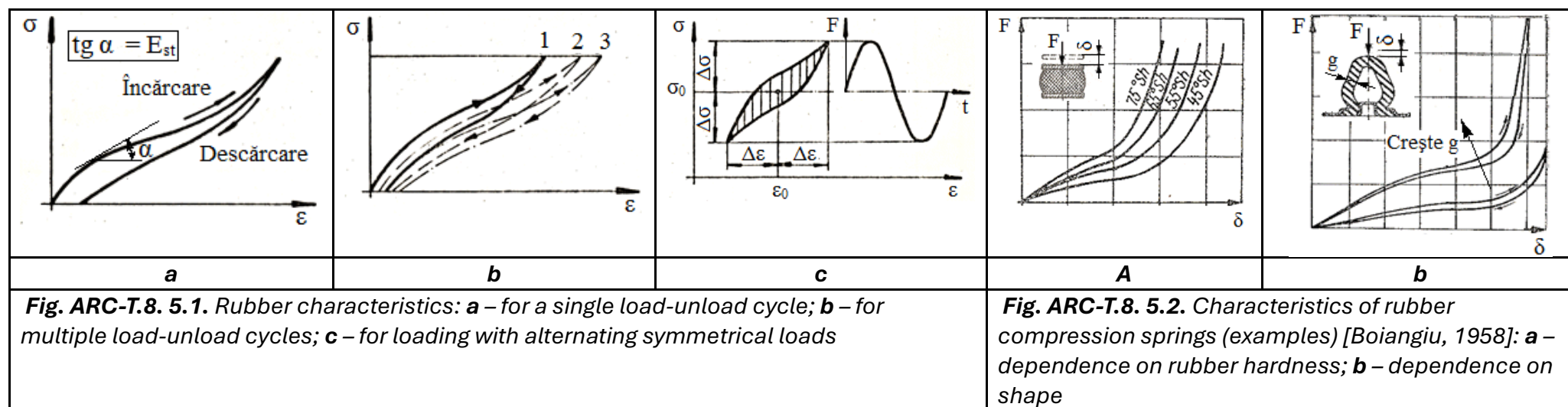
Rubber springs are frequently made from synthetic rubber (improved with additives) by vulcanization (pressing and heating in molds). By increasing the content of additives, the hardness and/or the degree of damping increases.

The characteristic of the material (rubber) is nonlinear (Fig. ARC-T.8. 5.1) with a variable static modulus of elasticity E_{st} (loading/unloading is done statically, at low speed); the unloading curve is located below the loading curve, forming a hysteresis loop whose surface determines the mechanical work of the internal frictions. Internal frictions are transformed into heat and therefore determine the heating of the spring.

Upon repeated loading and unloading, a shift in the spring characteristic occurs (Fig. ARC-T.8. 5.1, b) and, therefore, a decrease in the modulus of elasticity (stiffness).

In the case of prestressed (initially loaded) rubber springs subjected to cyclic variable loads (Fig. ARC-T.8. 5.1, c) the modulus of elasticity is different from the static one $E_{din} = \Delta\sigma/\Delta\varepsilon = k_{din} E_{st}$, where, $k_{din} = 1.5... 2$.

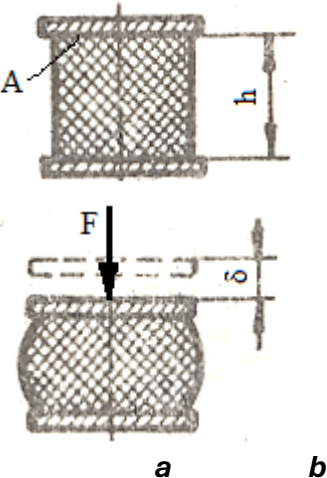
The characteristics of rubber springs depend on the hardness of the rubber (Fig. ARC-T.8. 5.2, a), its shape and dimensions (Fig. ARC-T.8. 5.2, b),



ARC-T.8.5. 1. Rubber compression springs

Compression rubber springs, often cylindrical or parallelepipedal in shape, are reinforced on the front faces with metal plates by vulcanization. These springs are frequently used as buffers to attenuate shocks and vibrations.

Tab. ARC-T.8.5. 1.1 Resistance and deflection calculation models

 b Fig. ARC-T.8. 5. 1.1 Calculation model: a – undeformed state; b – deformed state	<i>Simplifying points:</i> for <u>small deformations</u>, the linear characteristic is considered and Hooke's law is valid, $\sigma = E_c \varepsilon$ (E_c is the longitudinal modulus of elasticity of the rubber). <i>The compressive strength condition,</i> $\sigma_{cmax} = \frac{F}{A} = \frac{F}{\frac{\pi d^2}{4}} = \frac{4F}{\pi d^2} \leq \sigma_{ac} \quad (\text{ARC-T.8.5.1.1})$ where, A represents the area of the spring, σ_{ac} - the allowable compressive stress of the rubber ($\sigma_{ac} = 1...3$ MPa, for variable loads, $1...3$ MPa; $\sigma_{ac} = 3...5$ MPa, for static loads), d - the diameter of the spring with a circular section. <i>The arrow of the spring,</i> $\delta = \frac{Fh}{E_c A} = \frac{4}{\pi} \frac{Fh}{E_c d^2} = h \frac{\sigma_{cmax}}{E_c}, \quad (\text{ARC-T.8.5.1.2})$ where, h represents the length of the spring, $E_c = 3...5$ MPa; for $\sigma_{cmax} = \sigma_{ac}$ the maximum deflection is obtained from the condition of the compressive strength of the rubber. <i>The stiffness of the spring,</i> $k = \frac{F}{\delta} = \frac{\pi E_c d^2}{4h}. \quad (\text{ARC-T.8.5.1.3})$
--	---

ARC-T.8.5. 2. Rubber shear springs

Tab. ARC-T.8.5. 2.1 Strength and stiffness calculation models

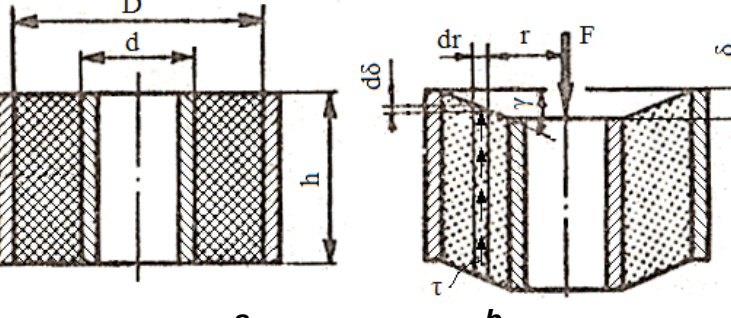
	<i>Simplifying points:</i> the bushing springs subjected to shear transmit the external force F through the vulcanized rubber on the two metal rings; the external force balances the sum of the shear stresses, considered uniformly distributed along the length, $F = A \tau = 2\pi r h \tau$; at <u>small deformations</u> they can be considered to have a linear characteristic and Hooke's law is valid, $\tau = G \gamma$, with G the transverse modulus of elasticity and $\gamma = F/GA$ - the angular deformation. <i>The maximum shear stress</i>, which appears on the contact (vulcanization) surface between the rubber and the inner metal ring ($r = d/2$) and <u>the shear strength condition</u>,
---	--

Fig. ARC-T.8. 4. 5.1 Calculation model: **a** – undeformed state; **b** – deformed state

$$\tau_{\max} = \frac{F}{\pi d h} \leq \tau_a,$$

(ARC-T.8.5.2.1)

where, τ_a

represents the allowable shear stress ($\tau_a = 1.5 \dots 4$ MPa).

The arc deflection, taking into account that, on the one hand, $\gamma \operatorname{tg} \gamma = d\delta/dr$ and, on the

other hand, $\gamma = F/GA$, is determined by the relationship, $\cong$

$$\delta = \int \frac{F}{GA} dr = \frac{F}{G} \int \frac{dr}{2\pi r h} = \frac{F}{2\pi h G} \int_{d/2}^{D/2} \frac{dr}{r} = \frac{F}{2\pi h G} \ln \frac{D}{d}. \quad (\text{ARC-T.8.5.2.2})$$

The stiffness of the spring,

$$k = \frac{F}{\delta} = \frac{2\pi h G}{\ln \frac{D}{d}}, \quad (\text{ARC-T.8.5.2.3})$$

where, G represents the transverse modulus of elasticity

ARC-T.8.5. 3. Rubber torsion springs

Tab. ARC-T.8.5. 3.1 Strength and stiffness calculation models

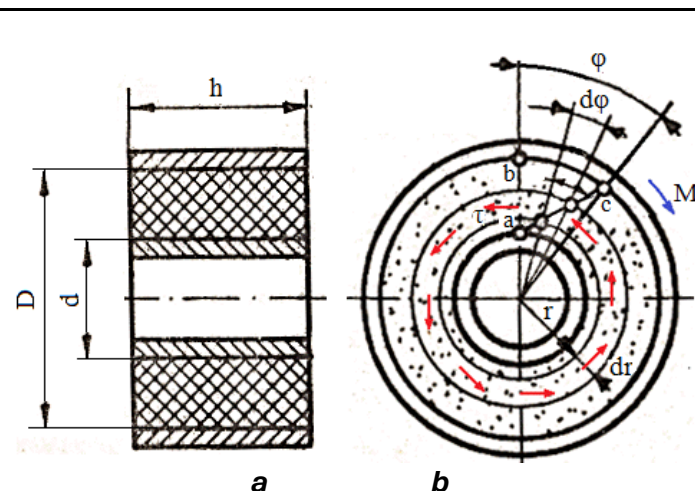


Fig. ARC-T.8. 5. 3.1 Calculation model: **a** – undeformed state; **b** – deformed state

Simplifying points: the torsionally stressed bushing springs transmit the external moment M_t through the vulcanized rubber on the two metal rings; the torsional moment balances the sum of the shear stresses, considered uniformly distributed on the circumference, $M_t = A \tau r = 2\pi r h \tau r = 2\pi r^2 h \tau$; at small deformations they can be considered to have a linear characteristic and Hooke's law is valid, $\tau = G \gamma$, with G the transverse modulus of elasticity and γ - the angular deformation.

The maximum shear stress, which appears on the contact (vulcanization) surface between the rubber and the inner metal ring ($r = d/2$) and the shear strength condition,

$$\tau_{\max} = \frac{M_t}{2\pi \left(\frac{d}{2}\right)^2 h} = \frac{2 M_t}{\pi d^2 h} \leq \tau_a, \quad (\text{ARC-T.8.5.3.1})$$

where, τ_a represents the allowable shear stress ($\tau_a = 1.5 \dots 4$ MPa).

The arrow (rotation) of the arc, taking into account that, on the one hand, $\gamma \approx \operatorname{tg} \gamma = r d\phi/dr$ and, on the other hand, $\gamma = \tau/G$ (according to the above hypothesis), is determined by the relationship,

$$\phi = \int \gamma \frac{dr}{r} = \int \frac{\tau}{G} \frac{dr}{r} = \int \frac{M_t}{2\pi r^2 G h} \frac{dr}{r} = \frac{M_t}{2\pi G h} \int_{d/2}^{D/2} \frac{dr}{r^2} = \frac{M_t}{\pi h G} \left(\frac{1}{d^2} - \frac{1}{D^2} \right). \quad (\text{ARC-T.8.5.3.2})$$

The stiffness of the spring,

	$k = \frac{M_t}{\varphi} = \frac{\pi \, h \, G}{\frac{1}{d^2} - \frac{1}{D^2}}$ (ARC-T.8.5.3.3)
--	--

CUP. MECHANICAL COUPLINGS

CUP-T. CONSTRUCTION AND MODELING OF MECHANICAL COUPLINGS

CONTENTS

CUP-T.1 DEFINITION	9-3
CUP-T.2 CONSTRUCTION STRUCTURES	9-3
CUP-T.3 AREAS OF USE, ADVANTAGES AND DISADVANTAGES	9-4
CUP-T.4 CLASSIFICATION	9-4
CUP-T.5 MATERIALS AND TECHNOLOGIES.....	9-6
CUP-T.6 FORMS AND CAUSES OF DISPOSAL OR INAPPROPRIATE BEHAVIOR	9-6
CUP-T.7 FUNCTIONAL AND CONSTRUCTIVE PARAMETERS	9-6
CUP-T.8 FUNCTIONAL AND CALCULATION MODELS.....	9-8
CUP-T.8.1 Permanent couplings	9-8
CUP-T.8.1.1 Fixed permanent couplings	9-8
CUP-T.8.1.2 Permanent rigid mobile couplings	9-11
CUP-T.8.1.2.1 Rigid mobile permanent couplings for compensation of axial deviations	9-11
CUP-T.8.1.2.2 Rigid mobile permanent couplings for compensating radial (transverse) deviations.....	9-11
CUP-T.8.1.2.3 Rigid mobile permanent couplings for compensation of angular or angular-axial deviations.....	9-13
.....	9-17
CUP-T.8.1.2.4 Rigid mobile permanent couplings for compensation of combined deviations (axial, radial and angular)	9-17
CUP-T.8.1.3 Permanent elastic mobile couplings	9-19
CUP-T.8.2 INTERMITTENT COUPLINGS	9-23
CUP-T.8.2.1 Controlled intermittent couplings	9-23

CUP-T.1 DEFINITION

Mechanical couplings are subassemblies that create a permanent or intermittent connection between two elements of one or two transmissions with the main function of *transmitting rotational motion and torque*, and with one or more secondary functions:

- *compensation of position deviations* (axial, radial, angular and combined) of connected elements due to execution and/or assembly errors;
- *shock and vibration damping, limiting the transmitted load* (torque);
- *speed limitation*;
- *coupling/decoupling* of linked elements;
- *interruption of load transmission* when changing direction.

CUP-T.2 CONSTRUCTION STRUCTURES

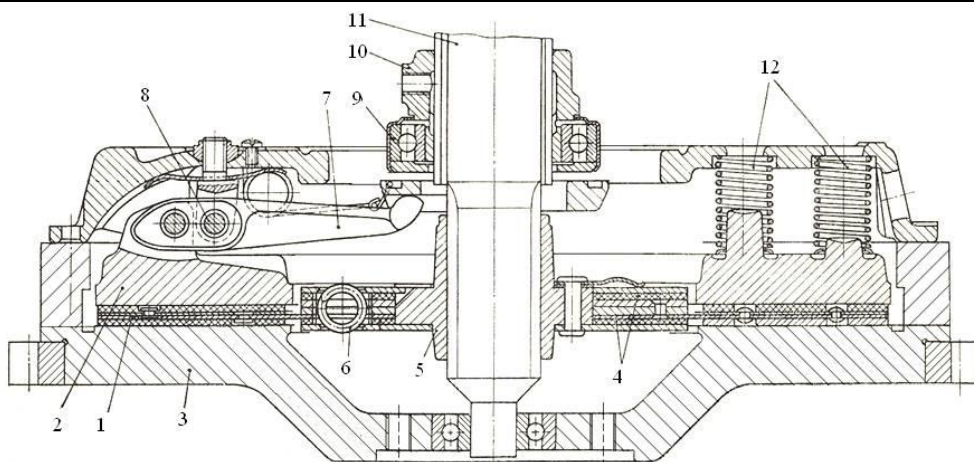


Fig. CUP-T.1 Construction structure of the mechanically controlled intermittent coupling (clutch)

Operation and construction. In the normal state, *engaged (engaged)*, the torque is transmitted *by friction* from the support disk 3 (usually the engine flywheel) to the driven disk 1 - pressed between the pressure disk (plate) 2 by the action of pre-compressed springs 12 - then, through the torsional damper, composed of tangential helical springs 6 and friction plates 5, to the hub 5 and through the spline to the output shaft 11 (usually the primary shaft of the gearbox).

Disengaging (disengaging) is achieved by axial movement of the sleeve 10, which, through the pressure bearing 7, actuates the ends of the release levers 7 which, by rotating around the bolts 8, cause the pressure disk (plate) 2 to be removed from the flywheel 3 and, finally, the interruption of the transmission of torque. The driven disc 1 is a subassembly composed, on the outside, of a steel support disc plated with discs of friction materials and, on the inside, of the helical springs 6, arranged equiangularly tangent to a circle, and the friction plates 4 which form the torsional damper which attenuates the oscillations of the torque that appear due to the shocks and non-uniformities induced by the external dynamic loads. Once

the action of the control coupling 10 ceases, the levers 7 are released and the springs 12 press the pressure disc (plate) 2, achieving *the coupling (clutching)*.

CUP-T.3 AREAS OF USE, ADVANTAGES AND DISADVANTAGES

Transmissions of motor vehicles, agricultural machinery, tractors, technological equipment, machine tools, etc. In general, they are used in almost all branches of mechanical engineering in a wide variety of constructive forms.

Tab. CUP-T.1 *Advantages and disadvantages of mechanical couplings*

Advantages *	Disadvantages *
- constructive simplicity; - overload protection; - requires low execution precision and relatively cheap assembly; - smaller dimensions	- limited transmission capacity; - slippage (in the case of force transmission through friction);
* compared to other types of couplings (hydraulic and electromagnetic)	

CUP-T.4 CLASSIFICATION

Tab. CUP-T.2 *Classification of mechanical couplings*

<i>Types of mechanical couplings</i>				<i>Description</i>
Mechanical couplings	Permanent	fixed		They transmit movement and torque permanently; they do not compensate for deviations; they do not dampen shocks and vibrations.
		Mobile	Rigid	With axial compensation (fig. CUP-T.4.1,b)
			Rigid	Transmits movement and torque permanently; compensates for axial deviations through relative movements between rigid elements; does not dampen shocks and vibrations
				With radial compensation (fig. CUP-T.4.1,c)
				With angular compensation (fig. CUP-T.4.1,d)
				Transmits movement and torque permanently; compensates for angular deviations through relative movements between rigid elements; does not dampen shocks and vibrations

				With combined compensation (fig. CUP-T.4.1,e)	Transmits movement and torque permanently; compensates for combined deviations (radial, axial and angular) through relative movements between rigid elements; does not dampen shocks and vibrations
			SPRING	With metal elastic element	They transmit movement and torque permanently; compensate for combined deviations (radial, axial and angular) by deforming elastic metallic elements; dampen shocks and vibrations
				With non-metallic elastic element	Transmits movement and torque permanently; compensates for combined deviations (radial, axial and angular) and deformation of non-metallic elastic elements; dampens shocks and vibrations
	Intermittent	ordered (clutches)	Mechanical control		Transmits movement and torque intermittently; mechanical control
			Hydrostatic control		Transmits movement and torque intermittently; hydrostatic control
			Pneumatically controlled		Transmits movement and torque intermittently; pneumatic control
			Electromagnetic control		Transmits movement and torque intermittently; electromagnetic control
		automatic	Load limiters		Transmits motion and torque intermittently; interrupts motion transmission when a set load is exceeded
			Speed limiters		Transmits motion and torque intermittently; interrupts motion transmission when a set speed is exceeded
			Meaning limiters		Transmits motion and torque intermittently; interruption of motion transmission when changing direction of rotation

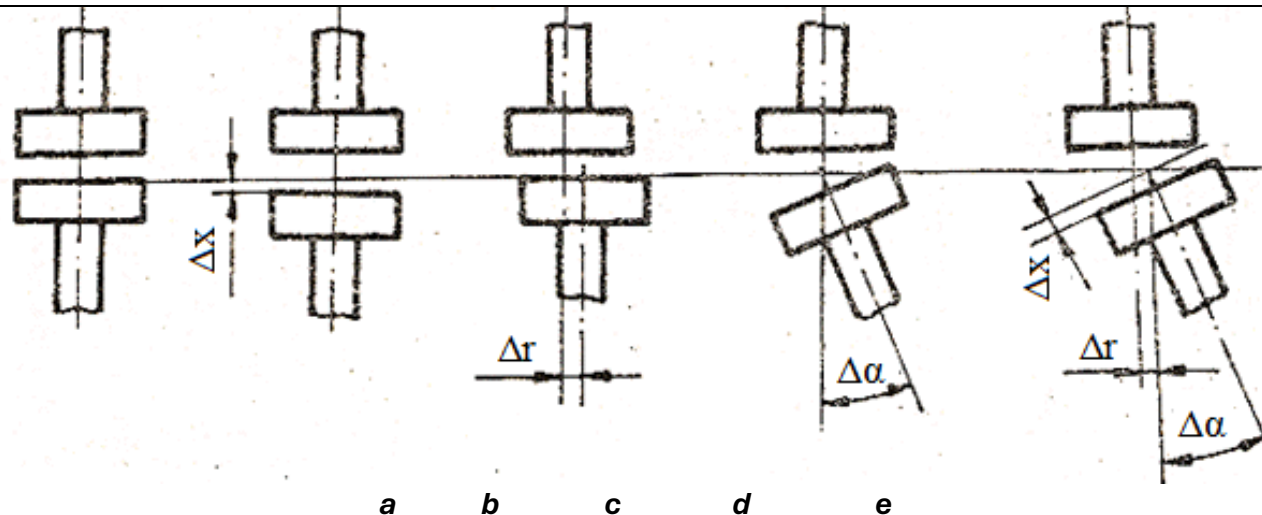


Fig. CUP-T.4.1 Deviations from coaxiality of shafts connected by couplings: **a** - reference position; **b** - axial; **c** - radial; **d** - angular; **e** - combined

CUP-T.5 MATERIALS AND TECHNOLOGIES

CUP-T.6 FORMS AND CAUSES OF DISPOSAL OR INAPPROPRIATE BEHAVIOR

Tab. CUP-T.3 Forms and causes of inappropriate behavior or removal from use

Shapes	Result	Appearance	Cause	Manifestation	Avoidance
Damage to active contact surfaces	Improper behavior (with irregularities, vibrations and noises)	Especially in couplings that transmit load through friction	Active wear of the blades	Uneven movements with vibrations; increased slippage (skating)	Limitation of permissible contact voltages by calculation; appropriate mounting.
Breaking of active elements	Disposal	Especially in couplings that transmit the load through the shape	Exceeding the static or fatigue strength of the material	Interruption of movement transmission	Limiting maximum static or fatigue stresses to values admissible by calculation; avoiding operating overloads

CUP-T.7 FUNCTIONAL AND CONSTRUCTIVE PARAMETERS

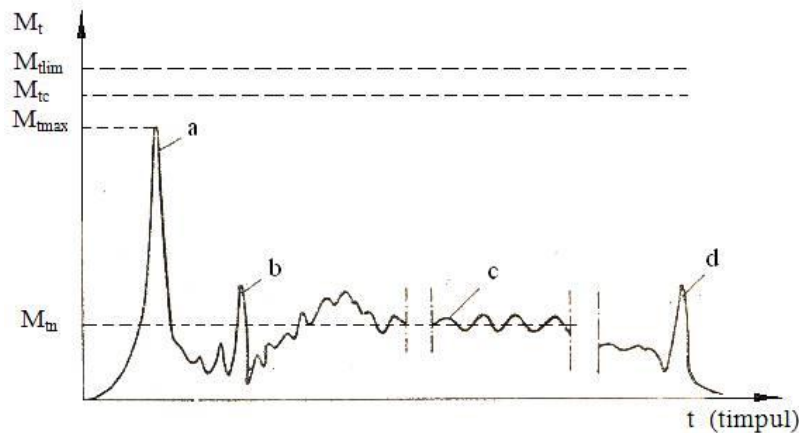


Fig. CUP-T.7.1 Variation of the torque

Main functional, constructive and calculation parameters

The torque transmitted by a coupling is based on a time-varying function (fig. CUP. T.7.1) determined by the real operating conditions which, in general, can be identified by the following periods: **a** – the start-up shock determined by the inertial starting loads (acceleration); **b** – passage through the resonance zone; **c** – continuous operation in stationary mode; **d** – the stop shock determined by the inertial deceleration loads. The nominal torque (load), M_{tn} , represents the highest average torque that occurs in steady-state continuous or intermittent operation. The design torque (choice), M_{tc} , represents the theoretical torque for design calculations that also takes into account additional loads to the nominal one: inertial, shocks, elastic forces due to forced deformation of the coupling components, internal friction forces between the moving components of the coupling; these loads, difficult to evaluate

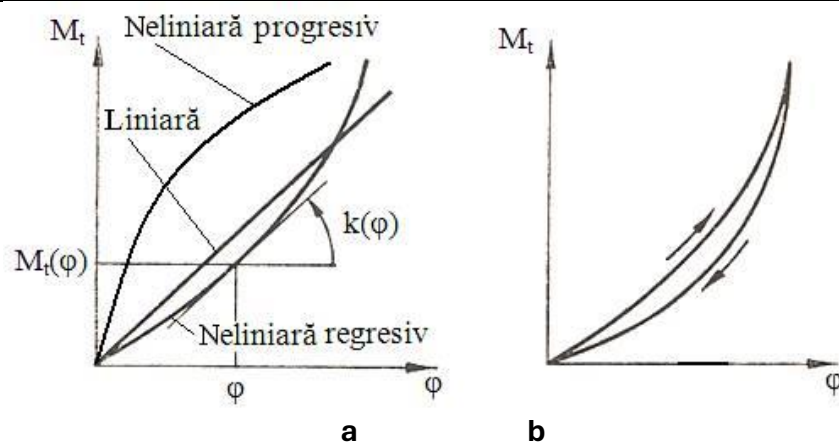


Fig. CUP-T.7.2 Characteristics of elastic couplings: **a** - without damping; **b** - with damping

quantitatively, are taken into account by means of a safety coefficient, K_s , which is determined based on experimental data taking into account the type of motor machine, the coupling, the type and operating mode of the driven machine. The calculation relationship for the design torque, $M_{tc} = K_s M_{tn}$, (CUP-T.1) contains the coefficient, $K_s > 1$, dependent on the specific operating cases (see fer. CUP.P.1).

The limit torque (only for safety couplings), as the maximum value for protecting the system, can generally be determined with the relationship, $M_{tlim} = K_{si} M_{tn}$, (CUP-T.2)

where, $K_{si} = 1.05 \dots 1.25$, represents the safety factor upon interruption which is determined depending on specific operating cases.

The elastic characteristic (only for elastic couplings), given by the variation of the torque as a function of the rotation angle, can be linear, with constant stiffness, or nonlinear (progressive or regressive, fig. CUP-T.7.2), with variable stiffness.

Torsional rigidity (only for elastic couplings), as a variation of the torsional moment in relation to the variation of the relative rotation angle of the coupling halves (fig. CUP-T.3), can be constant (linear characteristic, fig. CUP-T.3,a) and variable (non-linear characteristic) the same at loading and unloading (fig. CUP-T.7.2,a) or different (fig. CUP-T.7.2,b) at loading and unloading.

$$k(\phi) = \frac{dM_t(\phi)}{d\phi}$$

The capacity (degree of damping) is determined by the relationship,

$$d = L_a / L_e,$$

where: L_a represents the damping work (the area between the two curves in Fig. CUP-T.7.2,b) and L_e the elastic deformation work

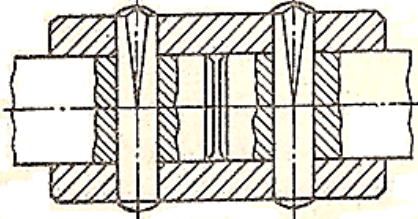
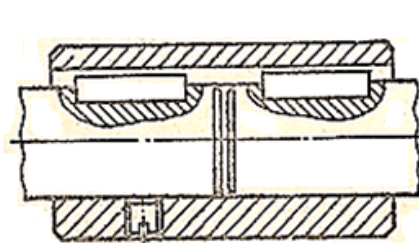
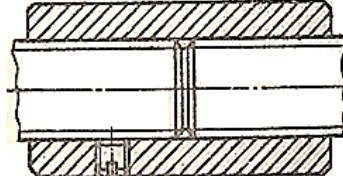
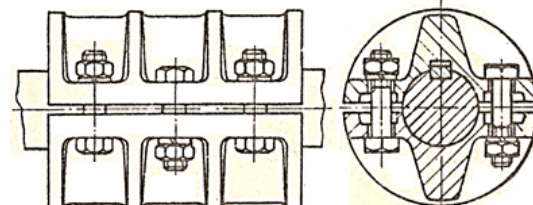
CUP-T.8 FUNCTIONAL AND CALCULATION MODELS

CUP-T.8.1 PERMANENT COUPLINGS

CUP-T.8.1.1 Fixed permanent couplings

CUP-T.8.1.1.1 Permanent fixed couplings with sleeves (muff couplings)

These couplings create a permanent connection between the coupled elements (shafts) and transmit the torque through a monobloc sleeve, by shape - through assemblies with transverse pins (fig. CUP-T.8.1.1.1.1,a), through parallel keys (fig. CUP-T.8.1.1.1.1,b), through splines (fig. CUP-T.8.1.1.1.1,c) etc. - or by friction (fig. CUP-T.8.1.1.1.2).

			
a	b	d	Fig. CUP-T.8.1.1.1.2 Permanently fixed coupling with assembled sleeve
Fig. CUP-T.8.1.1.1.1 Fixed permanent couplings with monobloc sleeve: a - with transverse cylindrical pins; b - with parallel keys; c - with splines;			

CUP-T.8.1.1.2 Permanent fixed couplings with flanges

Permanently fixed couplings with bolted flanges mounted with clearance	Permanently fixed couplings with bolted flanges mounted without play
--	--

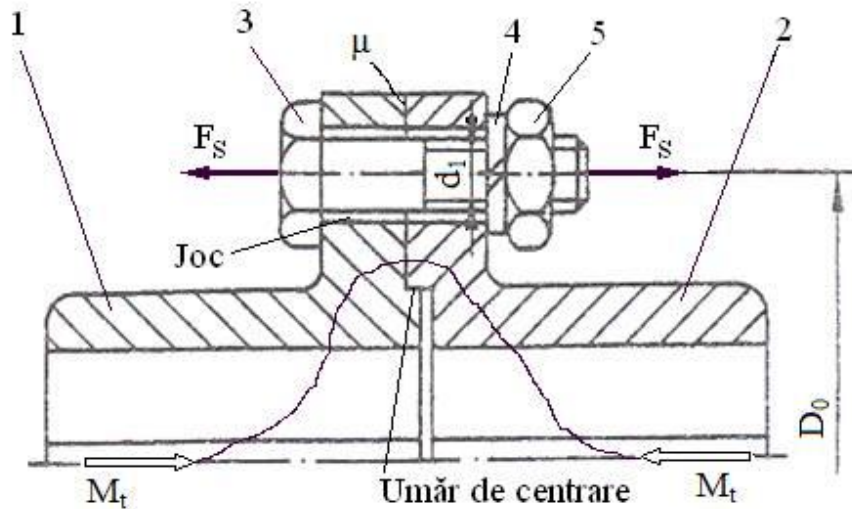


Fig. CUP-T.1.1.2.1 Flange coupling with bolts mounted with clearance

Function and construction: the torque is transmitted from the driving element by parallel key assembly to the driving half-coupling, 1, and by friction to the driven half-coupling, 2, and, likewise, by parallel key assembly to the driven element; the equiangularly arranged threaded assemblies are composed of the screw 3, mounted with clearance, and the nut, 5, which ensures the normal pressing force, F_s , necessary for transmitting the load by friction; the Grower washer 4 with the role of securing the assembly against unscrewing.

Condition for torque transmission through friction:

$$M_{tc} \leq M_f = \mu F_s z_s \frac{D_0}{2}, \quad (\text{CUP-T.3})$$

where M_{tc} represents the design torsion moment (see rel CUP-T.1), M_f – friction moment, μ – friction coefficient between the two half-couplings, F_s – tightening force of a threaded assembly, z_s – number of bolts, D_0 – diameter of the threaded assembly arrangement circle. From the relation (AEC-C.1) the force required for the assembly calculation can be determined,

$$F_s = \frac{2 M_{tc}}{\mu z_s D_0} \quad (\text{CUP-T.4})$$

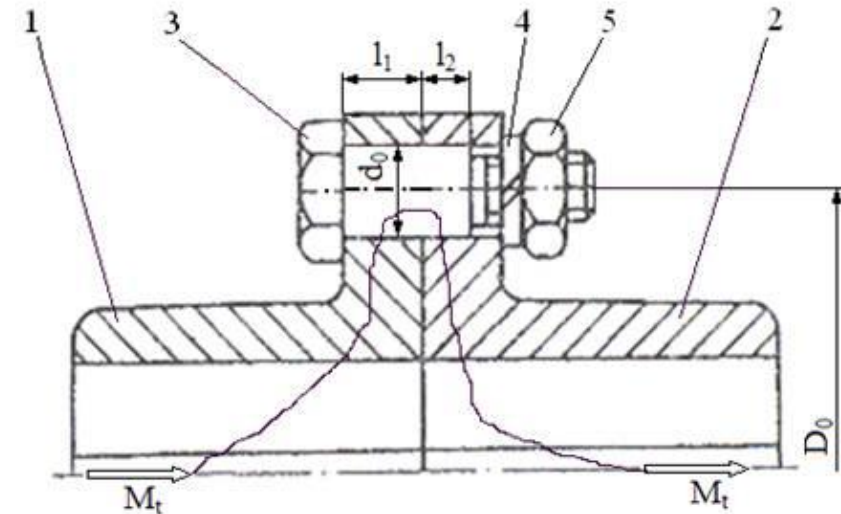


Fig. CUP-T.1.1.2.2 Flange coupling with backlash-free mounting bolts

Function and construction: the torque is transmitted from the driving element by parallel key assembly to the driving half-coupling, 1, and by shape to the screw rods 3 from which, likewise, by shape to the driven half-coupling 2, and, further, by parallel key assembly to the driven element; the equiangularly arranged threaded assemblies are composed of the screw 3, mounted without play (stepped), and the nut, 5, which ensures the minimum force to prevent the assemblies from disassembling during operation; the Grower washer 4 with the role of securing the assembly against unscrewing.

Condition for transmitting torque through the shape (crushing resistance):

$$\sigma_s = \frac{F_s}{A_s} = \frac{\frac{2 M_{tc}}{z_s D_0}}{d_0 \min(l_1, l_2)} = \frac{2 M_{tc}}{z_s D_0 d_0 \min(l_1, l_2)} \leq \sigma_{as}, \quad (\text{CUP-T.5})$$

where, M_{tc} is the design torsion moment (see rel. CUP-T.1), σ_s – the maximum crushing stress, σ_{as} – the allowable crushing stress of the material pair (half-couplings-screw), D_0 – diameter of the circle for arranging the threaded assemblies, d_0 – diameter of the screw (stepping) rod, l_{11} – length of the screw rod section in contact with the driving half-coupling, l_2 – length of the screw rod section in contact with the driven half-coupling, z_s – number of assemblies. From the

relation (CUP-T.5) it is possible to determine the screw stepping diameter d_0 or the minimum contact length for the assembly

calculation. σ_s σ_{as}

Shear verification condition of the screw rod:

$$\tau_f = \frac{F_f}{A_{af}} = \frac{\frac{2 M_t}{z_s D_0}}{\frac{\pi d_0^2}{4}} = \frac{8 M_{tc}}{\pi z_s D_0 d_0^2} \leq \tau_{af} \quad (\text{CUP-T.5'})$$

where τ_{af} represents the allowable shear stress of the bolt material

Note: Flange couplings with bolts mounted without clearance, at the same load, have smaller dimensions than those with bolts mounted with clearance; on the other hand, flange couplings with bolts mounted with clearance are cheaper, an advantage that leads to their frequent use in practice.

CUP-T.8.1.2 PERMANENT RIGID MOBILE COUPLINGS

CUP-T.8.1.2.1 Rigid mobile permanent couplings for compensation of axial deviations

CUP-T.8.1.2.2 Rigid mobile permanent couplings for compensating radial (transverse) deviations

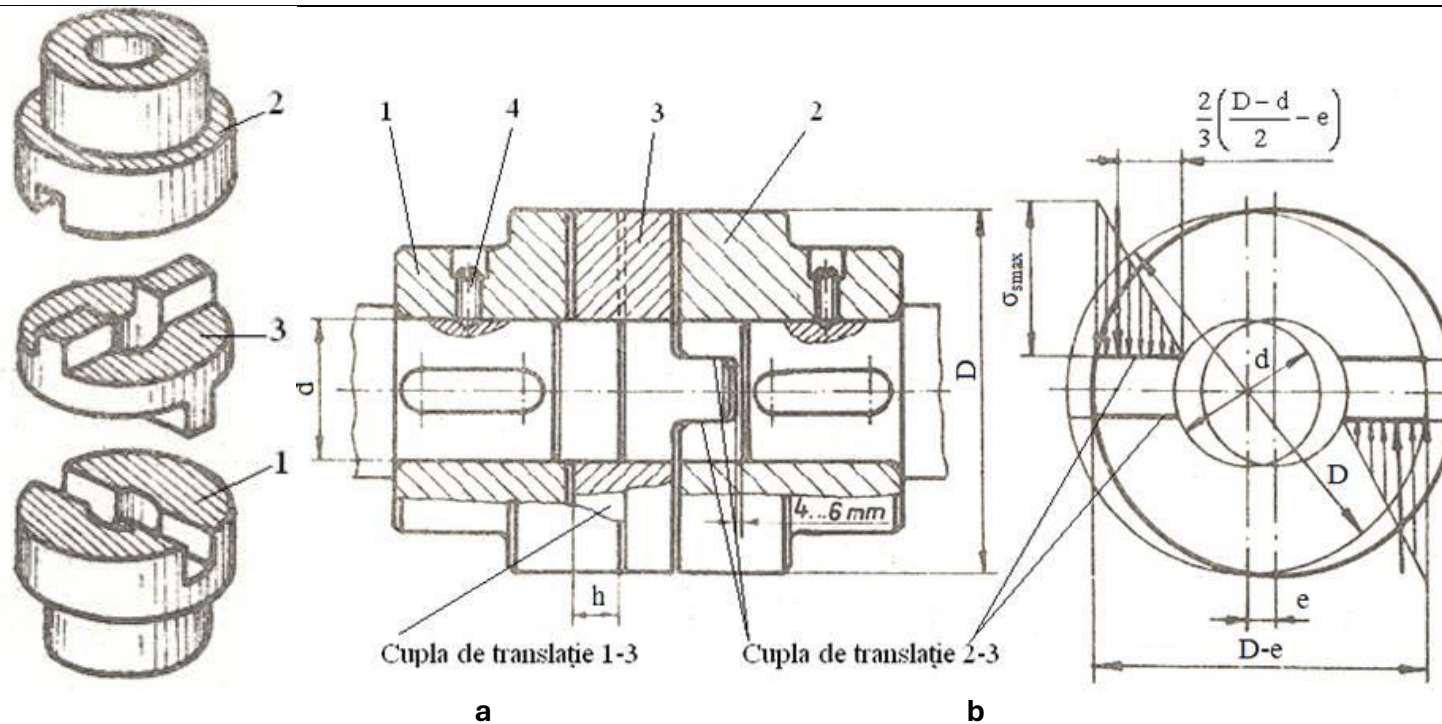


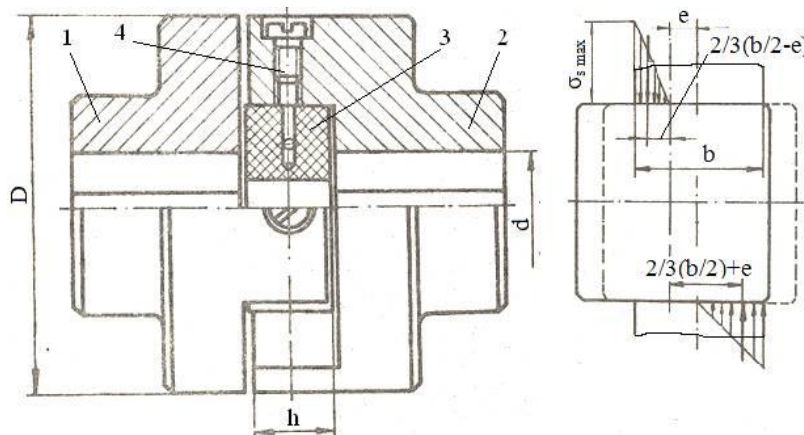
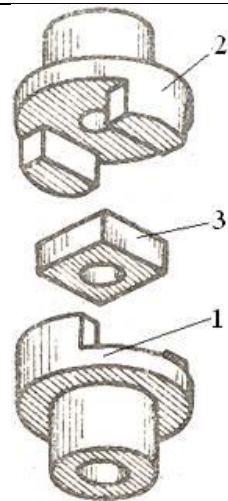
Fig. CUP-T.1.2.2.1 Transverse coupling (Oldham) with metal intermediate element

Operation and construction: the torque is transmitted *by shape* from the input shaft through a parallel key assembly to the driving half-coupling, 1, through the side faces of the translation coupling, 1-3, to the intermediate disc 3, then through the faces of the translation coupling 3 to the half-coupling 2 and from it through a parallel key assembly to the output shaft; during operation, the intermediate disc describes a planetary movement, its center describing a circle with a diameter equal to the eccentricity of the shafts, e ; the threaded pin 4 has the role of axially fixing the half-coupling.

Condition for transmitting torque through the shape (crushing resistance):

$$\sigma_{s \max} = 2 \frac{F}{A_s} = 2 \frac{\frac{M_{tc}}{\frac{2}{3} \left(\frac{D-d}{2} - e \right) + e + d}}{h \left(\frac{D-d}{2} - e \right)} = \frac{12 M_{tc}}{h (2D+d-e)(D-d-2e)} \leq \sigma_{as}, \quad (\text{CUP-T.6})$$

where, M_{tc} is the design torsion moment (see rel CUP-T.1), σ_s - the maximum crushing stress in the translational couplings, σ_{as} - the allowable crushing stress of the material coupling (half-couplings-disc), d - diameter of the input and output shafts, D - outer diameter, h - height of the translational couplings.



Operation and construction: the torque is transmitted *by shape* from the input shaft through a parallel key assembly to the driving half-coupling, 1, through the side faces of the translation coupling, 1-3, to the intermediate prismatic block 3, then through the faces of the translation coupling 3 to the half-coupling 2 and from it through a parallel key assembly to the output shaft; during operation, the intermediate disc describes a planetary movement, its center describing a circle with a diameter equal to the eccentricity of the shafts, e ; the screw 4 acts as a plug against the loss of lubricant introduced into the contact area.

Condition for transmitting torque through the shape (crushing resistance):

a	b	
Fig. CUP- T.1.2.2.1 Transverse coupling (Oldham) with non-metallic intermediate element		$\sigma_{s \max} = 2 \frac{F}{A_s} = 2 \frac{\frac{M_{tc}}{\frac{2}{3}(\frac{b}{2}-e) + \frac{2}{3}\frac{b}{2}-e}}{h(\frac{b}{2}-e)} = \frac{12 M_{tc}}{h(2b-e)(b-2e)} \leq \sigma_{as} \quad (\text{CUP-T.7})$ where, M_{tc} is the design torsion moment (see rel CUP-T.1), - the maximum crushing stress in the translational couplers, - the allowable crushing stress of the material couple (half-couplers-prismatic block), b – the contact length of the translational couplers, h – the height of the translational couplers. σ_s σ_{as}
Note: Due to the high friction in the translation couplings, it is necessary to use anti-friction materials (e.g. the prismatic block 3 in Fig. CUP-T.7 is made of textolite) and to lubricate with consistent greases; for these reasons, in some catalogs the efficiency is also indicated as a characteristic parameter.		

CUP-T.8.1.2.3 Rigid mobile permanent couplings for compensation of angular or angular-axial deviations

Angular and angular-axial mechanisms used for couplings

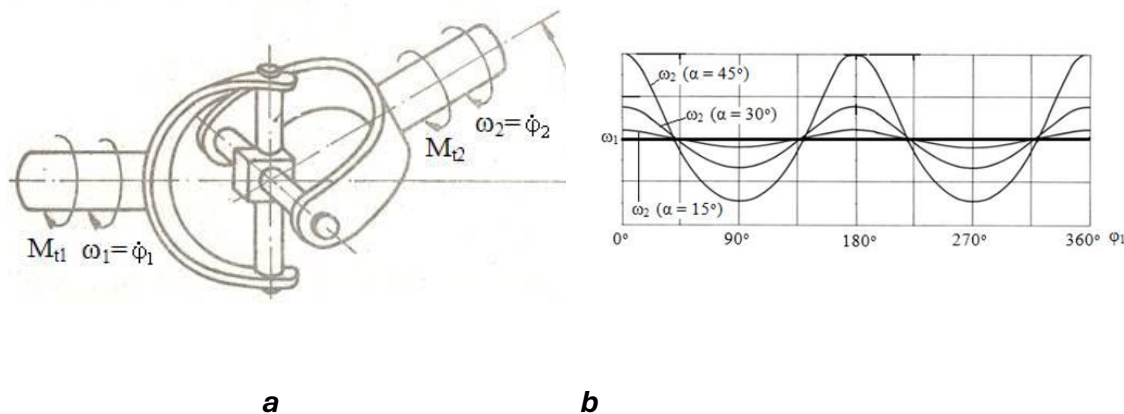


Fig. CUP-T.8.1.2.3.1 Cardanic angular mechanism: **a** – functional diagram; **b** – heterokinetics diagrams

Operation.

Cardan mechanisms (fig. CUP-T.8.1.2.3.1,a) which transmit and transform the rotational movement from a driving element to a driven one with concurrent axes, according to the relationship,

$$\omega_2 = \frac{\cos \alpha}{\cos^2 \phi_1 \cos^2 \alpha + \sin^2 \phi_1} \omega_1, \quad (\text{CUP-T.9})$$

where, ω_2 is the angular velocity of the output element, ω_1 – the angular velocity of the input element, α – the direction angle of the output element, ϕ_1 – the rotation angle of the input element.

From the diagrams in Fig. CUP-T.8.1.2.3.1,b it is observed that in the case of constant input motion (ω_1) the driven element motion (ω_2) results in a periodically variable altitude with increasing angle between axes α . Thus, the

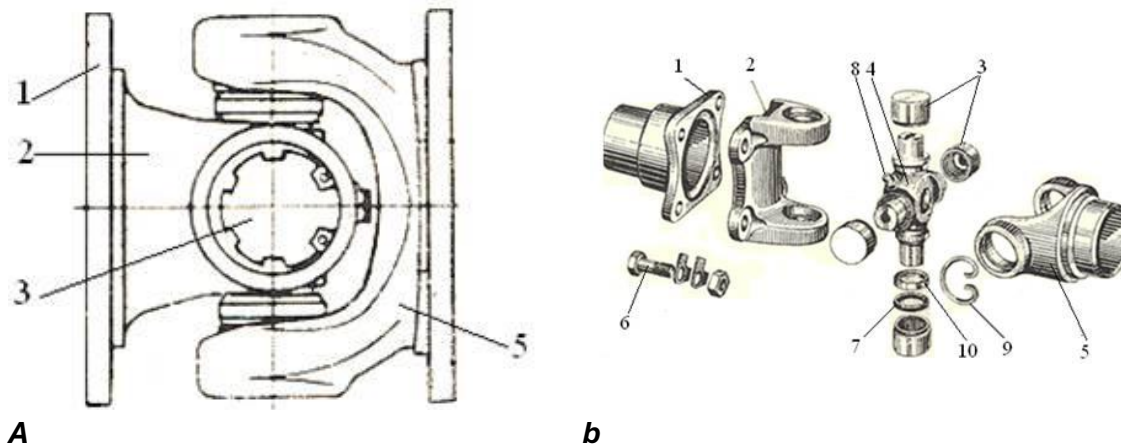


Fig. CUP-T. 1.2.3.2 The constructive solution of the cardan mechanism: **a** – side view; **b** – exploded structure

cardan mechanism can be coupled only at angles between the directions of reduced motion (e.g., $\alpha < 5$) for which heterokinetism is accessible (i.e., according to the definition in subchapter CUP-T.1, it transmits rotational motion, without performing its transformations).

Construction variants:

The solution in fig. CUP- T.8.1.2.3.2. The torque is transmitted from the driving element, the flange 1, through the assembly with screws 6, mounted with clearance, to the driving fork 2, then to the opposite arms of the universal joint 4 through needle bearings 3, and from the other rods, also through needle bearings to the driven fork 5; the lubrication of the needle bearings is done with consistent grease introduced through the plunger 8 into the channels in the universal joint body; the loss of consistent grease during operation is prevented by the sealing ring 7; the axial support of the needle bearings is through washers 10 and elastic rings 9.

The solution in fig. CUP- T. 1.2.3.3. The torque is transmitted from the driving shaft 1, through the contact between the profiles made in it and the intermediate elements (balls) 3, from them to the inner ring 5, also through direct contacts, and further through the assembly through splines to the driven shaft 2; the intermediate balls 3 are held in one plane by the cage 4; the inner ring 5 is fixed axially by means of the washer 7 and the elastic ring 6; to prevent the expulsion of the consistent grease from the active area, the protective bellows 8 was used.

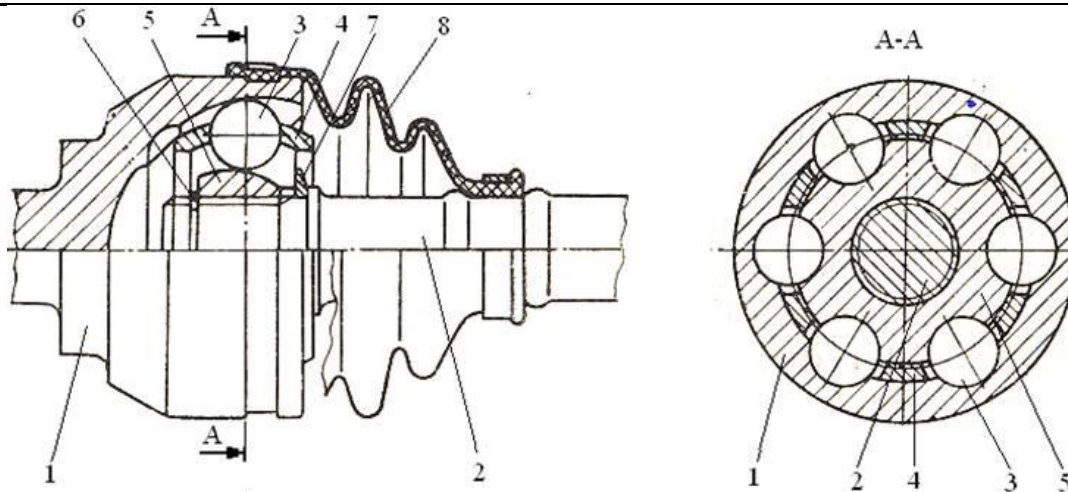


Fig. CUP- T. 1.2.3.3 The constructive solution of the angular mechanism with rolling elements (Rzeppa)

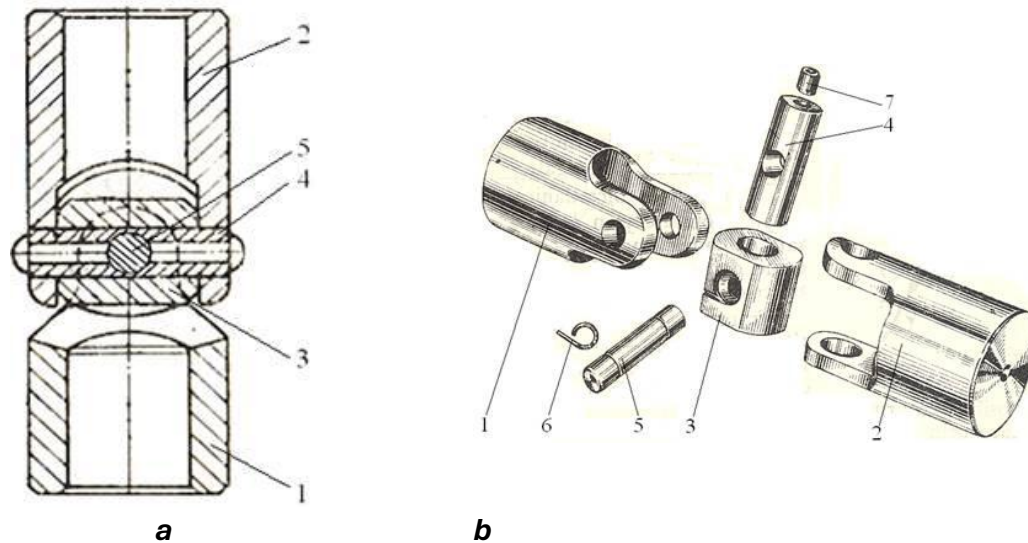


Fig. CUP-T. T. 1.2.3.4 The constructive solution of the angular cardan mechanism with central block and bolts (Curtis): **a** – axial section; **b** – exploded structure

The solution in fig. CUP-T. 1.2.3.4. The torque is transmitted through the form from the driving half-coupling, fork 1, to the pin 5, further, to the intermediate block 3, from it to the pin 4 and then to the driven half-coupling, fork 2; the axial fixation of the pin 5 is done by the elastic rings 6.

The solution in fig. CUP-T. 1.2.3.5. The torque is transmitted through the form from the driving half-coupling 1 with three axial guide channels, to the rollers 2, further, the tripod element (with three arms) 3, from it through the spline to the driving shaft 5; the axial fixation of the tripod element 5 is done by the elastic rings 5; to prevent the expulsion of the consistent grease from the active area, the protective bellows 5 was used.

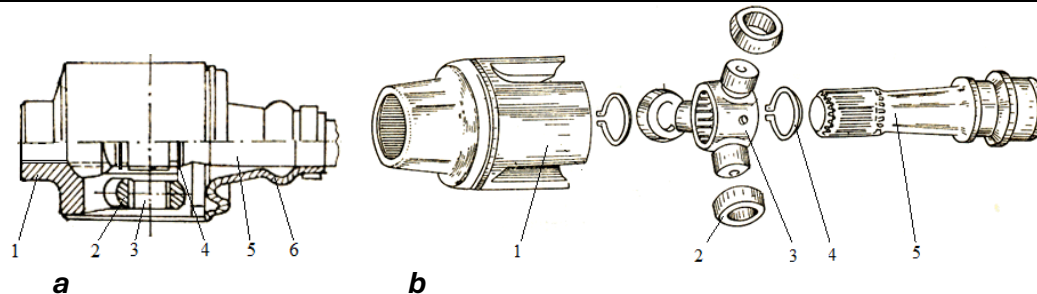


Fig. CUP-T. T. 1.2.3.5 The constructive solution of the tripod angular-axial mechanism: **a** – axial section; **b** – exploded structure

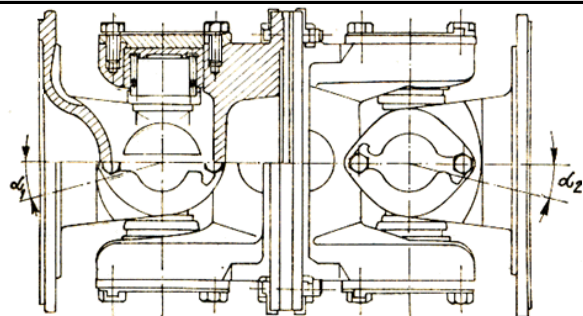
Angular couplings

Fig. CUP- T. 1.2.3.6 The constructive solution of the angular cardan coupling with two rigidly connected cardan mechanisms

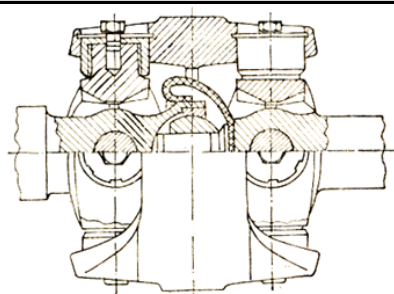


Fig. CUP- T. 1.2.3.7 The constructive solution of the angular cardan coupling with two cardan mechanisms connected with a spherical joint

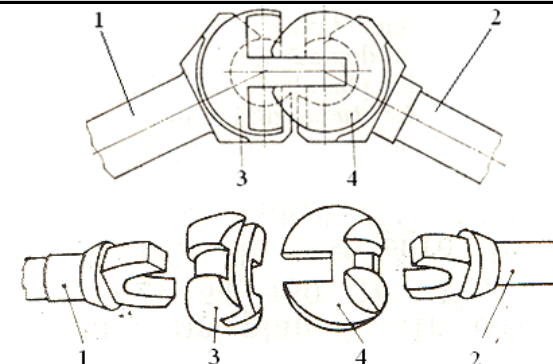


Fig. CUP- T. 1.2.3.8 The constructive solution of the angular coupling (Tracta) with a spatial mechanism composed of three series-connected rotation couplings: 1-3, 3-4, 4-2

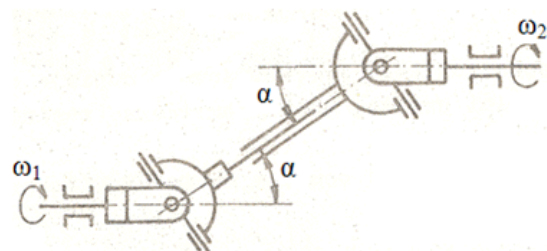
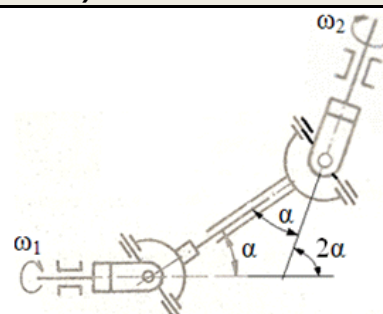
Angular-axial couplings (universal transmissions)**A****b**

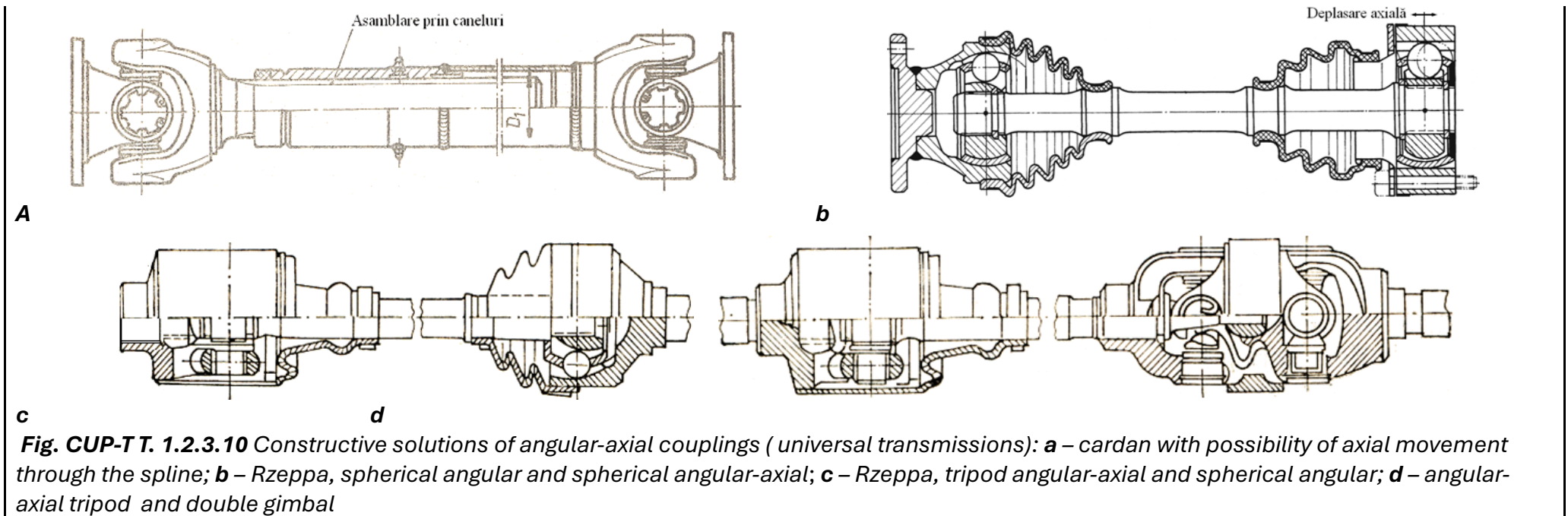
Fig. CUP- T. 1.2.3.9 Functional diagrams of angular-axial couplings with cardan and spline mechanisms: **a** – in *Z*; **b** – in *W*

Structural and assembly diagrams

Angular-axial couplings (universal transmissions) are obtained by serializing two cardan/angular mechanisms that have parallel/antiparallel input/output forks axes and the angles of the two mechanisms are equal (fig. CUP-T. 1.2.3.9, a,b). Thus, the heterokinetics of the first mechanism is canceled by the heterokinetics of the second and the synchronism of the input and output movements is obtained if the relative displacement between the mechanisms is ensured through splines (fig. CUP- T. 1.2.3.10,a) or within a mechanism (fig. CUP- T. 1.2.3.10,b,c,d).

Constructive variations

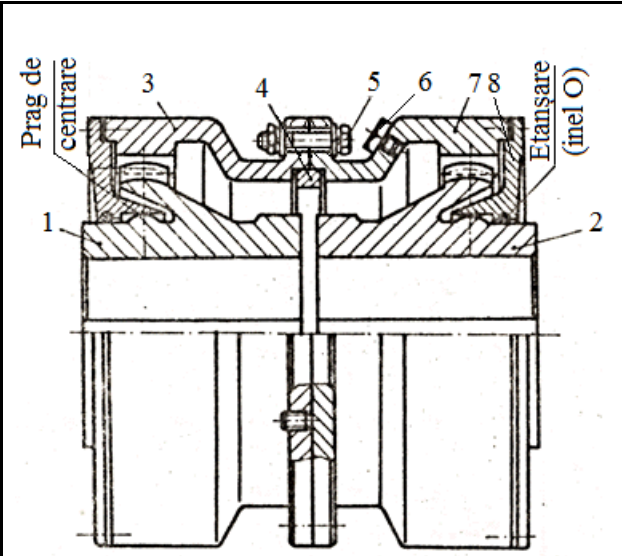
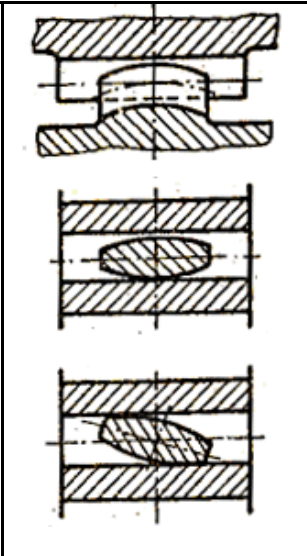
The solutions in Fig. CUP- T. 1.2.3.10 represent four angular-axial couplings (universal transmissions) obtained by serializing: two monocardan mechanisms with a cross (Fig. CUP- T. 1.2.3.10,a) by means of a spline assembly; two Rzeppa mechanisms (angular and angular-axial, Fig. CUP- T. 1.2.3.10,b); an angular Rzeppa mechanism with an angular - axial tripod mechanism (Fig. CUP- T. 1.2.3.10,c); a double cardanic coupling with a spherical joint with an axial-angular tripod mechanism (Fig. CUP- T. 1.2.3.10,c)



CUP-T.8.1.2.4 Rigid mobile permanent couplings for compensation of combined deviations (axial, radial and angular)

From the category of these representative couplings are the gear couplings (tab. CUP-T.8.1.2.4.1) which can simultaneously compensate for all types of deviations.

Tab. CUP-T.8.1.2.4.1 Gear coupling

		<i>Operation and construction</i> The torque is transmitted from the driving half-coupling 1 (with convex external teeth) through multiple contacts (form) to the driving sleeve 3 (with internal teeth), through the assembly with screws mounted with clearance 5 to the driven sleeve 7 (with internal teeth) and also through multiple contacts (form) to the driven half-coupling 2 (with convex external teeth). The convex shape (fig. CUP-T.8.1.2.4.1, b) of the teeth of the half-couplings mounted on the shafts maintains the contacts between the teeth permanently even in the case of large angular deviations. These couplings are capable of transmitting large torques with small overall dimensions, due to the simultaneous contacts between all the conjugate teeth of the connecting parts. The torque transmitted by the coupling decreases with increasing angular deviation, because in this case the mechanical friction work in the contacts also increases significantly, with consequences for wear and heating of the coupling. To reduce friction, the coupling is lubricated with thick grease (or for certain functional situations with oil) which is introduced into the contact area through the threaded hole (closed with plug 6) and which is prevented from being expelled by the covers 8 and the mobile seals with O-ring. Good operation (without noise), especially at high speeds, is ensured by increased precision of the teeth (with involute profile) and by the coaxial mounting of the internal teeth by means of the centering ring 4.
A Fig. CUP-T.8.1.2.4.1 The constructive solution of the toothed coupling	b	

CUP-T.8.1.3 Permanent elastic mobile couplings

CUP-T.8.1.3.1 Permanent elastic mobile couplings with metal deformable elements

Bar spring couplings (Forst)

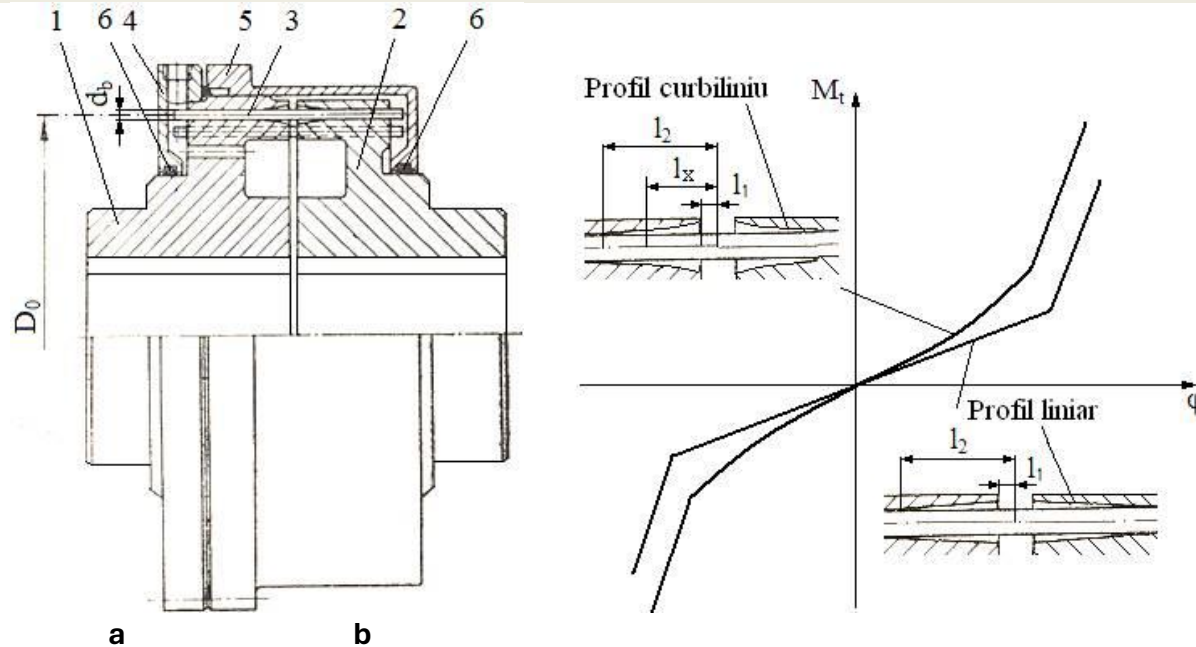


Fig. CUP-T.8.1.3.1.1 The constructive solution of the elastic coupling with bar springs (Forst): **a** – half-view and axial half-section; **b** – elastic characteristics

Function and construction. The torque is transmitted by shape from the driving half-coupling 1 to the driven half-coupling 2 by means of bar springs mounted in flared bores with a linear profile (conical bore with straight line generators) – the elastic characteristic composed of two straight segments with different slopes determined by the bending of the bars with the embedding lengths l_1 and l_2 – or curvilinear (conical bore with curvilinear generators) – the elastic characteristic composed of a nonlinear segment, determined by the bending with the variable embedding length $l_x \in [l_1, l_2]$, and a straight segment, determined by the bending of the bars with the embedding length l_1 .

These couplings can compensate for axial and angular deviations of the coupled shafts by relative displacements between the coupling halves and the bar springs. To reduce wear that occurs during operation, the bar spring area is lubricated with thick grease and to avoid its expulsion, protective housings 4 and 5 with sealing rings 6 are provided.

The coil spring coupling (Cardeflex)

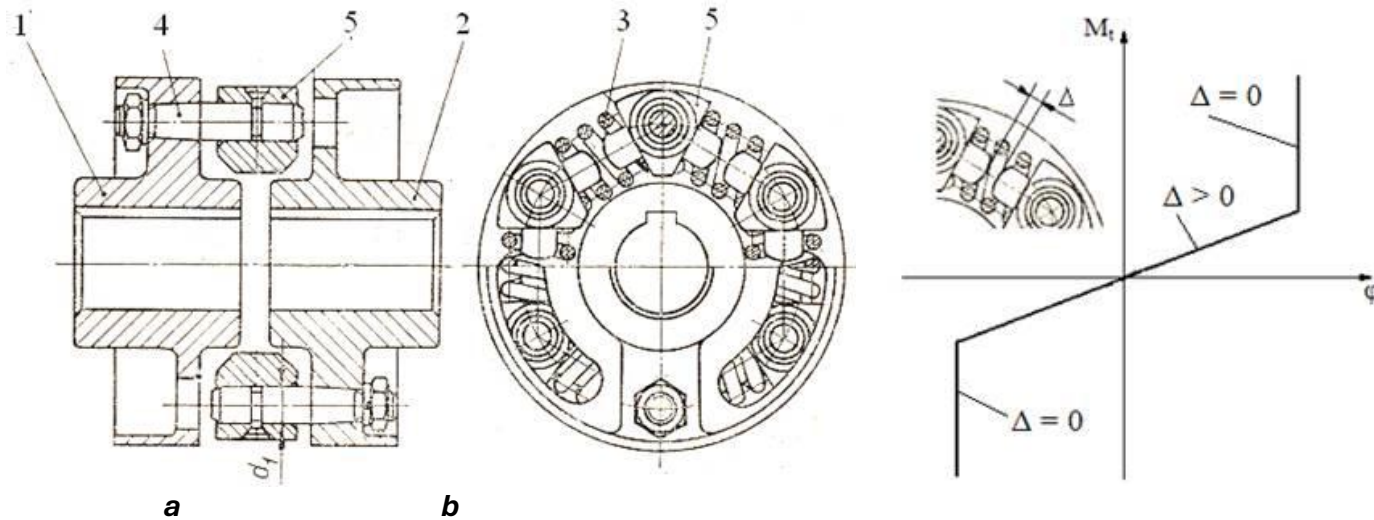


Fig. CUP- T.8.1.3.1.2 The constructive solution of the elastic coupling with helical springs (Cardeflex): **a** – axial section and front view; **b** – elastic characteristic

Operation and construction.

The torque is transmitted through the form from the driving half-coupling 1 through the bolts 4 to the segments 5 and then through the coil springs 3 to the segments 5 and the bolts 4 mounted on the driven half-coupling to the driven half-coupling 2. The elastic characteristic is composed of two straight line segments, one corresponding to the deformation period of the coil springs with the arrow $\Delta > 0$ (constant stiffness) and the other appearing at higher torques as a result of the direct contact between the guide pins of the segments 5 ($\Delta = 0$, infinite stiffness).

CUP-T.8.1.3.2 Permanent elastic mobile couplings with non-metallic elements

Elastic couplings with bolts

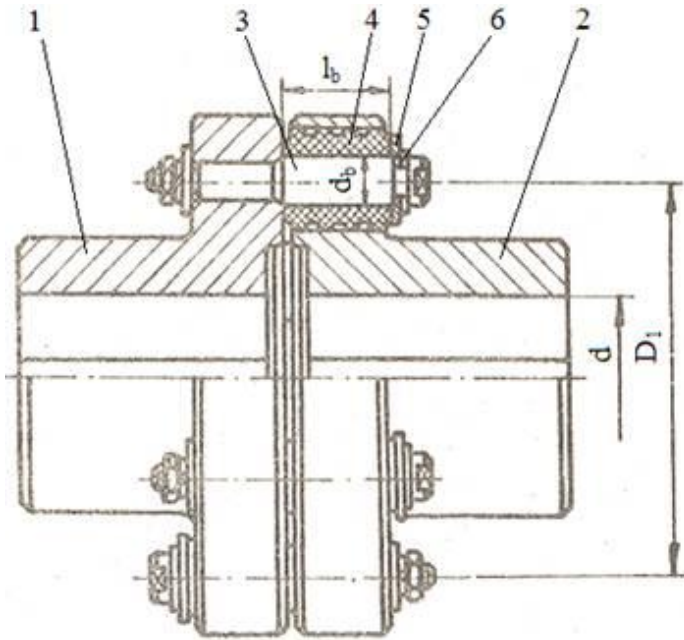


Fig. CUP- T.8.1.3.2.1 The constructive solution of the elastic coupling with bolts

Function and construction. The torque is transmitted from the input half-coupling 1 by shape to the bolts 3 and through the rubber sleeves (rings) 4 to the half-coupling 2. The rubber sleeves are axially fixed by means of washers 5 and elastic rings 6.

Use These couplings allow the compensation of radial deviations, $\Delta R = 0.3 \dots 0.6$ mm, angular deviations, $\Delta \alpha \leq 1^\circ$ and some axial deviations within the limits of maintaining contacts between the rubber sleeves and the bores of the coupling half 1.

Basic calculation relationships

The condition for the crushing resistance of the elastic element (the rubber sleeve), assuming uniform load distribution on the bolts, is given by the relationship,

$$\sigma_s = \frac{F_1}{d_b l_b} = \frac{\frac{2 M_{tc}}{z D_1}}{d_b l_b} = \frac{2 M_{tc}}{z D_1 d_b l_b} \leq \sigma_{as} \quad , \quad (\text{CUP-T.10})$$

where: M_{tc} represents the design torsion moment; F_1 – the force acting on a bolt; d_b , l_b – the diameter, respectively, the length of the sleeve bore, D_1 – the diameter of the bolts; σ_s – the maximum crushing stress; σ_{as} – the allowable crushing stress of the rubber (5...7 MPa).

The condition of the bending resistance of the bolt, under conditions of maximum angular deviations (the maximum embedment length, l_b , is considered),

$$\sigma_i = \frac{F_1}{W_z} = \frac{\frac{2 M_{tc}}{z D_1} l_b}{\frac{\pi d_b^3}{32}} = \frac{64 M_{tc} l_b}{\pi z D_1 d_b^3} \leq \sigma_{ai} \quad , \quad (\text{CUP-T.11})$$

where: σ_i – maximum bending stress, σ_{ai} – allowable bending stress (0.25...0.4) σ_{02} [MPa])

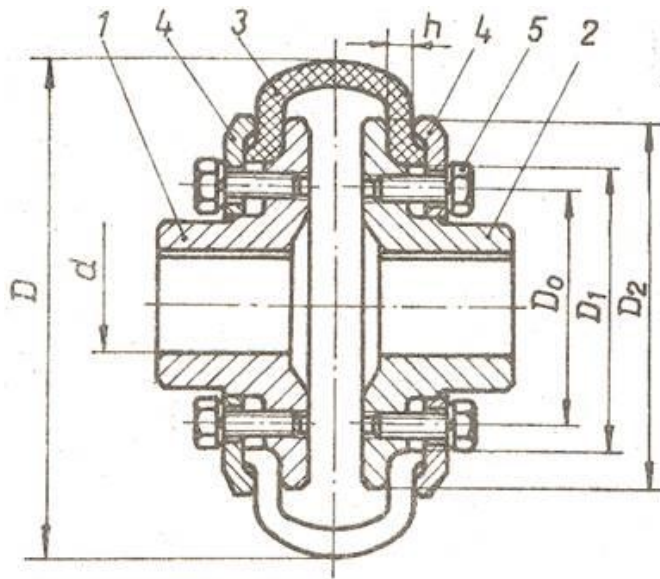
Elastic couplings with rubber band (Periflex)

Fig. CUP- T.8.1.3.2.2 The constructive solution of the elastic coupling with rubber band

Function and construction. The torque is transmitted from the input half-coupling 1 by friction to the band 3 and from it also by friction to the half-coupling 2. The rubber band is fixed to the two half-couplings by means of discs 4 tightened axially with screws 5.

Use These couplings allow the compensation of radial deviations, $\Delta R = 2 \dots 6$ mm, angular, $\Delta \alpha \leq 2^\circ \dots 6^\circ$ and axial $\Delta l = 3 \dots 6$ mm. In addition, they ensure good shock and vibration damping, having a high capacity for accumulating shock energy due to the increased volume of the elastic element.

Basic calculation relationships

The condition for transmitting torque through friction,

$$M_{tc} \leq M_f = \mu i z_{\text{с}} F_{\text{с}} \frac{D_1 + D_2}{4}, \quad (\text{CUP-T.12})$$

where, μ is the coefficient of friction between the tire (rubber) and the half-coupling or disc (steel), D_1 and D_2 are the minimum and maximum diameters of the friction surfaces, i – the number of friction surfaces (for the case in Fig. $i = 2$), $z_{\text{с}}$ - the number of bolts, $F_{\text{с}}$ – the tightening force of a bolt.

The condition of the bandage's crushing resistance,

$$\sigma_s = \frac{z_{\text{с}} F_{\text{с}}}{\frac{\pi}{4} (D_2^2 - D_1^2)} \leq \sigma_{as}, \quad (\text{CUP-T.13})$$

where, $F_{\text{с}}$ the tightening force of a screw, is determined from the relationship (CUP -T.12); σ_s – maximum crushing stress, σ_{as} – allowable crushing stress of the rubber (5...7 MPa).

The condition of shear strength of the bandage,

$$\tau_f = \frac{2 M_{tc}}{\pi D_2^2 h} \leq \tau_{af}, \quad (\text{CUP-T.14})$$

where, τ_f – maximum shear stress, τ_{af} – allowable shear stress of the rubber (0.3 ...0.5 MPa).

CUP-T.8.2 INTERMITTENT COUPLINGS

CUP-T.8.2.1 CONTROLLED INTERMITTENT COUPLINGS

CUP-T.8.2.1.1 Intermittent controlled couplings with form-fitting load transmission (direct contact)

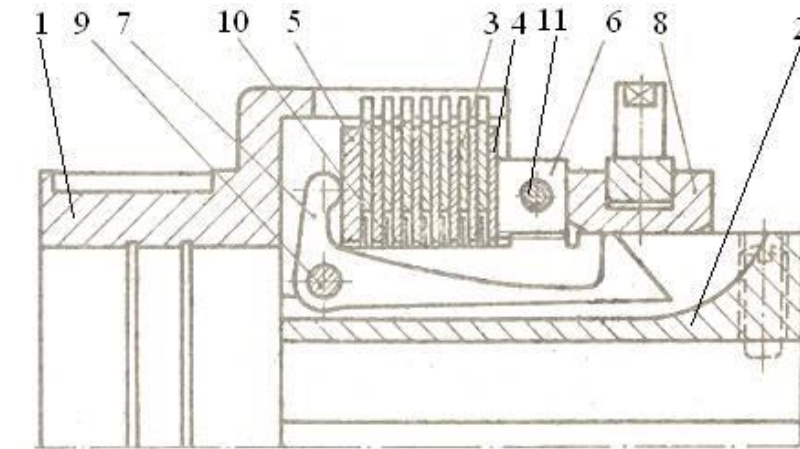
These couplings transmit torque through direct contact between the protrusions on the two coupling halves; representative of this are couplings with radial teeth (tab. CUP-T.8.2.1.1.1) and with front teeth.

Tab. CUP-T.8.2.1.1.1 Mechanically controlled intermittent coupling with radial teeth

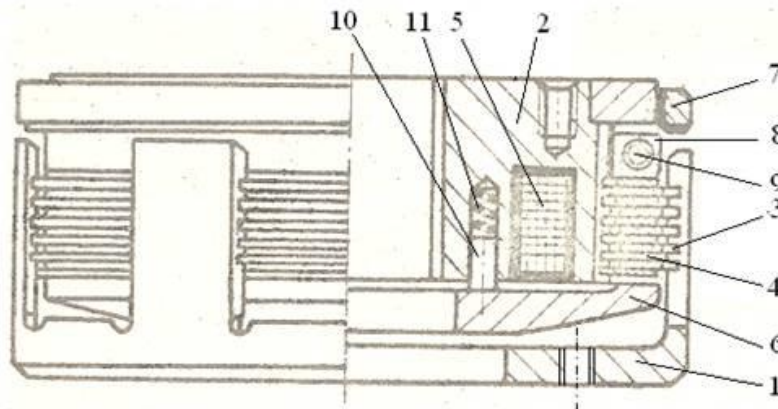
		Operation and construction In the engaged state, the torque is transmitted from shaft 1, through the spline to the externally toothed half-coupling 7, through the teeth (fig. CUP-T.8.2.1.1.2,d) to the socket 6, through the teeth to the gear wheel 3. In the disengaged state, the gear wheel 3 rotates freely on the shaft, supported radially by the bearing 6 and axially by the bearings 2 or 5. To facilitate coupling/decoupling, the frontal rounding of the teeth and the shortening of the length of every second tooth from the outside, as well as the elimination of every second tooth from the inside, are practiced (fig. CUP-T.8.2.1.1.2). If the coupling is carried out under conditions of synchronous movement of the coupled elements, torsional shocks occur, which lead to improper operation of the transmission as well as to additional stresses on it. The relative speed at which the coupling can take place is limited by the shocks that occur in the transmission; to increase the relative coupling speed, constructive solutions combined with friction synchronizers can be adopted.
Fig. CUP-T.8.2.1.1.1 Intermittent coupling with radial teeth	Fig. CUP-T.8.2.1.1.2 Coupling process: a – initial, uncoupled; b – intermediate, axial displacement without rotation; c – intermediate, axial displacement with rotation; d – final, coupled	

CUP-T.8.2.1.2 Intermittent controlled couplings with friction load transmission

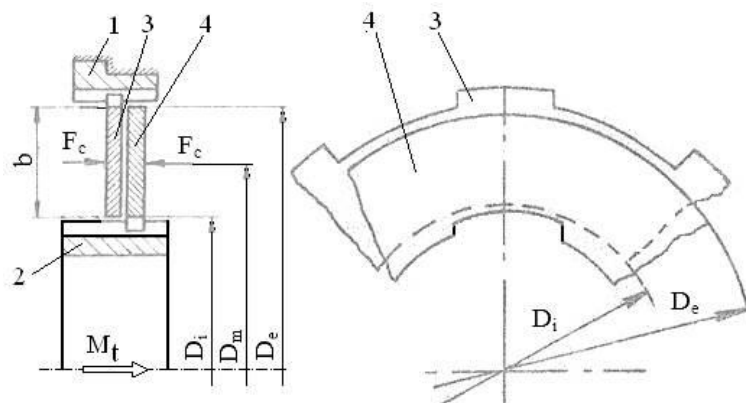
Multi-disc (flat-surface) friction - controlled couplings



A



b



Operation and construction.

The variant in Fig. CUP-T.8.2.1.2.1,a.

The torque is transmitted from the driving half-coupling 1 through the spline to the driving discs 3, through friction to the driven discs 4 and from them through the spline to the driven half-coupling 2. The pressing force (fig. c) is achieved by means of the pressure plate 5 under the action of the levers 9 supported by the bolts 9 and pressed by the control sleeve 8. The coupling/decoupling is done by the axial movement of the control sleeve 8 which acts/releases the levers 7. The adjustment of the pressing force (contact pressure) as well as the compensation of wear is done by means of the support disc 6 which has the shape of a sectioned nut and is locked with the tangentially mounted screw 11. The discs are detached upon uncoupling by the profiled springs 10.

The variant in Fig. CUP-T.8.2.1.2.1,b.

The torque is transmitted from the driving half-coupling 1 through the spline to the driving discs 3, through friction to the driven discs 4 and from them through the spline to the driven half-coupling 2. The pressing force (fig. c) is achieved by means of the electromagnetic field generated by the coil 5 powered by the slip ring 7. The coupling/decoupling is done by coupling/decoupling (manually or automatically) the coil power supply circuit. The adjustment of the pressing force (contact pressure) as well as the wear compensation is done with the support disc 8 which has the shape of a sectioned nut and is locked with the tangentially mounted screw, 9. The discs are detached upon decoupling by the pins 10 actuated by the springs 11.

Basic calculation relationships

The characteristic functional geometric parameters are associated with the annular friction surfaces with the inner diameter, D_i , the outer diameter, D_e , the average diameter, $D_e = (D_i + D_e)/4$, and the width, b .

Simplifying assumptions: the load is distributed equally on the friction surfaces; the pressure force of the discs is distributed equally on the pairs of discs; the forms of pressure variation and/or wear on the active surfaces can be synthesized in two variants:

- a. uniform pressure distribution, (v. **Obs.**)

c

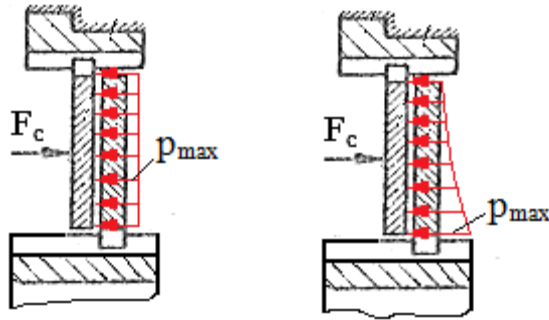


Fig. CUP- T.8.2.1.2.1 Controlled multi-disc friction clutches: **a** – mechanical control; **b** – electromagnetic control; **c** – calculation model

b. uniform distribution of wear (pressure is distributed unevenly, see **Note**).

The condition of crushing resistance of the active surfaces of the discs, can be verified with one of the relations:

$$p_{max} = \frac{F_c}{\pi \left(\frac{D_e^2}{4} - \frac{D_i^2}{4} \right)} = \frac{4F_c}{\pi(D_e^2 - D_i^2)} \leq \sigma_{as} (p_a) \quad , \quad (CUP-T.16)$$

for **case a.** (uniform pressure distribution on the active surfaces),

$$p_{max} = \frac{F_c}{\pi D_i b} \leq \sigma_{as} (p_a) \quad , \quad (CUP-T.17)$$

for **case b.** (uniform distribution of wear on the active surfaces), in which, p_{max} represents the maximum pressure on the active surfaces; F_c – normal pressing force (coupling), b – width of the friction surfaces, D_i – inner diameter, D_e – outer diameter; σ_{as} – permissible contact (crushing) stress of the pair of disc materials (e.g. steel - steel, sintered materials - hardened steel, cast iron - hardened steel; $\sigma_{as} = 1...3$ MPa, without lubrication (dry friction); $p_a = 0.2...0.5$ MPa - permissible pressure of the lubricant, for the case with lubrication.

The condition for transmitting torque through friction can be verified with one of the relations:

$$M_t = i F_f \frac{D_m}{2} = i \mu F_c \frac{D_m}{2} = i \mu \sigma_{as} \pi D_i b \frac{D_m}{2} = \frac{\pi}{2} i \mu \sigma_{as} \Psi (1 - \Psi) D_m^3 \geq M_{tc} \quad , \quad (CUP-T.18)$$

for **case a.** (uniform pressure distribution on the active surfaces) without lubrication,

$$M_t = i F_f \frac{D_m}{2} = i \mu F_c \frac{D_m}{2} = i \mu p_a \frac{\pi}{4} (D_e^2 - D_i^2) \frac{D_m}{2} = \frac{\pi}{2} i \mu p_a \Psi D_m^3 \geq M_{tc} \quad , \quad (CUP-T.19)$$

for **case a.** (uniform pressure distribution on the active surfaces) with lubrication,

$$M_t = i F_f \frac{D_m}{2} = i \mu F_c \frac{D_m}{2} = i \mu \sigma_{as} \pi D_i b \frac{D_m}{2} = \frac{\pi}{2} i \mu \sigma_{as} \Psi (1 - \Psi) D_m^3 \geq M_{tc} \quad , \quad (CUP-T.20)$$

for **case b.** (uniform distribution of wear on the active surfaces) without lubrication,

$$M_t = i F_f \frac{D_m}{2} = i \mu F_c \frac{D_m}{2} = i \mu p_a \pi D_i b \frac{D_m}{2} = \frac{\pi}{2} i \mu p_a \Psi (1 - \Psi) D_m^3 \geq M_{tc} \quad , \quad (CUP-T.21)$$

for **case b.** (uniform distribution of wear on the active surfaces) with lubrication,

where: M_t represents the torque transmitted by friction, M_{tc} – design torque; F_f – friction force corresponding to an active surface; i – number of friction surfaces; $D_m = (D_e + D_i)/2$ – average diameter, $\psi = b/D_m$ – width factor of the friction surfaces ($D_e = (1 + \psi) D_m$, $D_i = (1 - \psi) D_m$); μ – friction

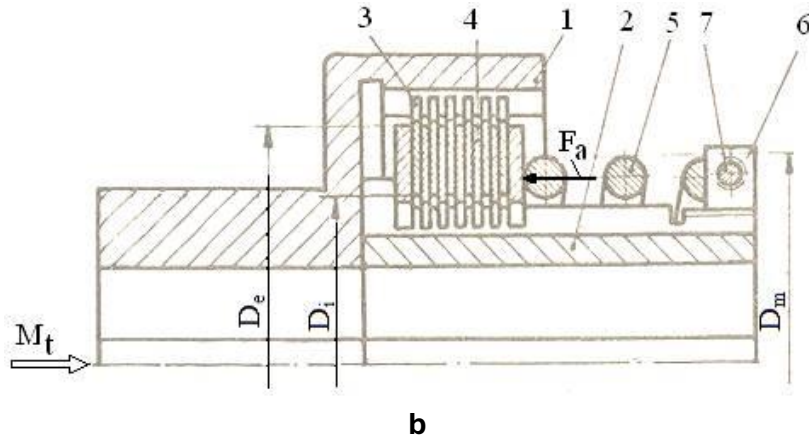
coefficient of the pair of materials in contact (e.g. for the steel-steel case, $\mu = 0.1-0.15$ with lubrication or $\mu = 0.15-0.4$ without lubrication; $\mu = 0.2-0.5$ for the case of a steel disc and a synthetic resin gasket with carbon fiber without lubrication (clutches))

Note.

1. The wear value is directly proportional to the pressure and is dependent on the speed; uniform wear leads to non-uniform pressure distribution with the maximum value at the minimum diameter (D_i).
2. Uniform pressure distribution is close to reality only in the case of friction surfaces in new condition or in the case of constructive solutions with an elastic element that maintains uniformly distributed pressure (e.g. the friction disc of the main clutch of the automobile).
3. The uniform wear hypothesis, in general, is closer to real processes, in the event that the coupling would still have a uniform pressure distribution, the resulting deviation consists of an oversizing in terms of the effective pressure - therefore a reduced wear - and an undersizing, of a small value - in terms of the capable moment, covered by the safety coefficient usually used [Achiriloaie, 1985].
4. The width coefficient, Ψ , has the optimal value $\Psi_{opt} = 0.27$ [Achiriloaie, 1985].
5. Considering that the friction forces that appear in the splines of the couplings and those of the discs determine the pressing force between the discs, so that in the case of a large number of discs, the latter are pressed too little against each other, and their contribution to the transmission of the torque is smaller than their influence on the increase in the weight of the coupling, its moment of inertia, etc.
6. Construction recommendations: the number of friction surfaces, $i < 6$, for dry operation and $i \leq 16$, for lubricated operation; in the case of $i > 1$, even values are adopted for i and the numbers of driven discs, $z_2 = i/2$, respectively, driving $z_1 = z_2 + 1$.
7. The value of the pressing force (coupling) F_c , determined from the condition of transmission of the torque through friction (see relations CUP-T.18... CUP-T.21) , will also check the condition of crushing resistance of the active surfaces (see relations CUP-T.16 and CUP-T.17)

CUP-T.8.2.1.3 Automatic intermittent couplings

Safety couplings	
a	The variant in Fig. CUP- T.8.2.1.3.1, a <i>Function and construction.</i> The torque is transmitted <i>by shape</i> from the driving half-coupling 1 through the pads 3, to the shear pins 3 and from them through the pads 5 to the driven half-coupling 2. The plugs 6 have the role of preventing the axial displacement of the subassembly of the pads 4 and the shear pin 3. <i>Basic calculation relationships</i> The condition for limiting the transmitted torque, $M_{tlim} = z F_{for} = z \tau_r A_{for} = z \tau_r, \quad (CUP-T.22) \quad \frac{D_0}{2} \frac{D_0 \pi d_1^2 D_0}{4 \cdot 2}$ where, $M_{tlim} = K_{sf} M_t$ is the limit torsional moment ($K_{sf} = 1.05...1.25$ - safety factor at failure); F_{for} – shear breaking force of a pin, A_{for} – shear breaking section area of a pin, D_0 – shear pin arrangement diameter, d_1 – shear pin



diameter, z – number of shear pins, τ_r – shear breaking stress of the pin material.

The variant in Fig. CUP- T.8.2.1.3.1, b

Function and construction. The torque is transmitted from the driving half-coupling 1 through the spline to the driving discs 3, then, through friction to the driven discs 4 and from them through the spline to the driven half-coupling 2. The pressing force, necessary to generate the friction forces, is achieved by the helical spring 5 precompressed by the sectioned nut 6, secured by the tangentially mounted screw 7.

Fig. CUP- T.8.2.1.3.1 Automatic safety couplings: **a** – with shear pins; **b** – multi-disc by friction

Basic calculation relationships

The condition for limiting the transmitted torque (see rel. CUP-T.18, CUP-T.19), can be verified with the relations:

$$M_{\text{tlim}} = \frac{\pi}{2} i \mu \sigma_{\text{as}} \Psi D_m^3, \quad (\text{CUP-T.23})$$

for the case of uniform pressure distribution without lubrication,

$$M_{\text{tlim}} = \frac{\pi}{2} i \mu p_a \Psi D_m^3, \quad (\text{CUP-T.24})$$

for the case of uniform pressure distribution with lubrication,

$$M_{\text{tlim}} = \frac{\pi}{2} i \mu \sigma_{\text{as}} \Psi (1 - \Psi) D_m^3, \quad (\text{CUP-T.25})$$

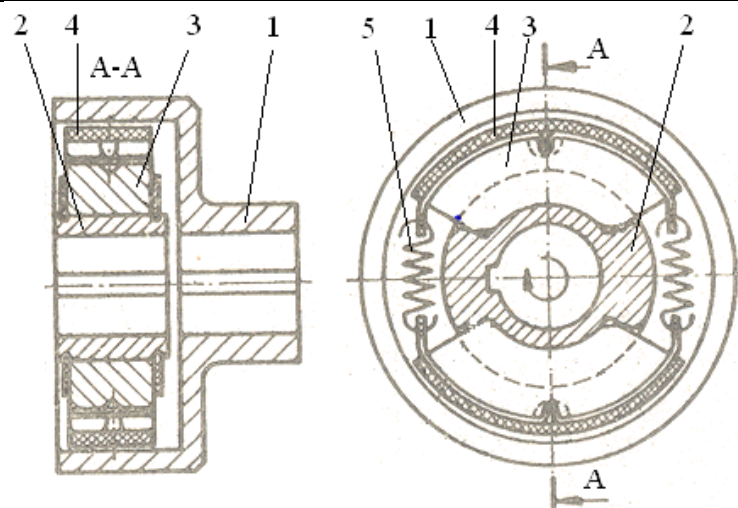
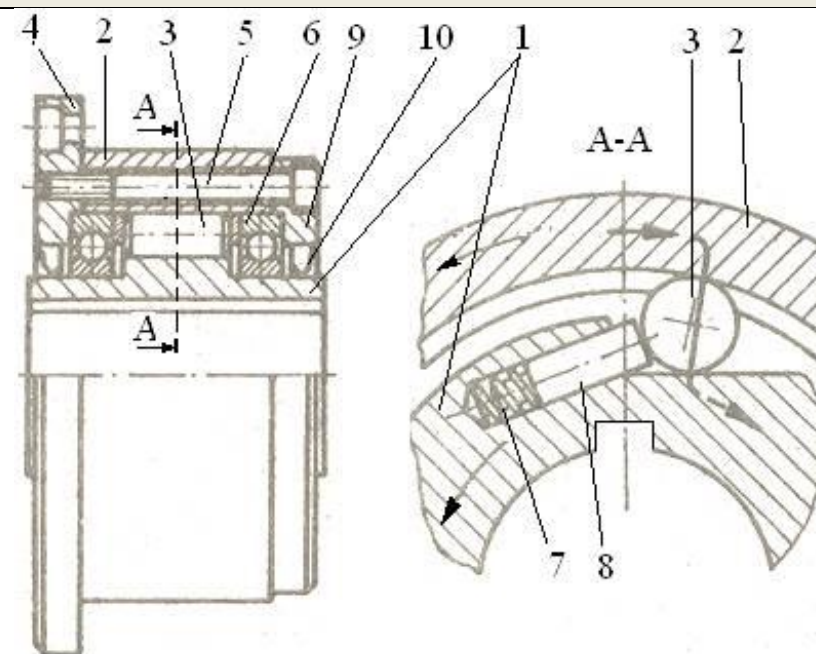
for the case of uniform wear distribution without lubrication,

$$M_{\text{tlim}} = \frac{\pi}{2} i \mu p_a \Psi (1 - \Psi) D_m^3, \quad (\text{CUP-T.26})$$

for the case of uniform wear distribution with lubrication, in which, $M_{\text{tlim}} = K_{\text{si}} M_t$ represents the limit torsional moment ($K_{\text{si}} = 1.05 \dots 1.25$ - safety factor at interruption); i – number of friction surfaces; μ – coefficient of friction of the pair of materials in contact (e.g. for steel-steel, $\mu = 0.1$ - 0.15 for the case with lubrication or $\mu = 0.15$ - 0.3 for the case without lubrication); D_i – inner diameter; D_e – outer diameter; $D_m = (D_e + D_i)/2$ – average diameter; $\Psi = b/D_m$ – the width factor of the friction surfaces, σ_{as} – the permissible contact (crushing) stress of the pair of disc materials (e.g. steel - steel, sintered materials - hardened steel, cast iron - hardened steel; $\sigma_{\text{as}} = 1 \dots 3$ MPa, without lubrication (dry friction); $p_a = 0.2 \dots 0.5$ MPa - the permissible pressure of the lubricant, for the case with lubrication.

The spring precompression force is determined by the relationship,

$$F_a = \frac{2M_{\text{tlim}}}{i \mu D_m} \quad (\text{CUP-T.27})$$

Speed and direction limiting couplings**Fig. CUP- T.8.2.1.3.2** Automatic speed limiter clutch**Fig. CUP- T.8.2.1.3.3** Automatic direction limiter coupling

The variant in fig. CUP- T.8.2.1.3.2 When starting up with the increase in the speed of the driving half-coupling 2, the centrifugal force acts on the shoes 3 and thus a pressing force is generated on the inner cylindrical surface of the half-coupling 1 and, therefore, the beginning of the transmission of the torque (its coupling). When the speed decreases, the centrifugal force acting on the shoes decreases and the transmission of the torque is interrupted. The springs 5 ensure firm decoupling in the case of adhesive bonding of the bandage 4 to the contact surface.

The variant in fig. CUP- T.8.2.1.3.3. For the direction of rotation indicated in the figure, the torque is transmitted from the driving half-coupling 1 through the cylindrical rollers 3 to the half-coupling 2, then through the assembly with the screws 5, mounted with clearance, to the flange 4. When changing the direction of rotation, the rollers 2 are released (uncoupled) and the transmission of the torque is interrupted, causing relative rotational movement between the two half-couplings (1 and 2) positioned radially by the bearings 6 fixed axially by the assembly with screws 5 between the flange 4 and the cover 9. To achieve a quick coupling (engagement), the rollers 3 are pressed by the pins 8 pushed by the springs 7

ETB. ELEMENTS OF TRIBOLOGY

ETB-T. ELEMENTS OF TRIBOLOGY: FRICTION AND WEAR

CONTENTS

ETB-T.1 DEFINITION	10-3
ETB-T.2 DESCRIPTION AND MODELING OF FRICTION PROCESSES	10-3
ETB-T.2.1 DEFINITIONS.....	10-3
ETB-T.2.2 CLASSIFICATION	10-3
ETB-T.2.3 MECHANICAL FRICTION CONNECTIONS	10-5
ETB-T.2.3.1 Moving links with relative movement (kinematic couplings) with friction (harmful)	10-5
ETB-T.2.3.2 Connections without relative movement (assembly, transmission) by friction (useful).....	10-7
ETB-T.2.4 MODELING OF FRICTION PROCESSES FROM KINEMATIC COUPLES	10-8
ETB-T.2.4.1 Dry friction modeling	10-8
ETB-T.2.4.2 Fluid friction modeling	10-11
ETB-T.2.4.3 Boundary and mixed friction modeling.....	10-13
ETB-T.3 DESCRIPTION AND MODELING OF WEAR PROCESSES	10-14
ETB-T.3.1 DEFINITIONS.....	10-14
ETB-T.3.2 Classification	10-14
.....	10-16
ETB-T.3.3 Modeling wear processes	10-16
ETB-T.4.1 DESCRIPTION AND MODELING OF LUBRICATION PROCESSES (LUBRIFICATION)	10-17
ETB-T.4.1.1 DEFINITIONS	10-17
ETB-T.4.1.2 CLASSIFICATION	10-17
ETB-T.4.1.3 LUBRICATION MODELING	10-18
ETB-T.4.2 LUBRICANTS	10-22
ETB-T.4.2.1 FUNCTIONS.....	10-22
ETB-T.4.2.2 CLASSIFICATION	10-22
ETB-T.4.2.3 PROPERTIES AND TYPES OF OILS.....	10-23
ETB-T.4.2.4 PROPERTIES AND TYPES OF GREASE	10-27
ETB-T.4.2.5 LUBRICATION WITH OTHER LUBRICANTS	10-28
ETB-T.4.2.6 LUBRICANT CHOICE	10-28
ETB-T.4.3 LUBRICATION METHODS AND DEVICES.....	10-28
ETB-T.4.3.1 OIL LUBRICATION METHODS AND DEVICES	10-28
ETB-T.4.3.2 GREASE LUBRICATION METHODS AND DEVICES	10-29
ETB-T.4.4 SEALING METHODS AND DEVICES	10-30
ETB-T.4.4.1 FUNCTIONS.....	10-30
ETB-T.4.4.2 MATERIALS FOR SEALING ELEMENTS	10-30
ETB-T.4.4.3 CLASSIFICATION AND TYPES OF SEALS.....	10-31

ETB-T.1 DEFINITION

Tribology is “the science and technology of the interaction of moving surfaces and the resulting applications” [D. Tabor, 1954]

Tribology is the science that deals with the study of the processes of friction, wear and lubrication that accompany the movements and/or interactions between the elements of mechanical technical products.

ETB-T.2 DESCRIPTION AND MODELING OF FRICTION PROCESSES

ETB-T.2.1 DEFINITIONS

Friction is “a complex process, of molecular, mechanical and energetic nature, which manifests itself between contact surfaces with relative motion” [Rădulescu, 1981].

Friction, according to DIN 50323, is the interaction between the real areas of bodies in contact and opposes relative motion or the tendency to move.

Friction, according to STAS 8069-87, is the physical phenomenon that occurs between two surfaces in contact, simultaneously subjected to a normal pressure force and a tangential force, which opposes the relative state of motion.

Note. Friction processes are energy dissipative and lead to *the generation of heat and the modification of the state of stresses and deformations* of the two elements in contact, contributing to the *deformation and/or breakage of a number of metal particles* on the surfaces in contact (wear). Reducing friction leads to a decrease in energy consumption (increased efficiency) and, at the same time, an increase in the life span of the products by reducing wear.

ETB-T.2.2 CLASSIFICATION

Criterion	Type	Description
Functional role	Harmful	<i>In anti-friction</i> processes/systems through the effects of heating, wear, energy loss (low efficiency), local heating and expansion, and vibrations leading to improper operation and, ultimately, to removal from service
	helpful	<i>In friction</i> processes/systems (brakes, friction transmissions, couplings) through the ability to transmit forces and moments under the negative effects of heating, wear and vibration
Area where it appears	outer	It appears in the areas of contacting surfaces of bodies and mechanical systems

	Intern	Inside materials (rubber); hysteresis behavior (energy storage)
Lubricant film thickness in the contact area (friction regimes depending on the lubrication condition)	Dry	Without lubricant
	Limit	Superficial boundary layers, resulting from oxidation, adsorption and/or chemisorption of molecules
	Mixed (semi-fluid)	Partial film of fluid and partial direct contact between bodies.
	Fluid	Complete separation by fluid film (hydrodynamic or elastohydrodynamic lubrication).
According to the nature of the relative motion between bodies	Sliding	<i>Sliding friction force</i> F_{fa} , at the contact between flat surfaces (fig. ETB-T.2.1,a) or cylindrical surfaces; <i>sliding friction moment</i> M_{fa} , on cylindrical surfaces (fig. ETB-T.2.1,b)
	Rolling	<i>The rolling friction moment</i> M_{fr} , when a cylinder rolls on a plane (fig. ETB-T.2.1,c) or when a sphere rolls on a plane (fig. ETB-T.2.1,e)
	Pivoting (spin)	<i>Pivoting friction moment</i> M_{fp} , at the frontal contact of a cylinder with a plane
	mix	Rolling with sliding (gears, bearings); rolling with pivoting (radial-axial bearings; fig. ETB-T.2.1,d); sliding with pivoting
The existence of relative motion	Static (resting)	There is a tendency for movement (prevents the occurrence of relative movement)
	Kinetic	There is relative motion.

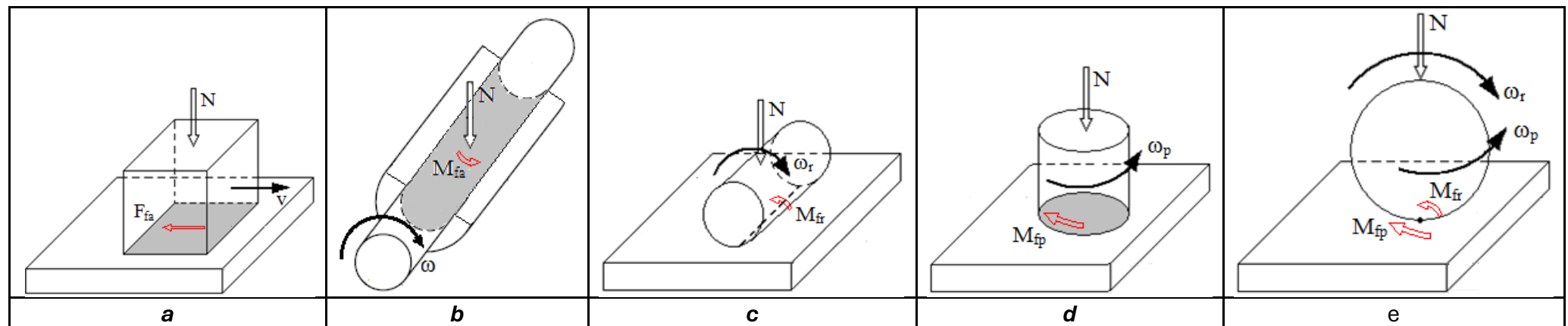


Fig. ETB-T.2.2.1 Types of friction: **a** – sliding friction on a flat surface; **b** – sliding friction on a cylindrical surface; **c** – rolling friction between a cylinder and a plane; **d** – pivoting friction; **d** – combined friction (rolling and pivoting)

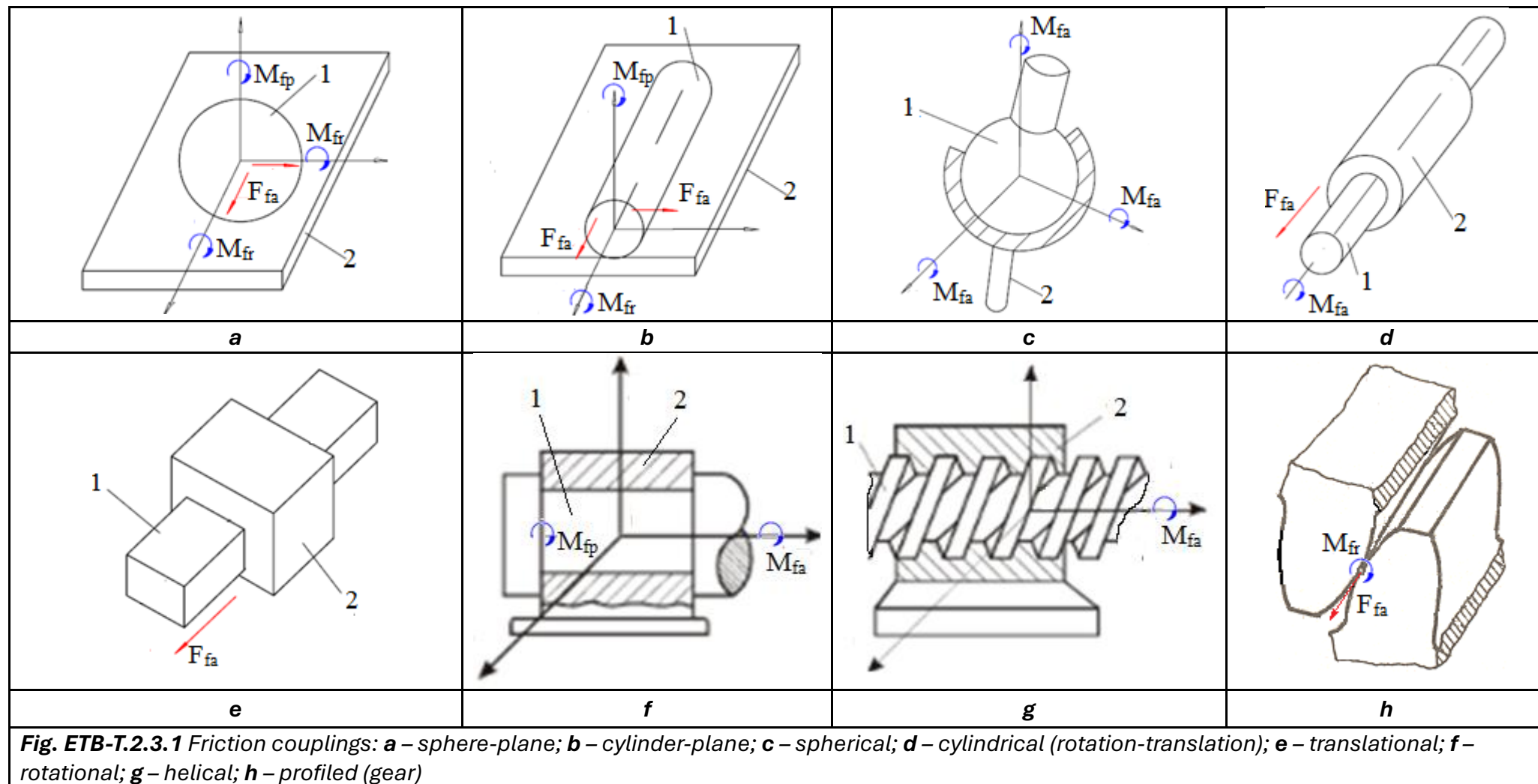
ETB-T.2.3 MECHANICAL FRICTION CONNECTIONS

ETB-T.2.3.1 Moving links with relative movement (kinematic couplings) with friction (harmful)

Movable connections (kinematic couplings) with friction are assemblies with two or more bodies in contact with relative movements (rotations and/or translations) that have the role of transmitting loads (forces and/or moments). Friction in these connections is harmful and, in order to increase efficiency, its reduction is sought, mainly by replacing sliding friction with rolling friction, by using antifriction materials and lubrication systems.

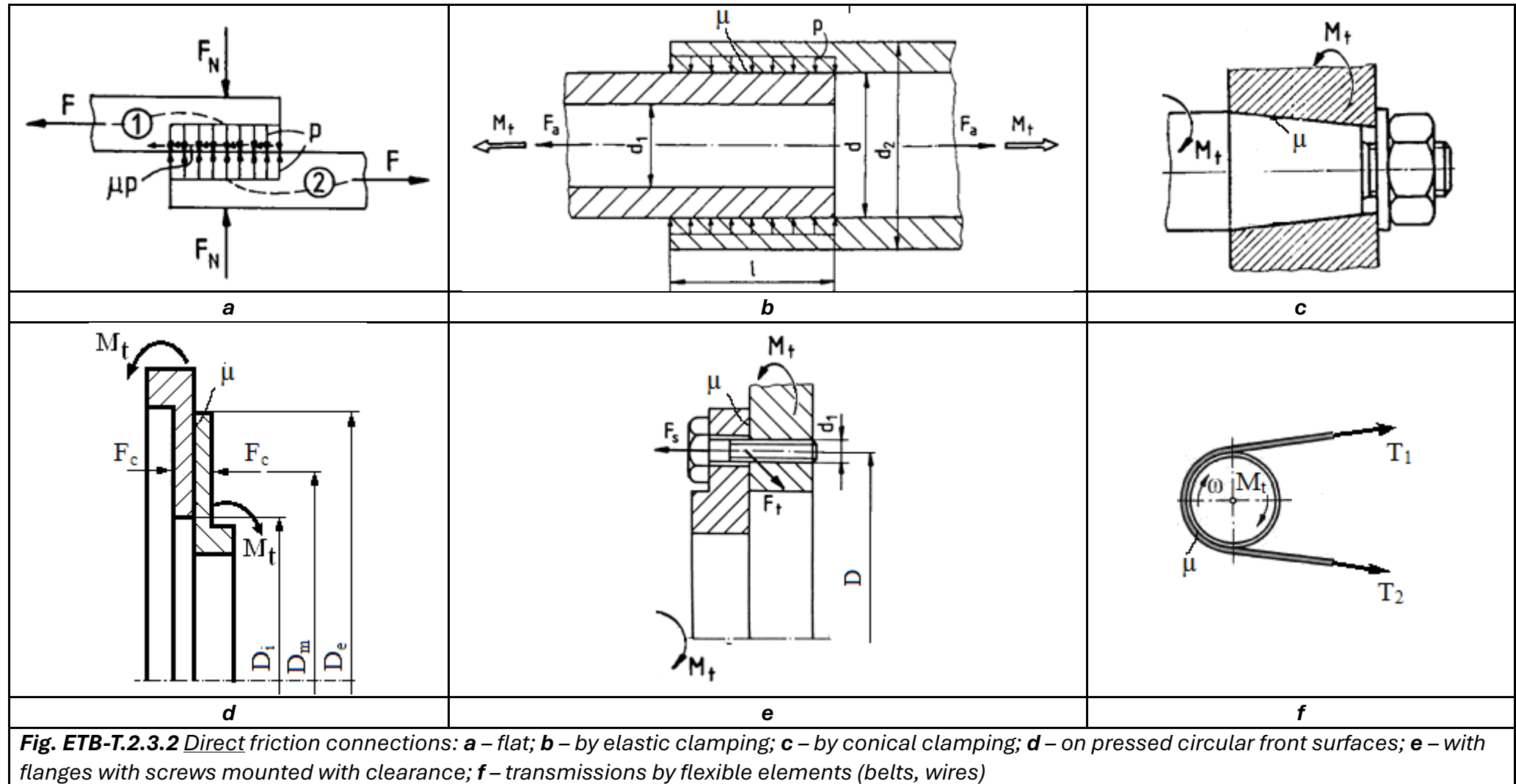
Criterion	Type	Observations
Type of contact surfaces	<u>Conform</u> (conjugate surfaces; lower couplings)	The surfaces of the contacting bodies “fit exactly ” and can deform together; the contact area is comparable to the dimensions of the bodies; e.g. sliding guides, plain bearings (fig. ETB-T.2.1, c, d, e, f)
	<u>Nonconforming</u> (nonconjugate surfaces; Hertzian contacts, higher couplings)	The surfaces in the unloaded state can be point-like or linear. When applying external loading the contact area has dimensions much smaller than the dimensions of the bodies; e.g. gears, bearings, variators (fig. ETB-T.2.1, a, b, h)
Number of suppressed relative movements	Class I (1 movement suppressed)	5 resistances (forces and moments) appear (fig. ETB-T.2.1,a)
	Class II (2 movements suppressed)	4 resistances (forces and moments) appear (fig. ETB-T.2.1,b)
	Class III (3 movements suppressed)	3 resistances (forces and/or moments) appear (fig. ETB-T.2.1,c)
	Class IV (4 movements suppressed)	2 resistances (forces and/or moments) appear (fig. ETB-T.2.1,d)
	Class V (5 movements suppressed)	1 resistance (force or moment) appears (fig. ETB-T.2.1,e)
Number of contacts	With single contact	Without lubricant
	With multiple contacts	Superficial boundary layers, resulting from oxidation, adsorption and/or chemisorption of molecules
Contact geometry	pencil	Non-conforming friction couplings or Hertzian couplings with elliptical point contact (ball bearings and screws, spherical roller bearings, speed variators with toroidal intermediate elements)
	Linear	Cylindrical plain bearings with relatively large clearance, cylindrical roller bearings, spur gears, cylindrical roller variators, chains, spur couplings, etc.
	FLAT	Thrust bearings, front seals, guides, brakes, etc.
	Cubic	Low clearance plain bearings, bushings for chains, wide belts, drum brakes, etc.),
	Conical	Tapered roller bearings with low clearance

	ball	Low -backlash ball joints
Type of friction	Sliding	Fig. ETB-T.2.1, c, e, g)
	Rolling	Fig. ETB-T.2.1, a, b, h)
	Pivoting	Fig. ETB-T.2.1, a,b)
	mix	Fig. ETB-T.2.1, a, h)



ETB-T.2.3.2 Connections without relative movement (assembly, transmission) by friction (useful)

Frictional *connections* are assemblies of two or more bodies in contact without relative motion, and their role is to transmit loads (forces and/or moments), usually *active* (main load flows in assemblies and transmissions). Friction in these connections is *useful* and its maximization is sought, mainly, by using contacts in which dry sliding friction occurs between surfaces made of materials with increased friction characteristics (*friction materials*).



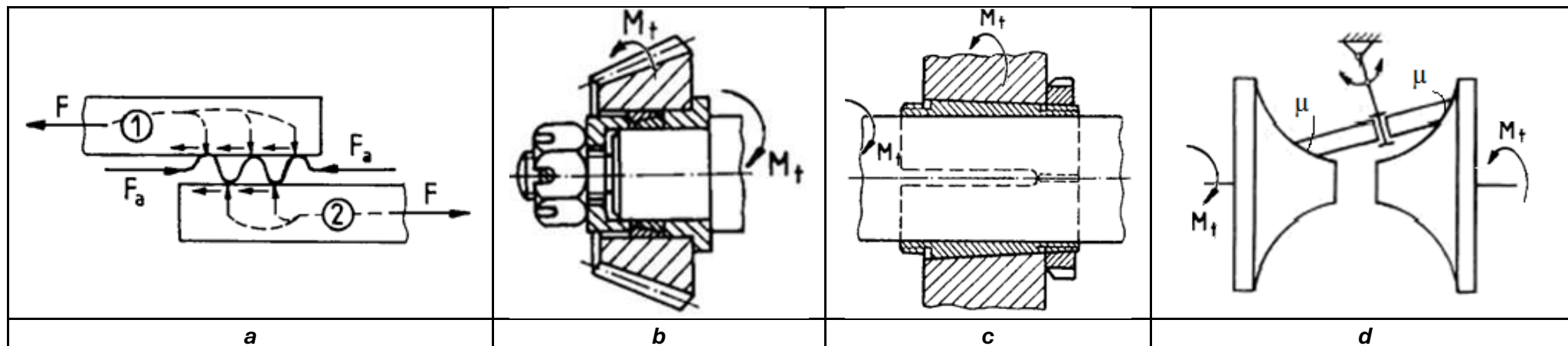


Fig. ETB-T.2.3.3 *Indirect friction connections (with intermediate elements): a – with corrugated elastic element; b – with frustoconical rings; c – with elastic bushing; d – toroidal speed variator*

ETB-T.2.4 MODELING OF FRICTION PROCESSES FROM KINEMATIC COUPLES

ETB-T.2.4.1 Dry friction modeling

ETB-T.2.4.1.1 Modeling dry sliding friction

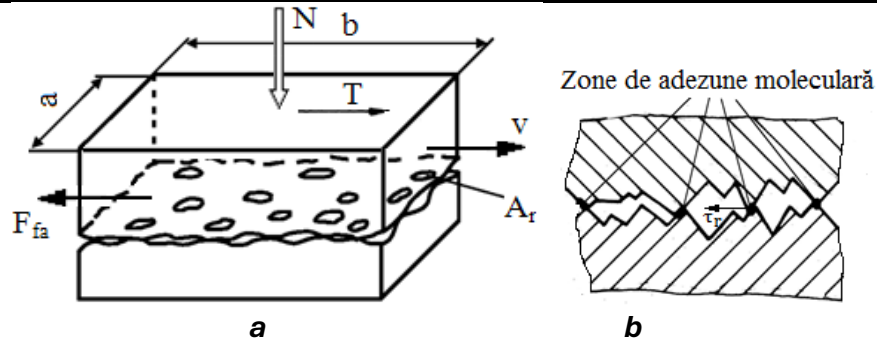


Fig. ETB-T.2.4.1 Dry sliding friction process: **a** – general scheme; **b** - detail

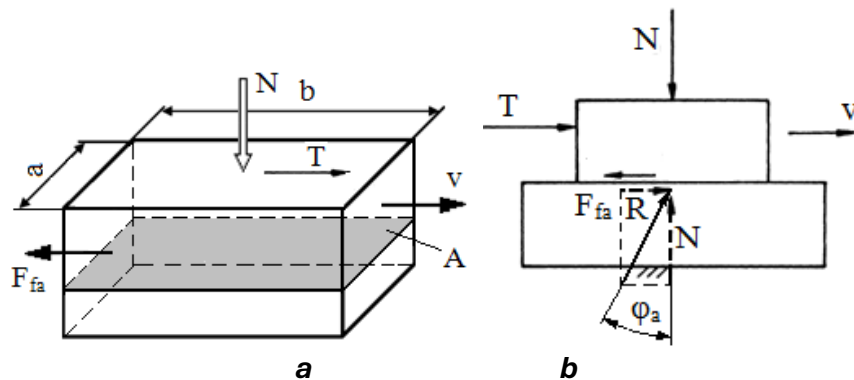


Fig. ETB-T.2.4.2 The classical model of dry sliding friction: **a** – general scheme; **b** - force scheme

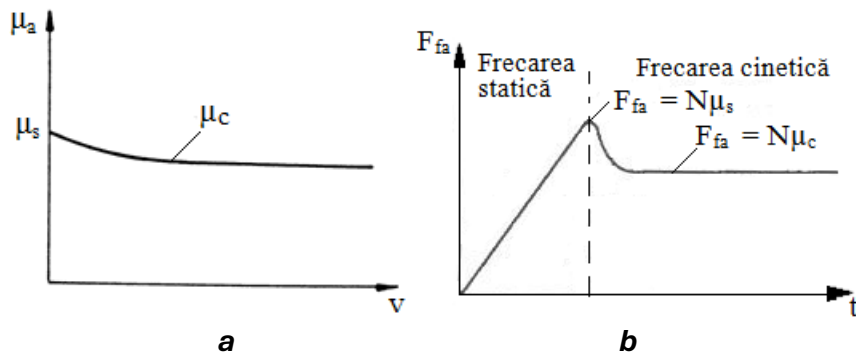


Fig. ETB-T.2.4.3: Variations of the characteristic parameters of friction: **a** – friction coefficient; **b** – driving force

The process of dry sliding friction. With the increase of the driving force, \$T\$, in the case of the existence of the normal pressure force, \$N\$, on the contacting surfaces the microirregularities deform elastically or plastically and the friction forces appear which are determined by the shear strength of the adhesion microzones (fig. ETB-T.2.4.1). In the case of relative movement the adhesion points break and consequently, dry friction is characterized by high friction coefficients and significant wear.

Amontons-Coulomb law,

$$F_{fa} = \mu_a N, \quad (\text{ETB-T.2.4.1})$$

where, \$N\$ represents the normal pressing force, \$F_{fa}\$ - the sliding friction force, \$\mu_a\$ - the sliding friction coefficient dependent on the types of materials and independent of the contact area \$A\$ and the relative velocity \$v\$.

The reaction slope (the resultant of the normal force \$N\$ and the friction force \$F_{fa}\$) can be determined from the relationship (fig. ETB-T.2.4.2),

$$\text{tg } \phi_a = \frac{F_{fa}}{N} = \frac{\mu_a N}{N} = \mu_a \quad (\text{ETB-T.2.4.2})$$

from which \$\phi_a = \arctan \mu_a\$, the angle of sliding friction.

Springden-Tabor law,

$$F_{fa} = \tau_r A_r = \tau_r \frac{N}{\sigma_c} = \mu_a N \quad (\text{ETB-T.2.4.3})$$

where, \$N\$ represents the normal pressing force, \$\mu_a = \tau_r / \sigma_c\$ - coefficient of sliding friction; \$\tau_r\$ - shear strength of the softer material (fig. ETB-T.2.4.1,b), \$\sigma_c\$ - compressive yield stress (resistance) of the softer material, \$F_{fa}\$ - sliding friction force equal to the breaking force of the protrusions that form the real contact area, \$A_r\$ (much smaller than the nominal area \$A\$).

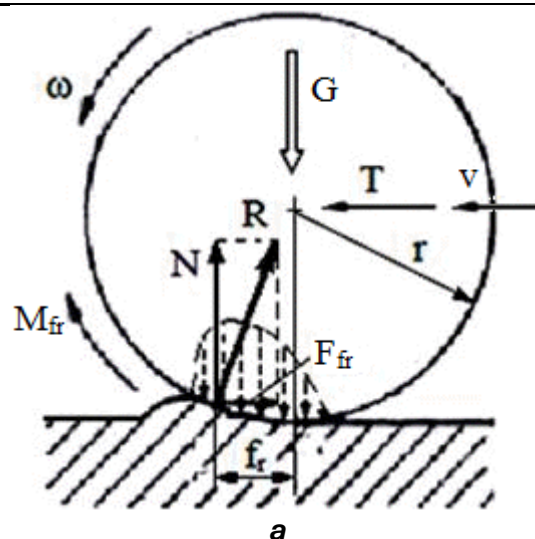
Note.

1. The coefficient of sliding friction is variable with the relative speed of movement of the bodies in contact (fig. ETB-T.2.3.3,a): the static friction

coefficient, μ_s , when there is no relative motion ($v = 0$); the kinetic friction coefficient, μ_s , when there is relative motion ($v > 0$).

2. The friction process over time (start-displacement) according to the variation in fig. ETB-T.2. 4.1,b: at small and increasing driving forces, T , the friction force increases until, $T = F_{fa} = \mu_s N$ and the angle ϕ between R and N becomes $\phi = \phi_a$; after this condition is met, the relative movement begins and the driving force T decreases as a consequence of the decrease in the value of the friction coefficient ($\mu_c < \mu_s$)

ETB-T.2.4.1.2 Modeling dry rolling friction



The process of dry rolling friction. In the real case, the bodies in contact are elastically deformable and the resistance to displacement under the action of the driving force T is determined by the resultant force, N , of the pressures on the contact surface which are concentrated at a point placed outside the plane of symmetry at a distance, f_r (fig. ETB-T.2. 4. 4).

From the equilibrium equations (fig. ETB-T.2.4.5),

$$T = F_{fr}; N f_r = T r, \quad (\text{ETB-T.2.4.1.4})$$

result,

$$T = F_{fr} = \frac{f_r}{r} N = \mu_r N, \quad (\text{ETB-T.2.4.1.5})$$

$$M_{fr} = f_r N, \quad (\text{ETB-T.2.4.1.6})$$

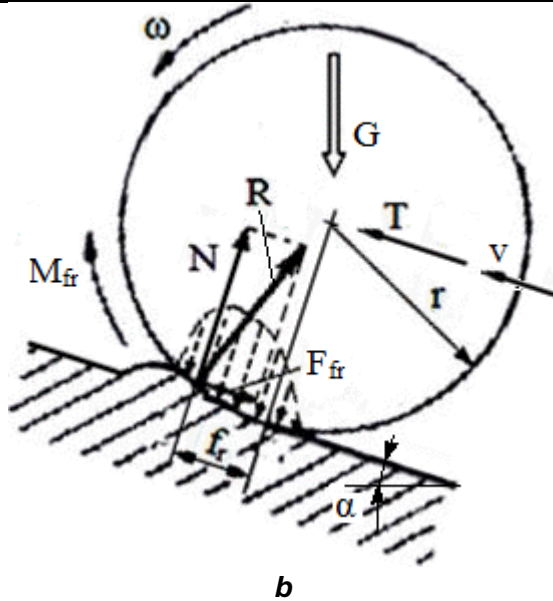
where: F_{fr} represents the rolling friction force (rolling resistance), $\mu_r = f_r / r$ - the coefficient (factor) of rolling resistance (friction), M_{fr} - the rolling friction moment, r - the wheel radius.

Possible movement states:

- Rest: $T < N f_r / r$, $T r < M_{fr}$;
- Sliding movement: $T > N f_r / r$, $T r < M_{fr}$;
- Rolling motion: $T < N f_r / r$, $T r > M_{fr}$, rolling will start more easily the lower the rolling friction coefficient (f_r / r) is;
- Combined rolling and sliding motion (real case): $T > N f_r / r$, $T r > M_{fr}$.

Note.

In the case of vehicle wheels, $N = G$ (horizontal rolling, fig. ETB-T.2. 4. 4,a), or $N = G \cos \alpha$, (uphill rolling, fig. ETB-T.2. 4. 4,b), the rolling resistance becomes, $F_{fr} = \mu_r G$ or, respectively, $F_{fr} = \mu_r G \cos \alpha$, where: G represents the weight on the wheel, α - the angle of the rolling track; c the rolling



resistance coefficient μ_r , frequently denoted by f , has the average value $f = 0.018$, for asphalt or concrete rolling track)

Fig. ETB-T.2.4. 4 Models of dry rolling friction: **a** – wheel in horizontal plane; **b** – wheel on slope

ETB-T.2.4.2 Fluid friction modeling

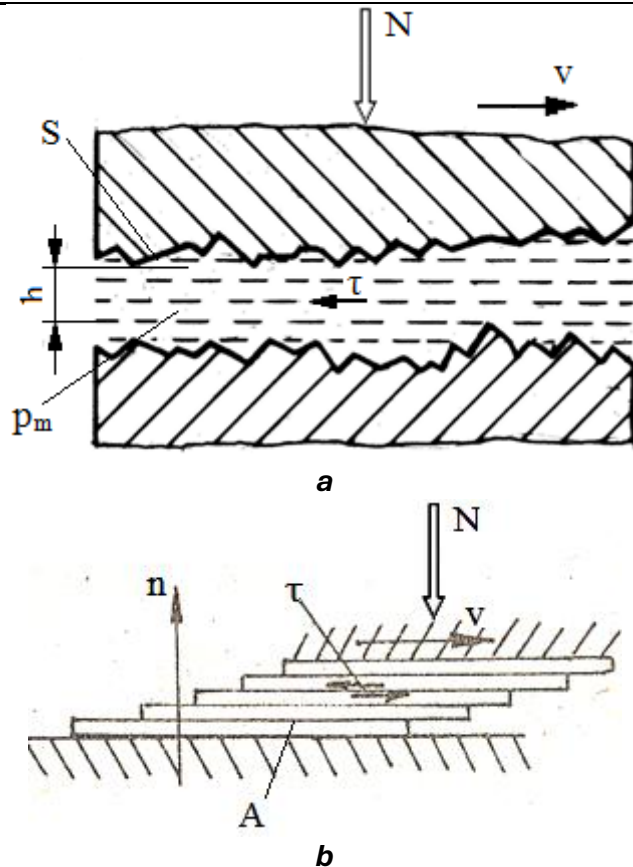


Fig. ETB-T.2.4.6 Fluid friction: **a** – diagram with parameters; **b** – simplified model

Fluid friction occurs when the contacting surfaces are separated by a film of lubricant h thicker than the maximum heights of the asperities (fig. ETB-T.2.4.6,a). The forces opposing the relative displacement are generated by the tangential stresses in the fluid, which according to Newton's law, are determined by the relationship,

$$\tau = \eta \frac{dv}{dn}, \quad (\text{ETB-T.2.4.1.7})$$

where, η [N s/m²] represents the dynamic viscosity of the lubricant; v [m/s] – relative velocity; n [m] – distance in the normal direction; at the limit considering: $dv = \Delta v = v$, $dn = \Delta h = h$, the simplified form of Newton's law results, $\tau = \eta v/h$.

The friction process inside the fluid is studied according to the theory of fluid mechanics.

Simplified calculation model

Conventionally, it can be considered that the friction force is the result of the internal shear of the fluid at the contact surface of area A (fig. ETB-T.2.4.6,b),

$$F_{ff} = \mu_f N = \tau A, \quad (\text{ETB-T.2.4.1.8})$$

From this relationship, taking into account the relationship (ETB-T.2.4.1.7) at the limit, the friction coefficient in the fluid is obtained,

$$\mu_f = \frac{\tau A}{N} \approx \frac{\eta \frac{v}{h} A}{N} = \frac{\eta v}{h \frac{N}{A}} = \frac{\eta v}{h p_m}, \quad (\text{ETB-T.2.4.1.9})$$

where p_m is the average pressure in the lubricant, h – the thickness of the lubricant film; practically, $\mu_f = 10^{-4} \dots 10^{-2}$.

The external normal force is taken over by the resultant of the pressure forces existing in the lubricant film and is called the bearing force or lift.

Fluid friction occurs in two distinct lubrication situations: hydrodynamic and hydrostatic lubrication.

ETB-T.2.4.3 Boundary and mixed friction modeling

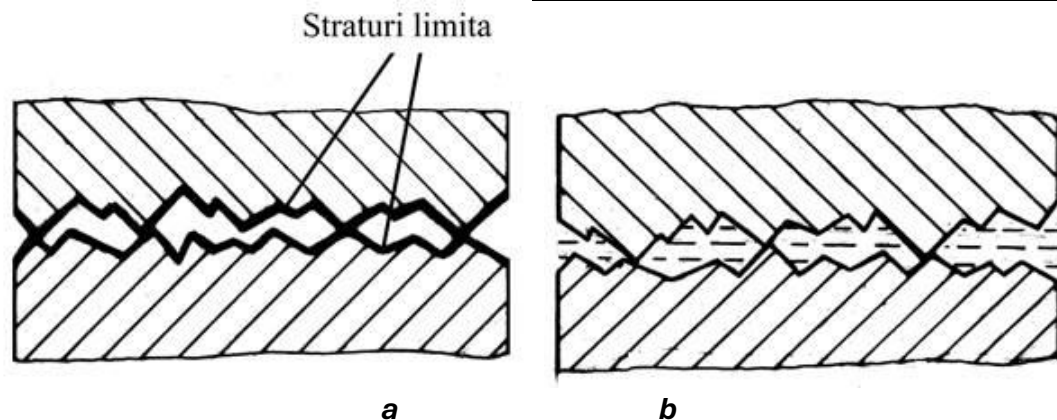


Fig. ETB-T.2.4.7 Intermediate friction schemes: **a** – boundary; **b** – mixed

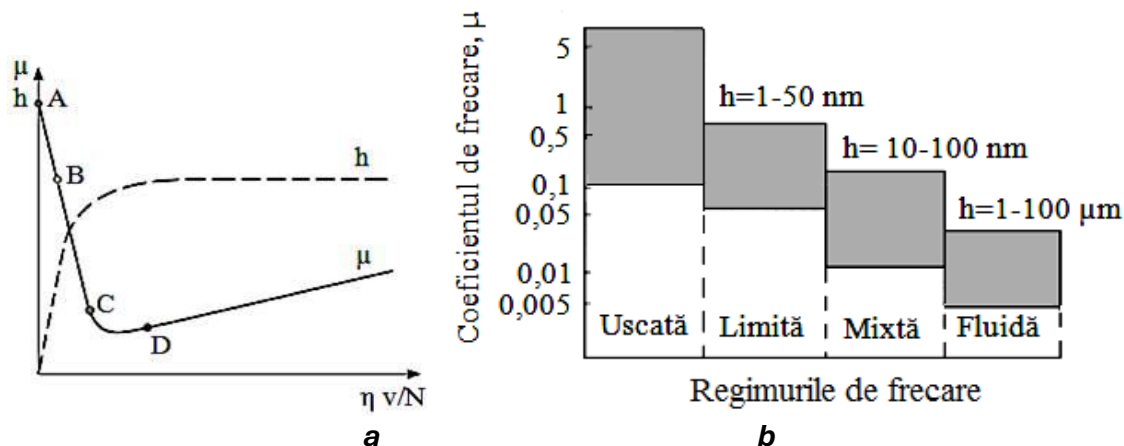


Fig. ETB-T.2.4.8 Variation of the friction coefficient: **a** - Stribeck curve; **b** - informative values [Wen, 2012]

Boundary (semi-dry) friction occurs when thin (boundary) molecular layers appear between the contacting surfaces (fig. ETB-T.2.4.7,a), formed as a result of physical (adsorption) and/or chemical (chemisorption) phenomena. Due to the much lower intermolecular (Van der Waals) bonds within this layer than in the case of molecular adhesion bonds (dry friction), the friction forces decrease by 2...3 times and wear can decrease by hundreds or even thousands of times. In the case of boundary friction, the laws of classical friction also apply.

Mixed (semi-fluid) friction occurs in situations where a lubricant film is formed interrupted by the roughness of the contacting surfaces (fig. ETB-T.2.4.7,b). Conventionally, friction forces are calculated as for dry friction taking into account friction coefficients (see fer. ETB.P.1.1) which are also dependent on the roughness of the surfaces. In the mixed friction regime, the stick-slip phenomenon frequently occurs, characterized by a rapid, periodic alternation between adhesion (static) and sliding friction, the movement being jerky.

In fig. ETB-T.2.4.8,a the variation of the friction coefficient, μ , and the thickness of the lubricant film, h , is presented, depending on the product of the dynamic viscosity (η) and the relative velocity (v) in relation to the pressing force (N). Thus, the following areas are identified:

- AB – dry or boundary friction;
- BC – mixed friction;
- CD – fluid friction

ETB-T.3 DESCRIPTION AND MODELING OF WEAR PROCESSES

ETB-T.3.1 DEFINITIONS

Wear is an energetic process of destruction of the surface layers of solid bodies in dry and/or fluid contact. *Frictional wear* occurs in the case of relative movement of bodies under the action of an external load. The process of destruction of surface layers involves, on the one hand, structural, chemical and physical changes and, on the other hand, deformations and detachments of material. There is a close interdependence between the wear and friction processes in the sense that wear is a consequence of friction, and the state of the surfaces resulting from wear influences friction.

ETB-T.3.2 CLASSIFICATION

Criterion	Type of wear	Forms of manifestation	Description
Chemical and physical phenomena	Adhesion	Friction (scoring)	In the areas of contact asperities, small transfers of material occur from one surface to another, which, following repeated frictional contacts, lead to <u>local deformations and hardening and to the breakage of small particles</u> .
		Seizure (Scuffing)	It occurs in the case of dry friction or local disappearance of the lubricant, when due to <u>increased pressures and frictions</u> , <u>local temperatures increase</u> and <u>plastic deformations and/or welding</u> occur at the level of the contact micro-asperities followed by their <u>breakage</u> . This sequence is repeated and deep marks appear in the form of scratches on the contacting surfaces which become unusable. In the extreme case, the welding bridges can no longer be sheared and <u>the kinematic coupling locks up</u> .
	Abrasion	Microchipping	It occurs due to <u>hard impurities in the contact area</u> that act as cutting elements
	Fatigue	pitting	It is characteristic of couplings with rolling motion with point contacts (ball bearings) or linear (roller bearings, gears) that are lubricated. After a large number of <u>time-varying stresses</u> in the areas of surfaces with microcracks or inclusions (from processing, heat treatment and/or cutting; fig. ETB-T.3.1) initial cracks appear, due to <u>material fatigue</u> , which, as a consequence of the <u>penetration of the lubricant and very high hydrostatic pressures</u> , enlarge, deform, break and finally material particles detach and microcraters form (<u>initial pitting</u>). Over time, these microcraters multiply and enlarge (<u>progressive pitting</u> , fig. ETB-T.3.2) and, ultimately, lead to damage to the surfaces in contact and therefore to improper operation of the coupling (with vibrations and/or noise).
		Exfoliation	It generally appears on the surfaces of <u>heavily loaded low-speed couplings</u> , especially in the case of defects (inclusions, inhomogeneities, microcracks) under the hardened surface layers (by hardening,

			cementation or nitriding) and is manifested, following time-varying stresses, by the appearance of <u>material dislocations</u> and voids (fig. ETB-T.3.3)
		cavitation	It occurs mainly on surfaces permanently immersed in fluids (including kinematic couplings) that operate at <u>very high speeds</u> (turbine blades, diesel engine cylinders, pump rotors). Low pressures and high temperatures can occur in the oil and therefore the formation of small <u>pockets of vapor and gas</u> (cavitation bubbles, fig. ETB-T.3.4) which, due to pressure variations and contacts with colder surfaces, <u>suddenly burst (by implosion)</u> and create <u>repeated pressure pulses</u> that cause <u>surface erosion</u> due to material fatigue.
	Corrosion	Cleaning	It results from the chemical action of water, oxygen or other aggressive substances from the surrounding environment and/or lubricants.
		mechano-chemical	It occurs in the case of relative movements of surfaces in contact, manifesting itself, on the one hand, by the removal of the corroded superficial layer, and, on the other hand, by changes in the superficial layer; these processes combined with those of friction (large releases of thermal energy, high local pressures, accumulations of electrostatic potential) make possible or accelerate chemical reactions at the level of surfaces in contact.

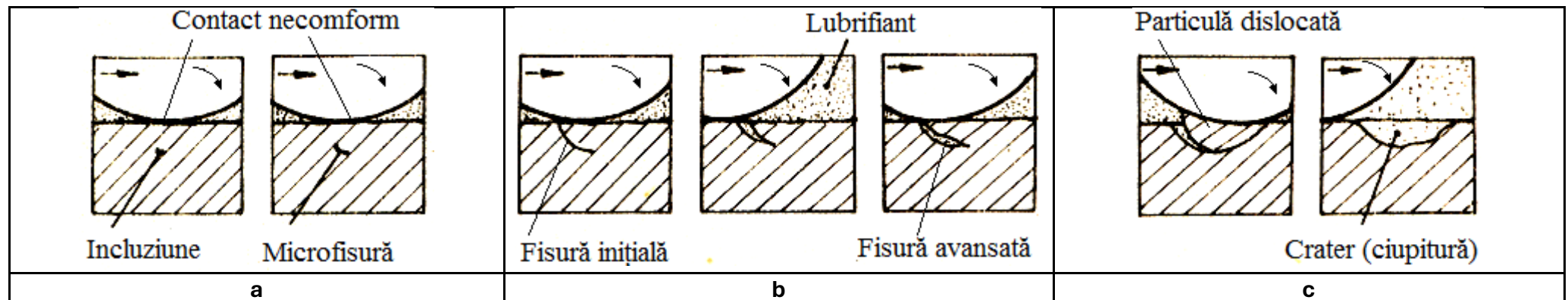
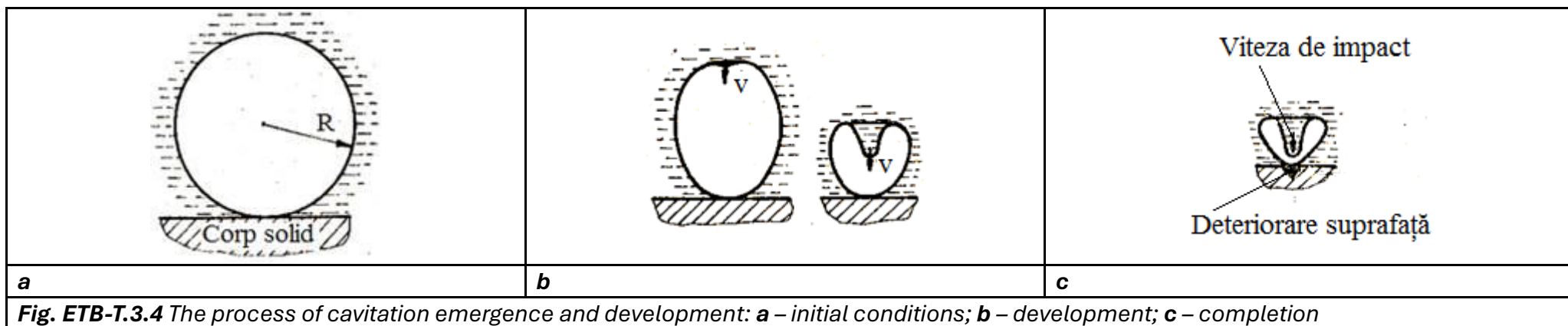
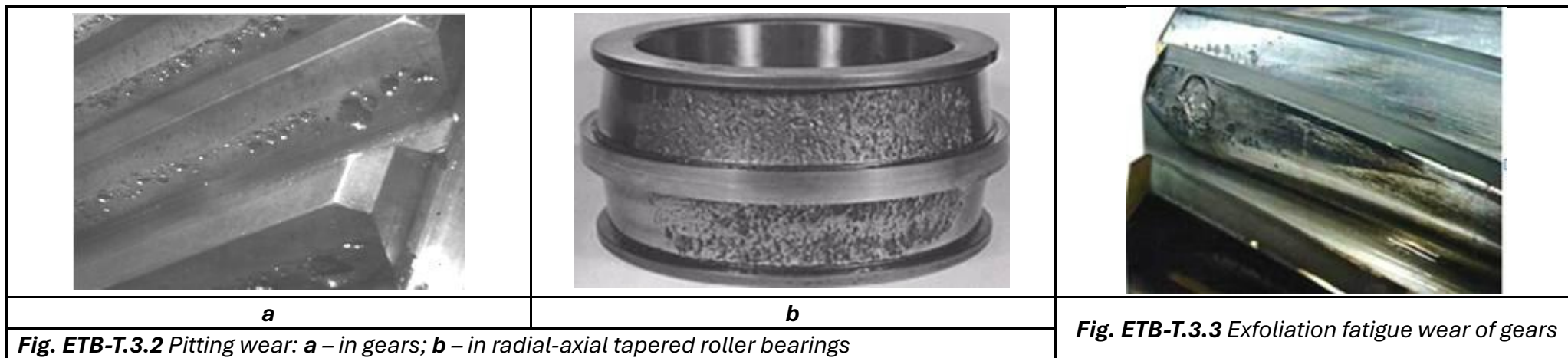


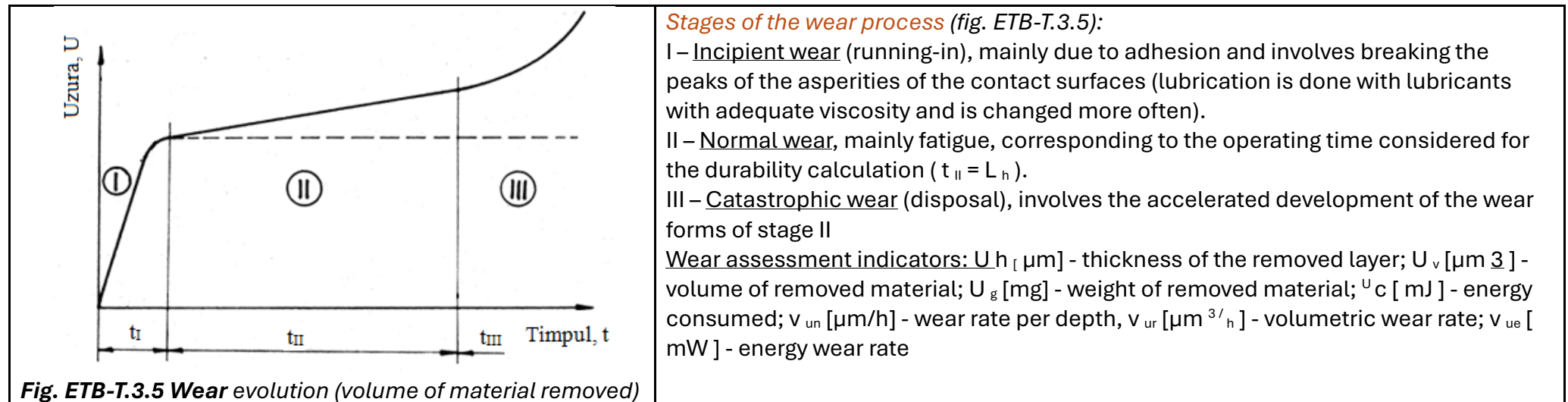
Fig. ETB-T.2.3.1 The process of pitting emergence and development: **a** – initial conditions; **b** – development; **c** – completion



ETB-T.3.3 MODELING WEAR PROCESSES

Operating period	Description	Observations
Running-in (Break-in)	Wear is accelerated (mainly abrasive) as a result of the mutual smoothing of new surfaces in contact (roughness decreases)	Running-in is performed under loads and at reduced speeds, progressively increasing, in special lubrication conditions (more fluid lubricant, with changes at short intervals)

Normal operation (operation),	Wear evolves very slowly due to one and/or another form of wear, the shapes and dimensions remain within the limits imposed by the correct functioning of the respective cup;	
Critical (obsolete)	when due to Rapid increase in wear rate, catastrophic wear and deviations in shape and dimensions of the contacting surfaces occur	Malfunction occurs and ultimately leads to the coupling being taken out of service.



ETB-T.4.1 DESCRIPTION AND MODELING OF LUBRICATION PROCESSES (LUBRIFICATION)

ETB-T.4.1.1 DEFINITIONS

Lubrication is the complex process of forming and maintaining the bearing lubricant film that ensures fluid friction.

Additional functional characteristics of lubrication: heat transfer, operation at high speeds, operation at low speeds, possibility of changing the operating position, ensuring fluid friction during intermittent movements, avoiding environmental contamination.

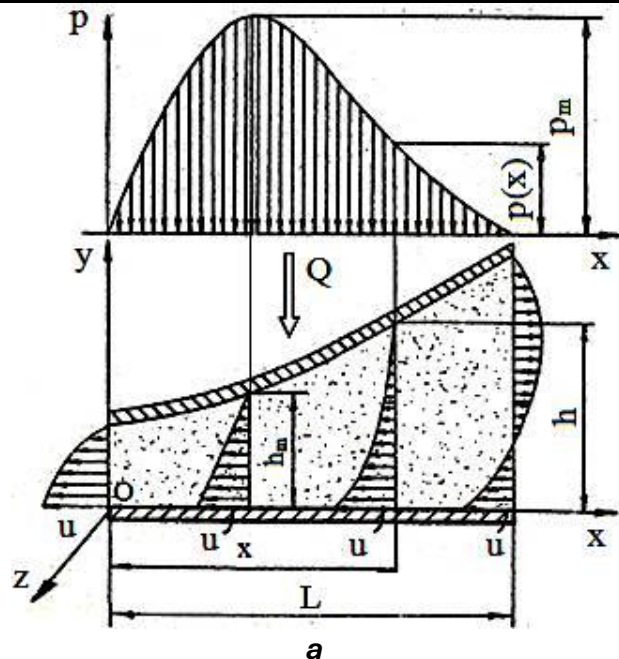
ETB-T.4.1.2 CLASSIFICATION

Criterion	Type	Description
Principle of generating the bearing lubricant film	Hydrodynamics/ Gasdynamics	It is achieved by introducing the lubricant, without pressure or with reduced pressure, into a key-shaped gap, which, with increasing relative speed, provides lift.
	Hydrostatic/ Gasstatic	The lubricant is forced in by an external pump at the required pressure and flow rate to permanently maintain the lubricant film.

Fluid type	Liquids (oils)	Liquids are incompressible lubricating media
	GAS	Gases are compressible lubricating media that provide very low friction resulting in good heat transfer and very low wear.
	Semi-solids	Thick greases
Considering the elastic deformability of the materials of the surfaces in contact	With rigid contact surfaces	If the contact areas are heavily hardened
	Elastohydrodynamic	The superficial contact areas are soft
Lubricant film thickness	With thin film	Lubricant film thickness 1...10 μm
	With thick film	Lubricant film thickness 10...100 μm

ETB-T.4.1.3 LUBRICATION MODELING

ETB-T.4.1.3.1 Hydrodynamic lubrication modeling



Calculation model of hydrodynamic lubrication with key effect

Assumptions of the calculation model based on the key effect (fig. ETB-T.4.1.3.1.1):

- the hydrodynamic process complies with Newton's basic law, according to rel. (ETB-T.2.4.1.7),
- the fluid flow is laminar,
- the fluid is practically incompressible,
- inertia forces are neglected,
- viscosity varies only with temperature,
- the pressure is constant over the thickness of the lubricant film,
- the lubricant adheres to the surfaces of parts in relative motion,
- the length in the z direction is infinite (flow is in one direction only).

The Reynolds equation is the basic one-way hydrodynamic equation.

$$\frac{\partial}{\partial x} \left(h^3 \frac{\partial p}{\partial x} \right) = 6 u \eta \frac{\partial h}{\partial x}, \quad (\text{ETB-T.4.1.3.1.1})$$

which through integration leads to the distribution of hydrodynamic pressures p depending on the variations in the thickness h of the lubricant film and the velocity u and viscosity η of the lubricant.

For the particular case (fig. ETB-T.4.1.3.1,b) of the interstitium with flat surfaces with thickness [Olaru, 2002],

$$h = h_1 + kx, \quad (\text{ETB-T.4.1.3.1.2})$$

where $k = \tan \alpha$, the relationship (ETB-T.4.1.3.1.1) becomes,

$$\frac{\partial}{\partial x} \left((h_1 + kx)^3 \frac{\partial p}{\partial x} \right) = 6 u \eta k$$

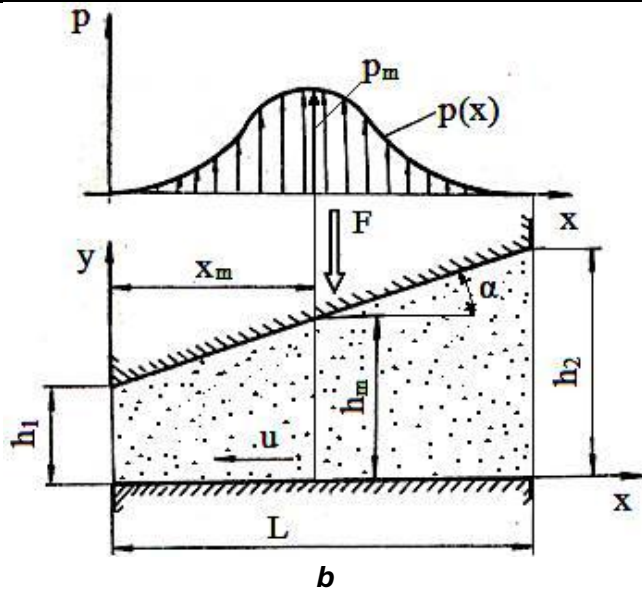


Fig. ETB-T.4.1.3.1.1 Hydrodynamic lubrication model based on the key effect: **a** – general case (any fixed surface); **b** – particular case (the fixed surface is flat)

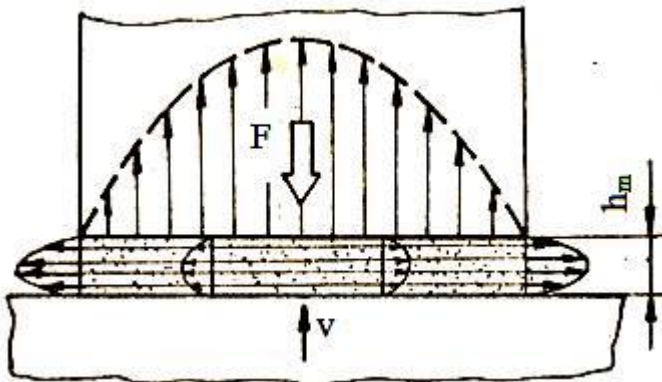


Fig. ETB-T.4.1.3.1.2 Hydrodynamic lubrication model based on the squeeze effect

By integrating this relationship, we obtain the relationship,

$$(h_1 + kx)^3 \frac{\partial p}{\partial x} = 6 u \eta k x + C_1, \quad (\text{ETB-T.4.1.3.1.3})$$

which by rearrangement becomes,

$$\frac{\partial p}{\partial x} = 6 u \eta k \frac{x}{(h_1 + kx)^3} + \frac{C_1}{(h_1 + kx)^3}. \quad (\text{ETB-T.4.1.3.1.4})$$

Understanding this relationship determines the expression of pressure,

$$p(x) = 6 u \eta k \int \frac{x}{(h_1 + kx)^3} dx + C_1 \int \frac{1}{(h_1 + kx)^3} dx + C_2, \quad (\text{ETB-T.4.1.3.1.5})$$

By imposing the boundary conditions, for \$x \in [0, L]\$, \$p = 0\$, which consider that both at the entrance and exit of the gap the lubricant pressure is zero quasi-equal to the atmospheric pressure, the integration constants \$C_1\$ and \$C_2\$ and the final form of the pressure expression are obtained (fig. ETB-T.4.3.1,b),

$$p(x) = \frac{6 \eta u}{k} \left(\left(\frac{h_1 h_2}{h_1 + h_2} \right) \frac{1}{(h_1 + kx)^2} - \frac{1}{h_1 + kx} + \frac{1}{h_1 + h_2} \right). \quad (\text{ETB-T.4.1.3.1.6})$$

$$\frac{\partial p(x)}{\partial x} = 0$$

From the condition \$\frac{\partial p(x)}{\partial x} = 0\$ the position of the pressure maximum results,

$$x_m = \frac{1}{h_1 + h_2/h_1} L. \quad (\text{ETB-T.4.1.3.1.7})$$

By replacing the relationship (ETB-T.4.1.3.1.7) in the relationship (ETB-T.4.1.3.1.6) the thickness of the gap (lubricant film) in the area of maximum pressure is obtained,

$$h_m = \frac{2 h_1 h_2}{h_1 + h_2} \quad (\text{ETB-T.4.1.3.1.8})$$

The load that the lubricant film can support (hydrodynamic lift) is determined by the relationship

$$F = - \int_0^L p(x) B dx \quad (\text{ETB-T.4.1.3.1.9})$$

where \$B\$ is the width of the flat surfaces. By substituting the relation (ETB-T.4.3.1.6) into the relation (ETB-T.4.3.1.9) and calculating the integral, we obtain,

$$F = - \frac{6 \eta B u}{k} \left(\frac{2 L}{h_1 + h_2} - \frac{1}{k} \ln \left(\frac{h_2}{h_1} \right) \right). \quad (\text{ETB-T.4.1.3.1.10})$$

Hydrodynamic lubrication with the effect of expelling (squeeze) the lubricant occurs when the active surfaces of the coupling approach at high speed in the normal direction (fig. ETB-T.4.1.3.1.2); it has a non-stationary character and is determined by the resistance that the lubricant opposes to its expulsion from the gap formed by the coupling surfaces; the permanence of the self-supporting film is ensured if the mobile surface has periodic movement (e.g. hydrodynamic dampers for flexural vibrations of shafts)

ETB-T.4.1.3.2 Hydrostatic lubrication modeling

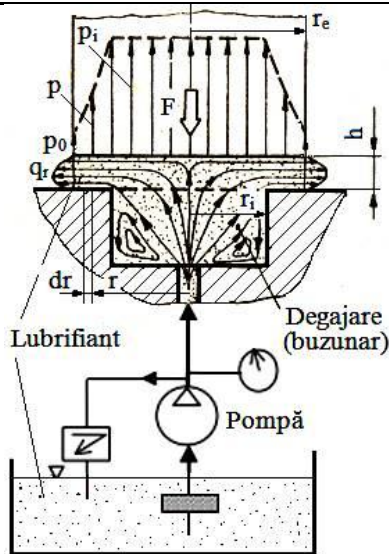


Fig. ETB-T.4.1.3.3 Hydrostatic lubrication model

The hydrostatic lubrication process

The lubricant film is ensured by force-feeding the friction area at the necessary pressure so that the resultant pressure field on the active surface of the coupling equals the external load (fig. ETB-T.4.1.3.3).

The calculation model

The lubricant flow rate in the radial direction is obtained by multiplying the unit flow rate, [Rădulescu, 1981], by the circle length $2\pi r$ (fig. ETB-T.4.3.3),

$$q_r = -2\pi r q_1 = -2\pi r \frac{h^3}{12\eta} \frac{dp}{dr} \quad (\text{ETB-T.4.1.3.2.1})$$

From this relationship, the pressure variation in the radial direction is obtained,

$$\int_{p_0}^p dp = -\frac{6\eta q_r}{\pi h^3} \int_{r_i}^r \frac{dr}{r}; \quad (\text{ETB-T.4.1.3.2.2}) \quad p = p_i - \frac{6\eta q_r}{\pi h^3} \ln \frac{r}{r_i}$$

$$\int_{p_0}^{p_i} dp = -\frac{6\eta q_r}{\pi h^3} \int_{r_i}^{r_e} \frac{dr}{r}; \quad (\text{ETB-T.4.1.3.2.3}) \quad p_0 = p_i - \frac{6\eta q_r}{\pi h^3} \ln \frac{r_e}{r_i}$$

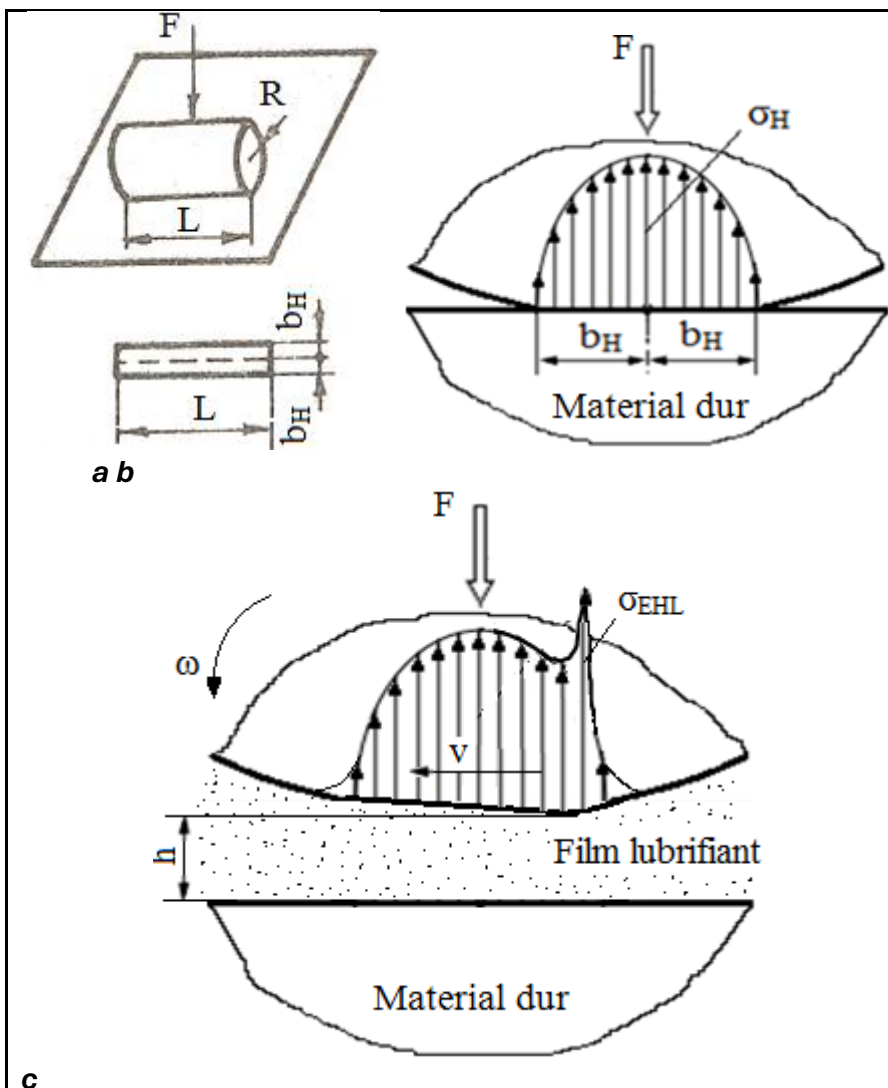
Thus, the expression for the lubricant flow rate becomes,

$$q_r = \frac{\pi h^3}{6\eta} \frac{p_i - p_0}{\ln \frac{r_e}{r_i}} \quad (\text{ETB-T.4.1.3.2.4})$$

The lift (force supported by the lubricant film) is calculated as the sum of the forces generated by the pressure p from the lubricant film existing on the annular active surface and the pressure p_i exerted on the annular circular surface of radius r_i :

$$F = \int_{r_i}^{r_e} p 2\pi r dr + \pi r_i^2 p_i = \int_{r_i}^{r_e} p_i 2\pi r dr + \int_{r_i}^{r_e} 2\pi r \frac{6\eta q_r}{\pi h^3} \ln \frac{r}{r_i} + \pi r_i^2 p_i = \pi r_e^2 p_0 +$$

ETB-T.4.1.3.3 Modeling of elastohydrodynamic lubrication



c

Fig. ETB-T.4.1.3.3.1 Model of elastohydrodynamic lubrication: **a** – macrogeometry; **b** – local elastic deformation without lubrication; **b** – local elastic deformation with lubrication

The process of elastohydrodynamic lubrication

Achieving hydrodynamic lubrication conditions involves transmitting forces through relatively large (conformal) contact surfaces, the pressures that occur on the contact surface being reduced and the lubricant film between the surfaces being continuous.

In practice, there are multiple situations (bearings, gears, cams, etc.) in which force transmission is done through friction couplings that do not conform to theoretical point or linear contacts that are heavily loaded (fig. ETB-T.4.1.3.3.1,a).

According to Hertz's theory, for the case in Fig. ETB-T.4.3.3.1,a) the stress distribution has the shape of an ellipsoid with semiaxes a , b , $p_{\max}$. In addition to the normal stresses, $\sigma_{H\max} = p_{\max}$ in the case of unlubricated Hertzian contact, there also appears the tangential stress whose maximum $\tau_{\max} = 0.3 \sigma_{H\max}$.

The presence of frictional forces modifies the distribution of contact stresses and leads to an increase in contact stresses.

The geometry, kinematics, and elasticity of the materials of non-conforming contacts (point-shaped or linear) ensure, in the presence of lubricant, conditions for achieving hydrodynamic lubrication, which at high loads as a result of superficial elastic deformations becomes elastohydrodynamic lubrication (EHL). The analysis of the elastohydrodynamic regime, for the general case, is a complex operation that takes into account the laws of hydrodynamic lubrication combined with the laws of elasticity theory, and analytical solution is very difficult, with only numerical solutions for specific concrete cases being achieved so far.

The calculation model of elastohydrodynamic lubrication

Theoretical and experimental studies have led to experimentally obtained models that establish the minimum thickness of the lubricant film,

$$h_{\min} = 0,035 (\eta v R_e)^{0,5} [\mu\text{m}] \quad (\text{ETB-T.4.3.3.1})$$

where, η is the dynamic viscosity in Poise, v – relative velocity in m/s and R_e – radius in mm.

For real cases, the influence of microirregularities (roughnesses) on the lubrication regime is taken into account by means of the parameter,

$$X_h = \frac{h_{\min}}{R_a} [\mu\text{m}], \quad (\text{ETB-T.4.1.3.3.2})$$

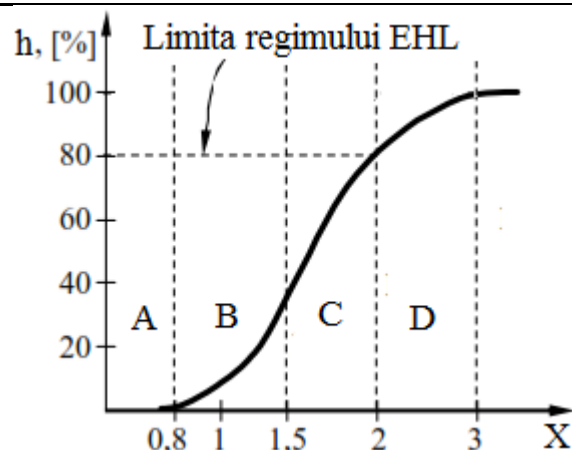


Fig. ETB-T.4.1.3.3.2 Friction regimes in elastohydrodynamic lubrication

where, $[\mu\text{m}]$ represents the equivalent roughness of the surfaces in contact with

$R_{a1,2} [\mu\text{m}]$ the roughnesses of the two surfaces in contact. $R_a = \sqrt{R_{a1}^2 + R_{a2}^2}$

Depending on the values of this parameter, the following friction regimes are highlighted (fig. ETB-T.4.1.3.3.2):

- limit or dry, $X_h < 1$ (zone A), is characteristic of couplings that transmit high loads, the damage to their surfaces being caused by abrasive wear or seizure;
- mixed, $X_h = 1.1 \dots 1.5$ (zone B), surface damage being characterized by pitting and exfoliation;
- partially elastohydrodynamic, $X_h = 1.5 \dots 3$ (zone C), shiny spots appear on the contacting surfaces, areas of interruption of the lubricant film.
- completely elastohydrodynamic, $X_h = 3$ (zone D), the lubricant film is equal to the contact surface, accidental wear is reduced

ETB-T.4.2 LUBRICANTS

ETB-T.4.2.1 FUNCTIONS

Main functions	Secondary functions
- ensuring the load-bearing film between the surfaces in relative movement capable of taking on external loads; - friction reduction; - reducing wear and tear;	- ensuring cooling; - protects against corrosion; - ensures the transfer of heat produced by friction to the outside; - attenuates shocks and vibrations

ETB-T.4.2.2 CLASSIFICATION

Aggregation status	Types	Examples	Description
solid	Plastics	Teflon, polyacetates	Independent or embedded in anti-friction layers
	Powdery materials	Graphite, mica, MoS ₂ , WS ₂	Embedded in lubricating pastes (mixed with oils or greases) or in the form of dry films (mixed with binders)
liquids	Oils	MINERAL	They are obtained from petroleum by refining. The additive is used to improve lubrication capabilities (reducing the decrease in viscosity with increasing temperature), anti-corrosion, anti-oxidation, anti-wear, extreme pressure, etc.

		synthetics	They are synthetic products created specifically for certain requirements. Synthetic oils perform better in terms of low-temperature pumpability, high-temperature stability, and surface protection.
	Unconventional	Water, melts, emulsions, acids	Molten materials: mercury, sodium, glass, etc.
Visco-plastic	Thick greases		With soaps based on Ca, Na, Li, Ba, Pb, Al
Gas	Air, steam, CO ₂ , He, CO ₂		

ETB-T.4.2.3 PROPERTIES AND TYPES OF OILS

ETB-T.4.2.3.1 Flow properties of oils

Name	Symbol, unit of measurement	Description
Dynamic viscosity	η [Ns / m ²] _{SI} or [Pa.s] _{SI} ; [P] _{CGS} (Poisie) or [cP] _{CGS} ; 1P = 1 Ns / cm ² ; 1cP = 10 ⁻³ Pa.s (dynamic viscosity of water at 20 °C)	Based on Newton's law (fig. ETB-T.2.4.6,b) <i>it</i> represents the proportionality factor between the tangential shear stresses of the lubricant layers and the velocity gradient in the normal flow direction; it is a measure of the internal friction in the lubricant and quantitatively defines the flow resistance.
Kinematic viscosity	ν [m ² / s] or [St] (Stokes); 1St = 1 cm ² / s; 1cSt = 10 ⁻⁶ m ² / s (kinematic viscosity of water at 20 °C)	It represents the ratio of dynamic viscosity to density (η / ρ)
Relative viscosity (Engler)	E [°] (Engler degrees)	The ratio between the flow time of 200 cm ³ of oil and the flow time of 200 cm ³ of distilled water at 20°C, through the calibrated orifice of the viscometer.
Density	ρ [kg/m ³] _{SI} or [kg/cm ³] _{CGS}	$\rho = (700...1200) \text{ kg/m}^3$
Specific heat	C [J/kg K] _{SI}	$c \approx 1670 \text{ J/kg K}$
Thermal conductivity	λ [W/ mK] _{SI}	$\lambda \approx 0.14 \text{ W/ mK}$

Note.

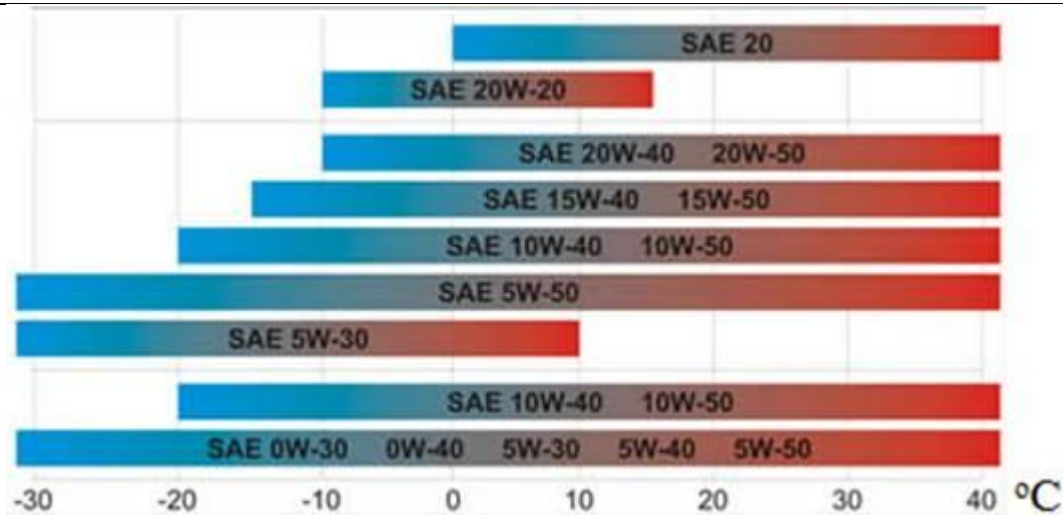
1. Viscosity characterizes the internal friction of the lubricant and quantitatively defines the resistance to flow; it determines the magnitude of the friction coefficient and, implicitly, the power losses, as well as the speed of its flow between metal surfaces.
2. The viscosity of oils decreases with increasing temperature (if the temperature increases from 20 °C to 70 °C the viscosity decreases by approx. 50 times); excessive decrease/increase in the viscosity of the oil leads to a decrease in its lubricating quality. The variation in the viscosity of oils with temperature is quantified by the viscosity index (40...200) which expresses the behavior of the lubricant with temperature compared to two standard oils, one with an index of 0 (very strong variation with temperature) and the other with an index of 100 (medium variation with temperature); the higher the viscosity index of an oil, the better it behaves at high temperatures without its viscosity decreasing too much and at low temperatures without it increasing too much.

ETB-T.4.2.3.2 Additional properties of oils

Property	Description
Unctuousness (lubricating capacity)	The adhesion property of the lubricant to solid surfaces; it is evaluated by determining the pressure that leads to the disruption of the lubricant film or by determining the antifriction capacity, measuring the friction coefficient
Flash point	The temperature at which a lubricant ignites spontaneously near an open flame (for mineral oils, 150... 210 °C).
Burning point	The temperature at which oil heated in open vessels, once ignited, burns completely. It is 20-30 °C higher than the flash point (210 °C)
Solidification (freezing) point	The temperature at which the oil no longer flows under its own weight; it is important for refrigeration machines, means of transport at low temperatures
Emulsibility	The property of a lubricant to mix with water and not separate from it; important for steam turbines
Acid content	Less than 0.7 ‰ to avoid chemically corroding the parts they lubricate.
Water content	Less than 0.5‰, water reduces oil absorption and adhesion and promotes corrosion.
Foaming	The property of oil to form and maintain foam on the surface as a result of the air or gas content in the oil mass; foaming is the cause of faults that can occur in transmissions

ETB-T.4.2.3.3 Types of automotive oils

Mineral oils		
For engines	M xx, (SAE xx)	Monograde oil; xx = 20,30,40 <u>viscosity grade (flow time through a given orifice) according to American SAE standards</u> (Society of Automotive Engineers, founded in 1911); the higher the grade, the more viscous the oil
	My yy W - xx (SAE yy W-xx)	Multigrade oil; yy = 0,5,10,15,20,25 <u>viscosity grade at low temperatures in winter</u> (oils with additives); xx = 20,30,40 viscosity grade according to American SAE standards; e.g. M20 W-40 is a multigrade additive oil that has a viscosity factor of 20 in winter, equivalent to an oil with a factor of 40 in summer; the lower the first number (yy), the better the oil behaves at low temperatures without increasing its viscosity much, and the higher the second number (xx), the better the oil is at high temperatures without decreasing its viscosity much.
For vehicle transmissions	T xx	Non-additive oils xx = viscosity index; e.g. T140 ;
	T xx EP y	Oils with extreme pressure (EP) additives; xx viscosity index; y – degree of additivity
		Special oils (automatic transmission oils,
For industrial transmissions	TIN xx EP	Oils with extreme pressure (EP) additives; xx kinematic viscosity in cSt at 50 °C; TIN 125 EP, TIN 42 EP, TIN 130 EP



Grad SAE	Timpul de scurgere (s)	Vascozitatea in CentiStokes la 100° C
10	< 14	4.00 (2 - 5)
20	15 la 24	7.45 (6 - 8)
30	25 la 34	10.90 (9 - 12)
40	35 la 44	14.40 (13 - 16)
50	peste 45	19.10 (17 - 21)

[www.uleiauto.ro]

Types of additives (molybdenum disulfide, boron nitride, etc.):

- antioxidants (resistance to oxidation);
- to increase the viscosity index (IV, makes it thicker in summer and thinner in winter) and with an anti-wear role when starting at low temperatures;
- antifriction;
- high pressure antiwear;
- anti-freezes (lower the freezing point);
- increase lubricity and EP (extreme pressure); increase the resistance of the oil film at high pressures, but if the normal concentration is exceeded, they can cause high wear;
- anti-rust;
- anti-foaming agents (break air bubbles in the oil);
- detergents (keep insoluble substances in solution and prevent their deposition in the form of varnishes);
- multifunctional: (synergistic action), the most common package: antioxidant, anti-foaming agent, detergent, dispersant, etc.

Additional specifications, American Petroleum Institute (API): Service categories (SA...SM), for gasoline engines; Commercial categories (CD, CH, CG, CH), for Diesel engines;

SA - for very old engines, without performance requirements
SB - for very old engines; used only when specifically requested by the manufacturer.
SC - for engines manufactured up to 1967.
SD - for engines manufactured up to 1971.
SE - for engines manufactured up to 1979.
SF - for gasoline engines manufactured after 1980, up to 1989; compared to SE, it provides better cleaning of the active parts of the engine and reduction of deposits, higher oxidation resistance and wear protection.
SG - for engines manufactured after 1980, up to 1993 inclusive; higher additivity compared to **SF**.
SH - For engines manufactured up to 1996 inclusive; compared to SG, it provides more stringent oil properties.
SJ - for all engines currently used; it is more effective than SH in terms of oxidation stability at high temperatures, catalyst compatibility, reduced foaming.
SL - introduced since July 2001
SM - introduced since July 2004; the highest performance class for gasoline engines; for all currently produced high-performance engines, with improved protection against deposits and operation at low temperatures; some oils have the " **Energy Saving** " specification

CA - CC - for diesel engines up to 1965.
CD - valid since 1965, lubricant for high-duty, supercharged diesel engines; Contains more efficient DD additives.
CE - for low and high-speed, high-duty, supercharged diesel engines; moderate additive, ensures low oil consumption.
CF - was introduced in 1994; For diesel engines with or without indirect injection, including those using fuel with sulfur above 0.5%; replaces the CD specification.
CF-2 - was introduced in 1994; for high-duty, two-stroke engines.
CF-4 - was introduced in 1990; for high-speed, four-stroke, naturally aspirated or supercharged diesel engines; can be used instead of API CD or CE.
CG-4 - was introduced in 1995; for high-duty, four-stroke diesel engines using fuel with less than 0.5% sulfur; CG-4 oils are recommended for engines that meet the 1994 emission standards; can be used instead of API CD, CE and CF-4.
CH-4 - was introduced on December 1, 1998; for high-speed four-stroke diesel engines that meet the 1998 emission standards; CH-4 oils are specially additived to be used with fuels with less than 0.5% sulfur; provides reduced wear on valve guides, more efficient soot handling and higher performance; can be used instead of API **CD, CE, CF-4** and **CG - 4**.
CI-4 - was introduced in 2002; for high-speed four-stroke diesel engines that will meet the 2002 emission standards in 2004; the **CI-4** standard is intended for engines equipped with an **exhaust gas recirculation** system and for Diesel engines using diesel fuel with a concentration of up to 0.5% sulfur; these oils can be used instead of oils that meet the **CD, CE, CF-4, CG-4, CH-4** standards; some oils may receive the **CI-4 Plus** specification.
CJ-4 - was introduced in September 2006; for 4-stroke Diesel engines, equipped with emission control systems with particulate filters and other modern systems.

Additional specifications, European Automobile Manufacturers Association (ACEA):

- A1** - Multigrade oil for gasoline engines, stable to shear at medium temperatures, provides fuel economy.
- A2** - Multigrade oil of normal viscosity for gasoline engines.
- A3** - Multigrade oil stable to shear, low volatility; very good cleaning properties, excellent stability.
- B1** - Oil for diesel engines, stable to shear at medium temperatures and low loads, provides fuel economy.
- B2** - Oil for light-duty diesel engines.
- B3** - Oil for light-duty diesel engines; exceeds B1, B2 requirements regarding wear conditions, deposits and soot formation.
- B4** - Advantageous, primarily, in direct injection diesel engines.
- E1** - Oil for normal drain interval, slightly turbocharged engines.
- E2** - Oil for normal drain interval, moderately turbocharged engines.
- E3** - Superior to E2-96 norm, SHPD oil, provides impeccable cleanliness and anti-wear protection; meets EURO 1 and EURO 2.
- E4** - More stringent standard than E3, ensures the longest oil change interval.
- E5** - Ensures better piston cleaning, high wear protection, better oxidation stability

ETB-T.4.2.4 PROPERTIES AND TYPES OF GREASE

Consistency greases are mixtures of mineral oils with metallic soaps of Na, Ca, Al, Ba, Li, Pb. During the operation of machines, due to the heating resulting from friction, they lose their consistency and are drawn between surfaces and provide lubrication.

ETB-T.4.2.4.1 Main properties of greases

Name	Description
Dropping point	The temperature at which the grease begins to flow under its own weight
Melting temperature	It should not be too high, to ensure lubrication through heating, but not too low, so that the grease does not melt on its own when the ambient temperature increases.
Consistency	Indicates the state of grease penetrability by a metal cone with standard geometry

ETB-T.4.2.4.2 Types of greases

Lubricants	Examples	Description
General purpose	U 75 Ca 2	U – general purpose grease 75 °C – dropping point, the soap used is Calcium (Ca), the consistency of the grease is grade 2
For bearings	RUL 100 Ca 3, RUL 145 Na 3	Greases with a dropping point of 100 and 145, respectively, with Calcium (Ca) or Sodium (Na) soap and consistency grade 3
Multifunctional	UM 160 Li-Ca-Pb 1, UM 170 Li-Ca-Pb 2	Greases with dropping point 160 and 170, respectively, with Lithium (L), Calcium (Ca) and Lead (Pb) soap and consistency grade 1 and 2 respectively

ETB-T.4.2.5 LUBRICATION WITH OTHER LUBRICANTS

ETB-T.4.2.5.1 Particularities of gas (air) lubrication

- at very high speeds, friction is significantly reduced, their viscosity being 100... 1000 times lower than that of liquids;
- damping because they are compressible, unlike liquids, which are practically incompressible;
- reduced friction also leads to a decrease in thermal levels, with gas-lubricated bearings being able to operate at speeds of up to 600,000 rpm, in the case of dental drills and precision equipment;
- low viscosity leads to reduced load-bearing capacity, especially in the case of gas-dynamic bearings;
- physico-chemically stable in various environments, gas lubrication can be used in nuclear technology, aerospace construction and other fields;
- gas lubrication (frequently, air) can be done gas-dynamically or gas-statically through specific supply and sealing systems.

ETB-T.4.2.5.2 Particularities of water lubrication

Water lubrication is used in the case of pumps that operate in a humid environment or are immersed in water and the use of conventional lubricants (oils, greases) is difficult and sometimes impossible.

ETB-T.4.2.6 LUBRICANT CHOICE

Recommendations:

- the main aim is to ensure viscosity conditions, stability over time and protection against corrosion;
- the oil viscosity must be chosen depending on the type and degree of wear of the coupling and the ambient temperature;
- the use of multigrade and synthetic oils makes it possible to use the same type of oil, regardless of the season;
- peripheral speed in the kinematic coupling: low speeds – thick greases, high speeds – oils.

ETB-T.4.3 LUBRICATION METHODS AND DEVICES

ETB-T.4.3.1 OIL LUBRICATION METHODS AND DEVICES

The main function of a lubrication device: supplying the necessary oil flow rate (determined by calculation) to the friction areas.

Lubrication type	Description
By drip	It is carried out at atmospheric pressure with devices that ensure conditions for regulating (dosing) the amount of oil through calibrated, adjustable orifices or retaining materials (ETB-T.4.3.1.1,a);

By bubbling (throwing) from the oil bath	It involves the partial immersion of rotating parts (ETB-T.4.3.1.1,b,c) in the oil bath from where it is thrown towards the friction coupling areas and onto the housing walls from where, by trickling, it can reach the surfaces that require lubrication.
By sprinkling	By entraining the oil into the friction zone by means of a ring (ETB-T.4.3.1.1,d) or disc (ETB-T.4.3.1.1,e)
By circulation	railway RUL-S.2
By injection (under pressure)	railway RUL-S.2
By spraying (oil mist)	railway RUL-S.2

Lubricant supply can be done manually, semi-automatically, automatically [Roloff, 2008]

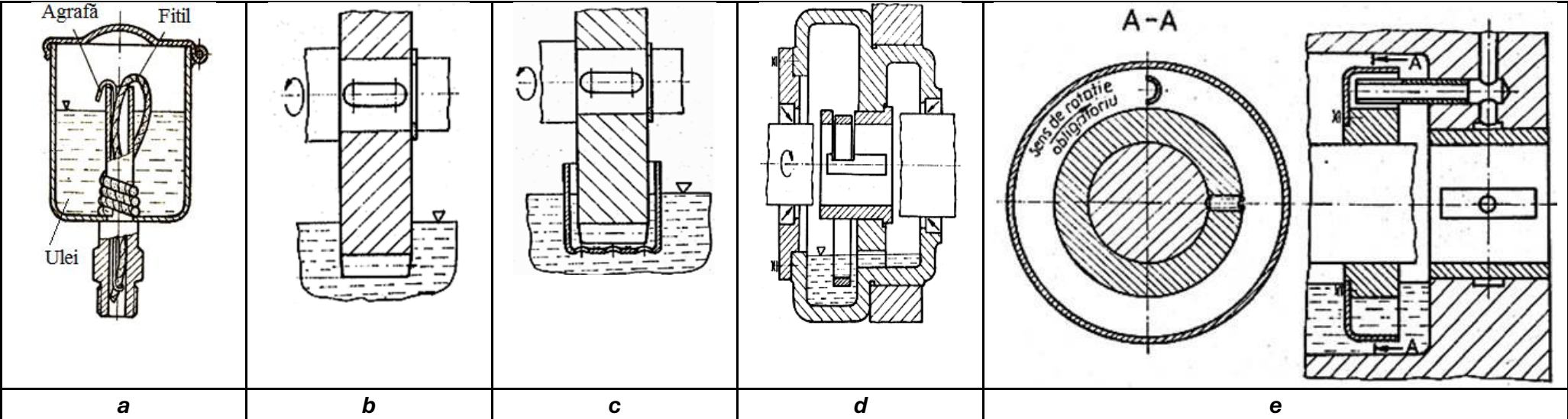
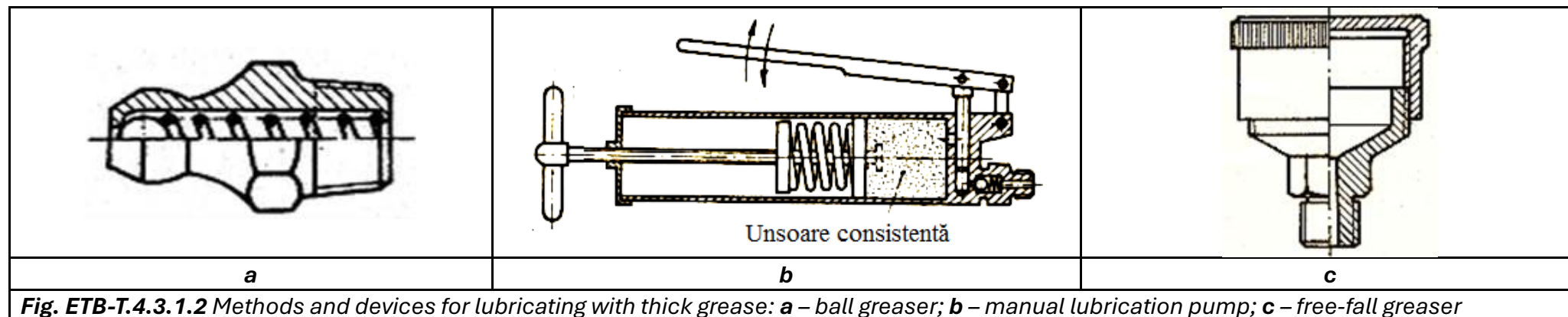


Fig. ETB-T.4.3.1.1 Methods and devices for oil lubrication: **a** – by dripping; **b** – bubbling at low peripheral speeds, $v < 15...25$ m/ s; **c** – bubbling at high peripheral speeds, $v < 30...100$ m/s; **d** – by ring spraying; **e** – by disc spraying

Note: In annex RUL-S.2, bearing lubrication solutions are presented.

ETB-T.4.3.2 GREASE LUBRICATION METHODS AND DEVICES

The main function of a lubricator: supplying consistent grease to friction areas.



ETB-T.4.4 SEALING METHODS AND DEVICES

ETB-T.4.4.1 FUNCTIONS

Functional role – separating two environments under different pressures, avoids lubricant losses from the sealed space and/or the penetration of impurities from outside into this space.

Requirements: increased durability and reliability, simple assembly and disassembly, easy maintenance, mechanical and chemical resistance, good thermal conductivity, minimal friction losses, compatibility with the sealed environment.

ETB-T.4.4.2 MATERIALS FOR SEALING ELEMENTS

Leather – has a good sealing capacity even on rough surfaces, also having the ability to absorb and retain lubricant; has good wear resistance and reduced friction coefficients in contact with metallic materials; maximum recommended speeds (peripheral and translational), 4 m/s; the working temperature up to which it functions normally is 100 °C.

Felt – has high lubricant retention capacities and low friction coefficients; low wear resistance.

Paper and cardboard – are used for fixed seals, at low pressures and temperatures up to 100 °C; before installation, they are impregnated with oil and resin solutions.

Cork – has a high coefficient of friction and low thermal conductivity; is impermeable to liquids at low pressures; is fragile and cannot be reused; e.g. sealing oil sump covers.

Elastomers – representative of this group is synthetic rubber; it withstands large deformations without generating appreciable stresses and easily adapts to the shapes of metal surfaces; there are various recipes with resistance to the type of sealing fluid (oil, steam, etc.).

Plastomers – thermoplastic synthetic materials (they become plastic when heated and solidify when cold); low friction coefficients.

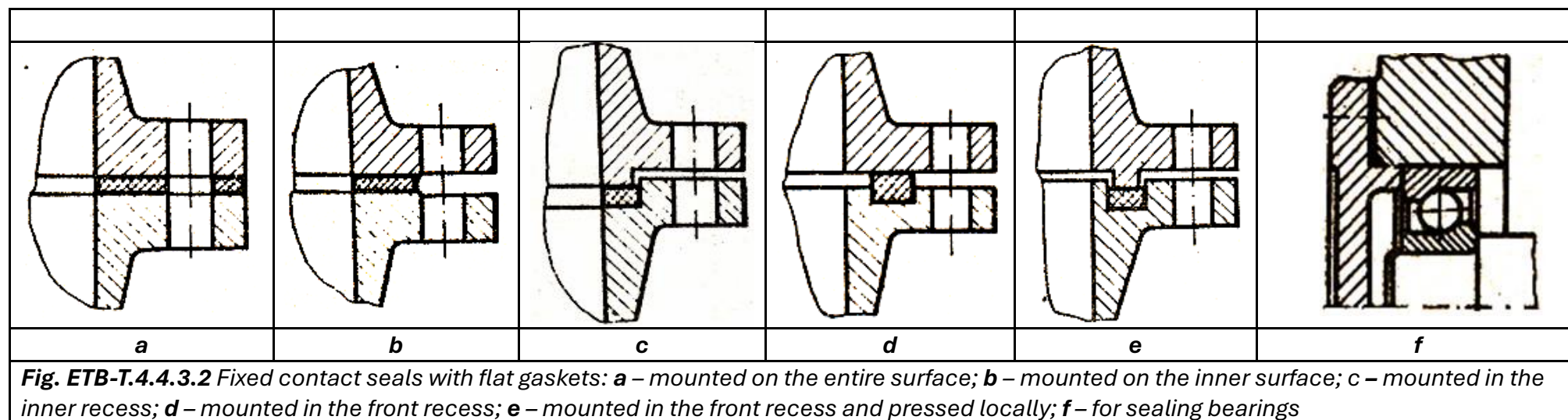
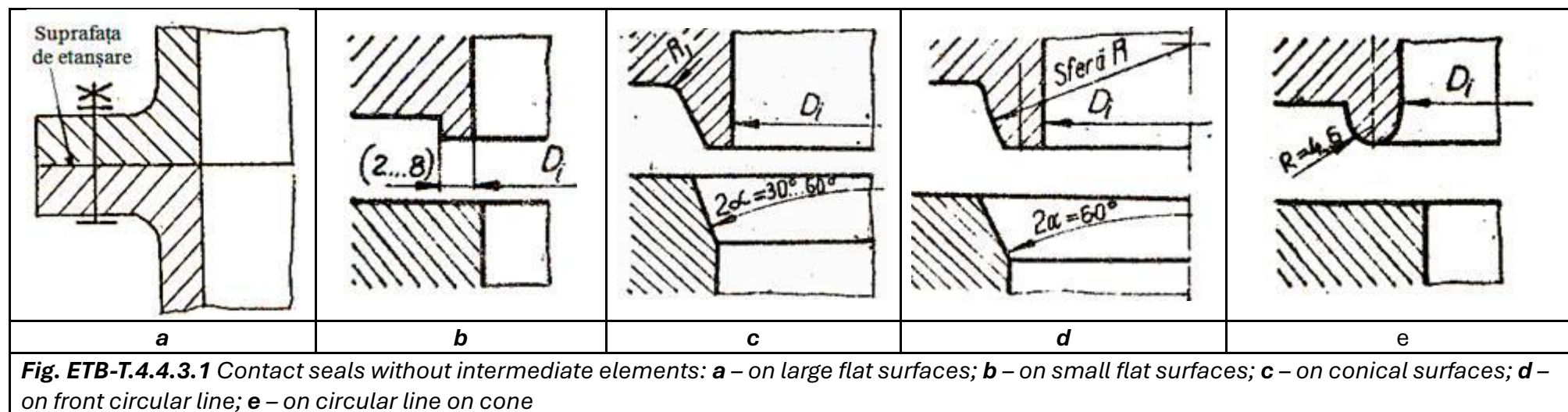
Metallic materials – soft lead (for acidic environments), soft aluminum (at low pressures), soft copper (at high temperatures), bronze and brass (high chemical resistance), gray cast iron (possibly with the addition of Si, for sealing the pistons of thermal engines); they are used in the form of flat or profiled gaskets, solid rings, etc.

ETB-T.4.4.3 CLASSIFICATION AND TYPES OF SEALS

Types of seals			Description
With contact	Fixed (no relative movements)	No intermediate elements	It is achieved by initially processing the surfaces that come into direct contact very finely, after which, by tightening, the seal is ensured (ETB-T.4.4.3.1); easy assembly and disassembly; it is achieved with pre-tightening so that in the event of operating pressure, a minimum tightening remains to ensure the seal.
		With intermediate element	The intermediate element, <u>the flat gasket</u> (ETB-T.4.4.3.2) or <u>profiled gasket</u> (ETB-T.4.4.3.3,f), is tightened between the sealed elements, so that when internal pressure occurs, the gasket remains sufficiently pressed for sealing; at high pressures, in order to avoid expulsion, the gasket is mounted in specially made housings (ETB-T.4.4.3.2,c,d,e; ETB-T.4.4.3.3,b); in the case of bearings, the sealing between the cover and the housing is made with flat gaskets (usually made of pressed cardboard) over the entire surface and the minimum thickness to compensate for processing and shape irregularities (ETB-T.4.4.3.2,f).
	Mobile (with relative movements)	Rotation	<u>With felt rings</u> ; they are cheap and simple and are used in the case of lubrication with consistent grease or oil at peripheral speeds ($v < 5$ m/s at the shaft level) and low temperatures ($T < 90^{\circ}\text{C}$); it is recommended that the shaft hardness in front of the felt ring be at least 45 HRC or that it be hard chrome-plated and polished; the rings with a rectangular section are mounted with a tightening in a channel with a trapezoidal section (ETB-T.4.4.3.4); friction losses are high ($\mu > 0.15$); sometimes, to increase the operating life and maintain contact between the felt ring and the shaft, the constructive solution in fig. can be adopted, which allows periodic tightening of the felt ring.
			<u>With rotary sleeve seals</u> (ETB-T.4.4.3.5); they are used for peripheral speeds of up to 10 m/s, at temperatures from -30°C to 200°C and at pressure differences between the sealed spaces of less than 0.05 Mpa; they generate high friction forces; they are used in the case of lubrication with oil but also with consistent grease; the surface of the shaft under the seal must be finely machined and the hardness must be at least 45 HRC; they can be used against the penetration of impurities from the outside and against lubricant leakage
		Translational	<u>With O-ring type seals</u> ; they are mounted in rectangular- section recesses made in bores (ETB-T.4.4.3.6) or shafts; to avoid the possibility of extrusion (crushing) of the ring in the gap between the parts in relative motion as well as for operation at higher pressures, lateral support rings are used (ETB-T.4.4.3.6,c); they are recommended for small diameters, short strokes and medium pressures, being used for hydraulic or pneumatic drives; they are simple in construction and are easy to assemble and disassemble

			<u>With sleeve-type seals</u> ; sealing by means of a sealing lip that must be directed towards the sealing medium under pressure; according to the profile, the sleeves can be <u>with a V-profile</u> (sealing of media with high pressures, low travel speeds, mounted axially pressed (ETB-T.4.4.3.9)), <u>with a U-profile</u> (mounted without axial pressing, (ETB-T.4.4.3.10)), <u>with an L-profile</u> (for sealing at the shaft level (ETB-T.4.4.5.7,b,c) or bores (ETB-T.4.4.5.7,a)), with a J-profile (for sealing at the shaft level (ETB-T.4.4.3.11,a) or bores (ETB-T.4.4.3.11,b))
			<u>With gland packings</u> (ETB-T.4.4.3.8); they have the possibility of adjusting the sealing after their wear occurs; the gaskets can be made of felt, hemp cord, rubber, etc.
			<u>With metal segments</u> (ETB-T.4.4.3.12)
No contact	With slot (ETB-T.4.4.5.13,a)		They are used in the case of lubrication with thick grease; optimal clearance 0.1...0.3 mm; sealing effectiveness increases with increasing gap length.
	With slot and channels	roundness	Grease expulsion and the penetration of impurities are prevented thanks to circular channels filled with consistent grease (ETB-T.4.4.3.13,b)
		Screw	They are used in the case of oil lubrication and when the shaft rotates in one direction to drive the oil inward; the helical channels can be made inside the cover (ETB-T.4.4.3.13,c) or on the shaft (ETB-T.4.4.3.13,d)
	With labyrinths	Thrust	The labyrinth is formed by a series of constrictions followed by spaces that widen suddenly; sealing is achieved as a result of the fact that the fluid velocity in the constriction area decreases through the vortices that form in the widened spaces (ETB-T.4.4.3.14,a); in axial labyrinths the slots are arranged parallel to the axis of rotation (ETB-T.4.4.3.14,b); in radial labyrinths the slots are arranged perpendicular to the axis of rotation (ETB-T.4.4.3.14,c) and can be for sectioned or unsectioned housings (caps) (ETB-T.4.4.3.14,d)
		spline	

Note.
1. Contact seals are used to seal elements without relative movement between them and have a separation or protection role.
2. Non-contact seals have the advantage of working practically without friction and without wear; they are used at high speeds and/or temperatures when more effective contact seals cannot be used.



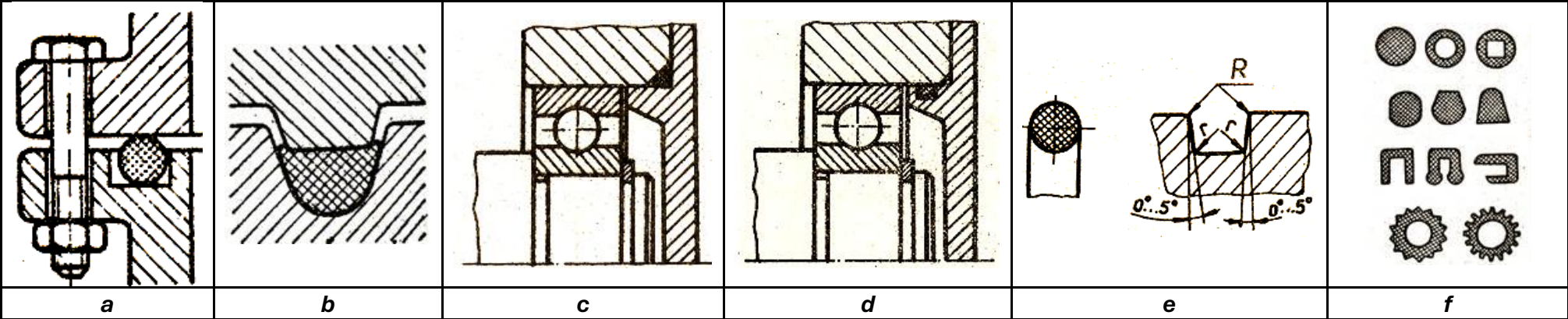


Fig. ETB-T.4.4.3.3 Fixed contact seals with profiled gaskets

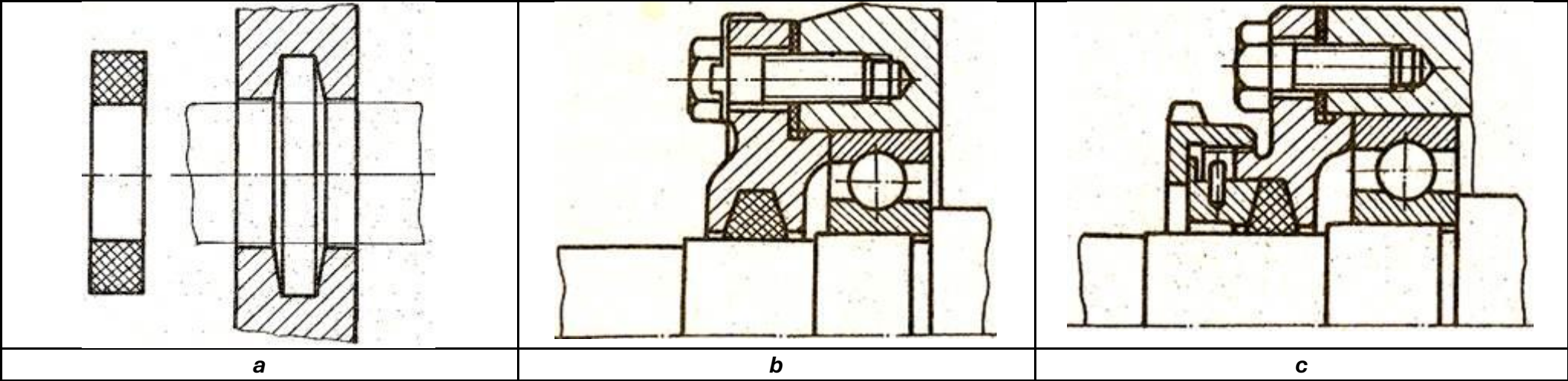


Fig. ETB-T.4.4.3.4 Rotary contact seals with felt rings

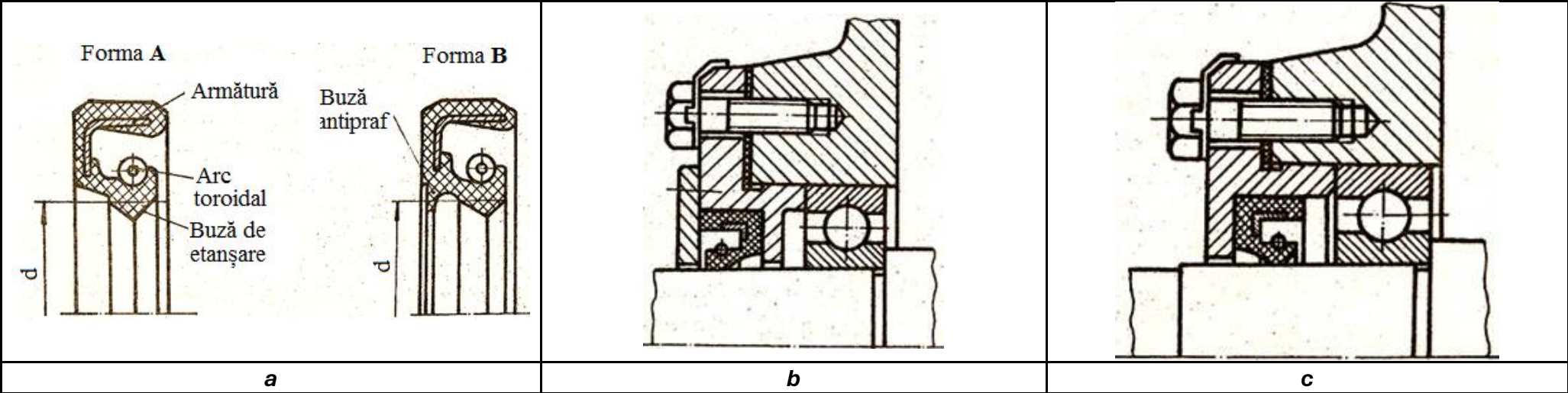


Fig. ETB-T.4.4.3.5 Contact seals with rotary sleeve seals

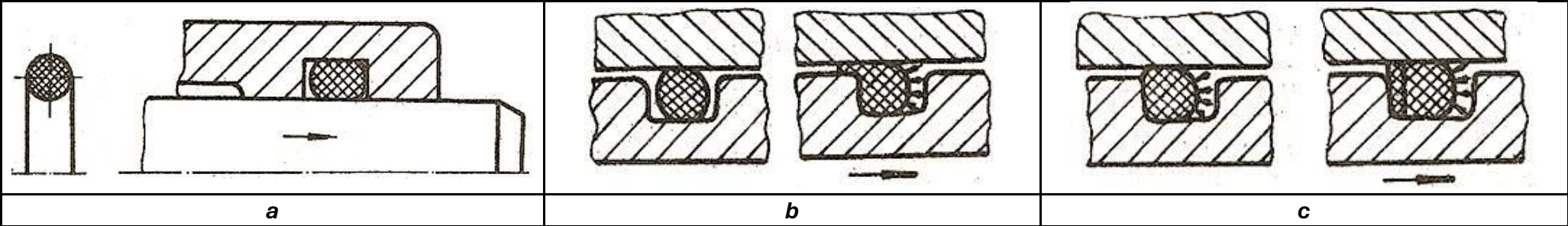


Fig. ETB-T.4.4.3.6 Translational contact seals with O -rings

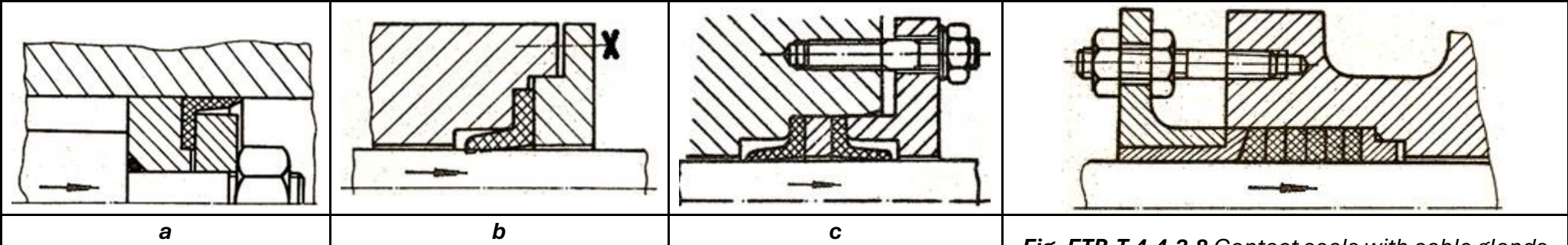


Fig. ETB-T.4.4.3.7 Translational contact seals with L-profile lip seals

Fig. ETB-T.4.4.3.8 Contact seals with cable glands

