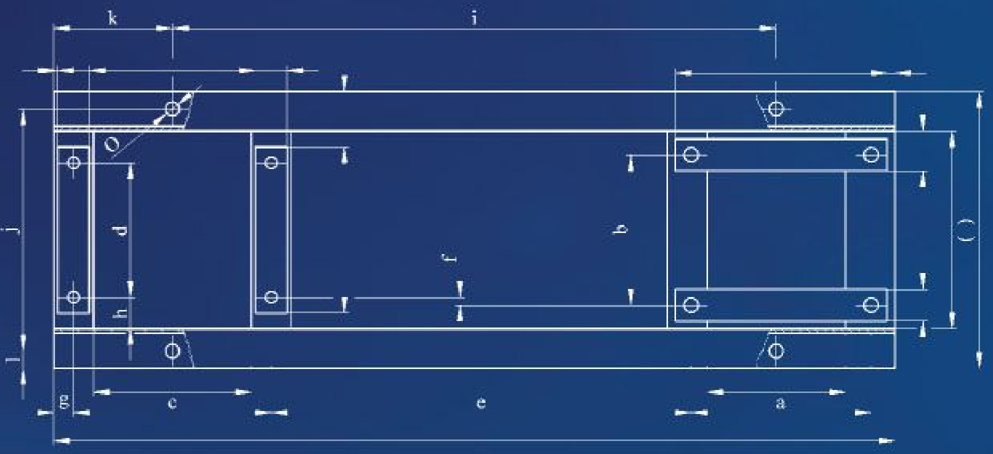
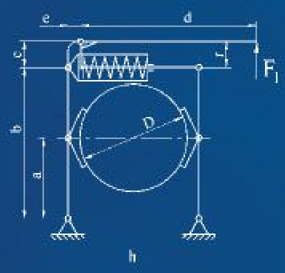
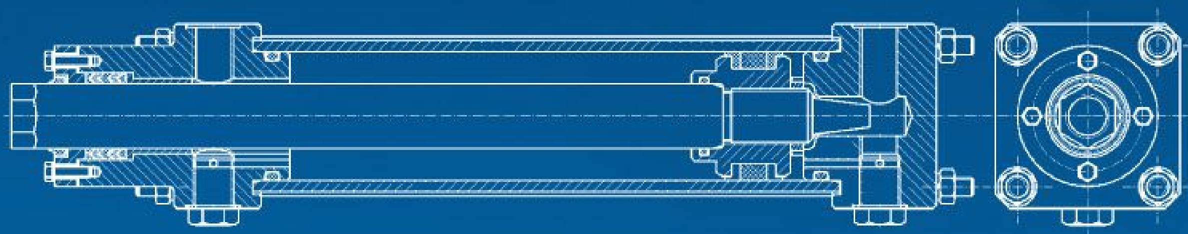
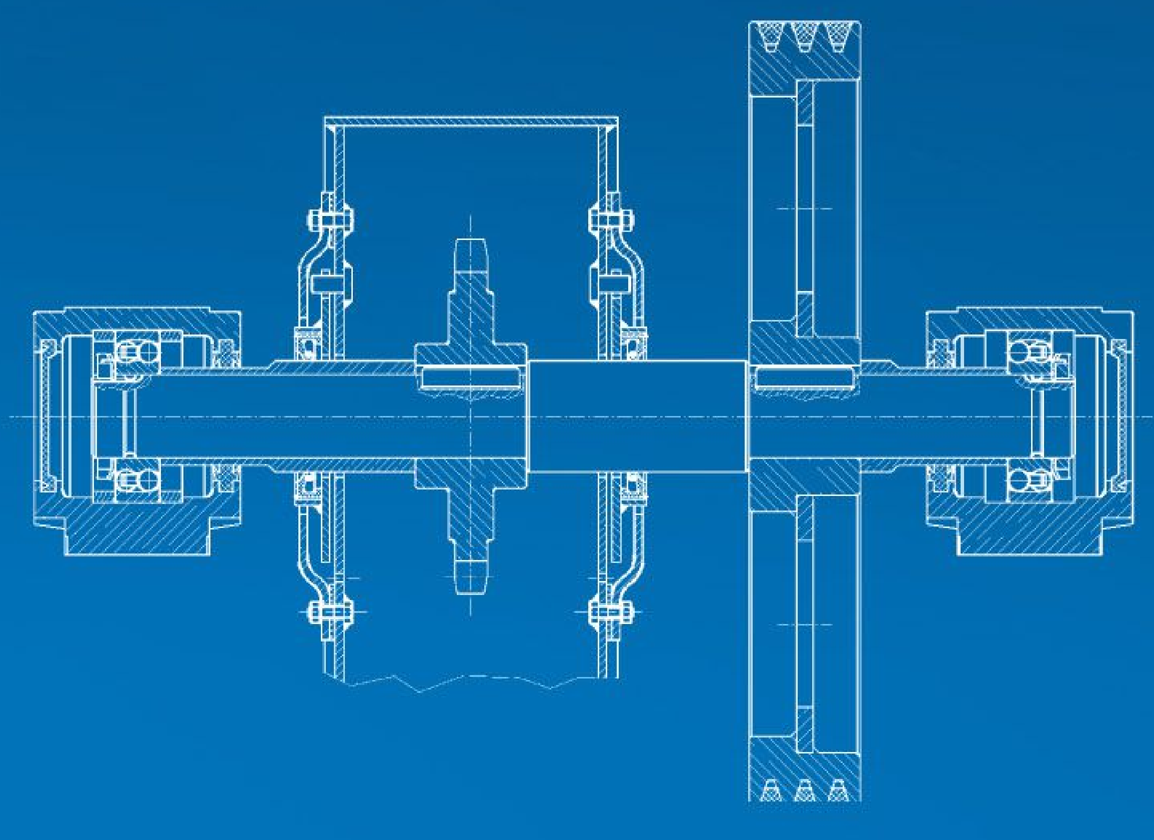




Zsolt Tiba

# Basic Constructions of Machine Design





UNIVERSITY OF DEBRECEN
FACULTY OF ENGINEERING

ZSOLT TIBA

# Basic Constructions of Machine Design



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Author:
ZSOLT TIBA

Professional Reader:
Imre Kocsis
University of Debrecen
Faculty of Engineering



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CONTENTS

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Figures:

Zoltán Budai M.Sc. Mechanical Engineering

Revised by:

Umar Ayub B.Sc. Mechanical Engineering

Dedicated to late Dr. József Tóth Laboncz who was a significant influence in the mechanical engineering program at the University of Debrecen and elaborated the design tasks in the machine elements course.

# Objective

Up-to-date machineries carrying out complicated technological tasks with high accuracy are controlled and supervised by computer program.

The machinery accordingly consists of a mechanism performing a form of motion, sensors, transducers, actuators and data acquisition and control system.

Comparing the mechanism of the up-to-date machinery with conventional one it is clear that both of them are built with the same standardized and well known machine elements and based on the same stressing and construction principles.

A good example for this is the internal combustion engine and transmission of a modern motor vehicle which contains the same crank mechanism, gear drive, joints, bearing supports and sealing compared to the one assembled 20-30 years ago.

The object of this book is to introduce basic constructions and dimensioning considerations with which the construction of any machine may be designed and a uniform engineering view may be developed. The book introduces the stressing, sizing and construction methods proceeding from the four design tasks involved in the mechanical engineering BSc training at the UD. The four tasks are: design of welded machinery base, design of hydraulic cylinder, design of external shoe thruster released drum brake and design of counter drive. The objective of the design tasks is to size and design basic constructions starting with free hand sketches from which the assembly drawing is constructed with rulers and any other drawing instruments and eventually elaborating the shop drawings of the parts for manufacturing. Neither the referred course nor this book contains the application of computer drawing programs. However the acquired knowledge and experiences prepare students for applying e.g. AutoCAD program as a constructor.

## Abbreviation of terms and marks

| | | |
|---|---|---|
| a | [mm] | centre distance |
| A | [$m^2$] | area |
| c | [$mmN^{-1}$] | spring compliance |
| $C_f$ | | surface finish factor |
| $C_r$ | | reliability factor |
| $C_z$ | | size factor |
| $C_t$ | | temperature factor |
| C | [N] | basic dynamic load rating |
| d,D | [mm] | diameter |
| e | [mm] | eccentricity |
| E | [$Nmm^{-2}$] | modulus of elasticity |
| E | [mm] | belt deflection |
| f | [mm] | displacement |
| f | [N] | load used to set belt tension |
| $F_g$ | [N] | excitation force |
| g | [$ms^{-1}$] | value of gravity |
| G | [$Nmm^{-2}$] | modulus of rigidity |
| $H_{(t)}$ | [N] load limit | |
| i | | ratio |
| I | [$cm^4$] | moment of inertia |
| $I_p$ | [$cm^4$] | polar moment of inertia |
| J | [$kgm^2$] | mass moment of inertia |
| $k_ü$ | | operation coefficient |
| $k_i$ | | starting coefficient |
| $k_T$ | | overload coefficient |
| K | [$cm^3$] | section modulus |
| $K_P$ | [$cm^3$] | section modulus for torsion |
| $K_f$ | | fatigue stress concentration factor |
| l | [mm] | length |
| L | [mm] | drive span length |
| L | | service life |
| m | [kg] mass | |
| M | [Nm] | moment |
| N | | load circle |
| n | [$s^{-1}$] | speed of revolution |
| n | | factor of safety |
| p | [$Nmm^{-2}$] | bearing stress |
| P | [kW] | power |
| P | [N] | equivalent dynamic load rating |
| $R_{eH}$ | [$Nmm^{-2}$] | yield stress |

| | | |
|---|---|---|
| $R_m$ | [$Nmm^{-2}$] | ultimate tensile strength |
| $R_{Dv}$ | [$Nmm^{-2}$] | completely reversed endurance limit |
| $R'_{Dv}$ | [$Nmm^{-2}$]] | completely reversed endurance limit |
| s | [$Nmm^{-1}$] | spring constant |
| $S_a$ | [N] | static shaft load |
| t | [s] | time |
| T | [N] | static tension |
| $T_a$ | [N] | load amplitude |
| $T_n$ | [N] | nominal load |
| $T_{(t)}$ | [N] | load |
| v | [$ms^{-1}$] | velocity |
| V | [$m^3$] | volume |
| $\alpha$ | [$rads^{-1}$] | angular frequency of vibration |
| $\gamma$ | [$rad(Nm)^{-1}$] | torsion spring compliance |
| $\mu$ | | friction coefficient |
| $\rho$ | [$kgm^{-3}$] | density |
| $\sigma_{meg}$ | [$Nmm^{-2}$] | allowed normal stress |
| $\tau_{meg}$ | [$Nmm^{-2}$] | allowed shear stress |
| $\varphi$ | [$\deg ree$] | angular displacement |
| $\omega$ | [$rads^{-1}$] | angular frequency |
| $\eta$ | | efficiency |

# Figures

## Tables

# Stressing and construction considerations

The book introduces the stressing, sizing and construction methods proceeding from the four design tasks involved in the mechanical engineering BSc training at the UD. The four tasks are:

A. Welded machinery base
B. Hydraulic cylinder,
C. External shoe thruster released drum brake
D. Counter drive.

The objective of the design tasks is to size and design basic constructions starting with free hand sketches from which the assembly drawing is constructed with rulers and any other drawing instruments and eventually elaborating the shop drawings of the parts for manufacturing. The design and stressing considerations of the four tasks start with introducing the design tasks to be elaborated and analysing the arising problems to be resolved. Unfortunately there is no way to present the step by step stressing and construction processes since it would contains a lot of sketches, drawings (assembling and shop drawing) however we present the methods how to figure out the construction of an operating device, machine. There are some design fields eg. high-pressure vessel and lifting device, where the calculation and construction methods are standardized and design norms ensure the proper design.
However in general cases there are no norms to be followed that’s why we have to acquire an engineering view to consider the design task as a complex one. It starts with determining the load necessary for stressing procedure [1], elaborating the basic constructions of the machine. In this book we assume the proficiency required for designing acquired from several subjects: statics, stress analysis, strength of materials, dynamics, machine elements etc. and this is in reference merely to the knowledge without derivations.
Now we are presenting this method through four design tasks to study how to build up logically the design process.

# CHAPTER A. Machinery base

Design a welded machinery base carrying a machine unit that is fixed to the basement. The machines of the machine unit can be selected according to the given task number [2].

Begin the design with making three free hand sketches representing different constructions. They may be constructed from eg. hot and cold rolled profiles and plates or the combination of them. Aim of the construction should be simple in design and production taking the requirements into account. Do not perform any stressing and vibration analysis.

Elaborate the assembly drawing of one of the three constructions selected by the instructor on a paper sheet. Apply appropriate scale for presenting its details and dimensioning.

The assembly drawing shall contain views and sectional views necessary to give overall, connecting and tolerated dimensions, to show the welding joints and its dimensions. Draw the machine unit (motor, coupling, gear, pump or fan) with thin line on the main view of the machinery base.

Give prescriptions regarding the welding joints and corrosion protection. Complete the title block with the parts list and give the necessary data regarding the material, dimension and standard number of them.

Although all the drawings are drawn in pensile, pay attention to applying the proper line thicknesses.

## Design, stressing and construction considerations

When designing a steel construction like a welded machinery base, the following details have to be clarified:

- task and operation requirements,
- construction in terms of producibleness, machinability, cost-effectiveness,
- load of the machinery base and stressing consideration.

# 1. Machinery base installations

A machinery base is used for fixing the machine unit on it and fixed to the basement. It may be produced by casting or constructed from eg.: hot or cold rolled steel profiles or plates or the combination of them. The construction should be simple in design and it should need only a few machining.

Fig. 1.1 shows a machinery base frame carrying a machine unit comprising of an electric motor, and a compressor. They are coupled with a flexible coupling. The machine unit constitutes an equipment group with the base frame fixed to the basement

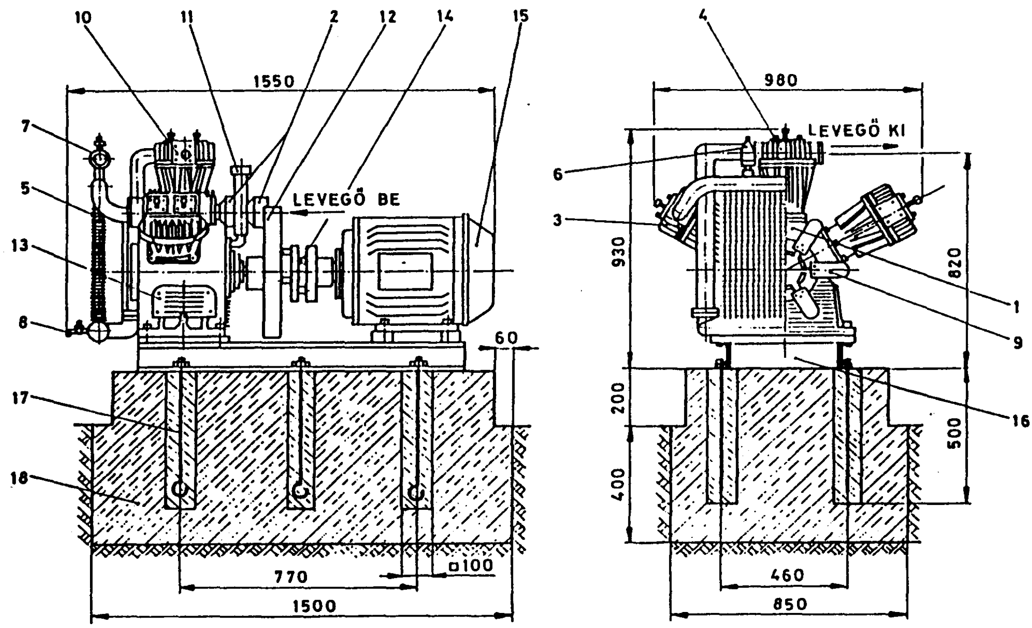



Figure 1.1 Compressor installation
Source: [2]

Fig. 1.2 shows a welded construction frame carrying a machine unit with centrifugal pump.

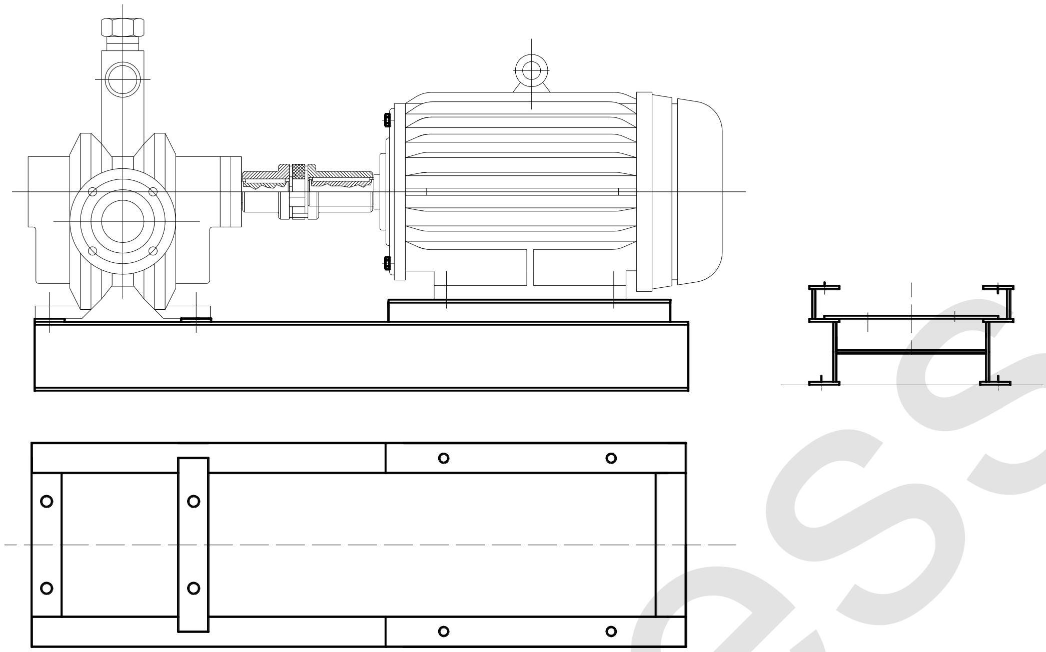


Figure 1.2 Centrifugal pump installation

The shaft height of the fan differs essentially from that of the motor in Fig. 1.3. Accordingly the motor is placed on mount of the frame.

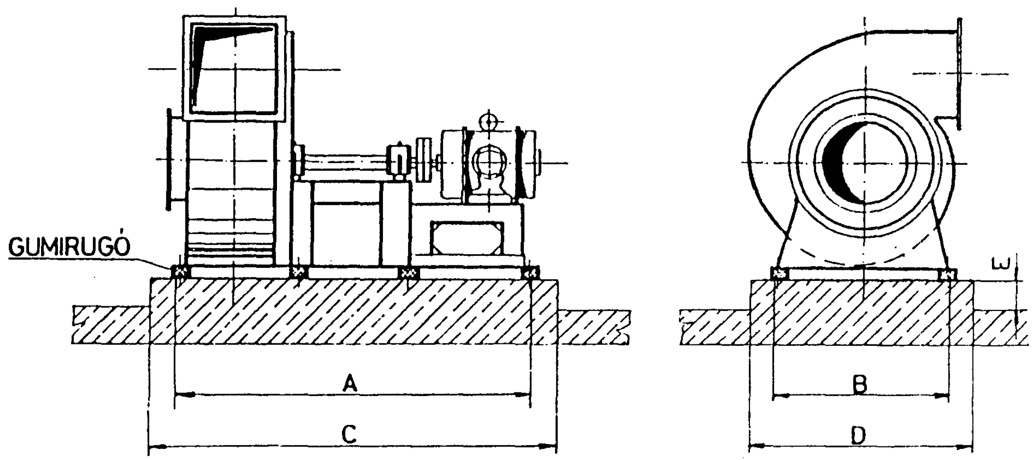



Figure 1.3 Fan installation
Source: [2]

Fig. 1.4 shows a double-stage helical gearbox driven by an electric motor. The rotating parts are covered with protecting cover.

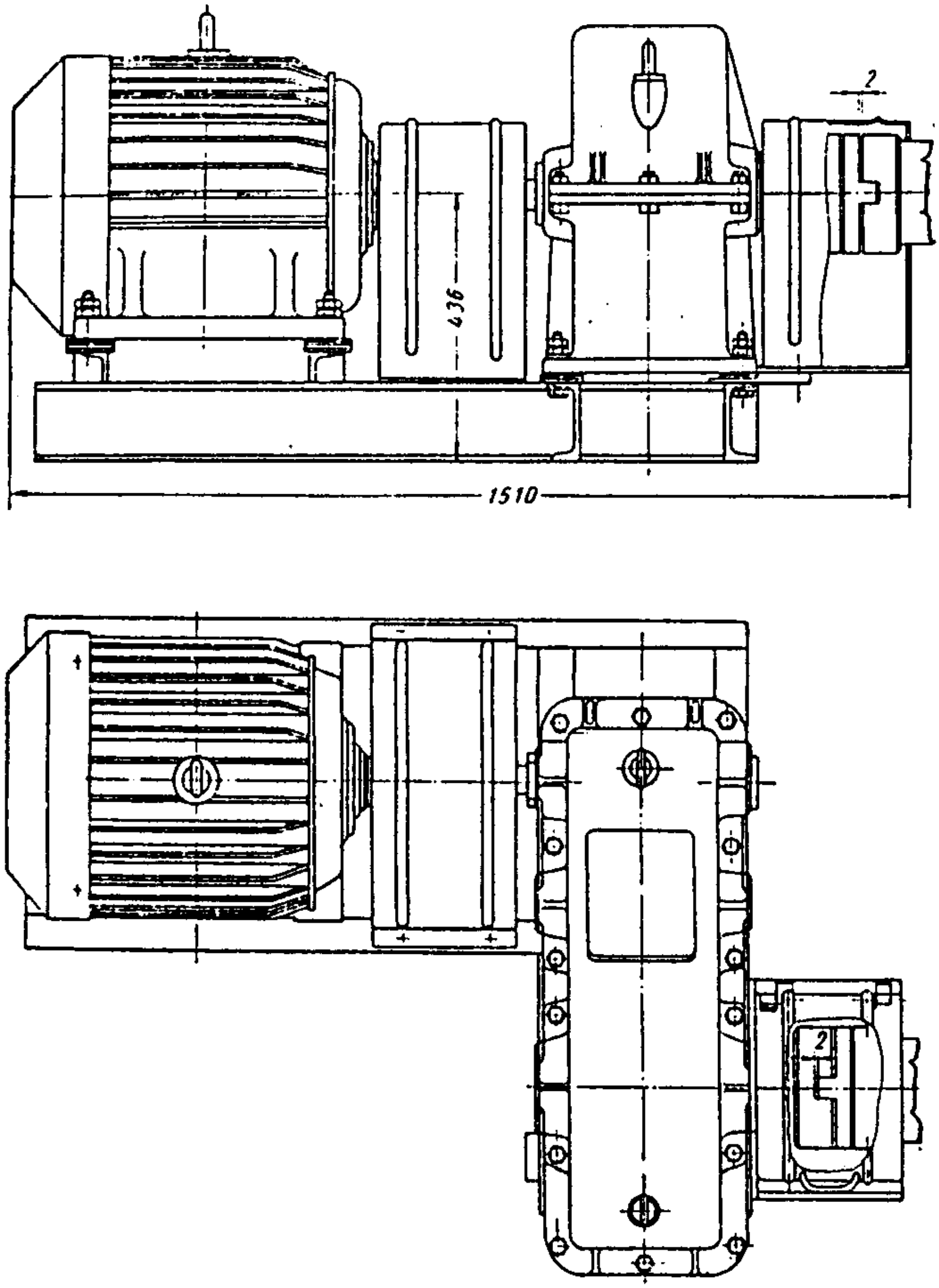



Figure 1.4 Machine group installation
Source: [2]

## 1.1. Construction requirements

The machinery base has to provide for the coaxiality of the centre lines of the prime mover and the coupled machine. Depending on the height difference between the shafts it may be achieved by a stepped construction. The coaxilarity of the coupled shafts during operation is provided by the rigidity of the framework. The bearing areas of the frame where the machines are assembled are in general machined if its overall dimension doesn't exceed the size of 1000x1000 mm. The height differences of the machined surfaces and their parallelism are tolerated by IT10 - IT12. For requiring the least machined bearing surface normally bosses (pads) are welded to the frame to be bearing surface. In the case of grater sizes thin gauge sheets are applied for adjusting the shafts to be collinear. For fixing the base frame to the basement DIN 529 anchor bolts with DIN 934 hex nuts may be used.

## 1.2. Machinery base constructions

Rarely machinery bases are cast but in general they are welded constructions built up from hot rolled or cold finished steel profiles, rolled plates or the combination of them [2], see Fig. 1.5 - Fig. 1.10.

If there is no height difference between the center lines the machinery base may be very simple in design, see Fig. 1.5.

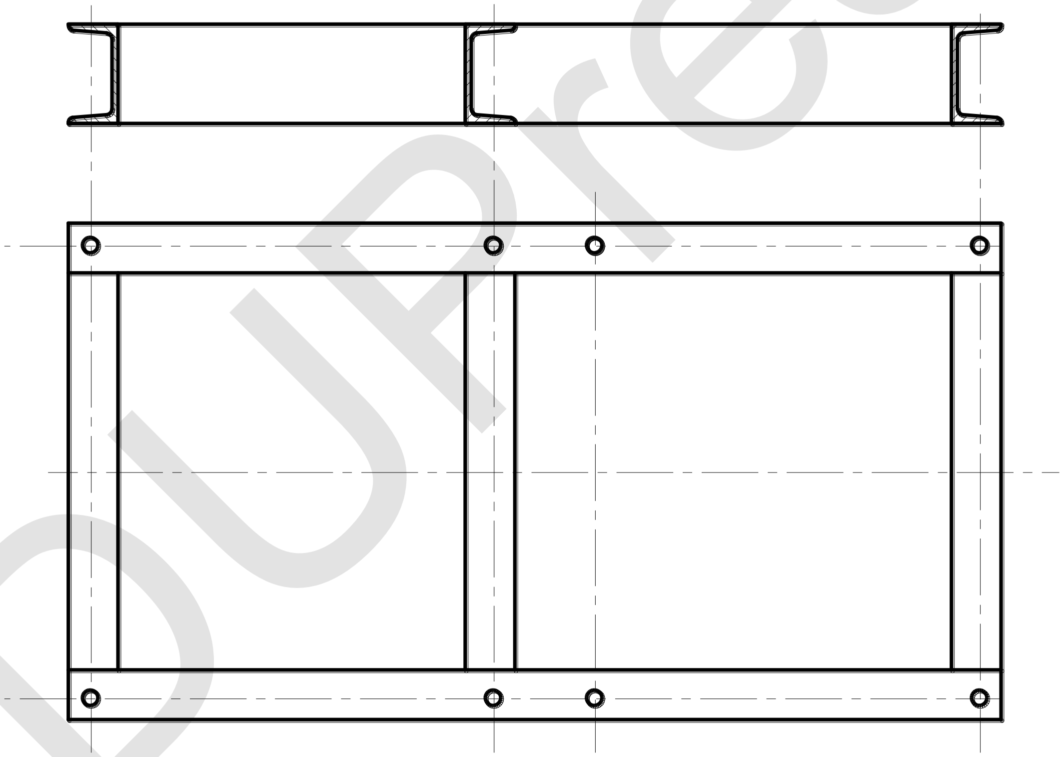


Figure 1.5 Welded construction base frame

In the case of bigger frame size corner truss may be used increasing the torsion stiffness of it, see Fig. 1.6.

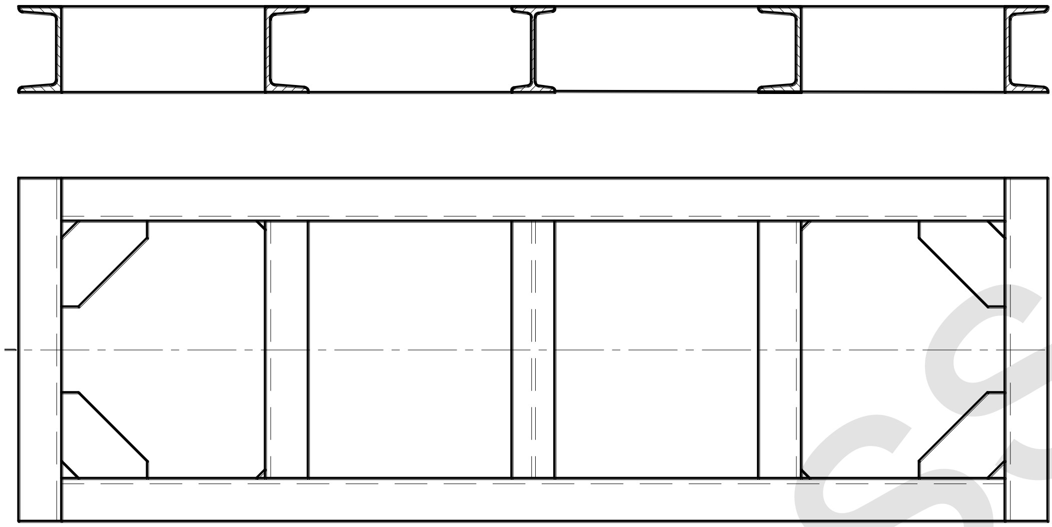


Figure 1.6 Corner truss construction

A very stiff machinery base shown in Fig. 1.7 is built up from cold finished steel sheet.

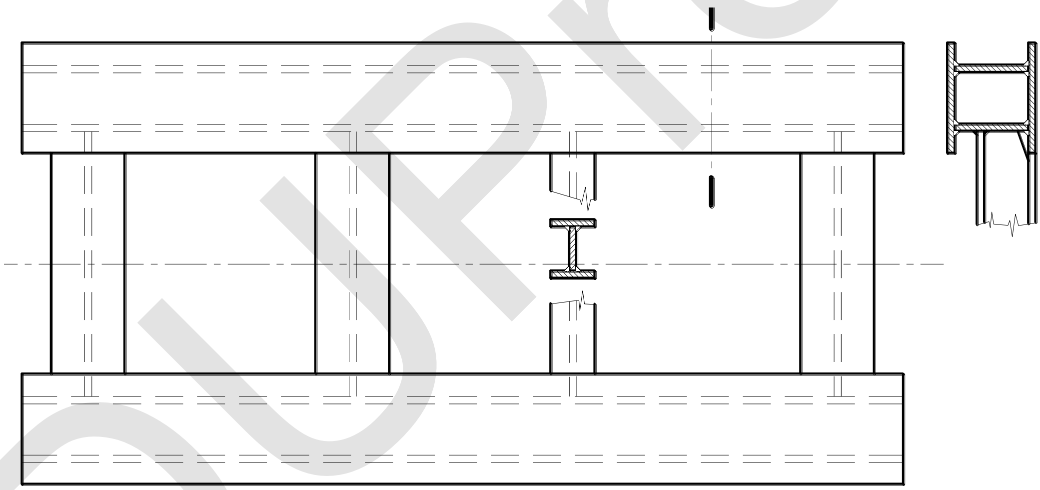


Figure 1.7 Cold finished steel construction

The form of machinery base depends on the machine unit layout: accordingly it may not be symmetric. Fig. 1.8 shows an offset construction carrying a double-stage helical gearbox represented in Fig. 1. 4.

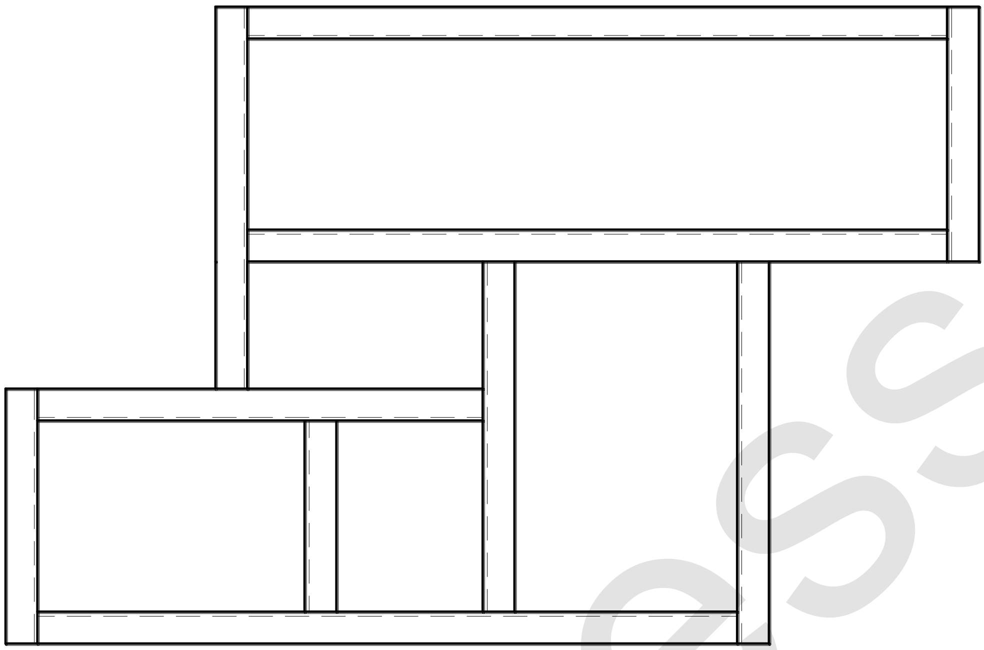


Figure 1.8 Offset construction

For light and smaller machine unit slender frame constructions may be applied, see Fig. 1.9 built up from U profile, box strut and flat bar, and Fig. 1.10 for achieving greater stiffness

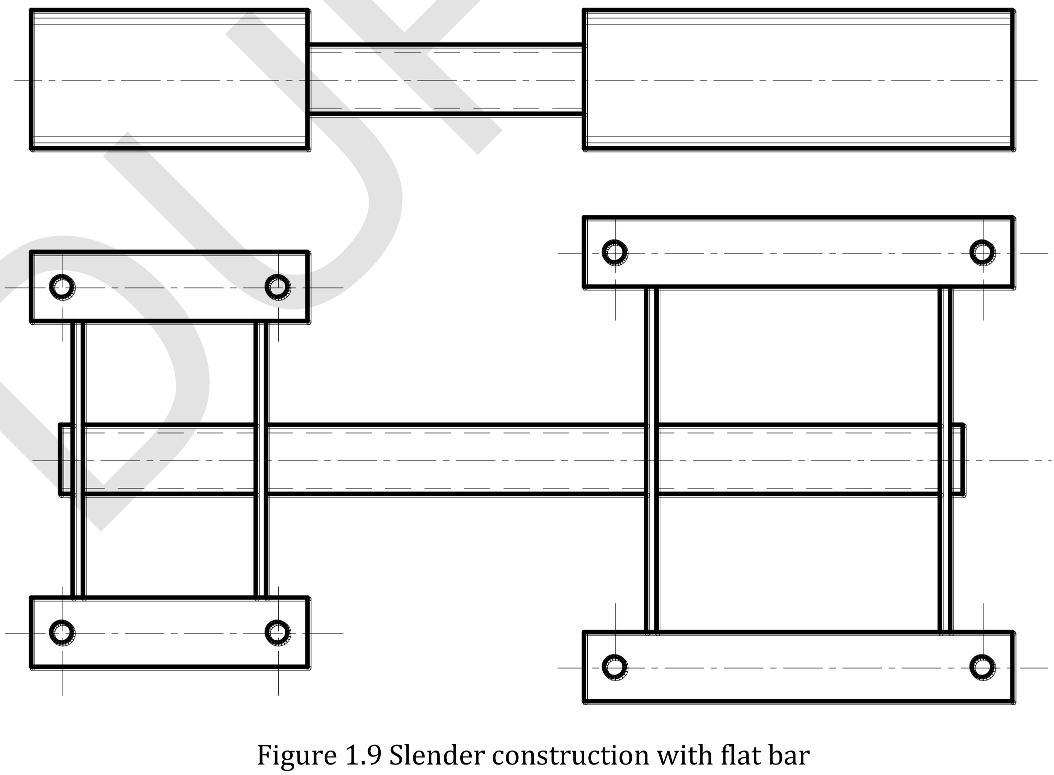


Figure 1.9 Slender construction with flat bar

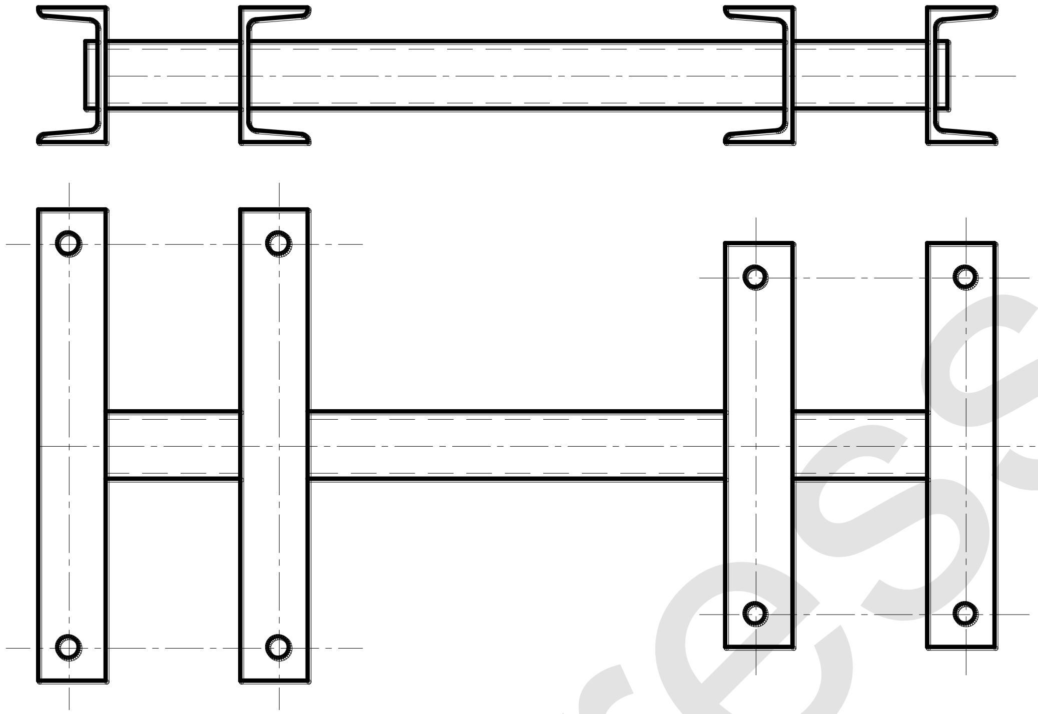


Figure 1.10 Slender construction with U profile

Fig. 1.11 shows a stepped construction where the required height difference is implemented by the clipped U profile. It may be applied for moderate height differences.

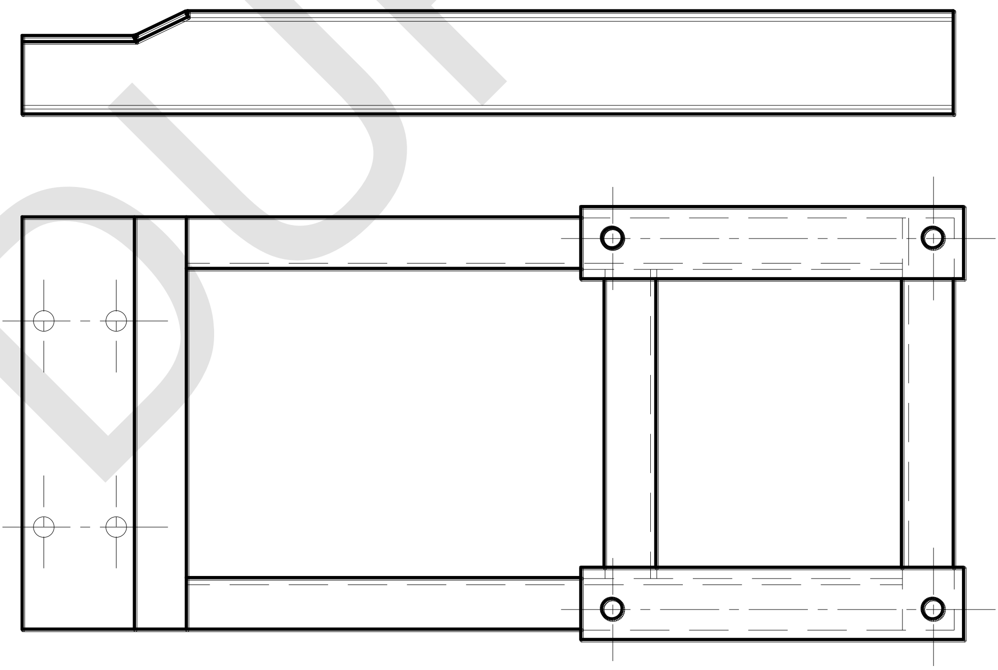


Figure 1.11 Stepped construction with clipped U profile

For significant height differences eg. driving a fan blower (see Fig. 1.3) two-deck base can be designed. The bottom carriage may be any construction mentioned before however the top deck is built up from hot or cold rolled plate, see Fig. 1.12.

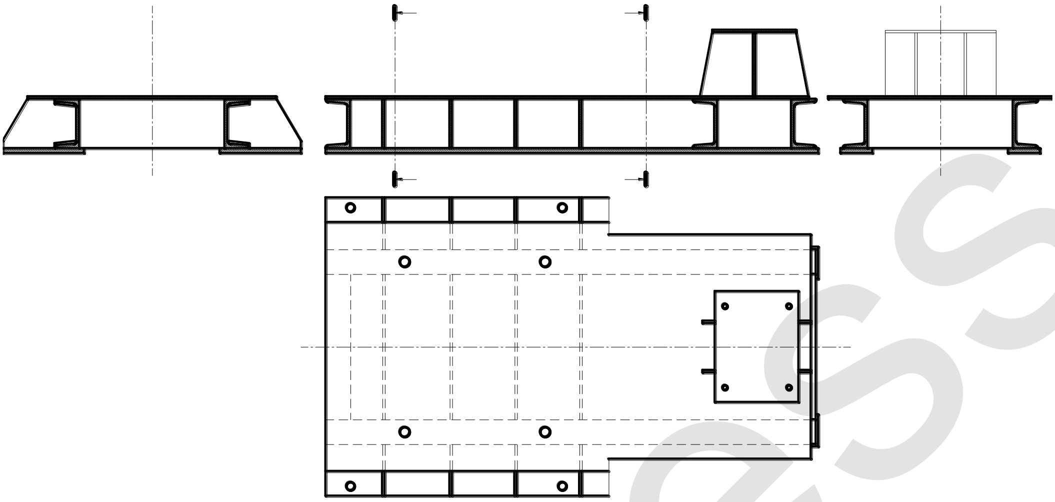


Figure 1.12 Two-deck frame construction

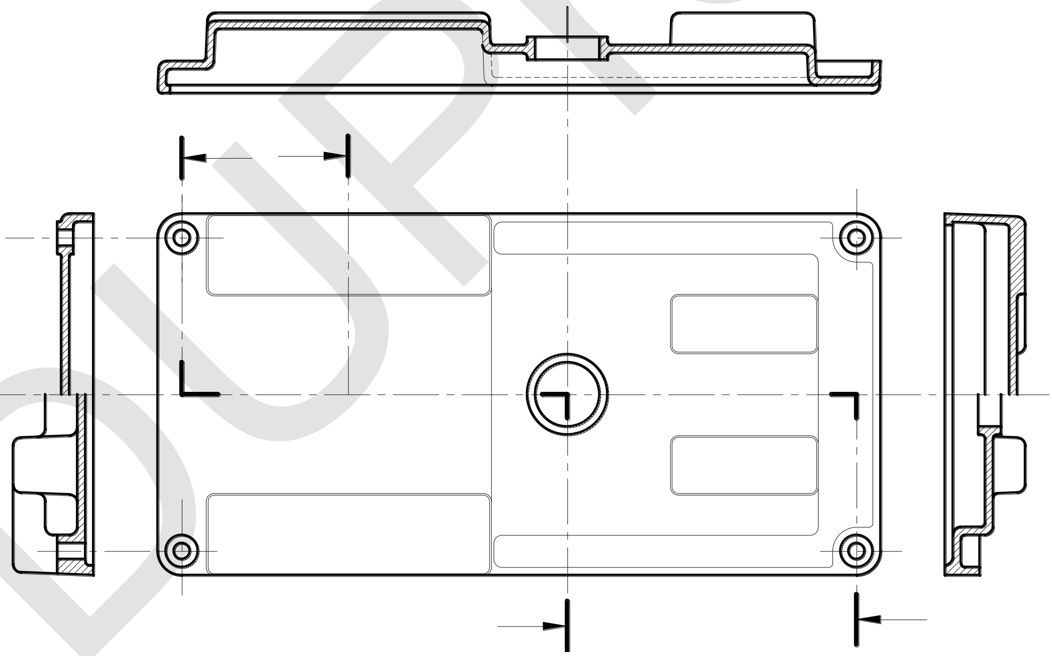


Figure 1.13 Cast machinery base

Cast machinery base may have special form and variable wall thickness and can achieve high stiffness, see Fig. 1.13. In mass production its cost can be low compared to the welded base. In small series production box-type welded machinery base can be designed instead of the cast one.

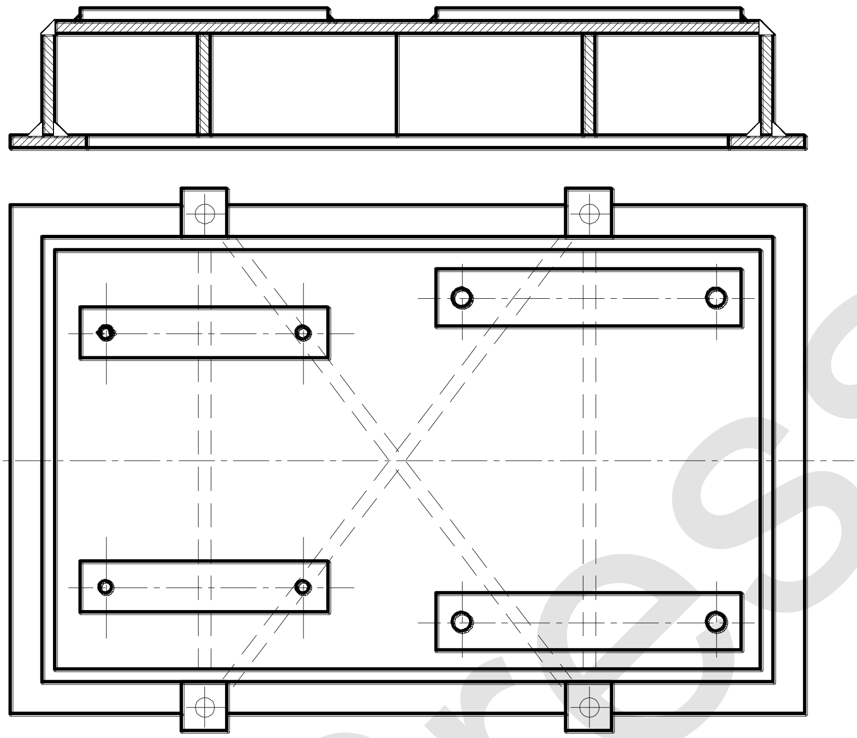


Figure 1.14 Box-type rolled plate base frame

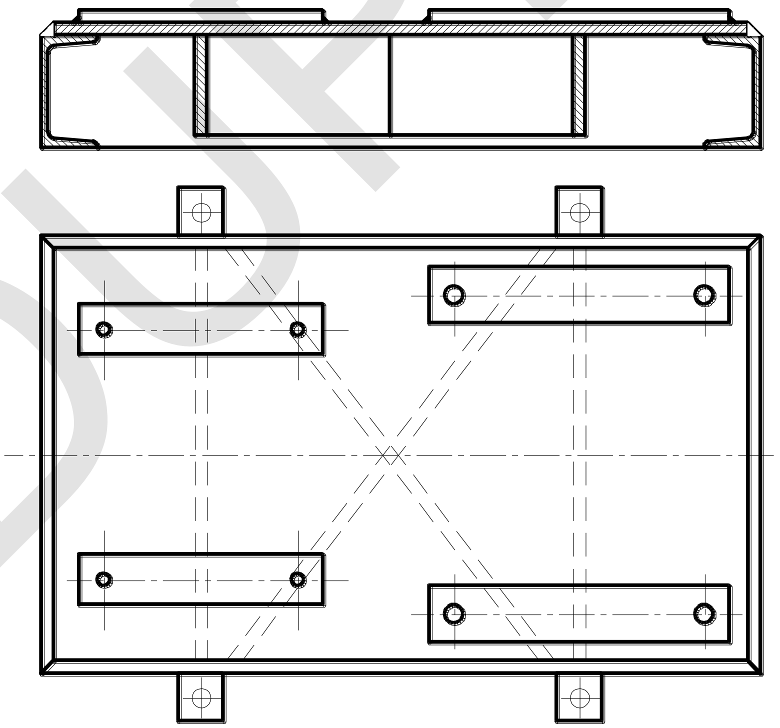


Figure 1.15 Box-type rolled plate base frame with U profile

Its construction is similar to the cast having its features and advantages. A box-type welded base built up from rolled plate is shown in Fig. 1.14 and applying U profile as well is shown in Fig. 1.15. The desired stiffness of the frame is implemented by cross braces in both cases.

# 2. Construction principles

Machinery base may be produced either by casting or by welding. Although both have their relative advantages and disadvantages, choice between casting and welding is often decided by cost. Casting is generally more economical if the part is to be produced in quantity because of the cost of patterns required for casting. However, welding is often advantageous when only a relatively few pieces are to be produced.

## 2.1. Welding design

Rolled plates and rolled structural members (beams, channels, angles, etc.) are commonly used to build up the necessary structure. The general rules for good weldment design are very much the same as for good casting design:

- Apply relatively thin sections, enlarged where necessary by welding in bosses, rims, etc.
- Provide bosses or machining pads wherever flat surfaces are to be machined.
- Strengthen and stiffen light sections with ribs.
- Use simple curves which can be easily obtained by bending plates.
- Parts to be welded are commonly formed by pressing or stamping, thus allowing more intricate shapes than those shown. Forging of the components prior to welding can also be used to advantage where more intricate shapes as well as high strength are required in large machine parts.

The appropriate design allows the welding arc or torch to reach the place to be welded. Welded structures tend to vibrate more than cast structures special when using excessively thin non-reinforced sections. To damp out this vibration, large thin sections may be stiffened with ribs. Welded machinery bases are commonly built by rolled structural members and plates. Warping during the welding process can be a serious problem unless proper holding facilities are applied.
Information and instructions have to be given in the drawing:

- connecting, overall and tolerated dimensions
- type, quality and dimensions of welded joints
- corrosion protection

## 2.2. Connections of structural members

The production of welded construction made of standardized rolled structural members (beams, channels, angles, etc.) is relatively simple. These structural members are produced in standardized length and after cutting into the application size the members may be connected together to create the base frame. The following figures show examples how to connect different profiles in different sizes; and the necessary connecting edge preparations [3].

**Crossbeam connections**

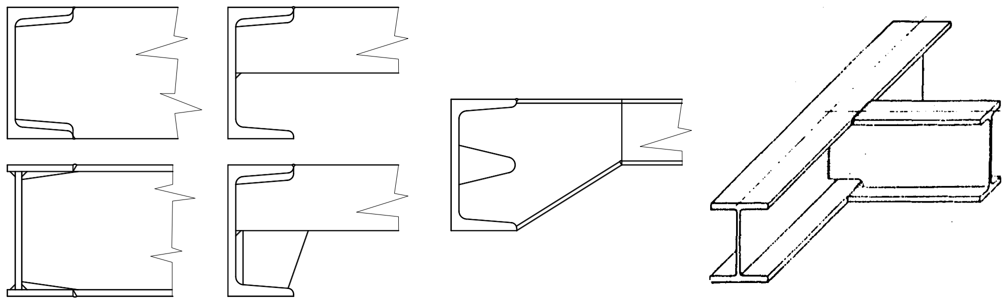
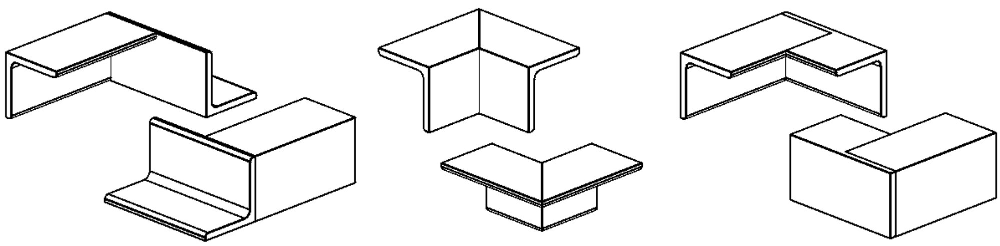


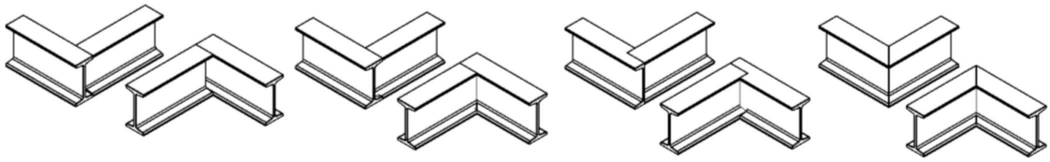
Figure 2.1 Crossbeam joints

**Corner connections**

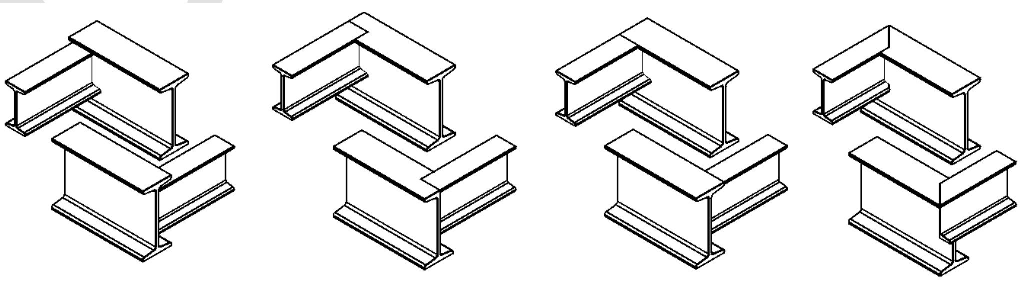
L profile

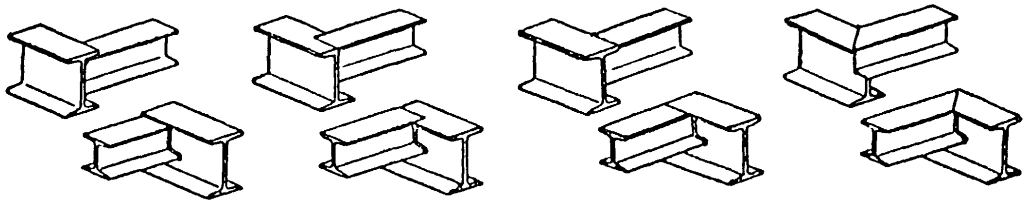


I profiles (same height)

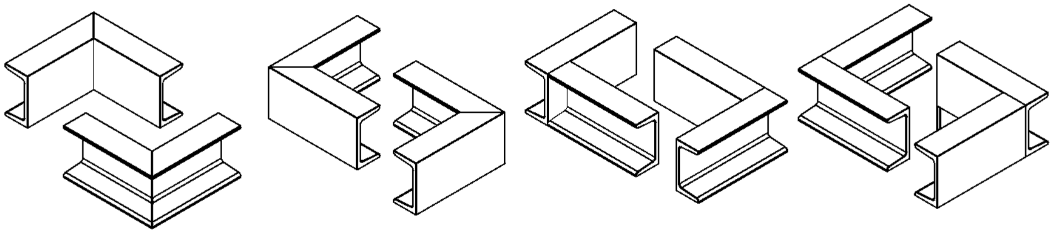


I profiles (different height)

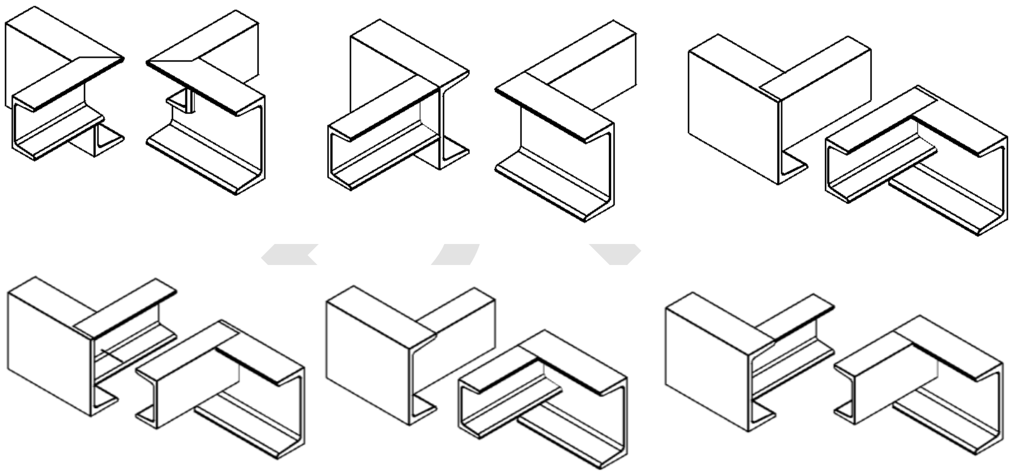




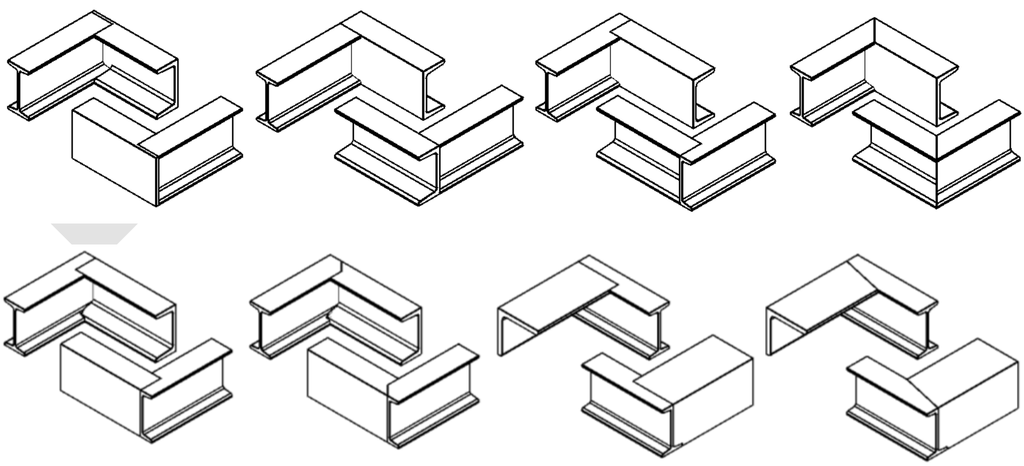
U profiles (same height)



U profiles (different height)



Corner joint of different profiles



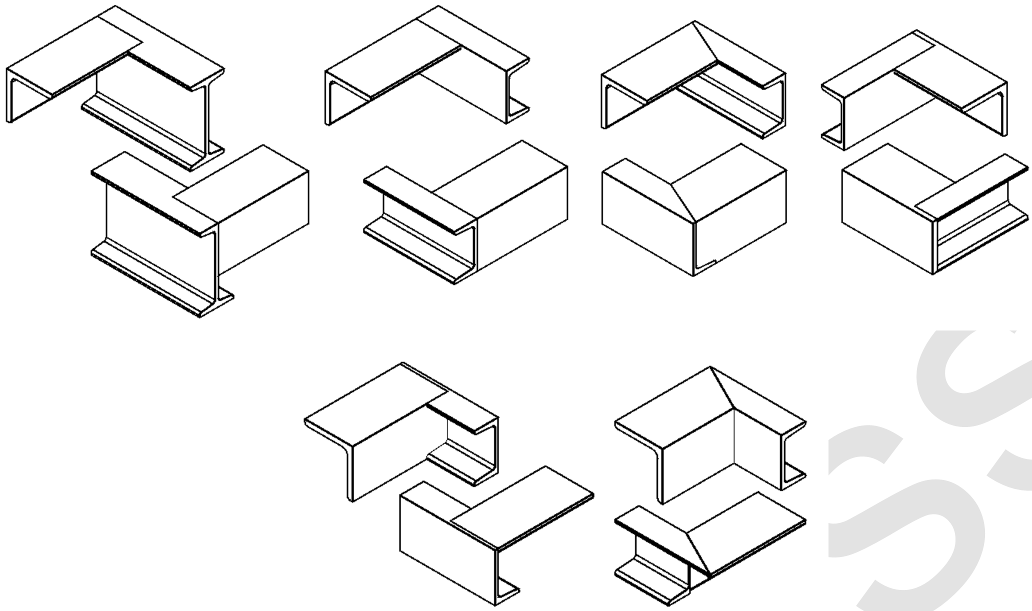


Figure 2.2 Corner connections
Source: [3]

# 3. Welded joints

There are a lot of welds and welded joints appropriate for fixing the members. However the cheapest welded joint is the fillet weld. For common application without special requirements it is applied for making lapping, tee and corner joint.
When applying butt weld eg. V weld, the necessary edges of the connecting parts must be machined prior to welding which is relatively expensive and sometimes requires special machine tools.
The butt weld size of a butt joint should equal the thickness of the thinner part. In the case of fillet weld, the minimal weld size is 3 mm, the maximal one is 0.7s, where s is the thickness of the thinner part. The symbols of welds and joints can be found in Fig. 3.1.

| Joint | | Symbol | Symbol | | |
|---|---|---|---|---|---|
| Designation | Section | | Angle | Width | symbol |
| Symmetrical end lap weld | | | - | Height x1 | |
| Asymmetrical end lap weld | | | - | x3/4 | |
| Square-butt weld | | | - | x1/2 | |
| Single-V butt weld | | | 90° | - | |
| Single-V butt weld with broad root face | | | 90° | - | |
| Single-V butt weld with bottom plate | | | 30° | x1 | |
| Single-U butt weld | | | - | x2/3 | |
| Single-bevel butt weld | | | 45° | - | |
| Single-bevel butt weld with broad root face | | | 45° | - | |
| Single-J butt weld | | | - | x1/2 | |
| Fillet weld | | | 45° | x1 | |
| Plug weld; plug or slot weld | | | 30° | x2 | |
| Spot weld | | | 30° | x2 | |
| Optional butt weld | - | | - | - | |

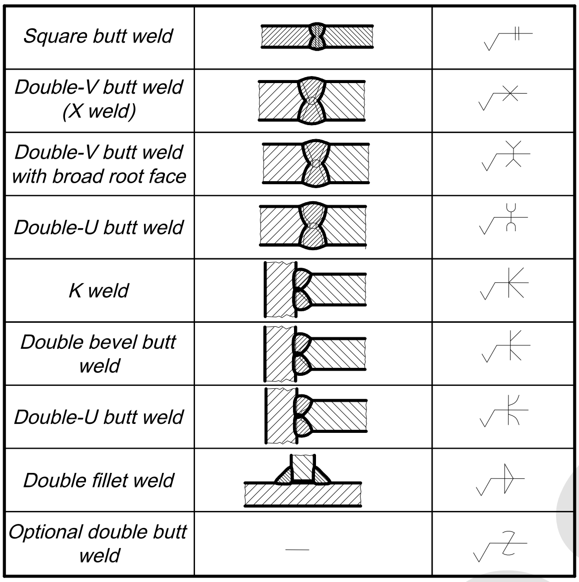



Figure 3.1 Welding symbols

# 4. Dimensioning of the machinery base

The machinery base is a welded construction, however the machine group is fixed on it with bolted joint and the machinery base itself is fixed to the basement with anchor bolts. The proper positioning of the welded frame's parts for welding and holes for fixing the machineries on the frame are provided by appropriate dimensioning. Accordingly there are three dimension systems needed to link with each other considering the producibleness, machinability, cost-effectiveness. To explain the method of dimensioning, we present the three dimensioning independently, and after that the three dimensioning will be linked in one dimension system represented in one figure.

Fig. 4.1 shows a base frame with dimensions positioning its parts, the overall dimensions and cutting length of the parts. We have to pay attention to giving each dimension only once.  Self-evidence dimensions may be given in parenthesis as an informative dimension however these dimensions cannot be used for manufacturing and inspections. Fig. 4.2 shows dimensioning of the hole system; the a x b and the c x d dimensions are the fixing one of the electric motor and the eg. pump; the "e" and "f" dimensions are the distances of motor's and pump's hole system in "x" and "y" directions; the "g" and "h" dimensions position the whole hole system to the base frame. The Fig. 4.3 shows the i x j dimension of the holes fixing the base frame to the basement by anchor bolts; the "k" and "l" dimensions position the hole system to the base frame. Eventually Fig. 4.4 shows the dimensioning system of the base frame containing all the dimensions.

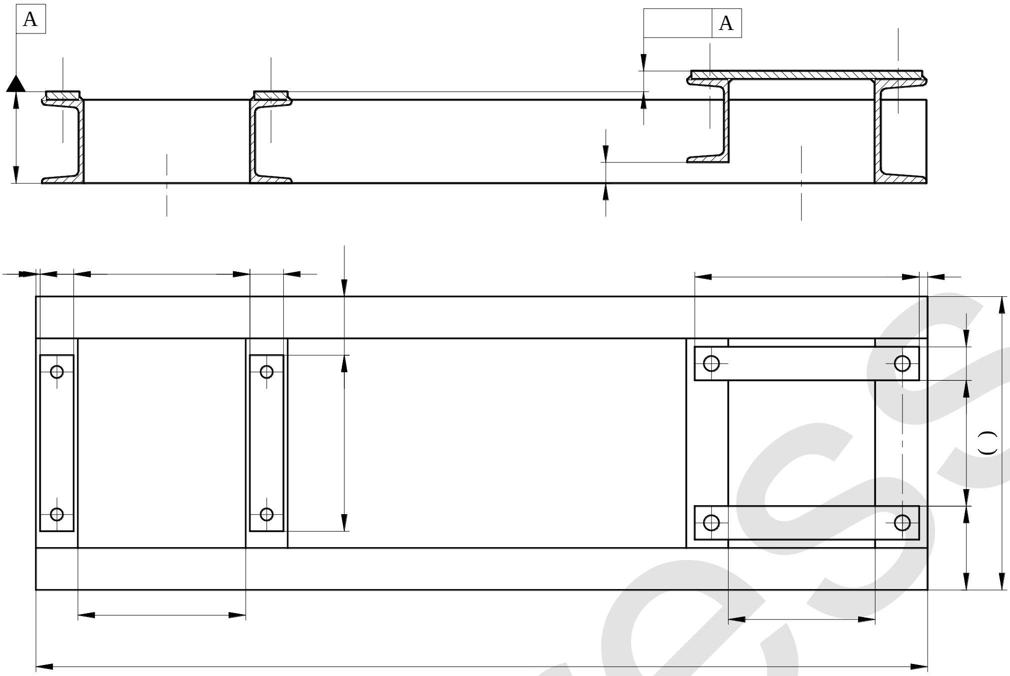



Figure 4.1 Dimensioning of the frame parts

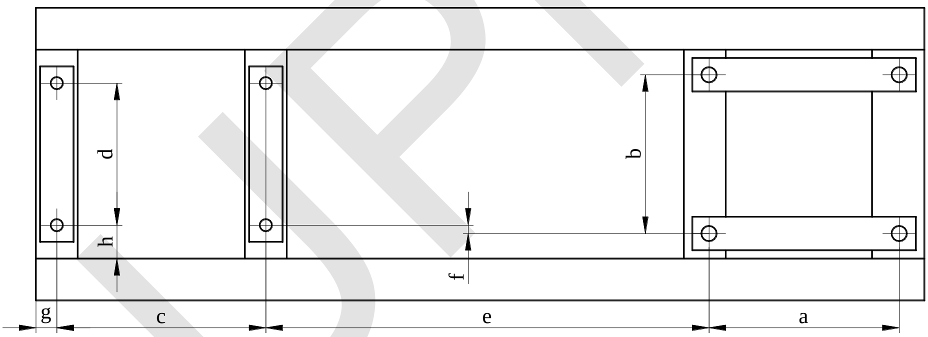



Figure 4.2 Dimensioning of the mahine group holes

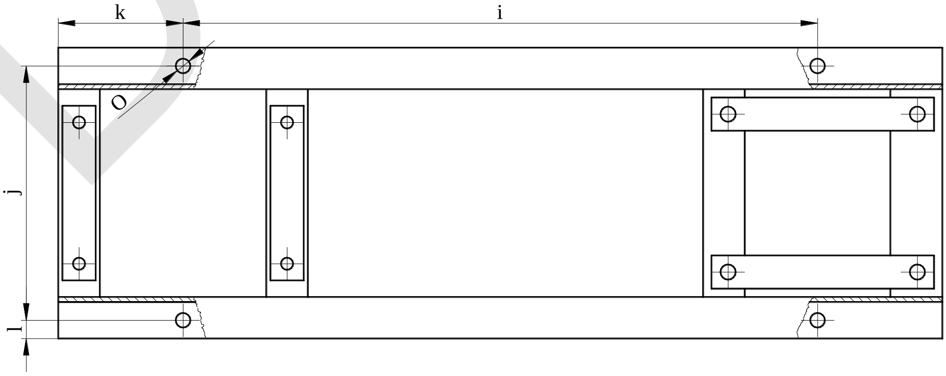



Figure 4.3 Dimensioning of the frame fixing holes

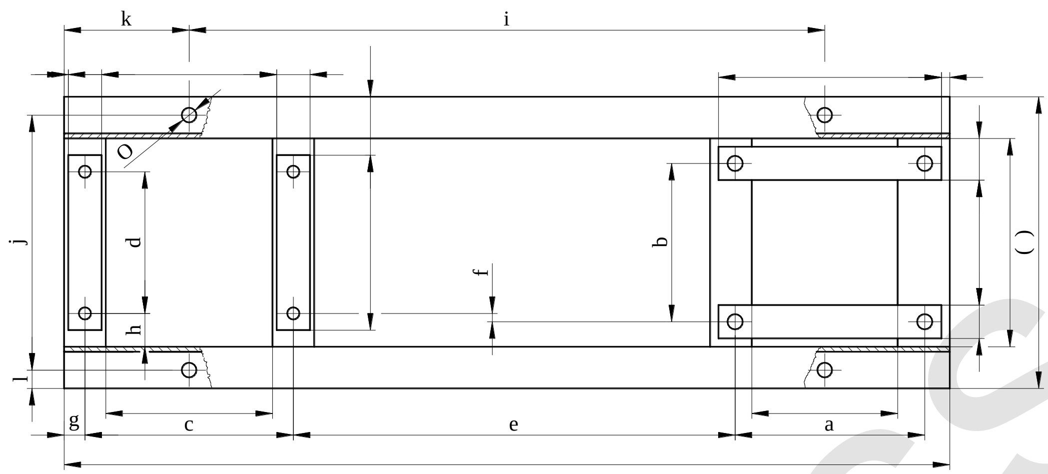



Figure 4.4 Base frame dimensioning

# 5. Load consideration, stressing

Machinery base is loaded by the own weight of the machine unit and the reaction forces of the drive. Base frames in common application are not stressed since it must be very stiff accordingly carrying out a stressing procedure would be pointless. In the case of special application checking for allowed elastic deformation may be prescribed. Now in common application case we do not carry out any stress or deformation calculation. Experienced engineers and technicians can guess sufficient size of the base frame without stress or deformation analysis. However excessive oversizing must be avoided. To gain experience in this field we can study well-tried constructions to feel the proportions (eg. the hole dimensions of the prime mover or gear to fix them compared to the size of the steel profile). It follows from this that if there is not enough room for assembling the bolted joint fixing the equipment unit to the base frame, the size of the steel profile is most likely small.

# 6. Corrosion protection

The most commonly applied corrosion protection of the welded bases is the painting. The contamination (rust, oil, grease, dust, …) has to be removed or cleaned from the surface. This procedure is called surface pretreatment comprising rust removal and degreasing. The service life of the coating depends fundamentally on the grade of the pretreatment. Protective coating contains primer, interstage and cover coat in some layers. The grades of rust removal from the surface are compiled in the Table 6.1 [2]. The grades of the different adherent contaminations and their explanation are compiled in the Table 6.2 [2].

Table 6.1 Grades of rust removal

| Grade of rust removal | Description | Condition of the surface |
|---|---|---|
| K0 | Clean to metal | The rust is completely removed from the surface. After dedusting the surface has uniformly metal polish. |
| K1 | Clean metal surface | The rust removal is limited. After dedusting the surface contains rusty stains in seizings. |
| K2 | Moderate clean | The rust removal is limited. After dedusting the surface is mat and contains rusty stains. |
| K3 | Moderate rusty surface | The rust removal is limited. After dedusting the surface is mat and contains metal polish stains. |

**Prescribing corrosion protection for operation under normal circumstances**

Example A

- surface pretreatment K2 - TY: moderate clean
- primer coating with PLUMBIM in two layers,
- one layer interstage coating with DUROL enamel paint
- cover coating with DUROL enamel cover paint

Example B

- surface pretreatment K1- TO: clean metal surface
- surface treatment: with WASH PRIMERREL paint in one layer
- primer coating with KORROMIM enamel paint in two layers,
- one layer interstage coating with TRINAT primer paint
- cover coating with TRINAT enamel cover paint

Table 6.2 Surface conditions

| Condition of the surface | |
|---|---|
| Grade | According to the adherent contaminations |
| TO | No impurities on the surface proved by water spreading and chemical reaction test and wiping with filter paper respectively. |
| TX | Impurities on the surface can be detected only by water spreading test. |
| TY | Impurities on the surface can be detected only by chemical reaction test. |
| TZ | Impurities on the surface can be detected only by water spreading and chemical reaction test. |
| TG | Only neutrality water can be detected on the surface. |

# CHAPTER B. Hydraulic cylinder

Design a hydraulic cylinder according to the given task number and the given construction represented in figures [9].

- Carry out stressing procedure relating to the wall thickness and determine the number and size of the screws or the size of the threaded tie rods fixing the cylinder head and cap to each other or to the flanges.
- Determine the prestressing force and the wrench torque necessary for fixing the cylinder cap.
- Check welding joints applied and the critical cross-sections of the piston rod subjected to fatigue load and check the hydraulic cylinder for buckling.
- Choose seals and pipe connectors from brand catalogues.
- Elaborate the assembly drawing of the hydraulic cylinder and the shop drawing of its parts on paper sheets. Apply the appropriate scale to represent the details.
- Compile the minutes of the calculations and stressing procedures carried out step by step.

The assembly drawing shall contains views and sectional views necessary to give overall, connecting and tolerated dimensions, to show the welding joints and its dimensions. Give prescriptions regarding the welding joints and corrosion protection. Complete the title block with the parts list and give the necessary data regarding the material, dimension and standard number of them. On the basis of the assembly drawing make the shop drawing of parts not available as a commercial product. It has to contain all the dimensions needed for manufacturing. Although the drawings are drawn in pencil, pay attention to applying the proper line thickness.

Design parameters

| Number | Type | p [MPa] | D [mm] | l [mm] |
|---|---|---|---|---|
| 1. | 1 | 4 | 50 | 200 |
| 2. | 2 | 6,3 | 60 | 250 |
| 3. | 3 | 10 | 70 | 300 |
| 4. | 4 | 16 | 80 | 400 |
| 5. | 5 | 25 | 90 | 500 |
| 6. | 6 | - | 100 | 550 |
| 7. | - | - | 110 | 600 |

**Comment**

The first figure of the task number refers to the construction of the hydraulic cylinder (see Fig. 1 - Fig.6), the second one refers to the working pressure, the third one refers to the cylinder bore diameter and the forth one refers to the working stroke, see it in Table 1. Choose the stroke velocity between 0.15 and 0.2 m/s.

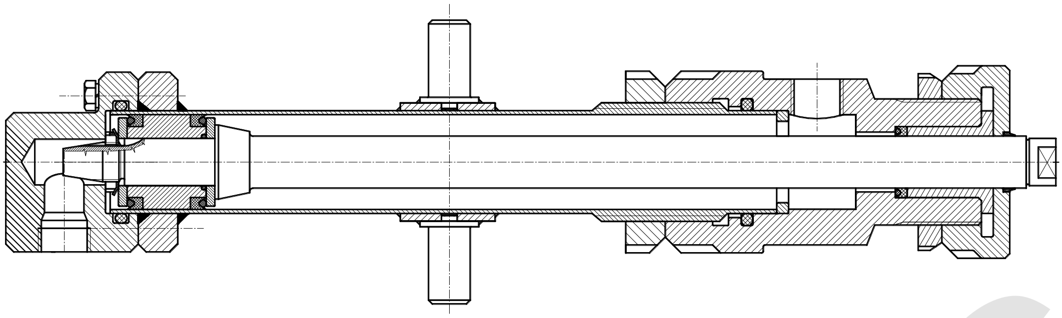


Figure 1

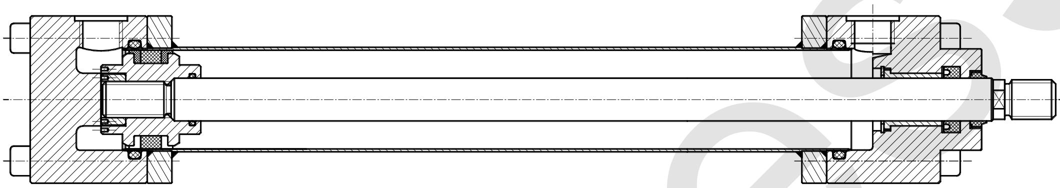


Figure 2

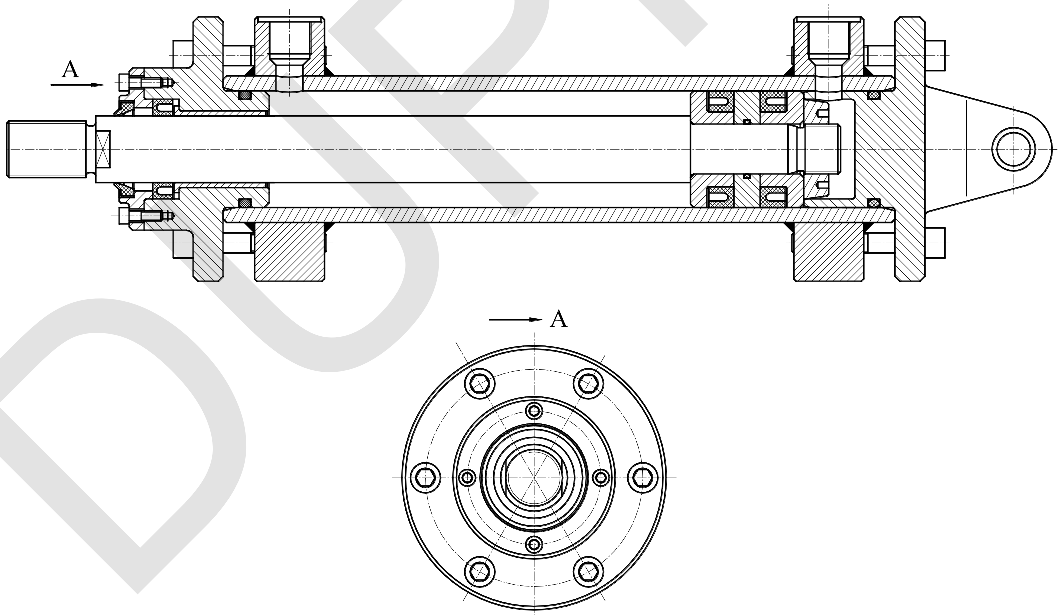



Figure 3

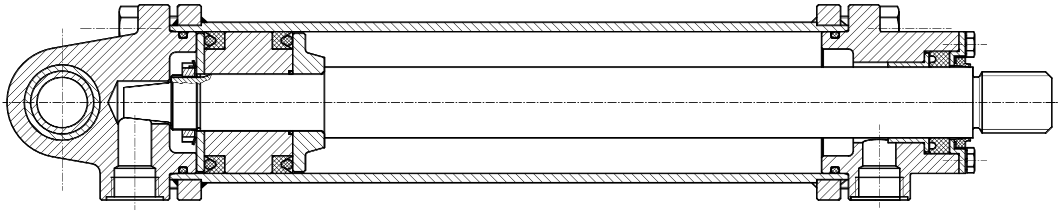


Figure 4

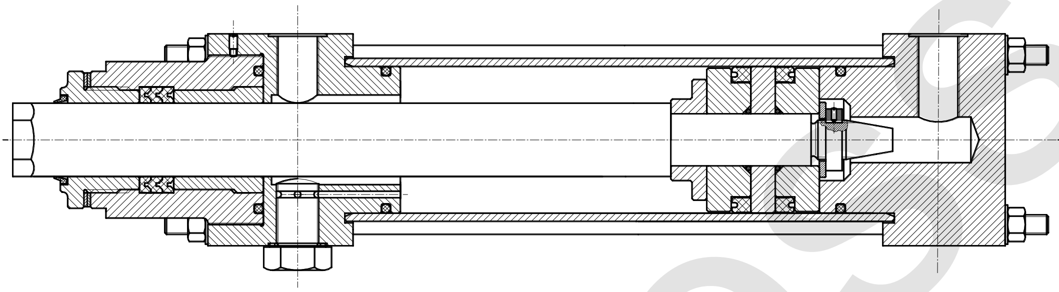


Figure 5

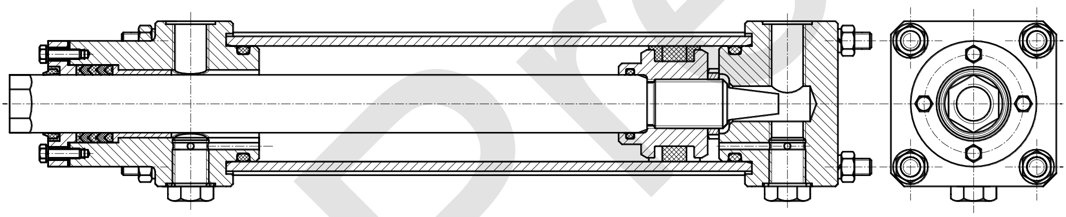


Figure 6

## Design, stressing and construction considerations

When designing a hydraulic cylinder, the following details have to be clarified:

- task and operation requirements,
- the piston rod moves out and into a high pressure room can be implemented by sealing,
- the piston (rod) moves between its terminal positions (along its stroke), at working velocity and pressure,
- contact moving sealing generate heat limiting its operation range,
- it is a high pressure cylinder covered by a head and cap fixed with screw joint, assembled with pretension force,
- moving out and in piston rod modifies the slenderness of the hydraulic cylinder,
- hydraulic cylinder may be the part of a mechanism or can be fixed independently to exert force,
- construction in terms of producibleness, machinability, cost-effectiveness,
- load of the parts of the hydraulic cylinder and stressing procedures.

# 1. Operation of a hydraulic cylinder

Hydraulic cylinders are single or double acting hydraulic motors with linear reversible motion. They are used as driving units of hydraulically operated equipment, like vehicle hoists, forest, agricultural, construction machinery, and in the mobile hydraulics generally. Wide variety of types and terminations of cylinders and pistons makes big variability of applications possible [4]. Hydraulic cylinders are designed for a nominal pressure up to 320 bars according to their type and size. The range of the cylinder bore is from 32 to 100 mm, the diameter of piston rods is from 22 to 80 mm, and maximal stroke is up to 1000 mm. Hydraulic cylinders perform straight-line motion in mechanisms when hydrostatic power of the fluid is turned to mechanical power by the travel of the piston or the cylinder.

## 1.1. Hydraulic cylinder types

Configurations:

a. Single action, see Fig. 1.1.a. The fluid pressure can thrust the piston only in one way. The force for backward motion is exerted by external force.
b. Double action achieved by the fluid pressure, see Fig. 1.1.b and Fig. 1.1.c (double rod).

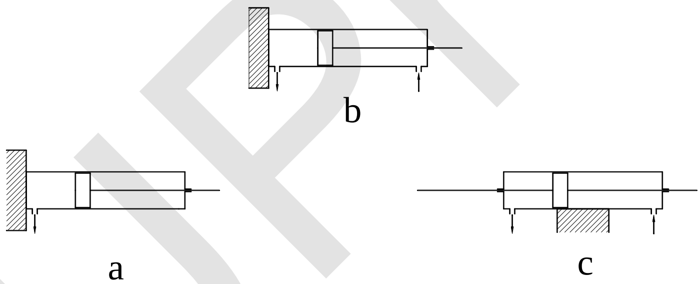



Figure 1.1 Hydraulic cylinder types

The most commonly applied hydraulic cylinder is the double action one with single rod, see Fig. 1.2.

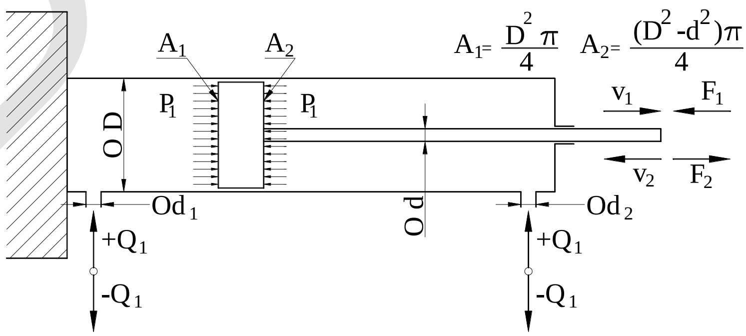



Figure 1.2 Double action operation

## 1.2. Operating forces

The equation of equilibrium on the bases of the draft, see Fig. 1.2.

**If** $p_1 > p_2$

- the piston moves into the $v_1$ direction
- the piston rod is subjected to compression load

$$A_1 p_1 - A_2 p_2 - F_f - F_1 = 0$$

The pressing force of the rod:

$$F_1 = A_1 (p_1 - \gamma p_2) - F_f$$

Where: $p_1$ operating pressure

$p_2$ pressure of the effluent oil

$A_1$ and $A_2$ effective piston area

$\gamma = \frac{A_2}{A_1}$ ratio of the areas

$F_f$ friction force between the piston and rod sealing

$p_1$ actuating pressure exerted by hydraulic pump,

$p_2$ hydraulic resistance of the effluent oil.

$F_f$ depends on:

- operating pressure
- type of the sealing applied
- oil viscosity
- stroke velocity

**If** $p_2 > p_1$

- the piston moves into the $v_2$ direction
- the piston rod is subjected to tensile load

The tensile force of the rod:

$$F_2 = A_1 (\gamma p_2 - p_1) - F_f$$

## 1.3. Operating features

Hydraulic cylinders may be installed for different applications.

a. Stroke velocity $v = 0$

Hydraulic cylinder is used to exert force without travel, see Fig. 1.3.

$F_1 = A_1 p_1 - F_f = A_1 p_1 \eta_{mech}$ since $p_2 \approx 0$

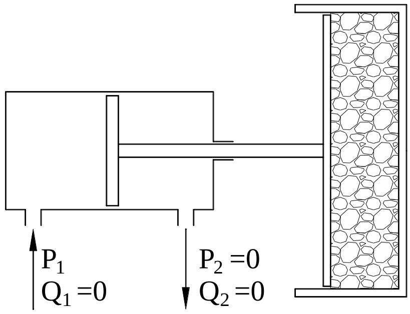



Figure 1.3 Hydraulic cylinder application

b. Stroke velocity $v < 0.2$ m/s.
   The hydraulic resistance of the effluent system is low therefore it can be ignored.
   The rod forces:

$$F_1 = A_1 p \eta_{mech} \text{ and } F_2 = A_2 p \eta_{mech}$$

The efficiency is approximately: $\eta_{mech} = 0.95 - 0.96$

$p$ : actuating pressure

The required oil flow rate in direction of $v_1$

$$Q_1 = v_1 A_1 \quad \left[\frac{l}{s}\right]$$

If the swallow is the same in both direction,

$$Q_1 = Q_2$$

than $v_2 = \frac{Q_2}{A_2} = \frac{Q_1}{\gamma A_1} \left[\frac{m}{s}\right]$

c. Stroke velocity is high.
   At high stroke velocity the actuating pressure has to cover the increasing resistance of the effluent oil too, hence less force gets to the piston rod. For calculating the piston rod force, diagrams are provided in catalogues about the hydraulic resistance depending on the stoke velocity.

If the hydraulic cylinder is used to move high-masses, the supplementary force due to the mass inertia of the moved parts has to be taken into account when the piston comes to its terminal position. The effects of the impact force can be reduced by cushioning system. It provides accurate adjustment and softer damped stops.

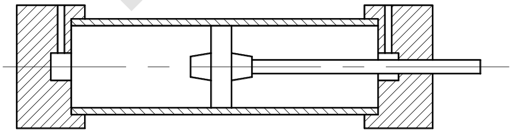


Figure 1.4 Cushioning construction

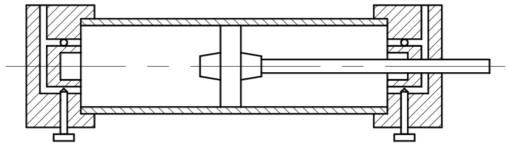


Figure 1.5 Cushioning with valves

Fig. 1.4 and Fig. 1.5 show two constructions for cushioning. The cushioning is achieved by the fluid resistance in the annulus between the bore in the head and the beveled part of the piston (see Fig. 1.4). Fig. 1.5 shows the application of a valve system to achieve cushioning.

# 2. Hydraulic cylinder constructions

The construction of the hydraulic cylinders varies according to the requirements of special installations: see Fig. 2.1 – Fig. 2.3. They may differ by the ways of fixing the cylinder head and the cap to the cylinder body or to each other. They may be threaded (Fig. 2.1 right side), fastened with screws to flanges (Fig. 2.2) of the cylinder body or the cylinder body may be clamped by the cylinder head and the cap by tie rods (Fig. 2.3) respectively [4].

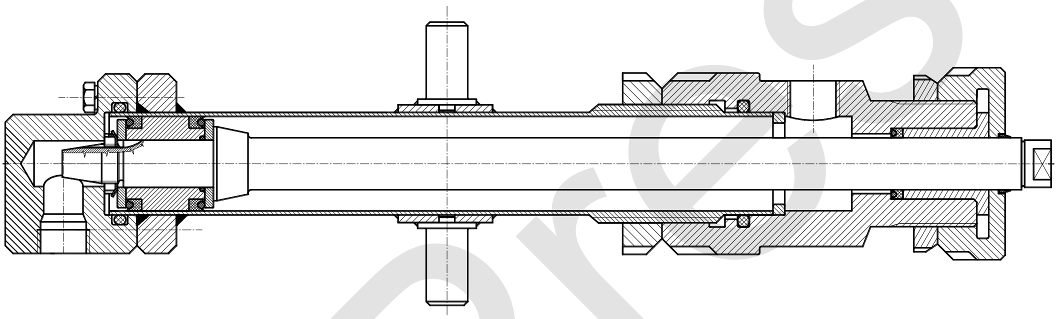


Figure 2.1 Threaded construction

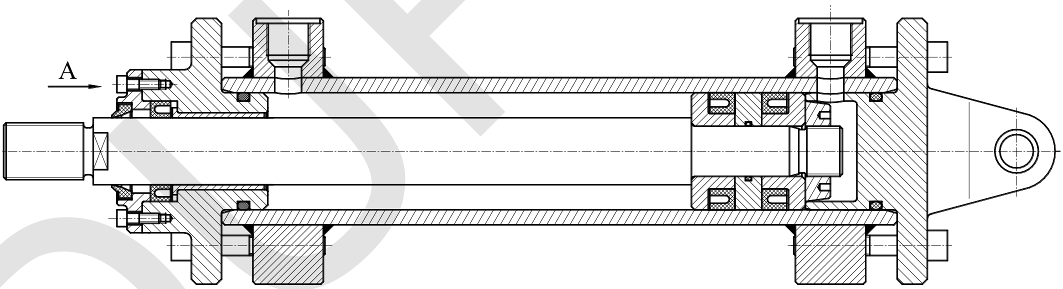



Figure 2.2 Screwed joint application

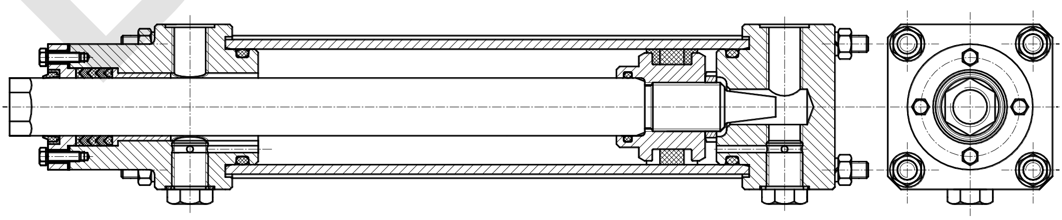


Figure 2.3 Tie rod application

## 2.1. Cylinder head, cap, piston and rod sealing

**Static seals**

In a static sealing application there is no movement between sealing surfaces or between the seal surface and its mating surface. In the hydraulic cylinder for body-to-head and cap sealing the "□", "☼" elastomer fabric and „O" elastomer rings are applied commonly as static seals [5].

**Dynamic seals**

They create a barrier between moving and stationary surfaces in rotary or linear applications such as rotating shafts and pistons. Commonly the elastomer **"U", „V"** sealing rings actuated by the pressure of hydraulic oil are applied for parts making oscillating motion. However for low speeds ($v_{max} < 0.1$ m/s) „O" ring, and "O" ring with back-up washer may be applied.

For higher loads and speeds ($v_{max} > 1$ m/s) special **piston seal kit** may be used comprising a piston bearing and piston seals. The piston bearing is a non-metallic piston wear band having a temperature resistance up to 200C°. The piston seals may be carbon graphite filled PTFE piston lipseals having an internal stainless steel spring that energizes both the dynamic and static seal lips to provide optimal sealing throughout the operating temperature range.

The hardness of the elastomers is in general 60, 70, 80 and 90 Shore A°. The recommended hardness of sealing materials between moving part:

- in pneumatics: $60 \pm 3$ Shore Aº
- in hydraulic system for
  - low pressure ($p_{max} < 1.6$ MPa) $70 \pm 3$ Shore Aº
  - medium pressure ($p_{max} < 6.3$ MPa) $80 \pm 3$ Shore Aº
  - high pressure ($p_{max} > 6.3$ MPa) $90 \pm 3$ Shore Aº

**Grooves for seals**

The actual groove shape and dimensions depend on the type and size of the seal ring cross section and the application field. Recommended groove dimensions are provided in the product catalogue. Since elastomeric materials are almost incompressible, it is necessary to provide sufficient room for the seal ring in the groove. The recommended groove dimensions provide that.

The requirements regarding the surfaces contacted by seal ring are the following:

for dynamic seals:

- the maximum surface roughness $R_a = 0.63$ μm, but preferably $R_a = 0.32$ μm
- the contact surface is treated to gain wear-resisting surface by nitride hardening, surface hardening, hard chromium plating

for static seals:

- the maximum surface roughness $R_a = 2.5$ μm but preferably $R_a = 1.25$ μm

**Clearance between the surfaces to be sealed**

The desired clearance between mating parts depends on the type of the seal, the hardness of the seal material and the operating pressure. In the case of hydraulic cylinders this requirement can be satisfied with H11/e9 fit for smaller sizes and H8/f7 fit for larger sizes.

## 2.1.1. O rings for reciprocating motion and static seal

O-ring may be applied as static and dynamic seal as well, depending upon the pressure and the speed. Standard sizes of O rings (Fig. 2.4) are compiled in product catalogues [6]. The size of the O ring is designated by the cross-section diameter d and the nominal inside diameter D.

The measure of clearance that can be permitted depends on the pressure difference across the O-ring and the ring material hardness. If the clearance is too big, the O-ring is extruded into the gap (Fig. 2.5). The reciprocating motion then tears away small pieces of the O-ring, which results in a leaking seal and contamination of the working fluid by the O-ring particles.

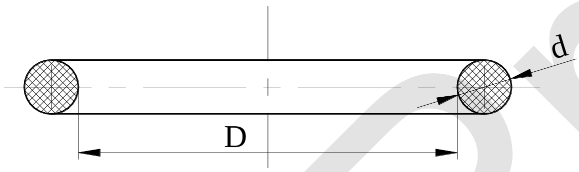



Figure 2.4 O-ring

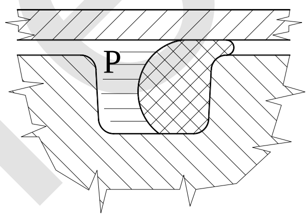



Figure 2.5 O-ring extruded

Without backup ring application the correlation between the rod speed and working pressure can be seen in Fig. 2.6 [6].

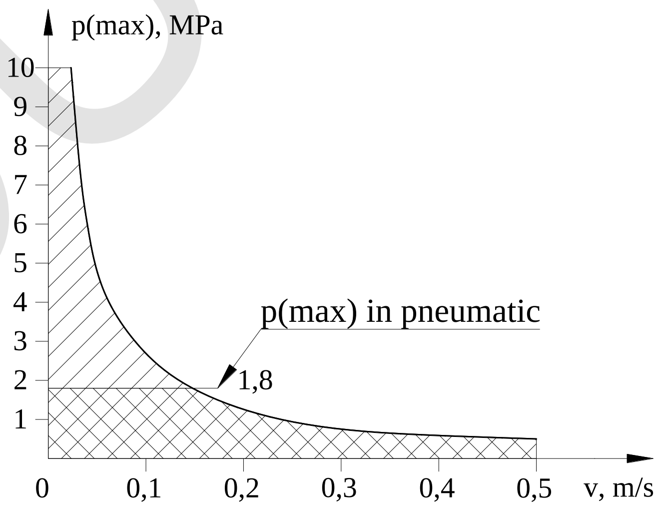



Figure 2.6 O-ring operation range

Due to the friction between the O ring and the contact surface heat is developing. The friction power is proportional to the product of the pressure and the velocity (pv), which results in a hyperbola function when considering the limited value of it. The temperature rise in this case is below the allowed value pertaining to the sealing material in continuous operation. Analysis of friction power is detailed in the brake design chapter. The application of O-ring is recommended up to 10 MPa in hydraulic system and up to 1.6 MPa in pneumatics.

**Groove for O rings**

Rectangle or trapezoidal groove dimensions for O rings on cylindrical surface are given in Fig. 2.7 and Fig. 2.8. Groove dimensions of different O ring cross-section diameters for sealing on static and reciprocating surfaces are summarized in Tab. 2.1 [6].

Table 2.1 Groove dimension for O-ring

| **Cross-section diameter d** | **Groove width** $b_0^{+0,2}$ | **Groove depth a** | | | **Fillet radius** $r_{max}$ | **Chamfer** $c_{min}$ |
|---|---|---|---|---|---|---|
| | | **Stationary surfaces** | **Reciprocating motion** | | | |
| | | | **Hydr.** | **Pneum** | | |
| 2.4 | 2.9 | 2 | 2.1 | 2.2 | 0.5 | 0.6 |
| 3 | 3.7 | 2.5 | 2.7 | 2.8 | 1.0 | 0.8 |
| 5.7 | 7.2 | 5 | 5.2 | 5.4 | 1.0 | 1.0 |

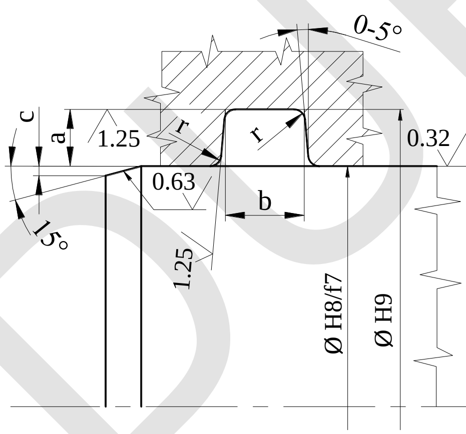



Figure 2.7 O-ring groove in the house

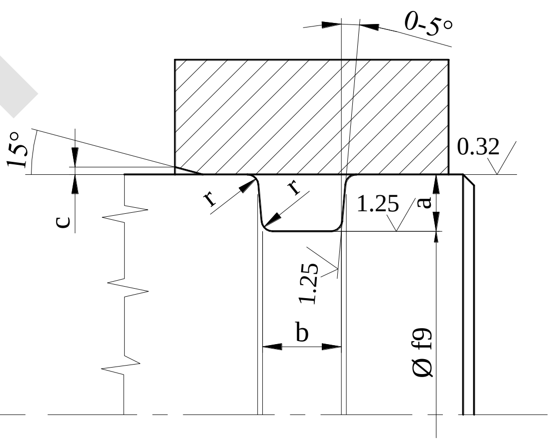



Figure 2.8 O-ring groove in the rod

Groove dimensions of small diameter (not standardized) O rings for sealing on cylindrical stationary surfaces are compiled in Tab. 2.2.

Table 2.2 Groove dimension for O-ring

| **Cross-section diameter d** | **Groove width** $b_0^{+0,2}$ | **Groove depth a** |
|---|---|---|
| 1.6 | 2.1 | 1.2 |
| 2 | 2.6 | 1.5 |

The trapezoidal groove dimension of flange sealing with „O" rings (see Fig. 2.9) for different cross-section diameters are compiled in Tab. 2.3.

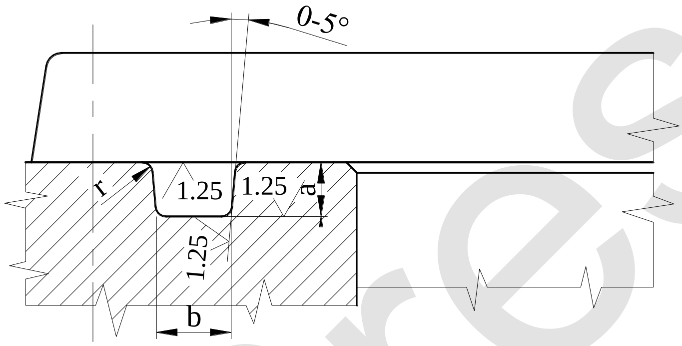



Figure 2.9 O-ring groove on end face

Table 2.3 Groove dimensions on end face for O-ring

| **Cross-section diameter d** | **Groove depth a** | **Groove width** $b_0^{+0,2}$ | **Fillet radius** $r_{max}$ |
|---|---|---|---|
| 2.4 | $1.7_{0}^{+0.1}$ | 3 | 0.5 |
| 3 | $2.2_{0}^{+0.1}$ | 4 | 1 |
| 5.7 | $4.4_{0}^{+0.1}$ | 7.6 | 1 |

Triangular groove dimension of flange sealing with „O" rings (see Fig. 2.10) for different cross-section diameters are compiled in Tab. 2.4.

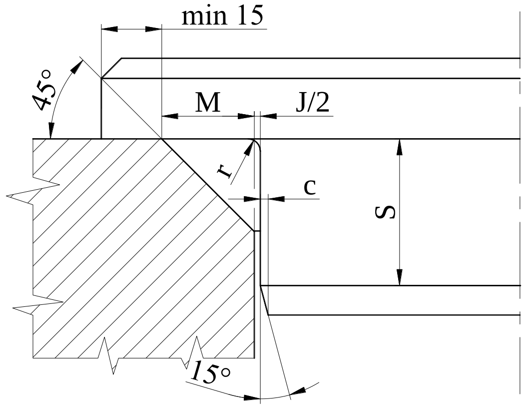



Figure 2.10 Triangular groove for O-ring

Table 2.4 Triangular groove dimensions for O-ring

| **Cross-section diameter d** | **Fillet width M** | **Fillet radius $r_{max}$** | **Length of the guide pin $s_{min}$** | **Chamfer $c_{min}$** | **Clearance $J_{max}$** |
|---|---|---|---|---|---|
| 2.4 | 3.3 | 1.3 | 5 | 0.7 | 0.12 |
| 3 | 4.2 | 2.0 | 6.0 | 0.8 | 0.15 |
| 5.7 | 7.8 | 3.0 | 10.0 | 1.2 | 0.18 |

Triangular groove dimensions of flange sealing with small diameter (no standardized) O rings are compiled in Tab. 2.5.

Table 2.5 Triangular groove dimensions

| **Cross-section diameter d** | **Fillet width** $M_{0}^{+0,08}$ |
|---|---|
| 1 | 1.45 |
| 1.6 | 2.13 |
| 2 | 2.7 |

The pressure limitations of O-rings can be overcome by the application of backup rings (Fig. 2.11) or other devices that prevent O-ring extrusion. Backup rings are made from leather, Teflon, plastics, or metal. The inelastic rings are split like a piston ring for assembly into the cylinder.

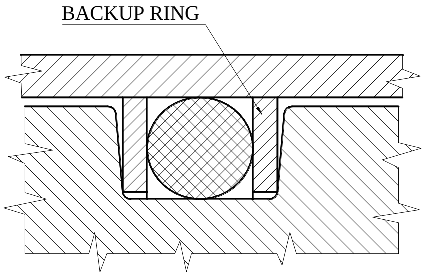



Figure 2.11 O-ring with backup rings

Tab. 2.6 contains the groove dimensions and the backup ring thickness of backup ring sealing application for different diameter groups, see Fig.2.13 [5].

Table 2.6 Groove dimensions for backup rings

| **"O" ring inner diameter** | **Cross-section diameter** | **Backup ring width** | **Groove width** $b^{+0.2}_{0}$ | |
|---|---|---|---|---|
| **D** | **d** | $m^{+0.05}_{-0.1}$ | **one backup ring** | **two backup rings** |
| 9.6 - 17.6 | 2.4 | 1.0 | 3.7 | 4.7 |
| 19.5 - 29.5 | 3.0 | 1.0 | 4.4 | 5.6 |
| 30.5 - 44.5 | | 1.5 | 4.9 | 6.4 |
| 44.3 - 79.3 | | 1.5 | 7.9 | 9.4 |
| 84.3 - 119.3 | 5.7 | 2 | 8.4 | 10.4 |
| 124.3 - 179.3 | | 2.5 | 8.9 | 11.4 |

Fig. 2.12 shows examples for backup ring installation of piston rod and piston sealing.

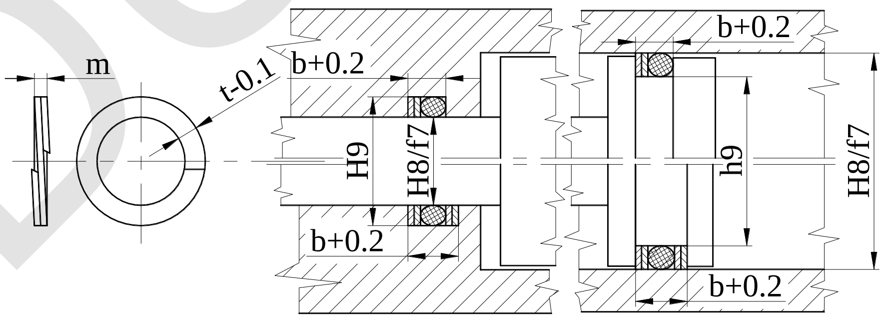



Figure 2.12 Backup ring installation

## 2.1.2. Symmetric profile elastomer „U” cup seal

Figure 2.13 shows elastomeric U-seals on a double-acting piston (Fig. 2.14). This seal may be applied as piston and piston rod seal (Fig. 2.15). They have approximately the same pressure limitations as O-rings, and for higher pressures backup rings are required. When U-seal is made of leather, filler is required between the lips to prevent collapse of the seal. The fluid pressure extends the cup outward against the cylinder wall and inwards against the piston and thus seals the piston in the cylinder. This action works from one direction hence a double-acting piston needs two cup seals to seal in both directions, see Fig. 2.14 [7].

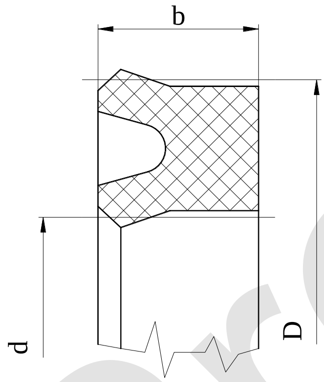
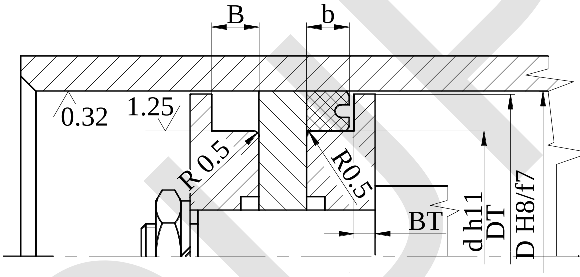



Figure 2.13 U cup seal



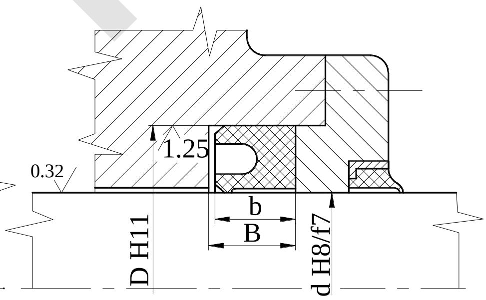




Figure 2.14 Piston sealing with U cup

Figure 2.15 Piston rod sealing with U cup

For nitrile caoutchouc material the maximal reciprocating velocity $v_{max} = 0.3$ m/s and the maximal working pressure is $p_{max} = 15$ MPa, for polyurethane material $p_{max} = 15$ MPa. The dimensions of symmetric profile elastomer „U” cup seal and the groove for it may be found in product catalogues.

## 2.1.3. Fabric-reinforced rubber “U” cup seal

It may be applied as piston and piston rod sealing. The dimensions of the U cup seal and the groove for it for piston and piston rod application may be found in product catalogues [6]. Its application as a piston and piston rod seal can be seen in Fig. 2.16.

A double acting piston requires two cup seals installed with the backs to each other. The maximal reciprocating velocity $v_{max} = 0.5$ m/sec, the maximal working pressure is $p_{max} = 25$ MPa.

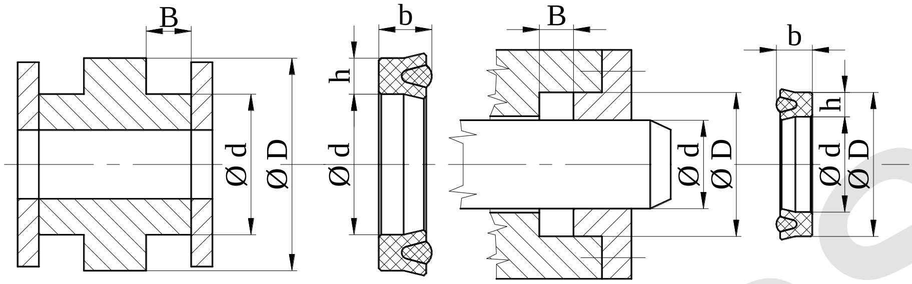



Figure 2.16 U cup installation in piston and on piston rod

## 2.1.4. Stuffing box with "V" packing

The stuffing box (Fig. 2.17) is used to seal fluids in the case of rotating shafts or reciprocating rods and pistons [7]. Seal between the packing and the rod or cylinder occurs as a result of radial expansion of the glands when they are tightened in axial direction. Friction between the packing and the moving rod causes wear, hence periodic tightening is needed. The ability to seal fluids under pressure depends on the type of packing material and number of packings used. The V-ring packing is considered superior to other lip types for sealing high pressures, specially above 345 MPa.

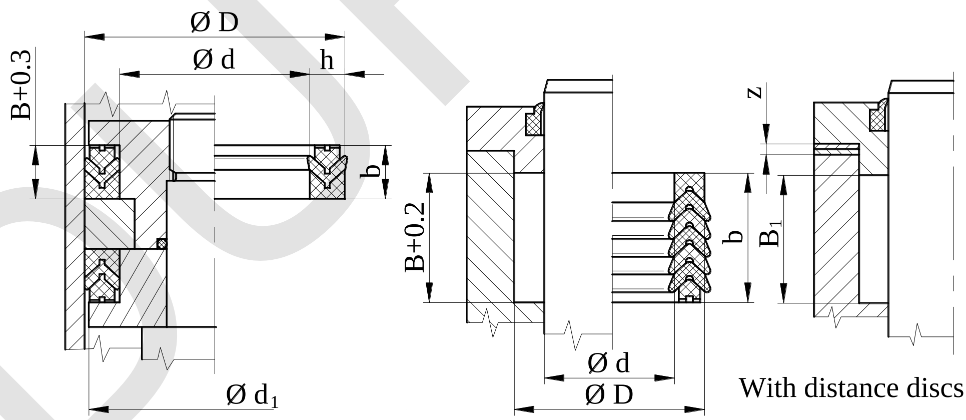



Figure 2.17 Stuffing box

When applying as a piston sealing, the packing leakage requirement is not so severe, hence the stuffing box comprises fewer packings and its length is shorter.

- for stuffing box comprising 3 packings, the maximal operating pressure is $p_{max} = 25$ MPa and reciprocating velocity is $v_{max} = 0.5$ m/sec
- the groove dimensions are compiled in catalogues.

When applying as a piston rod sealing (see Tab. 2.15) the packing leakage requirement is severe, hence more packing are used and the length of stuffing box is longer:

- the maximal working pressure and velocity for stuffing box comprising 7 packings recommended are the same like before.
- the groove dimensions are compiled in catalogues.

## 2.1.5. Compact piston seal kit

Compact piston seal kit comprises parts (see Fig. 2.18) having different function and made of different materials. Its dimensions are compiled in catalogues [7].

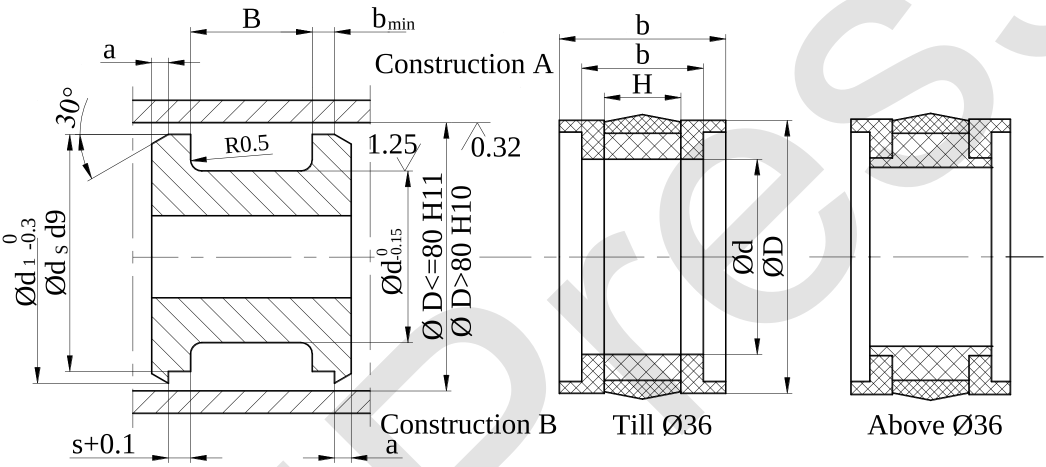



Figure 2.18 Compact piston seal kit

The piston lipseal is made of soft elastomer, the inner ring is made of Teflon and the back-up rings are piston bearings (nonmetallic piston wear band).
The piston seal kit seals in both directions. The maximal reciprocating velocity is $v_{\max} = 1$ m/sec.
The maximal working pressure depends on the working temperature: $p_{\max} = 31.5$ MPa at 60 ºC, $p_{\max} = 25$ MPa at 80 ºC, $p_{\max} = 16$ MPa at 100 ºC.

## 2.1.6. Rod wiper seals (scraper ring)

Rod wiper seals (scraper ring) are used on piston rods of hydraulic cylinders that are exposed to harsh environments. The purpose is to avoid mud, dust, and ice from the cylinder. A typical rod scraper ring is composed of a polyurethane element bonded to a metal shell which is pressed into the end cap of the cylinder. Molybdenum disulfide is sometimes added to the polyurethane to reduce friction. A rod scraper ring assembled in the end of the cylinder head is shown in Fig. 2.19 and its dimensions are compiled in catalogues. The wiper seal lip is pointed outward to remove foreign material when the piston rod is coming its retracted position.

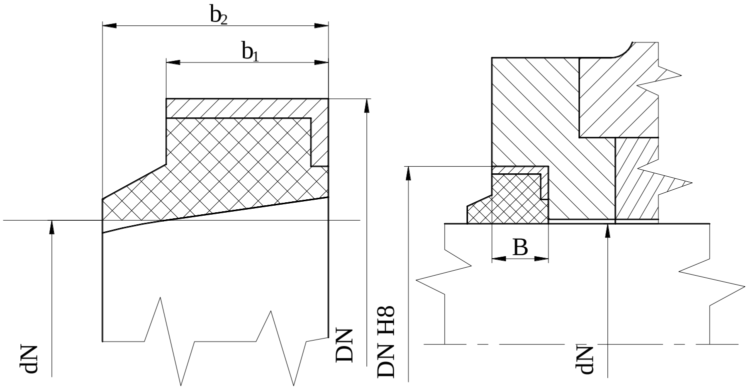



Figure 2.19 Scraper ring

## 2.2. Materials and technological aspects for designing

a. Materials of the cylinder
St 33, St 37 (DIN 17100 500) normalized, C35, C45 (DIN 17200) hardened and tempered, from cold drawn or hot rolled heavy-wall tube. The wall of light hydraulic cylinders may be made of hardened aluminium being polished.

The sliding surface of the cylinder is honed to $R_a$=0.32μm surface roughness. The same surface roughness pertains to the bearing surface of the piston having H8/f7 fit with the cylinder.

b. The section-shaped cylinder heads are made either by casting or die-forged. The common material of the cylinder head is Cast steel 40F, C45. The fit between the outside surface of the cylinder and the bore hole of the head is H8/f7.

c. The common materials of the piston rod are: C55, C60, Cm 35, 25 CrMo4 hardened and tempered. The sliding- sealing surface is hard chromium plated by electrometallization in the thickness of 0.02-0.03 mm and honed to $R_a$= 0.16-0.32 μm surface roughness.

d. The tie rods and screws are manufactured from C35, C45, Ck 22, Ck 35 or exceptionally from 34 CrMo 4 (DIN 17200) hardened and tempered if they are not commercial products.

e. In the case of series production, the outside surfaces of the hydraulic cylinder parts, not connecting to other parts are treated individually by black finishing, phosphating or anodizing in the case of aluminium.

In the case of unit production, the outer surfaces of the assembled hydraulic cylinder not connecting to other parts are paint coated (see in the welded machinery base chapter) inhibiting corrosion.

# 3. Hydraulic cylinder installation

The proper installation of the hydraulic cylinder influences the service life and its operation. For installation the overall and connecting dimensions must be given in drawing, see Fig. 3.1.

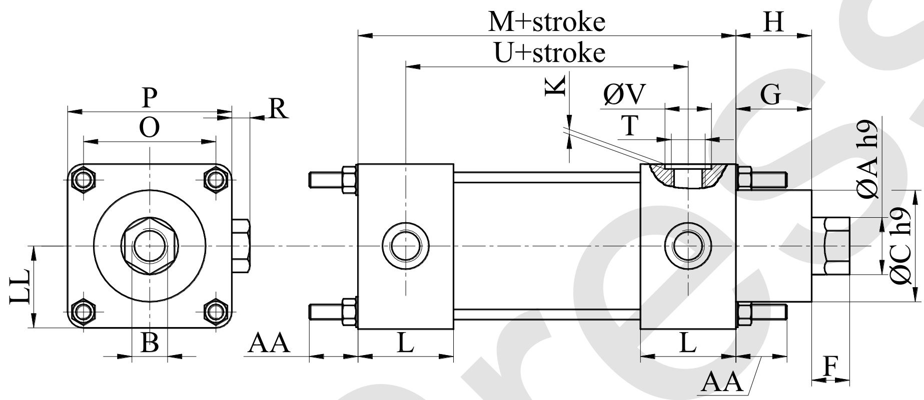



Figure 3.1 Connecting and overall dimensions
Source: [4]

## 3.1. Mounting styles

General consideration when selecting mounting style is to keep the cylinder thrust as close as possible to the centerline of the piston rod and free from misalignment or side thrust. Off-center thrust or side loads subtract substantially from the anticipated rod bearing and rod seal service life.
Mountings depend primarily upon the operating specifications of the application are generally one of the following three principles:

- Fixed centerline mountings: the thrust of the cylinder is focused on the centerline of the cylinder rod.
- Fixed non-centerline mountings: the thrust of the cylinder is aligned parallel to, but not on the centerline of the cylinder rod.
- Pivoted centerline mountings: the centerline of the cylinder may swing in one or more direction.

The above described operational features may be implemented by the following mounting types, see them in product catalogues:

- rectangular or square flange at head or at the cap
- Clevis mounting (Pin mounting)
- fixed pivot mounting

- pivot mounting with spherical bearing fitted into the pivot
- side lug, centerline lugs mountings and end lugs mountings
- side tapped mounting
- trunnion at head, at cap and at intermediate position
- extended tie rods mounting at both ends, or at cap, or at head

## 3.2. Stop Tube

In long cylinders which are pushing a load, internal stop tubes are used to prevent excessive bearing wear and jackknifing of the cylinder [4]. They are installed between the piston and the head, providing additional bearing support by increasing the distance between the piston and the head in the fully extended position (see Fig. 3.2).

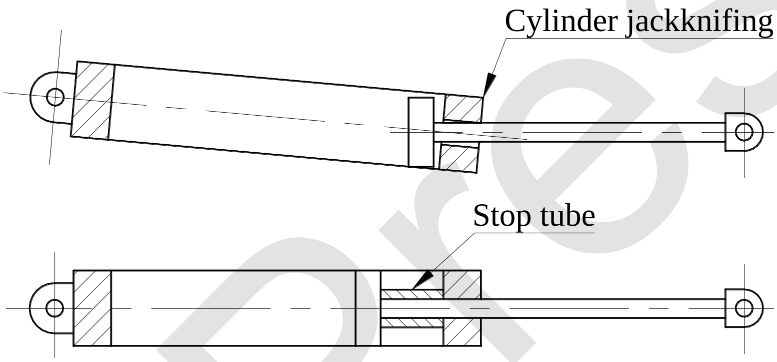



Figure 3.2 Cylinder jackknifing and stop tube application

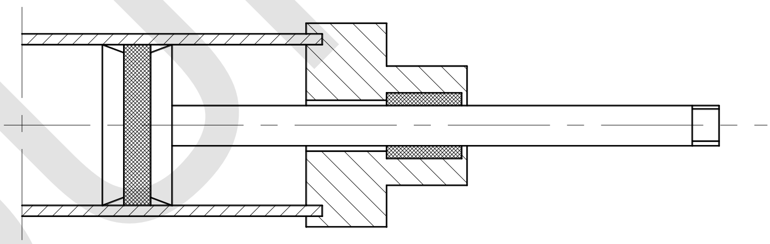


Figure 3.3 Unused stoke effect

The use of oversize rods to reduced bearing loads is not recommended because if misalignment occurs the additional rod stiffness will actually increase bearing loads.
The effect of a stop tube may be duplicated by providing additional unused stroke and stopping the cylinder extension by external means (see Fig. 3.3).
The guiding bush is made of bronze or polyamide. The fit between the guiding bush and the cylinder or the piston rod is H7/f6. The surface roughness of the contact surfaces should be: $R_a$ = 0.32-0.63 [μm]. If there is no side load, the guiding of the rod is provided by the sealing. In this case the fit between the piston and the cylinder bore is H8/e7.

## 3.3. Column strength consideration

For long push stroke cylinders, an oversize rod may be required to prevent column failure and rod bending. Total cylinder length extended is considered in column strength (see Fig. 3.4).

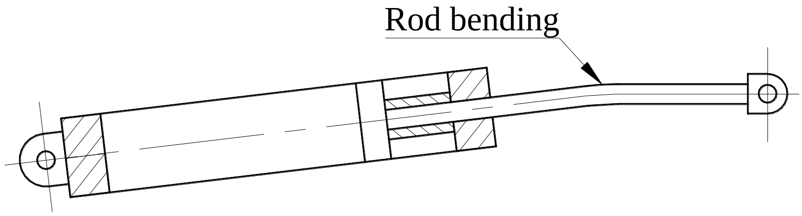



Figure 3.4 Piston rod bending

Additional off-center thrust and side loading may be caused by:

- cylinder body deflection because of flatness defect at fixing (Fig. 3.5)
- cylinder deflection because of not perpendicular backing surface to the piston rod (Fig. 3.6)
- rod deflection of long cylinders requiring additional body support (Fig. 3.7)

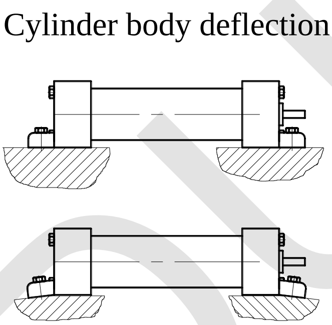



Figure 3.5 Flatness defects

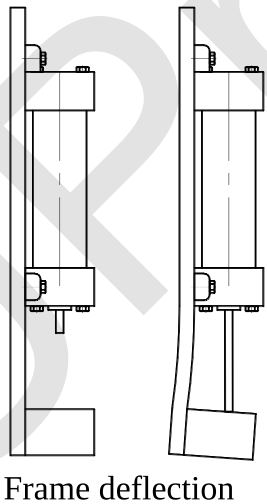



Figure 3.6 Backing defect

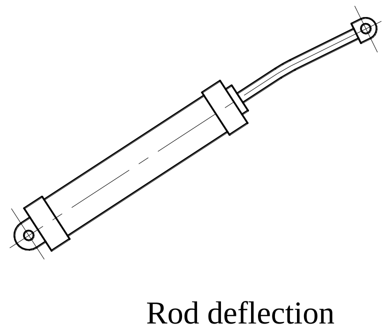



Figure 3.7 Additional body support

# 4. Stressing of hydraulic cylinders

During the preliminary design the following stressing have to be carried out:

- wall thickness of the cylinder
- flexible stability of hydraulic cylinder (buckling)
- piston rod

- screws and tie rods fixing the cylinder head and cap to each other or to the flanges
- choosing rod accessories (rod clevises, rod eyes, spherical rod eyes, eye brackets, clevis brackets, pivot pins) and their dimensions

## 4.1. Hydraulic cylinder dimensions and stressing

The hydraulic cylinder is a pressurized heavy-wall pipe, which expansion in radial direction is restrained by parts mounted on it. Since taking the effects of restraining mentioned before into consideration is very complicated, as a simplification, the model of heavy-wall pipe may be applied for calculating the stresses.
According to this model the stresses arising on the inside surface of the cylinder are as follow:

$$\sigma_1 = \sigma_t = p\frac{1+a^2}{1-a^2} \qquad \sigma_2 = \sigma_a = p\frac{a^2}{1-a^2} \qquad \sigma_3 = \sigma_r = -p$$

where: $a = \dfrac{D}{D_{out}}$

| | | |
|---|---|---|
| D | [mm] | inside diameter |
| $D_{out}$ | [mm] | outside diameter |
| p | [MPa] | operating pressure |

The equivalent stress (maximum distortion energy theory):

$$\sigma_e = \sqrt{\frac{1}{2}\left[(\sigma_1-\sigma_1)^2+(\sigma_2-\sigma_3)^2+(\sigma_3-\sigma_1)^2\right]}$$

The wall thickness is appropriate, if:

$$\frac{S_y}{\sigma_e} \geq n_{pre}$$

$n_{pre} = 2-3$ factor of safety prescribed

When preliminary stressing, after choosing the cylinder material, the minimum wall thickness $v_{\min}$ may be calculated by the following simplified form (maximum shear theory) [9]:

$$v_{\min} = \frac{D}{2}\left[\sqrt{\frac{R_{eH}}{R_{eH}-2n_{pre}p}}-1\right][mm] \tag{4.1}$$

## 4.2. Flexible stability of hydraulic cylinder (buckling)

The rod subjected to compression load may buckle. Buckling of rod is its large lateral deflection due to a small increase of compressive load. The cylinder, the piston and

the piston rod create a complicated flexible set-up, and cannot be modelled by a simple column. Therefore it is expedient to apply a simplified model based on experiences. Tests were conducted on hydraulic cylinders which specific length ($l_i$) is minimum 24 times of the diameter of their piston rod.
The specific length and the factor of safety of flexible stability depend on the fixing and rod connection of the hydraulic cylinder, see Fig. 4.1.

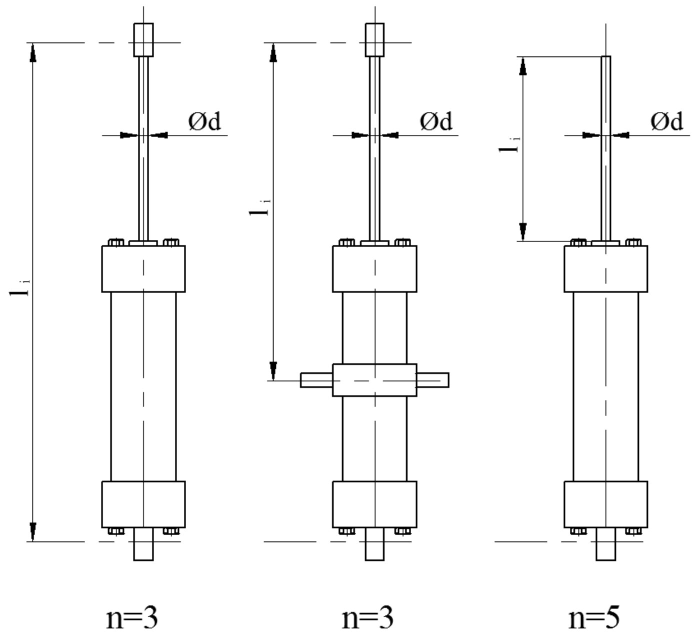



Figure 4.1 Specific length of the cylinder
Source: [9]

The collapsing force of the hydraulic cylinder having specific length $l_i$ can be calculated by the following form:

$$F_{coll} = \frac{\pi^2 EI}{l_i^2} \tag{4.2}$$

where: I [mm⁴] moment of inertia for circular cross-section area

$I = \frac{d^4\pi}{64}$ for solid circular cross-section

$I = \frac{\left(D^4 - d^4\right)\pi}{64}$ for hollow circular cross-section

The maximum piston force:

$$F_D = p\frac{D^2\pi}{4} \tag{4.3}$$

The factor of safety of flexible stability:

$$n = \frac{F_{coll}}{F_D} \geq n_i \tag{4.4}$$

The ratio of the piston diameter and the diameter of the solid piston rod may be taken for

$$\frac{D}{d} = 1.6 - 2.3 \tag{4.5}$$

The ratio of 2.3 pertains to $p = 4$ MPa, the ratio of 1.6 pertains to $p = 25$ MPa hydraulic pressure. The interstate values vary proportionally. The actual dimension of the piston rod diameter is determined by the size of the rod sealing and the rod wiper seals.

Ratios for hollow piston rods customary are the following:

$$\frac{D}{d} = 1.8.....2.5 \quad \text{and} \quad \frac{d}{d_b} = 1.3 \tag{4.6, 4.7}$$

where: $d_b$ inside diameter of the rod

## 4.3. Fixing the cylinder head and cap

Depending on the hydraulic cylinder construction the cylinder head and cap are fixed either to the flanges welded to the cylinder or to each other. The screws and tie rods are prestressed at assembly to produce an installation force, so called preload [10]. It is needed to avoid separation of the cylinder head from the cap or the flange and remaining sufficient clamping force in the parts connected when acting the operation (loosening) force due to the operation pressure.

The prestressing force is adjusted with the wrench torque.

The total torque pertaining to the preload (installation force) needed to overcome the friction torque on the thread and on the collar, and the slope torque of the thread respectively can be calculated by the following formula:

$$M_{wrench} = M_{up} + M_{collar} = F_{inst}\frac{d_2}{2}tg\left(\alpha + \rho'\right) + F_{inst}\frac{d_m}{2}\mu_{collar} \tag{4.8}$$

where: $\frac{d_2}{2}F_{inst}\alpha$ slope torque

$\frac{d_2}{2}F_{inst}tg\rho'$ thread torque

$\frac{d_c}{2}F_{inst}\mu_{collar}$ collar torque

The calculation method of $F_{inst}$ depends on the action point of the operation force (loosening force) due to the working pressure: In the simplest cases two basic models may be applied: the operation (loosening) force arising acts either:

- external loosening model: the operation force acts on the outer surfaces of the cap and the flange, see Fig. 4.2
- internal loosening model: the operation force acts on the contact surface of the cap and the flange, see Fig. 4.2

In this simplest cases when the cylinder is clamped by the head and cap with tie rods, it may be modeled by external loosening one, see Fig. 2.4 however if the head and the cap is fixed to the flange separately, may be modeled by the internal loosening one, see Fig. 2.3.

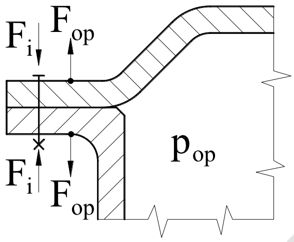



Figure 4.2 Acting point of operation force

## 4.3.1. External loosening model

Fig. 4.3 shows the free-body diagram of the tie rod connection (see the construction in Fig. 2.4) modelled by springs when applying the operation force on the outer surfaces of the cap and the flange (external loosening). The free-body diagram, represents the strain of the bolts applied (normally four pieces) and the compression strain of the cylinder.

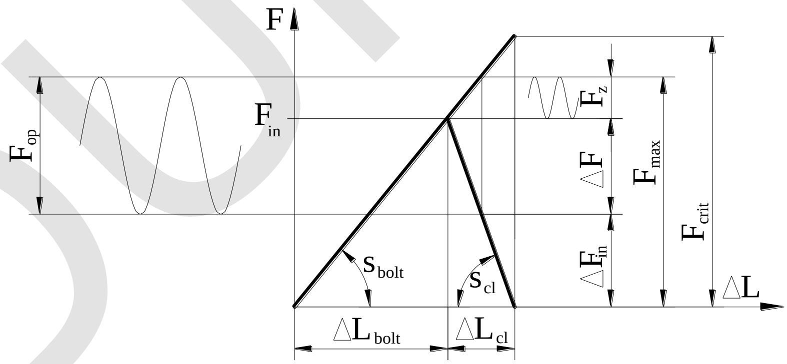



Figure 4.3 Free body diagram of the joint

**Assembly**

The tie rods (bolts) are tightened at assembly in order to realize the $F_{in}$ initial preload of the head and the cap. The screw shanks elongate due to the $F_{in}$ initial preload, and the clamped cylinder (the deformation of the head and cap are neglected) is pressed since it provides the reaction force for $F_{in}$.

$$\Delta L_{bolts} = \frac{F_{in}}{zs_{bolt}} \quad \text{and} \quad \Delta L_{cl} = \frac{F_{in}}{s_{cl}} \tag{4.9, 4.10}$$

where: $s_{bolt} = \frac{AE}{l}$ [N/mm] spring stiffness of the bolt

| | |
|---|---|
| l | length of the shank loaded |
| A | cross-section area of the bolt |
| $s_{cl} = \frac{A_{cl}E}{l}$ [N/mm] | spring stiffness of the clamped cylinder |
| l | length of the cylinder loaded |
| $A_{cl}$ | cross-section area of the cylinder |
| $z$ | number of the bolts applied |

$E = 2.1\ 10^5\, MPa$

**Operation**

When acting the operation pressure, the operation force in the threaded joint is carried by the bolt and the clamped parts. The alteration of the working pressure causes fatigue load on the bolts. The additional load on the bolts:

$$F_{z(t)} = F_{op(t)} \frac{zs_{bolt}}{zs_{bolt} + s_{cl}} \tag{4.11}$$

where: $F_{op} = p\frac{D^2\pi}{4}$

D [mm] diameter of the piston

The additional load carrying one bolt:

$$F_{z\ bolt(t)} = \frac{F_{z(t)}}{z} \tag{4.12}$$

where z: number of bolts applied

The maximum load subjects the joints:

$$F_{\max} = F_{in} + F_z \tag{4.13}$$

The pressing force decreases in the clamped parts moreover it can cease. This happen when $F_{op} = F_{crit}$

To avoid the separation of the head and the cap from the cylinder the preload $F_{in}$ has to be adjusted at assembly appropriately.

Accordingly $F_{op} n_{sep} = F_{crit}$ has to be ensured.

The $n_{sep} = 1.1 - 1.15$ is the factor of safety against separation.

The correlation between the initial preload $F_{in}$ and the operation force $F_{opt}$:

$$F_{in} = n_{sep} F_{op} \frac{s_{cl}}{z s_{bolt} + s_{cl}} \tag{4.14}$$

The installation force of a bolt at assembling:

$$F_{inst} = \frac{n_{sep} F_{op}}{z} \frac{s_{cl}}{z s_{bolt} + s_{cl}} \tag{4.15}$$

**Checking the bolted joint against fatigue**

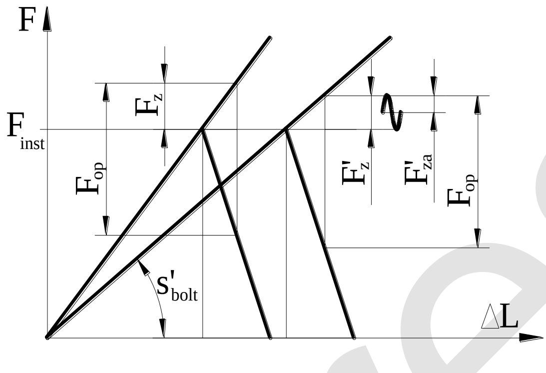



Figure 4.4 Supplementary force on joint

$F_{z_a}$ decreases if $s_{bolt}$ is decreasing.

(slenderness of the bolt shank is appropriate)

It may be seen from Fig. 4.4 that $F_{op}$ effects less $F_z$ additional force on the bolt shank when applying smaller spring stiffness bolt shank. If $F_{op}$ is fluctuating then $F_z$ fluctuates too. Necessarily, if $F_{op}$ is altering with the time, $F_z$ is altering as well. The smaller the $F_{z_a}$ the less susceptible for fatigue. This is why eg. the cylinder-head screws in the internal combustion machines are slender, the length of it is long compared to its diameter.

**Stressing bolted joint**

The bolt size may be chosen proportional to the typical dimensions of the hydraulic cylinder eg. diameter of the cylinder, piston rod, ect. The material of the bolt is chosen appropriately for satisfying the stressing requirements with respect to the safety.

I. At assembly:

The preliminary safety factor against plastic deformation is $n_{pre} = 2.5 - 3$. At assembly the bolt shank is subjected to static load: tension and torsion (no operation force acts).

$M_{wrench}$ => $F_{inst}$ => $\sigma = \frac{F_{inst}}{A_{min}}$ tensile stress

$$M_{thread} + M_{slope} =>$$

$$\tau = \frac{\left(M_{thread} + M_{slope}\right)}{K_p} = \frac{F_{inst}\dfrac{d_2}{2}tg\left(\alpha + \rho'\right)}{\dfrac{d_2^{\;3}\pi}{16}} \quad (4.16)$$

$$\sigma_{eq} = \sqrt{\sigma^2 + 3\tau^2} \quad (4.17)$$

$$n = \frac{S_y}{\sigma_{eq}} \le n_{pres}$$ K: section modulus, $K_p$: polar section modulus

II. In operation:

$$F_{\max} = F_{inst} + F_{z\ bolt} => \sigma_{\max} = \frac{F_{inst}}{A_{\min}} + \frac{F_{z\ bolt}}{A_{\min}} \quad (4.18)$$

a. Static loading

$$n = \frac{S_y}{\sigma_{\max}} \le n_{pres}$$ (checking against plastic deformation)

b. Under dynamic load

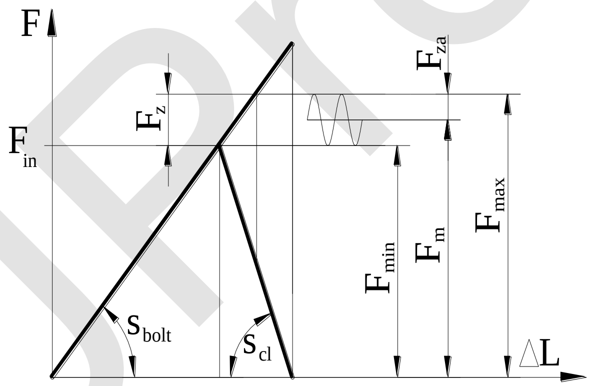



Figure 4.5 Fatigue load on the joint

Goodman criteria against fatigue [11]:

$$\frac{\sigma_{ba}}{S_e} + \frac{\sigma_{bm}}{S_u} = \frac{1}{n} \quad (4.19)$$

where: n = 1.2……2.2

$$\sigma_{ba} = \frac{F_{z\ bolt\ a}}{A_{\min}} \quad (4.20)$$

$$\sigma_{bm} = \frac{F_{inst} + F_{z\ bolt\ a}}{A_{\min}} \quad (4.21)$$

$$S_e = C_f C_r C_s C_t \left( \frac{1}{K_f} \right) S_e' \qquad (4.22)$$

$S_e' = 0.45 S_u$ for axial loading

$C_s = 1$ for axial load

$C_f = 1$

$C_r$ reliability factor

$C_t$ temperature factor

$$S_e = C_r C_t \left( \frac{1}{K_f} \right) 0.45 S_u$$

$K_f$ for steel threaded members:

| Metric class | Rolled thread | Cut thread | Fillet |
|---|---|---|---|
| 3.6 - 5.8 | 2.2 | 2,8 | 2.1 |
| 6.6 - 10.9 | 3.0 | 3.8 | 2.3 |

## 4.3.2. Internal loosening model

If the cylinder head and the cap are fixed to the flange by screwed joint (see the construction in Fig. 2.3), the internal force may be assumed that it acts on the contact surface (internal loosening). The free-body diagram of this situation is shown in Fig. 4.3.

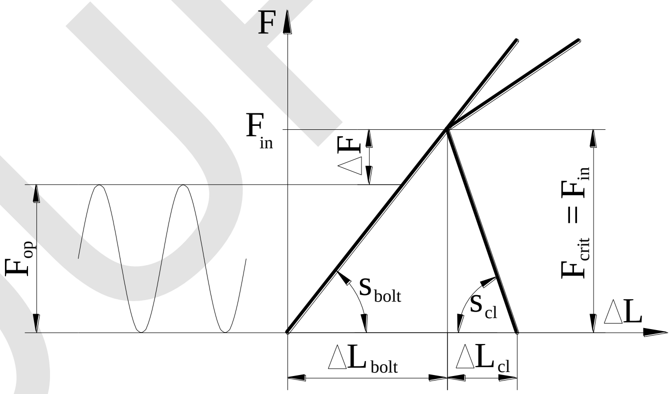



Figure 4.6 Free body diagram of the external loosening model

The stiffness of the clamped part may be derived in the case of internal loosening model from the deformation of the head or cap. The mechanical model of it is complicated, this is why we apply one of the empirical ones.

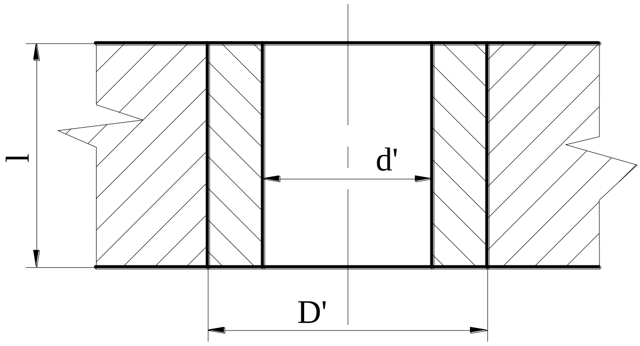



Figure 4.7 Clamped part of cap

The stiffness of the deformed and the whole clamped part:

$$s_{bush} = \frac{A_k E}{l} \qquad s_{cl} = z s_{bush} \tag{4.23, 4.24}$$

where: $A_k = \frac{\left(D^{12} - d^{12}\pi\right)}{4}$

$D' \simeq 1.2d' + 0.14l$

$d'$ hole diameter

l grip

E modulus of elasticity

The head and cap are considered as z pieces of bushes connected in parallel. The screw shanks are not subjected to additional force until the operation force reaches the initial preload $F_{in.}$ When these forces are equal, the cap and the head separate: $F_{op} = F_{crit}$, accordingly the screw shank is not subjected to fatigue load. The required installation force:

$$F_{inst} = \frac{n_{sep} F_{op}}{z} \tag{4.25}$$

**Stressing screw joints**

The screws are subjected only to static load hence no additional load acts until separation. The checking procedure is the same as one of the external loosening case at assembly.

## 4.4. Checking the piston rod against fatigue

The piston rod in the cross section at which the rod eye is connecting to it is subjected to tensile and compression load (results in reversed stress pattern) under working condition (see chapter 1) that can cause fatigue. However the rod load in the cross section at the piston rod shoulder where the piston is supported in axial direction is fluctuating tensile force. Namely the compression force is transmitted by the shoulder but the cross section is prestressed by the nut fixing the piston on the rod. The calculation of the tension force induced by the nut is detailed in chapter 4.4.1.

After calculating the mean and amplitude stresses in the particular cross sections the checking can be carried out.

The factor of safety against fatigue in the case of uniaxial stress:

$$n_f = \frac{S_u}{\sigma_m + \frac{S_u}{S_e}\sigma_a} \tag{4.26}$$

The factor of safety $n_f$ should be between 1.8 and 2. Stress concentration must be taken into account for calculating the modified endurance limit $S_e$.

$$S_e = C_f C_r C_s C_t \left(\frac{1}{K_f}\right) S_e' \tag{4.27}$$

where: $C_f$ surface finish factor

$C_r$ reliability factor ($C_r = 1$ for 50% survival rate)

$C_z$ size factor

$C_t$ temperature factor

($C_t = 1$ for room temperature)

$k_f$ fatigue stress concentration factor

The values of these factors see in engineering directives.

If there are more items in terms of fatigue stress concentration in the particular cross section, the effects of them are adding up. Commonly they are considered by increasing the larger factor by 50%. If the particular part is moreover threaded, it can be increased by 70%.

## 4.4.1. Initial force and tightening torque of fasteners

The fasteners are classified according to the property class that defines its strength and material (according to MSZ 229/2). The material properties are compiled in Tab. 4.1.

Tensile force in the threaded joint adjusted by the prescribed tightening torque has to correspond to the operation requirements of the joint and must be high enough to avoid loosening. After calculating the required tensile force the bolt size depends on the material properties of the chosen bolt material. The tightening torque eventually has to be prescribed for assembling considering the 90% of yield strength of threaded fasteners.

Table 4.1 Material properties of fasteners

| **Property class** | **Tensile stress** $R_m$ [MPa] | **Yield stress** $R_{eH}$ [Mpa] |
|---|---|---|
| 3.6 | 300 | 180 |
| 4.6 | 400-500 | 240 |
| 4.8 | | 320 |
| 5.6 | 500-700 | 300 |
| 5.8 | | 400 |
| 6.8 | 600 | 480 |
| 8.8 | 800 | 640 |
| 10.9 | 1000 | 900 |

Without derivation, Tab. 4.2 contains the correlation between the $F_{in}$ installation force (for 90% utilization of $\sigma_{02}$ or $\sigma_s$) the thread dimension, the property class of the threaded fastener, the $\mu_G$ friction coefficient between the thread profiles [12]. It may be realized that with increasing $\mu_G$ the allowed installation force $F_{in}$ is degreasing since at tightening the screw shank is subjected to increasing torque as well and the normal stress component of the allowed resultant stress and by this means the resulting installation force decrease.

Coefficients of friction $\mu_K$ and $\mu_G$ for different surfaces and lubrication conditions varying in a range are compiled in Tab. 4.3 [12]. Since $\mu_K$ and $\mu_G$ influence the correlation between the tightening torque and the installation force, a K coefficient may be defined and its values can be taken from Tab. 4.4 [12]. The K values apply to standard threads of sizes M 1.4 to M 42. From M16 to M42 threads the K values are to be reduced by 5%.

Table 4.2 Tension forces for standard threads

| $F_{in}$ ($10^3$ N) for coefficients of friction $\mu_G$ in the threads | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Dimension | Property class | 0.06 | 0.08 | 0.10 | 0.12 | 0.14 | 0.16 | 0.18 | 0.20 | 0.24 | 028 |
| M 6 | 4.8 | 5.2 | 5.1 | 4.9 | 4.7 | 4.5 | 4.3 | 4.1 | 4.0 | 3.7 | 3.4 |
| | 5.8 | 6.6 | 6.3 | 6.1 | 5.9 | 5.6 | 5.4 | 5.2 | 5.0 | 4.6 | 4.2 |
| | 8.8 | 10.5 | 10.1 | 9.8 | 9.4 | 9.0 | 8.6 | 8.3 | 7.9 | 7.3 | 6.7 |
| | 10.9 | 14.7 | 14.2 | 13.7 | 13.2 | 12.7 | 12.1 | 11.7 | 11.2 | 10.3 | 9.5 |
| | 12.9 | 17.7 | 17.1 | 16.5 | 15.8 | 15.2 | 14.6 | 14.0 | 13.4 | 12.3 | 11.4 |
| M 8 | 4.8 | 9.6 | 9.3 | 8.9 | 8.6 | 8.3 | 7.9 | 7.6 | 7.3 | 6.7 | 6.2 |
| | 5.8 | 12.0 | 11.6 | 11.2 | 10.8 | 10.3 | 9.9 | 9.5 | 9.1 | 8.4 | 7.7 |
| | 8.8 | 19.2 | 18.6 | 17.9 | 17.2 | 16.5 | 15.7 | 15.2 | 14.6 | 13.4 | 12.4 |
| | 10.9 | 27.0 | 26.1 | 25.2 | 24.2 | 23.2 | 22.3 | 21.4 | 20.5 | 18.9 | 17.4 |
| | 12.9 | 32.4 | 31.3 | 30.2 | 29.0 | 27.9 | 26.8 | 25.7 | 24.6 | 22.6 | 20.9 |
| M 10 | 4.8 | 15.3 | 14.8 | 14.2 | 13.7 | 13.2 | 12.6 | 12.1 | 11.6 | 10.7 | 9.9 |
| | 5.8 | 19.0 | 18.5 | 17.8 | 17.1 | 16.5 | 15.9 | 15.1 | 14.5 | 13.4 | 12.3 |
| | 8.8 | 30.5 | 29.5 | 28.5 | 27.4 | 26.3 | 25.3 | 24.2 | 23.2 | 21.4 | 19.7 |
| | 10.9 | 42.9 | 41.5 | 40.1 | 38.5 | 37.0 | 35.5 | 34.1 | 32.7 | 30.1 | 27.7 |
| | 12.9 | 51.5 | 49.8 | 48.1 | 46.2 | 44.4 | 42.6 | 40.1 | 39.2 | 36.1 | 33.2 |
| M 12 | 4.8 | 22.2 | 21.5 | 20.8 | 20.0 | 19.2 | 18.4 | 17.7 | 16.9 | 15.6 | 14.4 |
| | 5.8 | 27.8 | 26.9 | 25.9 | 25.0 | 24.0 | 23.0 | 22.1 | 21.2 | 19.5 | 18.0 |
| | 8.8 | 44.5 | 43.0 | 41.5 | 40.0 | 38.4 | 36.8 | 35.3 | 33.9 | 31.2 | 28.7 |
| | 10.9 | 62.5 | 60.5 | 58.4 | 56.2 | 54.0 | 51.8 | 49.7 | 47.7 | 43.8 | 40.4 |
| | 12.9 | 75.0 | 72.6 | 70.0 | 67.4 | 64.8 | 62.2 | 59.6 | 57.2 | 52.6 | 48.5 |
| M 14 | 4.8 | 30.5 | 29.6 | 28.5 | 27.4 | 26.4 | 25.3 | 24.3 | 23.3 | 21.4 | 19.7 |
| | 5.8 | 38.1 | 36.9 | 35.6 | 34.3 | 32.9 | 31.6 | 30.3 | 29.1 | 26.8 | 24.7 |
| | 8.8 | 61.0 | 59.1 | 57.0 | 54.9 | 52.7 | 50.6 | 48.5 | 46.5 | 42.8 | 39.5 |
| | 10.9 | 85.8 | 83.1 | 80.1 | 77.1 | 74.1 | 71.2 | 68.3 | 65.5 | 60.2 | 55.5 |
| | 12.9 | 103 | 99.7 | 96.2 | 92.6 | 89.0 | 85.4 | 81.9 | 78.5 | 72.3 | 66.6 |
| M 16 | 4.8 | 41.8 | 40.5 | 39.2 | 37.7 | 36.3 | 34.8 | 33.4 | 32.1 | 29.5 | 27.2 |
| | 5.8 | 52.3 | 50.7 | 48.9 | 47.2 | 45.4 | 43.6 | 41.8 | 40.1 | 36.9 | 34.0 |
| | 8.8 | 83.6 | 81.1 | 78.3 | 75.5 | 72.3 | 69.7 | 66.9 | 64.2 | 59.0 | 54.4 |
| | 10.9 | 118 | 114 | 110 | 106 | 102 | 98.0 | 94.1 | 90.2 | 83.0 | 76.5 |
| | 12.9 | 141 | 137 | 132 | 127 | 122 | 118 | 113 | 108 | 99.6 | 91.9 |
| M 20 | 4.8 | 65.3 | 63.3 | 61.2 | 59.0 | 56.7 | 54.5 | 52.2 | 50.1 | 46.1 | 42.5 |
| | 5.8 | 81.7 | 79.2 | 76.5 | 73.7 | 70.9 | 68.1 | 65.3 | 62.7 | 57.7 | 53.2 |
| | 8.8 | 131 | 127 | 122 | 118 | 113 | 109 | 105 | 100 | 92.3 | 85.1 |
| | 10.9 | 184 | 178 | 172 | 166 | 159 | 153 | 147 | 141 | 130 | 120 |
| | 12.9 | 220 | 214 | 206 | 199 | 191 | 184 | 176 | 169 | 156 | 144 |

Table 4.3 Friction coefficients of threads and bearing surfaces

| **Surface on mating party:** Bearing face on part ($\mu_K$) or nut thread ($\mu_G$) | | Lubrication condition | **Fastener surface** Bearing face of fastener head or bottom surface of nut ($\mu_K$) or fastener thread ($\mu_G$) | | |
|---|---|---|---|---|---|
| | | | Steel, oil-quenched or zinc phosshated | | Steel zinc-elect-roplated 6μm |
| | | | pressed rolled | turned cut | |
| Stell, | rolled | Lightly oiled | 0.13 - 0.19 | 0.10 - 0.18 | 0.10 - 0.18 |
| | planed, milled, turned, cut | | 0.10 - 0.18 | - | 0.10 - 0.18 |
| | ground | | 0.16 – 0.22 | 0.10 – 0.18 | 0.10 – 0.18 |
| Gray cast iron | planed, milled turned, cut | | 0.10.. 0.18 | - | 0.10 – 0.18 |
| All-black malleable iron | ground | | 0.16 - 0.22 | 0.10 - 0.18 | 0.10 - 0.18 |
| Steel, | cadmium-electroplated, 6μm | | 0.08 – 0.16 | 0.08 – 0.16 | - |
| | zinc-electroplated, 6μm | | 0.10 – 0.18 | 0.10 – 0.16 | 0.16 – 0.20 |
| | zinc-electroplated female threads | | - | - | 0.10 - 0.18 |
| | ground, rolled, phosphated | | 0.12 – 0.20 | | - |
| | machined, phosphated | | 0.10 - .0.18 | - | - |
| Al-Mg alloys, processed, cut | | | 0.08 - 0.20 | - | - |
| Steel, | cadmium-electroplated, 6μm | Dry | 0.08 - 0.16 | - | - |
| | cadmium-electroplated, 6μm female thread | | 0.08 – 0.14 | - | - |
| | zinc-electroplated, 6μm | | 0.10 - 0.18 | - | 0.20 - 0.30 |
| | zinc-electroplated, female thread | | 0.08 - 0.16 | - | 0.12 - 0.20 |

Table 4.4 K coefficient

| | | Coefficient of friction for fastener head and nut bearing face $\mu_K$ | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | 0.04 | 0.06 | 0.08 | 0.10 | 0.12 | 0.14 | 0.16 | 0.18 | 0-20 | 0.24 | 0.26 |
| Coefficient of friction for threads $\mu_G$ | 0.008 | 0.094 | 0.108 | 0.120 | 0.134 | 0.148 | 0.162 | 0.176 | 0.190 | 0.204 | 0.232 | 0.260 |
| | 0.10 | 0.104 | 0.118 | 0.132 | 0.146 | 0.158 | 0.172 | 0.186 | 0.200 | 0.214 | 0.242 | 0.270 |
| | 0.12 | 0.114 | 0.128 | 0.142 | 0.156 | 0.170 | 0.184 | 0. 196 | 0.210 | 0.224 | 0.252 | 0.280 |
| | 0.14 | 0.124 | 0.138 | 0.152 | 0.166 | 0.180 | 0.194 | 0.208 | 0.222 | 0.234 | 0.262 | 0.290 |
| | 0.16 | 0.134 | 0.148 | 0.162 | 0.176 | 0.190 | 0.204 | 0.218 | 0.232 | 0.246 | 0.272 | 0.300 |
| | 0.18 | 0.146 | 0.160 | 0.172 | 0.186 | 0.200 | 0.214 | 0.228 | 0.242 | 0.256 | 2.284 | 0.312 |
| | 0.20 | 0.156 | 0.170 | 0.184 | 0.198 | 0.210 | 0.224 | 0.238 | 0.252 | 0.266 | 0.294 | 0.322 |
| | 0.24 | 0.176 | 0.190 | 0.204 | 0.218 | 0.232 | 0.246 | 0.260 | 0.274 | 0.286 | 0.314 | 0.342 |
| | 0.28 | 0.198 | 0.212 | 0.224 | 0.238 | 0.252 | 0.266 | 0.280 | 0.294 | 0.308 | 0.336 | 0.362 |

For getting the maximum installation force $F_{in\ max}$ the maximum possible installed torque $M_{in\ max}$ must be prescribed. It is calculated using the K-value for the lowest values for $\mu_K$ and $\mu_G$ (since bigger part of the installed torque is devoted to higher tension force).

$$M_{in\ max} = K_{min} F_{in\ max} d \tag{4.28}$$

The lowest installed tightening torque $M_{in\ min}$ is a result of the quality of the tightening process and the equipment used. Performing the tightening process by hand using torque wrench, the actual torque can usually be achieved within a variation $\pm 10\%$ of the nominal one. The lowest installed tightening torque is then:

$$M_{in\ min} \approx 0.8 M_{in\ max} \tag{4.29}$$

The lowest installed clamping force $F_{in\ min}$ which then may occur is calculated using the K-value for the maximum values of the two coefficients of friction $\mu_K$ and $\mu_G$ as follows:

$$F_{in\ min} = \frac{M_{in\ min}}{K_{max} d} \tag{4.30}$$

**Example:**
An M 10-8.8 bolt or screw (pressed and rolled, phosphated) is used to fasten a ground steel part to an AlMg housing with blind threaded holes; the parts are lightly oiled [12].

From Tab 4.3

$\mu_K$ = 0.16 - 0.22

$\mu_G$ = 0.08 - 0.22 → in accordance with Tab 4.2 for $\mu_G$ = 0.08

$F_{In\ max}$= 29,500 N

From Tab 4.4

$K_{min}$ = 0.176 and $K_{max}$ = 0.280

$M_{in\ max} = K_{min} F_{in\ max} d$ = 0.176X x 9,500 x 0.010 = 51.9 Nm

$M_{in\ min} \approx 0.8 M_{in\ max}$ = 41.5 Nm

$F_{in\ min} = \frac{M_{in\ min}}{K_{max} d}$ = 41.5/(0.280 x 0.010) = 14,800 N

# 5. Design and stressing steps of the hydraulic cylinder

- Calculation of the wall thickness (Eq. 4.1).
- Choosing the pipe dimension on the basis of the inner diameter and the wall thickness (see product catalogue).
- Determining the piston rod (solid or hollow) diameter (Eq. 4.6, Eq. 4.7).
- Checking the flexible stability of the hydraulic cylinder depending upon the fixing and end conditions (Eq. 4.4).
- Selecting the seal for the piston, piston rod (chapter 2.1.2, chapter 2.1.3, chapter 2.1.4 and chapter 2.1.5).
- Selecting the seal between the cylinder and the head and the cap (chapter 2.1.1).
- Selecting the scraper for the piston rod (chapter 2.1.6).
- Checking the connecting dimensions of the seals and designing the piston and the seal housing, recess in the head and cap (see product catalogue).
- Selecting the port and pipe connections (see product catalogue).
- Selecting the spherical rod eye (see product catalogue).
- Constructing the hydraulic cylinder in the prescribed mounting style.
- Stressing the threaded joint fixing the head and the cap. Use the external loosening model if tie rods are applied and the internal loosening model when screws are applied (chapter 4.3).
- Check the piston rod for fatigue in two cross sections: where the piston is supported with a shoulder and at the connection with the rod eye (chapter 4.4).
- Draw the shop drawing of the cylinder, piston, piston rod head and the cap.

# CHAPTER C. External shoe drum brake

Design a thruster released linkage type drum brake for hoisting-gear according to the given task number and the given construction represented in Fig. 1.3 [13].

Determine by calculation:

- the necessary brake drum diameter standardized and the dimensions of shoes on the basis of allowed bearing stress and the heat generation,
- the operation range of the linkage mechanism,
- the free body diagram of the linkage mechanism and the type of the brake thruster,
- the parameters of the brake spring and stress it,
- the necessary pin dimensions.

Design and draw:

- the assembly drawing of the brake (designed as a welded construction) on A1 paper sheet. Apply the appropriate scale to represent the details.
- shop drawing of the spring.

The assembly drawing shall contains views and sectional views necessary to give overall, connecting and tolerated dimensions, to show the welding joints and its dimensions.
Give prescriptions relating to the welding joints and corrosion protection. Complete the title block with the parts list and give the necessary data regarding the material, dimension and standard number of them.
Although all the drawings are made by pensile pay attention to applying the proper line thicknesses.

| Number | Brake type | $M_{brake}$ [Nm] | n [1/s] |
|---|---|---|---|
| 1. | a | 100 | 25 |
| 2. | b | 200 | 16.7 |
| 3. | c | 250 | 12.5 |
| 4. | d | 350 | |
| 5. | e | 400 | |
| 6. | f | 500 | |
| 7. | g | 600 | |
| 8. | h | 700 | |
| 9. | i | 800 | |

## Design, stressing and construction considerations

When designing a drum brake, the following details have to be clarified:

- task and operation requirements,
- analyzing the linkage mechanism determining its moving range and ratios,
- actuating possibilities and solutions,
- bearing force of the parts and stressing procedures,
- mechanical brake converts kinetic energy into heat energy by friction work,
- heat equilibrium determining the operation whether it is continuous or intermittent,
- construction in terms of producibility, machinability, cost-effectiveness,

# 1. Task and operation requirements of brakes

Thruster brake is a device used to retard the speed of moving machinery and to stop it accurately to the desired position. The braking force is applied to the brake shoes by a preloaded compression spring. The shoes are pressed on the rotating brake drum retarding its speed and finally stop it. The releasing of the brake drum by compressing the spring is done by thruster.

## 1.1. Assembly and operation of the brake

Fig. 1.1 shows a spring applied, thruster released linkage type drum brake [14]. The standard version is equipped with manual lining wear compensation and braking torque is externally adjustable. The release mechanism is operated by means of electrohydraulic thruster. The function and the operation of the brake can be followed in the Fig. 1.1. A thruster shoe brake has a pair of cast iron shoes (1) which are lined up with friction pads (2). The shoes are hinged on main arm (3) and side arm (4) of the brake, each of them have a hinge pin (5) fitted in the base. They are connected to each other on top by a tie rod (6), which is hinged in the crank lever (7) and to the swivel block (8) in the side arm, locked by lock nuts (9). A crank lever is hinged on the main arm and the other end is fixed to the top clevis (11) of the thruster by a hinge pin (12). A brake spring is assembled in bracket hinged on the base and is pre-loaded by locknuts (13, 14). The pre-tension in this spring provides the braking torque. The thruster is fitted on the base by a hinge pin (15). When the thruster is not energized, the brake shoes are pressed on the brake drum (16) fitted on the drive motor shaft and hold it under the effect of braking force provided by the spring. In such condition, the brake is applied and the drum cannot rotate. As the top clevis travels upwards the angle lever turns, pushes the brake arms from the drum and compresses the brake spring stored its energy for the next cycle. Simultaneously one of the brake arms is retracted. When the first arm (either the main or the side) reaches the stop-screw on the brake base member the other brake arm is retracted, releasing the brake drum.

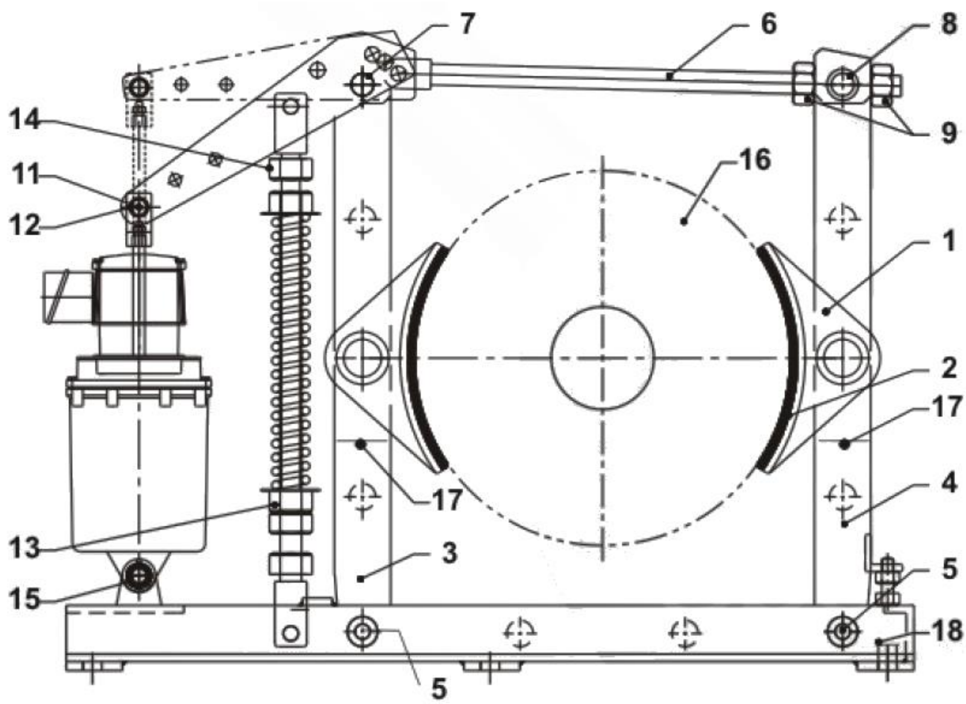



Figure 1.1 Build-up of the brake
Source: [14]

### 1.1.1. Thruster with integrated springs

The basic components of a thruster, i.e. electric motor and closed hydraulic system are coaxially assembled to form a functional unit (see Fig. 1.2). The working fluid of the hydraulic system serves as the operating medium for the generation of thrust [15]. In the switched-off state (de-energized), the hydraulic piston with the piston rod is at its lower limit (when it is disconnected) or at its some of low position determined by the arm linkage mechanism. In the switched-on state, the centrifugal pump delivers the working fluid under the piston and produces there hydraulic pressure, i.e. the thrust of the thruster. As a result of this pressure, the piston travels along its path against the internally fitted braking spring. Thus, the piston can either travel the total stroke distance or the externally reduced stroke lengths. When the thruster is in the disconnected state the piston returns to its original position under the impact of the internal spring force. Except for the starting and running down phases of the motor, the lifting and lowering speeds are linear. The response times obtained depend on the magnitude of the load as well as on the viscosity of the working fluid injected affected by the ambient temperature.

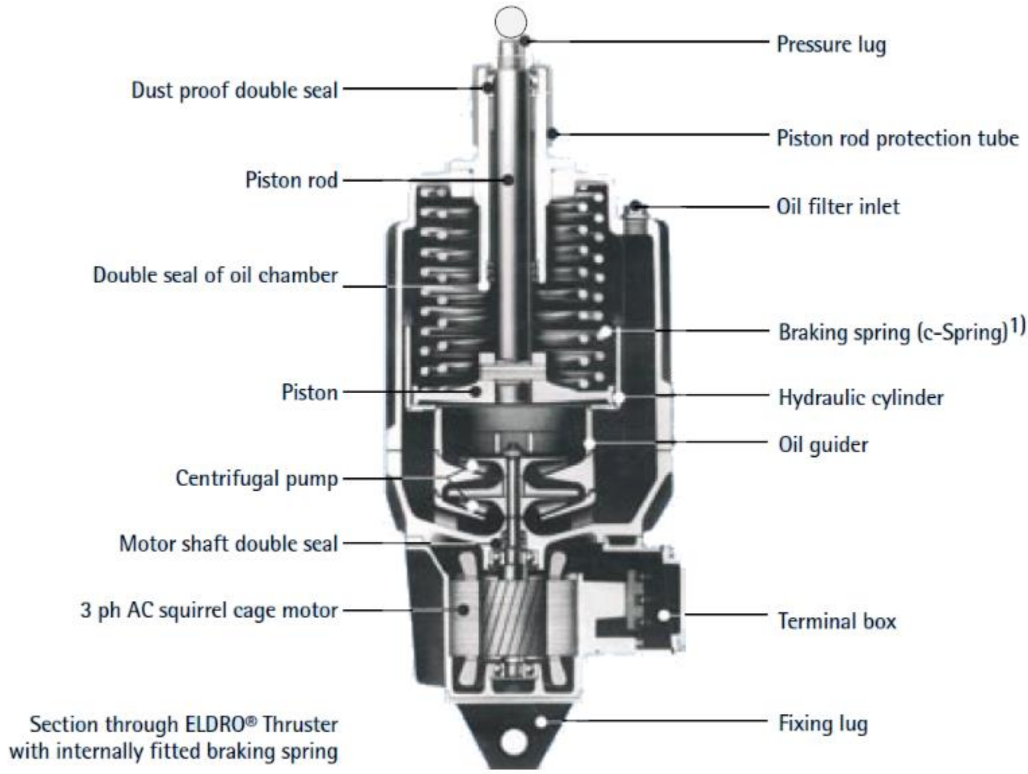



Figure 1.2 Components of thruster
Source: [16]

### 1.1.2. Thruster without internal springs

The thruster doesn't contain integrated spring to force the piston into the bottom position. Accordingly in the switched off state (de-energized, the piston's position is determined by the arm linkage mechanism due to the external load (braking spring or weight). When energized, the thruster has to overcome the load on the eye-lug exerted by the external spring or weight. This operation is the same as the one with integrated spring.

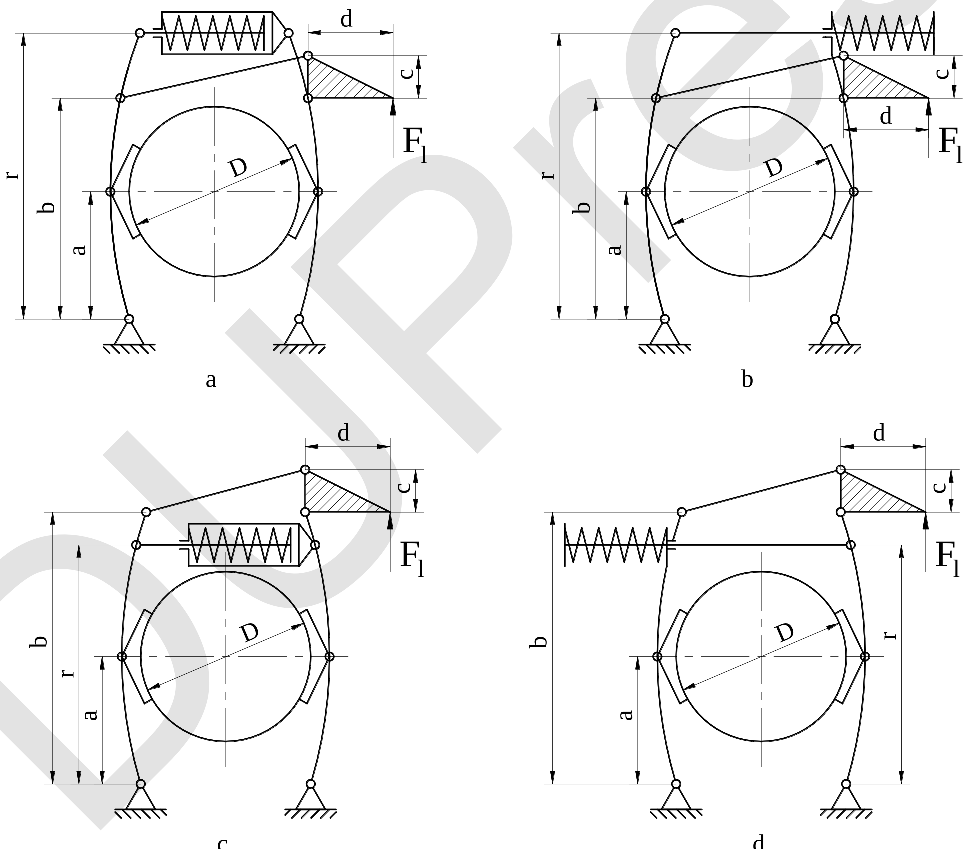
## 1.2. External double-shoe thruster released drum brake

Fig. 1.3 shows some sketches of the external shoe drum brakes [13]. The common features of them are that the frictional force is provided by a spring and the release is provided by electrohydraulic brake thruster. As it can be seen in Fig. 1.3, constructions from "a" to "h" contain brake springs. The task of brake thruster is only to release the brake (spring is not involved in the device). The brake thruster contains the integrated spring only in brake construction "I".
In crane hoist, brakes are applied in order to control the speed of the raised load (stop, retard or maintain speed. Accordingly the brake may be retarding brake (in

general until standing), catch brake or regulating brake. The technical requirements of these brakes are detailed in MSZ 19171/1-36.
In terms of safety engineering, the brake can only be a negative operation method in hoisting gear. In normal position (closed condition) the electro-hydraulic thruster is de-energized, the brake is applied by applying the brake shoes to the drum. The braking force is provided commonly by spring force or seldom by weight-load. When operating the hoisting gear the brake is released by an energized brake thruster.
The main parts of this brake (see Fig. 1.1):

- brake drum with the brake shoes or brake disk with the brake pads between which the frictional force is arising
- brake spring or weight-load
- brake thruster releasing the brake
- arm linkage mechanism

**Sketches of the external double-shoe drum brake**



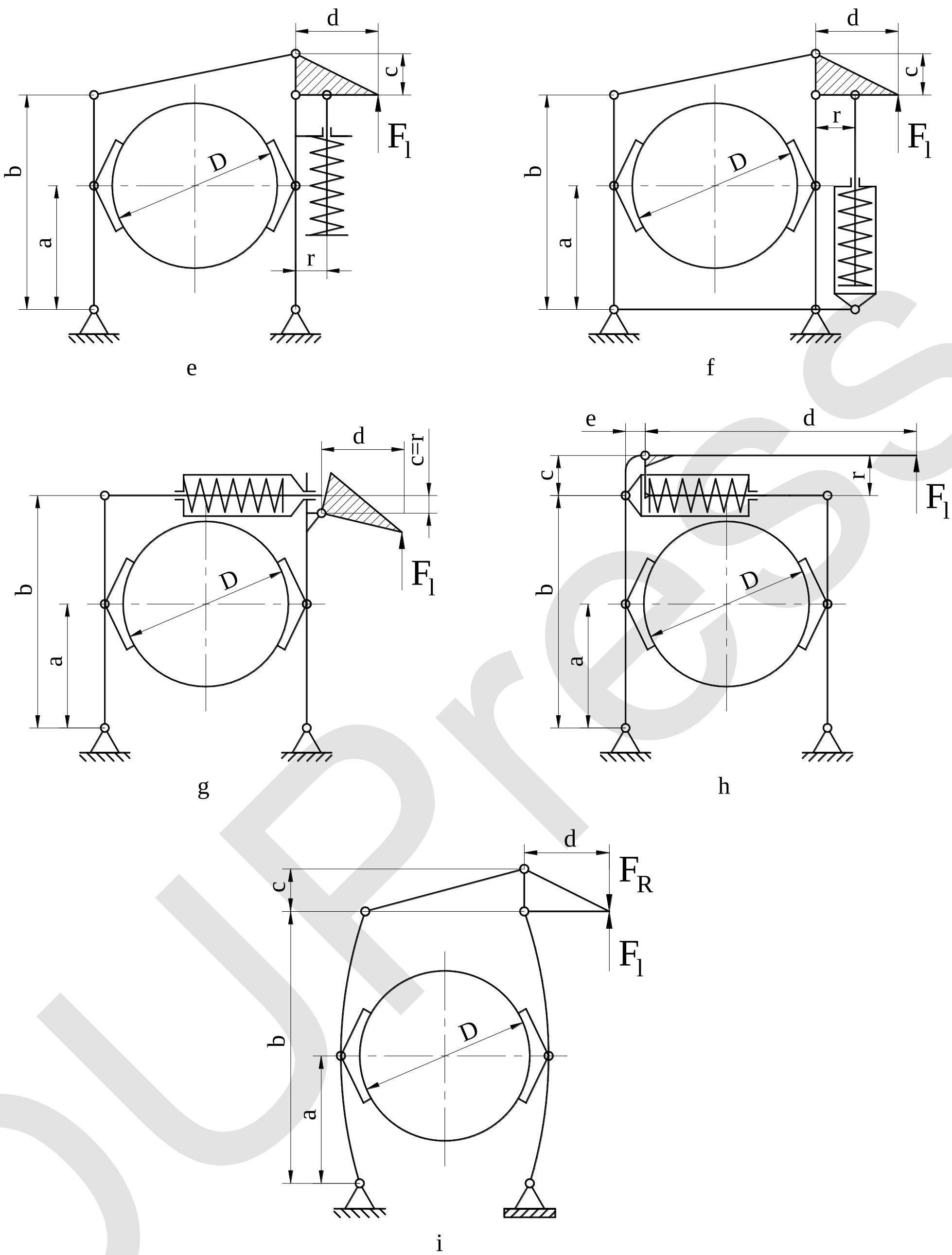



Figure 1.3 Brake constructions

## 1.3. Energy absorption and dissipation

The task of a mechanical brake in terms of energy equilibrium is to transform the kinetic energy of a moving system into heat energy. The generating heat energy heats up the brake and the connecting parts which can limit the operation of the brake. This is why the generated heat must be dissipated into the environment in order to control the warming-up of the parts. The quality of heat dissipation depends on the size, shape and condition of surface of the parts and the flow of the surrounding air. With increasing temperature the coefficient of friction of the lining material decreases. This phenomenon is the fading when the effectiveness of the brake may be deteriorated. The satisfactory brake performance requires that the heat generation should not exceed the heat dissipation capability.

Energy sources: $E_k = \frac{1}{2} m v^2$ kinetic energy of translation (1.1)

$E_k = \frac{1}{2} J \omega^2$ kinetic energy of rotation (1.2)

$E_p = mgh$ potential energy (1.3)

where: E [J] energy
m [kg] mass
J [kgm²] mass moment of inertia about its axis of rotation
V [m/s] velocity
$\omega$ [rad/s] angular velocity
h [m] height

## 1.3.1. Energy equilibrium of braking

When braking, the kinetic energy of the system comprising rotating and translating parts is transformed into heat energy. The time rate of heat energy developed is $P_{br}$ braking power. Based on the energy equilibrium the brake can be either intermittent or continuous operation.

Heat dissipation capability:

$$P_{diss} = CA\Delta t_{allow} \tag{1.4}$$

where: $P_{diss}$ [W] heat dissipation
$\Delta t$ [C°] allowed temperature rise (depends on the lining material)
A [m²] surface area of parts
C $[\frac{W}{m^2 C^o}]$ heat transfer coefficient
C=7.5 for still air
C=8.5 for average air circulation

On the basis of the energy equilibrium two situations can be specified, the continuous and the intermittent operations.

## 1.3.1.1. Continuous operation braking

$$P_{diss} \;\; \text{capability} > P_{br}$$

The brake parts dissipate the heat energy generating into the surroundings, hence the brake can operate continuously assuming an operating v speed. In practice either the maximum density of heat flux is specified (see chapter 2.1.2)

$$q = \mu p v \qquad [\frac{W}{m^2}] \tag{1.5}$$

or the maximum value of

$$(pv)_{max} \qquad [\frac{N}{m^2}\frac{m}{s} = \frac{W}{m^2}] \tag{1.6}$$

The mechanical properties of common friction materials are compiled in Tab. 2.1.
The maximum value of q depends on the cooling conditions:

( $\mu$ = 0.25-0.50, $p_{allow}$ = 0.5-2 MPa, $t_{allow}$ = 200-260 C°):

| **Operation** | **heat dissipation** | $q_{max}$ |
|---|---|---|
| continuous | poor | 1.05 |
| continuous | good | 3.00 |

## 1.3.1.2. Intermittent operation braking

$$P_{diss.} \text{capability} < P_{br}$$

The brake parts cannot dissipate the heat energy generating into the surroundings, this is why the brake cannot operate continuously. When braking, the heat energy is stored in the parts. After releasing the brake, parts need sufficient time for dissipating the heat energy for cooling down.

Energy absorption results in temperature rise:

$$\Delta t = \frac{H_{abs}}{cm} \tag{1.7}$$

where: $H_{abs}$ [J] absorbed heat energy

c $[\frac{J}{kgC^o}]$ specific heat

$c \approx 500 \frac{J}{kgC^o}$ for steel

m [kg] mass of the parts absorbing the energy

Energy equilibrium of an intermittent operation brake:

$$W_{br} + W_r + W_m = \frac{1}{2}m\left(v_1^2 - v_2^2\right) + \sum \frac{1}{2}J\left(\omega_1^2 - \omega_2^2\right) + mg\left(h_1 - h_2\right) \tag{1.8}$$

| | | | |
|---|---|---|---|
| where: | $W_{br}$ | [J] | work done by the brake |
| | $W_r$ | [J] | work done by the resistance (rolling friction, bearing friction, air resistance) |
| | $W_m$ | [J] | work done by the drive motor |

As a simplification, in intermittent operation the $W_{br}$ may be regarded as the $H_{gen}$ stored in the brake parts ($H_{gen}$ is the generated heat during the braking process).

# 2. External double-shoe drum brake design

The braking moment of the double shoe external drum brake required can be provided by different drum diameters and the appropriate friction forces as peripheral force.

$$M_{br} = 2\mu F_n \frac{D}{2} \text{ [Nm]} \tag{2.1}$$

where: $\mu$ coefficient of friction on the interface of the drum and the shoe lining

$F_n$ [N] normal force between the drum and the shoe

D [m] diameter of the drum

## 2.1. Drum diameter and the shoe dimensions

The diameter of the drum can be determined from the necessary bearing area of the shoe namely there is a correlation between the shoe and the drum dimension. The necessary bearing area of the shoe follows from the allowed bearing stress due to the $F_n$ normal force and from the required heat dissipation capability.

### 2.1.1. Checking for allowed bearing stress

The $F_n$ normal force can be calculated on the basis of the allowed bearing stress of the brake lining (see Table 2.1) and the dimensions of the shoe (see Fig. 2.1 and Table 2.2) [13].

$$F_n \le a_1 b_1 p_{allow} \text{ [N]} \tag{2.2}$$

where: $a_1$ and $b_1$ [mm] dimensions of the shoe

On the basis of the Fig. 3.1:

$$a_1 = D \sin\frac{\alpha}{2}$$
$$b_1 = B - 2c_1 \tag{2.3}$$

Where: $\alpha$ = 60° - 90° angle of contact

($\alpha$ = 70°)according to the DIN 15435)

$c_1$ dimension depends on the diameter of the drum

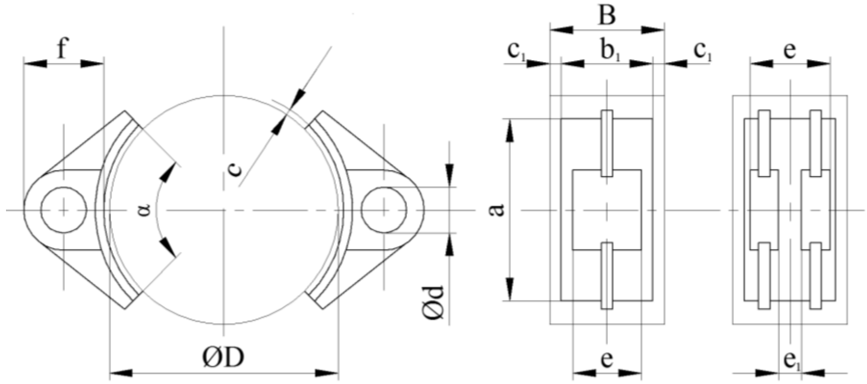



Figure 2.1 Dimensions of brake shoe

The brake drum is often constructed as a part of a flanged flexible coupling, hence its main dimensions are standardized (MSZ.11720, DIN 15431, etc.).

Table 2.1 Material properties of some brake lining

| **Lining material** | **Drum material** | **Type** | **Coefficient of friction** | **$Pa_{all}$ [MPa]** | **$q_{all}$ [w mm$^2$]** | **Tmax [°c]** | **Comment** |
|---|---|---|---|---|---|---|---|
| Woven (dry) | Cast iron or steel | BA | 0.35 – 0.41 | 0.3 | 2.5 | 125 | 'fperodo'[1] |
| | | AS 10 | 0.32 – 0.42 | 0.3 | 2.5 | 125 | |
| | | AH 11 | 0.30 – 0.46 | 0.3 | 2.5 | 125 | |
| | | MZ 41 | 0.30 – 0.44 | 0.4 | 3.0 | 150 | |
| | | UR 41 | 0.27 – 0.46 | 0,.5 | 3.0 | 150 | |
| | | Retinax B EM – 2 | 0.35 – 0.5 | 0.4 | 3.0 | 140 | G03ZT-15960 |
| Molded (dry) | | VG 95/1 | 0.20 – 0.33 | 0.35 | 3.0 | 225 | "Ferodo" |
| | | MS 21 | 0.30 – 0.40 | 0.35 | 3.0 | 175 | |
| | | Retinax B EM - 1 | 0.3 – 0.4 | 0.3 | 3.0 | 200 | G0JZT-15960 |

Table 2.2 Shoe dimensions

| **D** | **B** | **$b_1$** | **c** | **d** | **e** | **$e_1$** | **f** | $\delta_{min}$ | **z** | **$d_1$** |
|---|---|---|---|---|---|---|---|---|---|---|
| 160 | 60 | 55 | 5 | 20 | 45 | - | 30 | 0.7 | 4 | 5 |
| 200 | 70 | 65 | 5 | 20 | 50 | - | 35 | 0.8 | 4 | 5 |
| 250 | 80 | 70 | 8 | 25 | 55 | - | 40 | 1.0 | 8 | 6 |
| 300 | 90 | 80 | 8 | 30 | 60 | - | 42 | 1.0 | 8 | 6 |

| | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| 350 | 100 | 90 | 8 | 30 | 85 | 50 | 45 | 1.2 | 10 | 8 |
| 400 | 120 | 105 | 10 | 35 | 100 | 60 | 50 | 1.2 | 10 | 8 |
| 500 | 140 | 125 | 10 | 40 | 120 | 70 | 60 | 1.5 | 12 | 8 |
| 600 | 160 | 140 | 12 | 45 | 130 | 80 | 63 | 1.5 | 18 | 8 |

Comments (dimensions are in mm):

$\delta_{min}$ the minimal shoe to-drum clearance

z recommended number of the rivets fixing the lining

$d_1$ diameter of the hollow rivet according to MSZ 10813

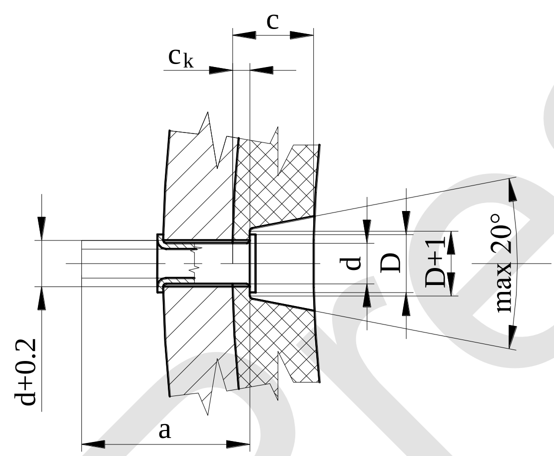



Figure 2.2 Riveted joint of brake shoe lining

## 2.1.2. Checking for heat developing

The coefficient of friction of lining material has a small variation on changes in pressure, velocity and temperature. The actual value of the coefficient of friction depends on the material of the drum and the lining, the bearing stress, the sliding speed, the temperature, the condition of the surfaces (eg. contamination).

The expected values of $\mu_{min}$ and $\mu_{max}$ for the most widely applied lining materials are compiled in Tab. 2.1. It has to be emphasized that these values are valid up to the given $T_{max}$ temperature limit. If the temperature of the lining exceed long the temperature limit, the lining material damages. Hence brakes are designed and operated with respect to the heat developing as well.

**Energy equilibrium of the external double shoe drum brake**

The heat energy generated during braking $H_{gen}$ can be derived from the energy equilibrium of the braked system. If the braking takes an actual braking time $t_{br}$, the generated heat energy can be calculated as follow:

$$H_{gen} = \int_0^{t_{br}} P_{br(t)} dt \ [J] \tag{2.4}$$

$$P_{br} = M_{br}\omega_{(t)} \qquad [W] \tag{2.5}$$

where: $P_{br(t)}$ [W] braking power

$M_{br(t)}$ [Nm] actual braking moment

$\omega_{(t)}$ [rad/s] actual angular velocity of the drum

The generated heat energy during the braking:

$$H_{gen} = \int_0^{t_{br}} M_{br(t)}\omega_{(t)} dt \qquad [J] \tag{2.6}$$

The heat flux, or the generated heat energy in unit time that is equal to the braking power:

$$\phi = \frac{dH_{gen}}{dt} \qquad [W] \tag{2.7}$$

$$\phi = M_{br(t)}\omega_{(t)} \qquad [W] \tag{2.8}$$

The density of heat flux is the heat flux on unit area:

$$q = \frac{\phi}{A_{bearing}} \le q_{allowed} \qquad [Wmm^{-2}] \tag{2.9}$$

When calculating the maximum density of heat flux, as a simplification the braking moment and the angular velocity are assumed constant with the time (as a continuous operation brake).

$$q_{max} = \frac{\phi}{A_{bearing}} = \frac{M_{br}\omega_0}{A_{bearing}} = \frac{\mu F_n D\omega_0}{2a_1b_1} = \mu_{max} p v_0 \ [Wmm^{-2}] \tag{2.10}$$

where: $v_o$ [$ms^{-1}$] peripheral velocity of the drum

p [MPa] bearing stress of the lining

From the Eq. 2.9 there fallows the $A_{bearing}$, the dimension of the shoe and eventually the diameter of the drum through an iterative calculation. Consequently the brake is good with respect to the warming if the actual density of flux is less than allowed one for the given lining material (see Tab. 2.1).

From the Eq. 2.10 it can be seen that the maximum density of heat flux is proportional to the product of v and p. Consequently from Eq. 2.10 there follows:

$$q_{max} = \frac{M_{br}\omega}{A_{bearing}} = \frac{M_{br} 2\pi n}{A_{bearing}} \Rightarrow M_{br} n = A_{bearing} q_{allow} \frac{1}{2\pi} = k_{br}$$

$$M_{br} n = k_{br} \text{ [W]} \tag{2.11}$$

where: $k_{br}$ [W] brake warming constant

$$k_{br} = \frac{q_{allow}}{\pi} a_1 b_1 \text{ [W]} \tag{2.12}$$

The Eq. 2.11 can be depicted by a hyperbola (in log scale it is depicted by a straight line) in the service diagram of a continuous operating brake (see Fig. 2.3) [13].

Since the maximum normal force is limited by the allowed bearing stress (see Eq. 2.2), the maximum brake moment is limited too, hence the curve is cut by $M_{br\ max}$. In this range the peripheral velocity of the drum is low, therefore heat development is negligible.

$$M_{br.max} = \mu_{min} p_{allow} a_1 b_1 D \text{ [Nm]} \tag{2.13}$$

In the rotating drum peripheral stress arises. Its allowed value arises at the $v_{max}$ peripheral velocity of 40 $ms^{-1}$.

$$n_{max} = \frac{v_{max}}{D\pi} \text{ [s}^{-1}\text{]} \tag{2.14}$$

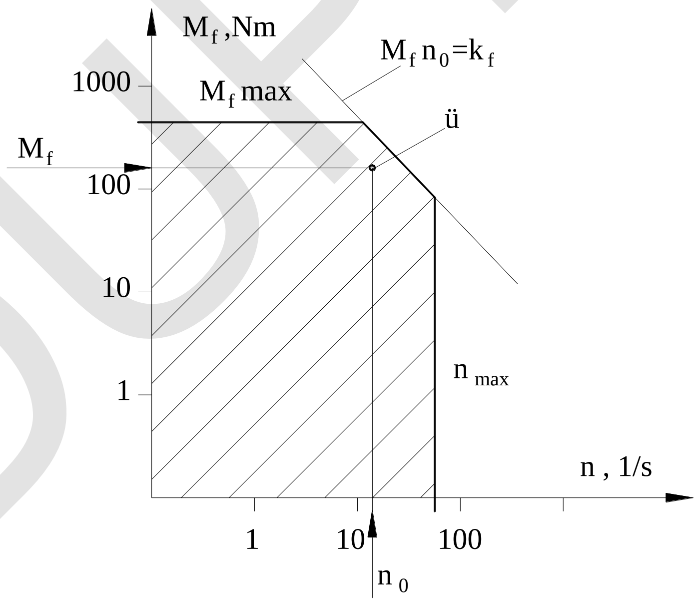



Figure 2.3 Service diagram of the brake

If the operating point (ü) of the brake determined by the brake moment and the drum rpm is in the hatched area, the brake appropriate for allowed bearing stress, warming and rpm. Since the brake moment is determined by the minimum coefficient of friction and the density of flux is determined by the maximum coefficient of friction, the possible alteration of the coefficient of friction must be considered when calculating the brake warming constant.

$$k_{br} = \frac{q_{allow}}{\pi} \frac{\mu_{min}}{\mu_{max}} a_1 b_1 \text{ [W]} \tag{2.15}$$

When designing a drum brake we have to endeavor to implement the smallest drum diameter and the smallest angle of contact of the shoe. The application of oversized brake should be avoided since it has unfavorable dynamic effects.
The basic data when designing a brake are the $M_{br\,max}$ and the rpm. For calculating the preliminary diameter of the drum the following empirical formula may be applied (see Eq. 2.16) after choosing the material of the lining and the angle of contact of the shoe for 90° [13].

$$D_{pre} = 17 \sqrt[3]{\frac{M_{br\,max}}{\mu_{min} p_{allow}}} \text{ [mm]} \tag{2.16}$$

Eq. 2.16 is an empirical formula. To get the $D_{pre}$ in [mm], the $M_{br\,max}$ in [Nm] and the $p_{allow}$ in [MPa] are substituted respectively. The preliminary diameter of the drum can be rounded to the nearest standardized diameter compiled in Tab. 2.2. After that the minimum value of the angle of contact can be calculated on the basis of Eq. 2.1, Eq. 2.2 and Eq. 2.3. The calculated value should be rounded to the nearest ten degree. After checking the actual bearing stress whether it is less than the allowed one, the brake warming constant may be calculated and the brake service diagram may be drawn (see Eq. 2.15). The brake can be checked for bearing stress and for warming after depicting the operating point in the diagram. The brake drum is commonly made of cast iron or cast aluminium, but in the case of unit production it may be a welded one. The lining is fixed to the shoe by bounding, riveting or rarely by bolted joint. The advantage of fixing by bounding is that the contact area of the shoe is not reduced by the rivet head. The disadvantage of it is that the lining can be removed only by machining.
The widespread fixing method is the riveting. The applied rivets: countersunk and hollow rivet, see in Fig. 2.2. The advantage of riveting is the easy change of the lining, its disadvantage is the reduced effective surface area that must be taken into consideration when designing.

## 2.2. Brake actuation

The brake in a winch crane is a negative operation method. Namely the brake is applied in normal position. The braking force is provided commonly by spring force or seldom by weight-load. When operating the winch crane, the brake is released by a brake thruster.

The brake shoes hinged on the brake arms are pressed to the drum in order to produce the $F_n$ normal force necessary for the required friction force. $F_n$ normal force can be calculated from Eq. 2.1 that is provided by the brake spring pre-loaded. The connection between the shoe and the brake spring is implemented by linkage. The proportion of this mechanism is $i_{s,ac}$ (see Fig. 2.4).

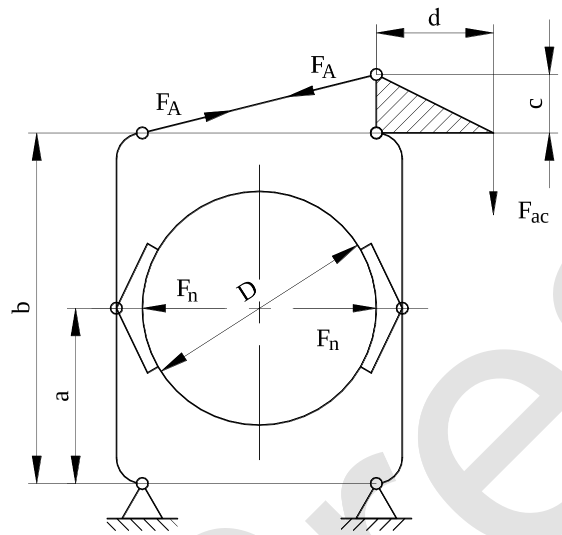



Figure 2.4 Linkage mechanism of the brake

$$bF_{ax} = aF_n \quad \Rightarrow \quad F_{ax} = \frac{a}{b}F_n$$

$$cF_{ax} = dF_{ac} \quad \Rightarrow \quad F_{ax} = \frac{d}{c}F_{ac}$$

$$F_{ac} = \frac{a}{b}\frac{c}{d}F_n\frac{1}{\eta} = \frac{F_n}{i_{s,ac}\eta}$$

$$\text{where: } i_{s,ac} = \frac{b}{a}\frac{d}{c} \tag{2.17}$$

$$F_n = i_{s,ac}F_{ac}\eta \tag{2.18}$$

$$M_{br} = \mu F_n D$$

$$F_{ac} = \frac{M_{br}}{\mu D i_{s,ac}\eta} \tag{2.19}$$

where: $F_{ac}$ [N] activating force provided by eg. brake spring
$F_n$ [N] normal force of the shoe
$i_{s,ac}$ proportion of lever arms between the shoe and the spring
$\eta$ mechanical efficiency of the linkage

$\eta \approx 0.9$ for considering the friction of pin

**Brake spring operated thruster brake**

When pre-loading the spring the necessary spring deflection is $f_{pre}$, if the spring constant is s [N/mm],

$$f_{pre} = \frac{F_{ac}}{s} \qquad [mm] \qquad (2.20)$$

This spring force must be at disposal in closed position. However, when releasing the brake, a $\delta_{min}$ shoe to drum clearance will develop (see Tab.3.2) causing an additional deflection of the spring which magnitude is:

$$\Delta f^{+} = 2\delta_{min} i_{s,ac} \xi \ [mm] \qquad (2.21)$$

where: $\xi$ = 1,1 for considering the clearance fit of pins

The maximum spring force accordingly:

$$F_{acmax} = F_{ac} + s\Delta f^{+} \ [N] \qquad (2.22)$$

During operation the brake lining wear yields a reduced spring force due to less spring deflection. The brake lining wear may be checked by the actual shoe to drum clearance in released position.

The allowed maximum value of clearance is $\delta_{max}$, from which the lining wear is:

$$\delta_{wear} = \delta_{max} - \delta_{min} \ [mm] \qquad (2.23)$$

Due to the arm linkage mechanism between the shoe and the brake spring, the grater shoe to drum clearance caused by the wearing results in a less spring deflection when the brake is applied that corresponds to a decreased spring force. This decrease in spring force causes a decrease in the braking moment.

The spring force applying to the worn lining:

$$F_{acmin} = F_{ac} - s\Delta f^{-} \qquad [N] \qquad (2.24)$$

where: $\Delta f^{-}$ [mm] reduction of the spring deflection

$$\Delta f^{-} = 2\delta_{wear} i_{s,ac} \xi \qquad (2.25)$$

The normal force of the shoe and the braking moment:

$$F_{n\ min} = F_{ac\ min} i_{s,ac} \zeta \qquad [N] \qquad (2.26)$$

$$M_{br\ min} = \mu_{min} F_{n\ min} D \qquad [Nm] \qquad (2.27)$$

If no automatic lining wear compensation device is applied, the brake has to be designed with respect to the allowed maximum braking moment decreasing due to the wearing. The decreasing can be maximum 15% of the nominal braking moment. This requirement may be complied with brake spring having the least spring constant.
The pre-loading of the brake spring cannot be adjusted when applying thruster with internal springs. For adjusting the brake moment the proportion of lever arms must be designed adjustable between the shoe and the thruster.
The parts of the brake mechanism are made of cast iron in the case of series production. In the case of unit production the parts are welded from hot rolled or cold finished steel profiles and cold finished steel plate or the combination of them. When designing the brake it is expedient to match the brake mechanism to the shaft height of the electric motor or gear box. The height dimensions of the brake are resulted from the drum diameter and the proportion of lever arms required. The transversal dimensions are considered with respect to the stiffness requirements.

The connecting rod between the crank lever and the side arm is threaded and this way its operating length is adjustable. The connecting rod is subjected to pulsating tension and has to be checked against fatigue. The thruster released linkage type drum brake mechanism is an instable construction of which position is determined by the drum when the brake is applied and by the stop-screw either in the side arm or in the main arm in released position. When releasing the brake the instable mechanism starts to overbalance and the gap develops between the drum and one of the shoes. By applying the stop-screw in the appropriate arm and the nuts on the connecting rod respectively, the equal gap between the drum and the shoes can be adjusted. The other stop-screw must be removed otherwise the pressure lug of the thruster could not travel to the upper limit. The setting can be done with feeler gauge. The recommended gap between the lining and the drum is compiled in Tab. 2.2. As it was mentioned, the clearance between the pins and bushes influences the actual displacement of arms. To reduce this effect small clearance should be applied.

Table 2.3 Allowed bearing stress of arms

| **Material** | | **$p_{allow}$ [MPa]** | **Comment** |
|---|---|---|---|
| **Arm or bushing** | **Pin** | | |
| Cast iron | Steel | 5.0 | run dry |
| Cast iron | Steel | 7.0 | lubricated |
| Steel | Steel | 15.0 | run dry |
| Steel | Steel | 25.0 | lubricated |
| Bronze ()bzö 12 | Steel | 15.0 | lubricated |
| Danamid | Steel | 10.0 | run dry |
| Teflon | Steel | 7.0 | run dry |

The recommended fit between the pin and the bush is H9/e8 or H9/f8. By applying appropriate lubrication and pin dimensions the bearing stress may be low and consequently the pin wearing may be small. The pins are checked for bearing stress. The allowed bearing stresses are compiled in Tab. 2.3.

# 3. Choosing the thruster

When releasing the brake, the energized thruster overcomes the spring force and the shoes move clear off the drum. The basic principal of choosing is its nominal thrust $F_{thr\ n}$ along the nominal stroke $L_{n.}$ If the proportion of lever arms between the spring and the thruster is $i_{ac,thr}$, the maximum load exerted by the thruster arises in released condition of the brake.

$$F_{thr\ max\ calc} = \frac{F_{ac\ max}}{i_{ac,thr}}\ [N] \tag{3.1}$$

The thruster is chosen appropriately if the following stipulation is fulfilled:

$$0{,}85F_{thrn} \le F_{thr\ max\ calc} \le 0{,}95F_{thrn} \tag{3.2}$$

If the proportion of lever arms between the shoe and the thruster is $i_{s,thr}$, the travel of the pressure lug of the thruster and the thrust can be determined (see Fig. 3.1) respectively.

$$i_{s,thr} = i_{s,ac} i_{ac,thr} \tag{3.3}$$

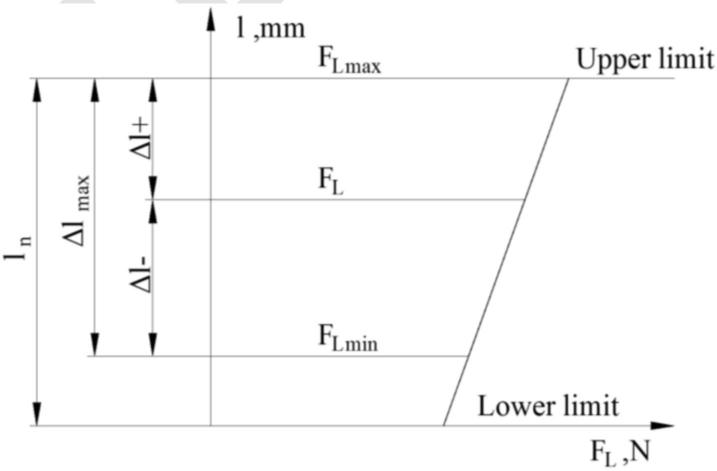



Figure 3.1 Operation range of the thruster

When the brake is applied, the pressure lug is travelling from the upper limit downwards and takes $\Delta l$ distance until applying the brake shoes to the drum.

$$\Delta l = 2\delta i_{s,thr} \xi\ [mm] \tag{3.4}$$

Consequently, the actual stroke of the thruster depends on the actual shoe to drum clearance. It means that $\Delta l^{+}$ pertains to $\delta_{min}$, $\Delta l^{-}$ pertains to $\delta_{wear}$ and $\Delta l_{max}$ pertains to $\delta_{max}$ and the actual strokes may be calculated by Eq. 3.4. The pressure lug must not get close to the lower limit because in this position the arm linkage mechanism would be backed and the shoes could not apply to the drum and as a consequence the braking moment cannot develop. To avoid this situation the following requirement must be complied:

$$\Delta l_{max} \leq 0.8 l_n \tag{3.5}$$

Considering the Eq. 3.4 the maximum allowed clearance is as follows:

$$\delta_{max} = \frac{0.8 l_n}{2 i_{s,thr} \xi} 1 \tag{3.6}$$

# 4. Lever arms proportions

The main feature of the thruster released linkage type drum brake is the proportions of lever arms which are defined between the shoes and the spring ($i_{s,ac}$), the shoes and the thruster ($i_{s,thr}$) and the spring and the thruster ($i_{ac,thr}$).
When ignoring the pin friction these proportions are the following:

$$i_{s,ac} = \frac{F_n}{F_{ac}}, \quad i_{s,thr} = \frac{F_n}{F_{thr}}, \quad i_{ac,thr} = \frac{F_{ac}}{F_{thr}} \tag{4.1}$$

The formulas of proportions of lever arms for different brake types are compiled in Tab. 4.1 (see Fig. 1.4).

Table 4.1 Proportion of lever arms

| **Figure sign** | $i_{s,ac}$ | $i_{s,thr}$ | $i_{ac,thr}$ |
|---|---|---|---|
| a-d | $\frac{r}{a}$ | $\frac{b}{a}\frac{d}{c}$ | $\frac{b}{r}\frac{d}{c}$ |
| e-h | $\frac{b}{a}\frac{r}{c}$ | $\frac{b}{a}\frac{d}{c}$ | $\frac{d}{r}$ |
| i | $\frac{b}{a}\frac{d}{c}$ | $\frac{b}{a}\frac{d}{c}$ | 1 |

The proportion of lever arms depends on the brake constructions. However tried constructions have proportion between the shoe and the spring:

$$i_{s,ac} = 2 - 4 \tag{4.2}$$

and $i_{s,thr}$ proportion between the shoe and the thruster should provide $\delta_{max} = (2 - 2.5)\delta_{min}$ shoe to drum clearance. The measure of $i_{s,thr}$ depends on the stroke magnitude but in general it results:

$$i_{s,thr} = 12 - 16 \tag{4.3}$$

Of course any other proportions may be applied if necessary in order to adapt the brake to custom-build conditions.

# 5. Brake spring design

The brake spring has to be designed according to the standard of MSZ 4333-81 or MSZ 5737-80. When designing the brake spring, the difficulty is caused by the fact that the overall dimension of the spring must match to other brake parts like as brake arm (see Fig. 1.4) and depending upon the brake type, sometimes the spring must be inserted between the arms. To comply with the requirement that the maximum braking moment decreasing because of the wearing can be maximum 15%, the proportions of lever arms and sometimes the drum diameter have to be modified. The brake is designed appropriately if the maximum allowed brake clearance (see Eq. 3.6) with respect to the Eq. 3.5 can be utilized, accordingly the brake clearance will result in less than 15% braking moment decreasing.

From Eq. 2.24 and 2.25 the necessary spring constant may be expressed:

$$0.85F_{ac} = F_{ac} - s\Delta f^{-} \quad \Rightarrow \quad s = \frac{0.15F_{ac}}{\Delta f^{-}} \tag{5.1}$$

The spring constant of a helical compression spring can be calculated according to the Eq. 5.2.

$$s = \frac{Gd^4}{8D^3\left(N_t - N_{gr}\right)} \tag{5.2}$$

where: $s$ [N/mm] spring constant

$G$ [N/mm$^2$] modulus of rigidity

$G$ = 78000-82000 N/mm$^2$ for spring steel

$d$ [mm] spring wire diameter

$D$ [mm] spring coil diameter

$N_t$ number of total coils

$N_{gr}$ number of grounded coils

$C = \frac{D}{d}$ spring index: ratio of the mean coil diameter to wire diameter

$C$ = 6 -12 in majority of springs

As it can be seen from Eq. 5.2, any spring constant may be implemented by choosing its parameters appropriately but the spring must match to the brake construction.

The maximum shear stress arising in the spring wire:

$$\tau_{max} = K_s \tau_{nominal} = K_s \frac{8DF_{acmax}}{d^3 \pi} \tag{5.3}$$

where: $K_s$ stress concentration factor, accounts for the effect of spring wire curvature

$$K_s \cong 1 + \frac{0.615}{C}$$

The allowable shear stress for preliminary calculation can be calculated according to the Eq. 5.4 or can be found in Fig. 5.1 for cold-worked wire (d<10mm) and in Fig. 5.2. for hot-worked wire
(d > 10mm).

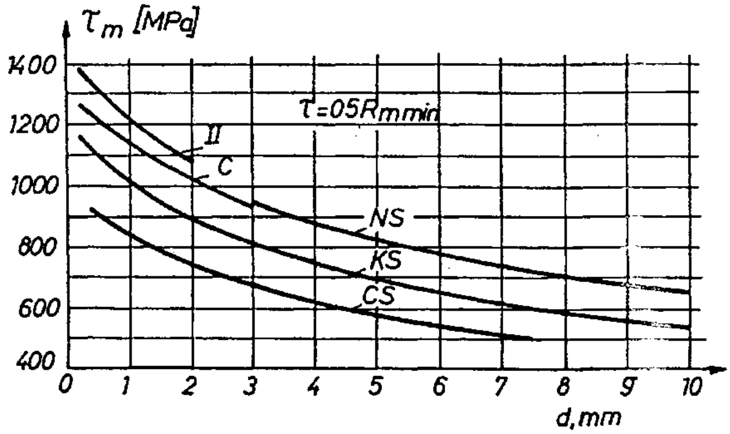



Figure 5.1 Allowed stress of cold-worked wire
Source: [13]

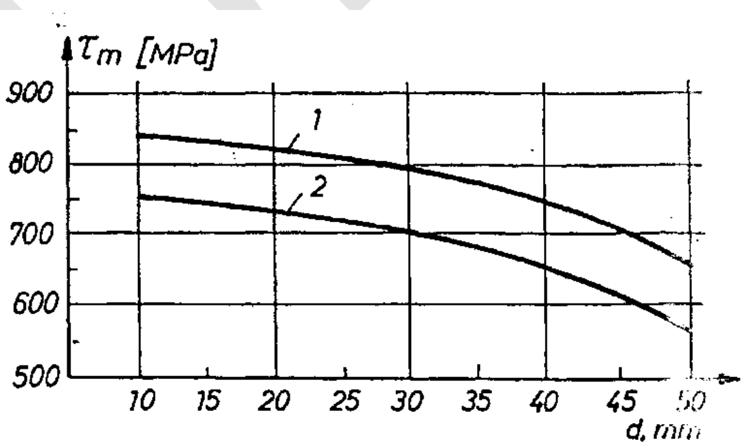



Figure 5.2 Allowed stress of hot-worked wire
Source: [13]

$$\tau_{allow} = 0.5 S_u \tag{5.4}$$

where: $S_u$ [N/mm²] ultimate stress ($R_m$)

The wire diameter involved in Eq. 5.2 must be greater than $d_{min}$ derived from Eq. 5.3.

$$d_{min} = \sqrt[3]{8\frac{K_s}{\pi}\frac{DF_{acmax}}{\tau_{allow}}} \tag{5.5}$$

The solid height (shut height) of the spring under solid load:

$$H_s = \left(N_t + 1 - N_{gr}\right)d \tag{5.6}$$

The working deflection of the spring:

$$f_n = \frac{F_{acmax}}{s} \tag{5.7}$$

The solid deflection of the spring:

$$f_s = f_n + \delta_c \tag{5.8}$$

The free height of the spring:

$$H_f = H_s + f_s \tag{5.9}$$

The free height of the spring must be checked whether it may be inserted when assembling the brake. The clash deflection (the sum of the least clearances between the active coils) is calculated according to the Eq. 5.10 and 5.11.

$$\delta_c = ydN_a = yd\left(N_t - N_{cl}\right) \quad \text{for general application} \tag{5.10}$$

$$\delta_c = 1.5yd\left(N_t - N_{cl}\right) \quad \text{for impact load} \tag{5.11}$$

where: Y specific spring clearance coefficient (see Fig. 5.3 for cold-worked wire and Fig. 5.4 for hot-worked wire)

$N_{cl}$ number of closed coils

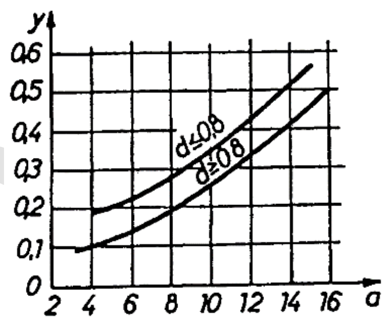



Figure 5.3 Y for cold-worked wire
Source: [13]

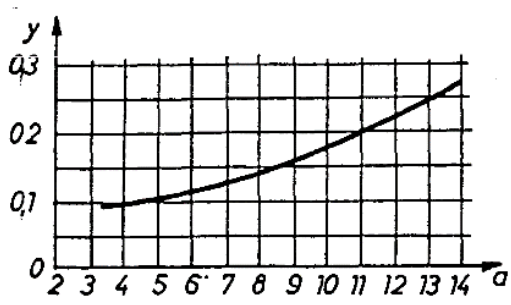



Figure 5.4 Y for hot-worked wire
Source: [13]

The coil pitch:

$$P = \frac{f_s}{N_t - N_{cl}} + d \tag{5.12}$$

# 6. Thruster and brake constructions

The characteristic and technical data of the ELDRO thrusters and they dimension drawings may be found in production catalogues.
Some drawings of different constructions of external double-shoe thruster released drum brake with or without brake spring built in the thruster can be found in product catalogues [16].

# 7. Design and stressing steps of the brake

- Determine the preliminary drum diameter: Eq. 2.16
- Calculate the $F_n$ normal force: Eq. 2.1
- Choose brake lining material and check the brake for bearing stress, warming and peripheral velocity: Eq. 2.2, Eq. 2.3, Eq. 2.10
- Draw the service diagram: Eq. 2.11, Eq. 2.12, Eq. 2.13, Eq. 2.14, Eq. 2.15, Fig. 2.3
- Choose preliminary proportion of lever arms on the basis of Fig. 9.1 - Fig. 9.12: Eq. 4.2 and calculate the proportion of lever arms: Tab. 4.1
- Calculate the necessary actuating force: Eq. 2.19
- Choose thruster: Eq. 3.1
- Prescribe the $\delta_{\min}$ : Tab. 2.2; calculate $\delta_{\max}$ : Eq. 3.6 and $\delta_{wear}$ : Eq. 2.23
- Determine the necessary spring constant: Eq. 5.1 and design the spring: Eq. 5.2 – 5. 12

# CHAPTER D. Counter drive

Design a counter drive for driving a double shift operation conveyor to convey bulk material [17].

Data of the electric motor, the drive ratio and the angle of drive layout are compiled in the Table. The conveyor is driven by a short-circuited asynchronous motor through a counter shaft that is driven by a belt drive. The power is transmitted from the counter shaft to the conveyor by a chain drive. The chain drive must operate in a casing.

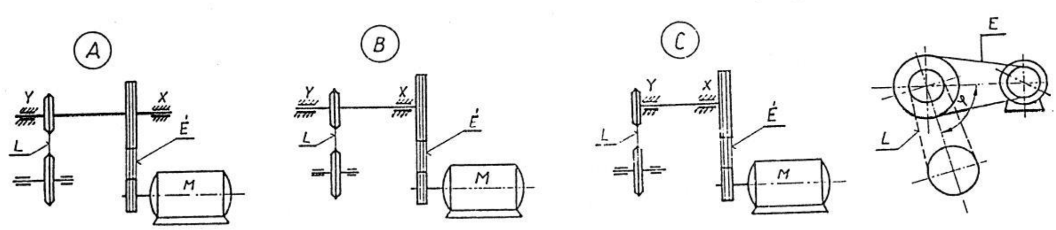
Tasks:

- Design the power transmission by dividing total drive ratio into the belt drive and the chain drive. Determine the main parameters of the belt and the chain drives relating to its dimensions and forces.
- Stress the counter shaft against fatigue and permanent deformation on the basis of the load diagram.
- Design the shaft bearing by selecting the appropriate rolling bearings.

Elaborate the following drawings:

- The assembly drawing of the counter drive (without the electric motor and the conveyor). The assembly drawing must be detailed enough for drawing the shop drawing of any parts.
- The figure of the kinematic layout of the drive in the scale of 1:10 with the specific dimensions, placed on the assembly drawing above the title block.
- The shop drawing of the pulley and the sprocket assembled on the counter shaft.

The assembly drawing shall contains views and sectional views necessary to give overall, connecting and tolerated dimensions, to show the welding joints and its dimensions of the casing. Although the drawings are made by pencil, pay attention to applying the proper line thickness.




Task variations

| Number | $P_n$ [kW] | $n_m$ [1/s] | $i_{res.}$ | φ° |
|---|---|---|---|---|
| 1 | 3 | 12 | 12 | 0 |
| 2 | 4 | 16 | 10,5 | 45 |
| 3 | 5,5 | 18 | 9 | 90 |
| 4 | 7,5 | 24 | 7,8 | 135 |

| Number | $P_n$ [kW] | $n_m$ [1/s] | $i_{res.}$ | φ° |
|---|---|---|---|---|
| 5 | 10 | 27 | 6,9 | 180 |
| 6 | 13 | 32 | 6,9 | 225 |
| 7 | 17 | 38 | 5,9 | 270 |
| 8 | 22 | 48 | 5 | |

## Design, stressing and construction considerations

When designing a counter drive, the following details have to be clarified:

- task and operation requirements,
- features and advantages of the belt drive and the chain drive,
- dividing the drive ratio for the belt drive and the chain drive,
- designing the belt and chai drives; the way how to choose belt and chain from brand catalogues,
- determining the load necessary for stressing the shaft from the belt and chai drive,
- designing the bearing support of the counter shaft and choosing bearings,
- chain lubrication requirements, implementation of them,
- construction in terms of producibility, machinability, cost-effectiveness.

**Operation requirements**

For implementation of counter drive there are a number of means to mechanically change speeds and transmit power satisfying the requirements of the driven machine. There may be a lot of demands facing the drive in order to get an optimized drivetrain [1]. In this chapter we apply a belt and a chain drive endeavoring the inexpensive solutions taking the drive elements from product catalogues and designing parts with the fewest working.

V-belt and chains drives are methods of transmitting power between two shafts separated by a wide distance, and they are used over a wide range of speed ratios. V-belts are quiet and require very little maintenance. Chain drives are positive engaging drives requiring lubrication however no initial tension is necessary.

The prescribed drive ratio is divided for the bel and the chain drive. The belt drive will transmit power from the electric motor to the counter shaft while the machine will be driven by the chain drive from the counter shaft.

In the following chapters we study these dives to acquire their operation features and parameters for designing and stressing the counter shaft.

# 1. Belt drives generals

A belt drive is a low cost means of transmitting rotary motion and power from one shaft to another. Belt drives are smooth running, quiet operation, and resistant to start-up or momentary overloads. Recently improved reinforcing materials have made belt drives more practical where formally only chain drives would have been reliably employed [18]. Belt drives are used when large distances between shafts make gears impractical or when the operational speed is too high for chain drives.

Most common applied belt types:

- Flat belts: Usually a composite construction with cord reinforcement, suited for high speeds and relatively low power. Flat belt drives produce very little noise and absorb more torsion vibration from the system than either V-belt drive. It has an efficiency of around 98%, just like a gear drive.
- Round belts: Used in agricultural machinery drives and light duty or appliance drives such as vacuum cleaners. Round belts are similar to V-belts and they run in V sheaves.
- V-belts require less tension than flat belts do, because they have more surface area contacting the pulley and therefore more friction. V-belts are comprised of a load carrying cord tensile member located at the pitch line, embedded in a relatively soft matrix which is encased in a wear resistant cover. The wedging action of a V-belt in a pulley groove results in a drive which is more compact than a flat belt drive, but short centre V-belt drives are sensitive to shock. V-belt drive can transmit more power than the flat belt drive. Its efficiency varies between 70 and 96%. The V-belt speed should be in the range of about 20 m/s.
- Timing or synchronous belts are a specific class of belts that contain toothed members similar to spur gears. The teeth on the belt mesh with the teeth on the pulley providing for a positive, non-slip rotational drive assembly. The efficiency of toothed belt drive ranges from about 97 to 99%. It can efficiently operate up to 80 m/s belt speed.

In the counter drive designing task the power transmission between the electric motor and the counter shaft has to be implemented with V-belt drive hence it is detailed in the followings.

## 1.1. V-belt types, dimensions and power ratings

The parts of the drive (pulleys, belts) are made in series production. The dimensions and qualities of them are standardized in order to provide the interchangeability. Hence the design of the drive can be carried out by suitable selection of the parts from product catalogues.

**Narrow V-Belts**

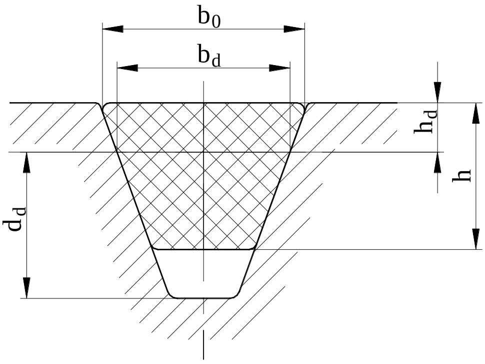
Narrow V-belts have a belt width to belt height ratio of approx. 1:1.2.

The maximum belt speed of $v_{\max} = 42$ m/s should not be exceeded.

The permissible flex rate of $v_{\max} = 42$ 1/s.

The narrow (wedge) V-belt with the sections SPZ, SPA, SPB and SPC, developed specially for all industrial applications from lightly loaded drives to heavily loaded drives, provides more tensile member support than the classical V-belt. Its dimensions are set forth in DIN 7753 Part 1 and ISO 4184 standards (see Fig. 1.1)

[19]. The datum length $L_d$ of the belt is the length measured at the position of the datum width and is the basis of standardization.

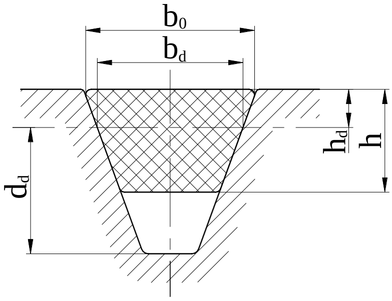



| Section | | SPZ | SPA | SPB | SPC |
|---|---|---|---|---|---|
| Belt lop width | $b_0$ | 9.7 | 12.7 | 16.3 | 22 |
| Datum width | $b_d$ | 8.5 | 11 | 14 | 19 |
| Belt height | h | 8 | 10 | 13 | 18 |
| Distance down to datum line | $h_d$ | 2 | 2.8 | 3.5 | 4.8 |
| Recommended minimum pulley datum diameter | $d_{d\,min}$ | 63 | 90 | 140 | 224 |

Figure 1.1 Narrow V-belt dimensions
Source: [19]

**Classical V-belts**

Its dimensions are set forth in DIN 2215 Part 1 and ISO 4184 standards. Classical V-belts have a belt width to belt height ratio of approx. 1:1.6. The maximum belt speed $v_{max} = 30$ m/s should not be exceeded. The permissible flex rate is considerably lower than that for narrow belts, at $f_{B\max} = 80$ 1/s. Classical V-belts are employed primarily as replacements on industrial drives. For new drives, it is almost always recommended that narrow belts be specified for reasons of space and cost. The dimensions of the classical V-belts are compiled in Fig. 1.2. It includes sections 5, 8, 20 and 25 from an earlier edition of DIN 2215. The ISO Standards specify the datum length for measuring and identifying the belt length. The datum length is the circumferential length of the belt measured at a datum width $L_p$.




| Section | DIN 2215 | (5) | 6 | (8) | 10 | 13 | 17 | (20) | 22 | (25) | 32 | 40 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | ISO 4184 | - | Y | - | z | A | B | - | C | - | D | E |
| Belt top with $b_o$ | | 5 | 6 | 8 | 10 | 13 | 17 | 20 | 22 | 25 | 32 | 40 |
| Datum with $b_d$ | | 4.2 | 5.3 | 6.7 | 8.5 | 11 | 14 | 17 | 19 | 21 | 27 | 32 |
| Belt height h | | 3 | 4 | 5 | 6 | 8 | 11 | 12.5 | 14 | 16 | 20 | 25 |
| Distance down to datum line $h_d$ | | 1.3 | 1.6 | 2.0 | 2.5 | 3.3 | 4.2 | 4.8 | 5.7 | 6.3 | 8.1 | 12 |
| Recommended minimum pulley datum diameter $d_{d\,min}$ | | 20 | 28 | 40 | 50 | 71 | 112 | 160 | 180 | 250 | 355 | 500 |

Figure 1.2 Classical V-belts dimensions
Source: [19]

The datum lengths pertaining to different belt sections are contained in production catalogues.

## 1.2. V-belt pulleys

In the case of series production pulleys are predominantly made from GG 20 cast iron and are available pilot bored, finished bored or with a taper bush. Pulleys may be made of any other materials if the dimension tolerances of the pulley can be provided and the deformation occurring is under the limit when operating. The cast pulleys are made as a solid pulley in the case of smaller sizes $(d_d \leq 140mm)$, as a plate pulley in the case of medium-sized $(140 \leq d_d \leq 280mm)$ and as a spoked pulley in the case of large sizes (see Fig. 1.3).

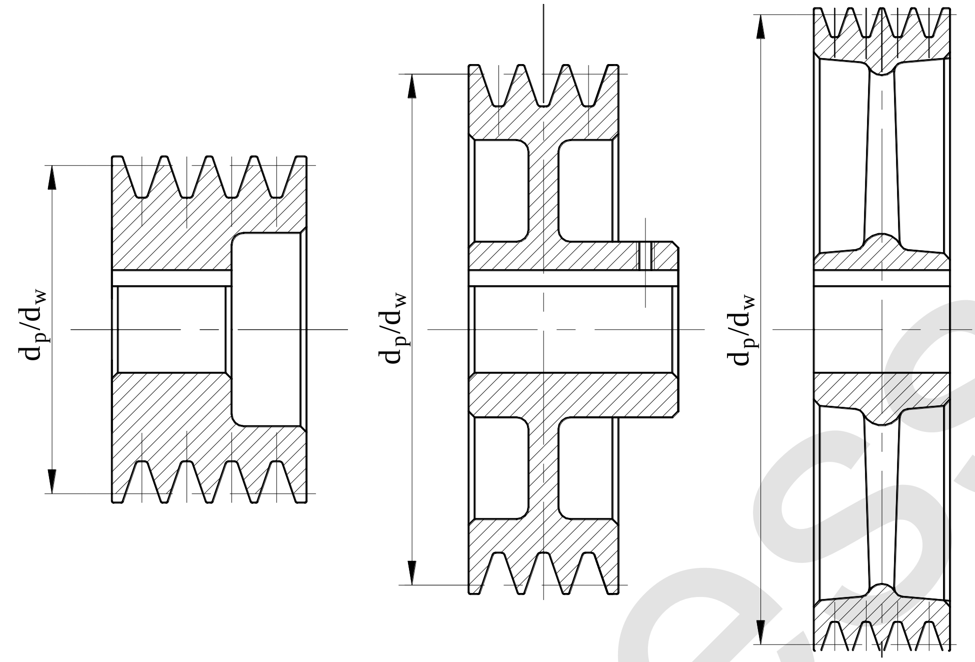



Figure 1.3 Cast pulley constructions

In the case of unit production pulleys may be welded construction, see Fig. 1.4. The number of plates in the plate pulley welded depends on the width of the crown. If necessary, web may be used between the plates but its resistance to air flow and its effects must be taken into consideration.

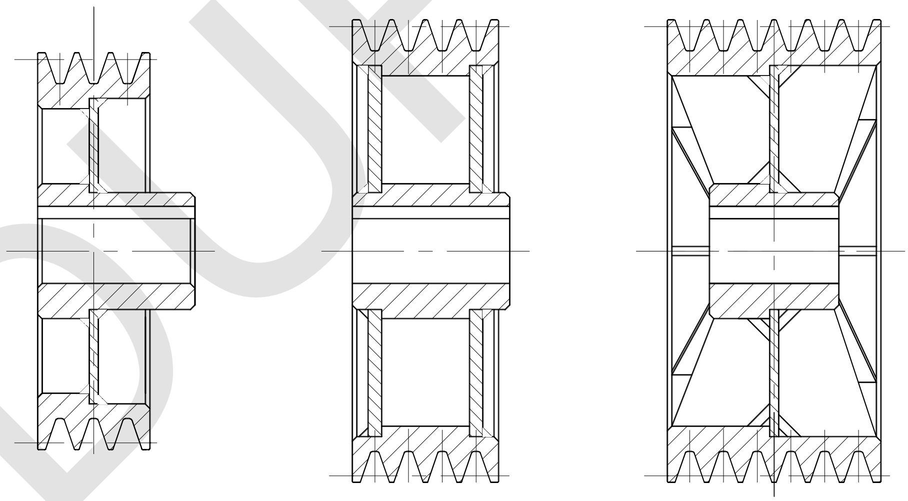


Figure 1.4 Welded pulley constructions

The torque between the shaft and the hub may be transmitted by keyed joints (DIN 6885/1) or e.g. with taper bushes. The taper bush can be selected from product catalogues on the basis of nominal torque; the keyed joint must be checked for bearing stress.

Between the shaft and the hub transition fits are applied (H7/m6, H8/n7). The axial positioning of the hub at the shaft end is provided by a shaft shoulder and a back-up washer with a threaded joint (see Fig. 1.5 a). A hub placed not at the shaft end may be backed up by a distance piece or positioned by a radial fixing screw (Fig. 1.5 b).

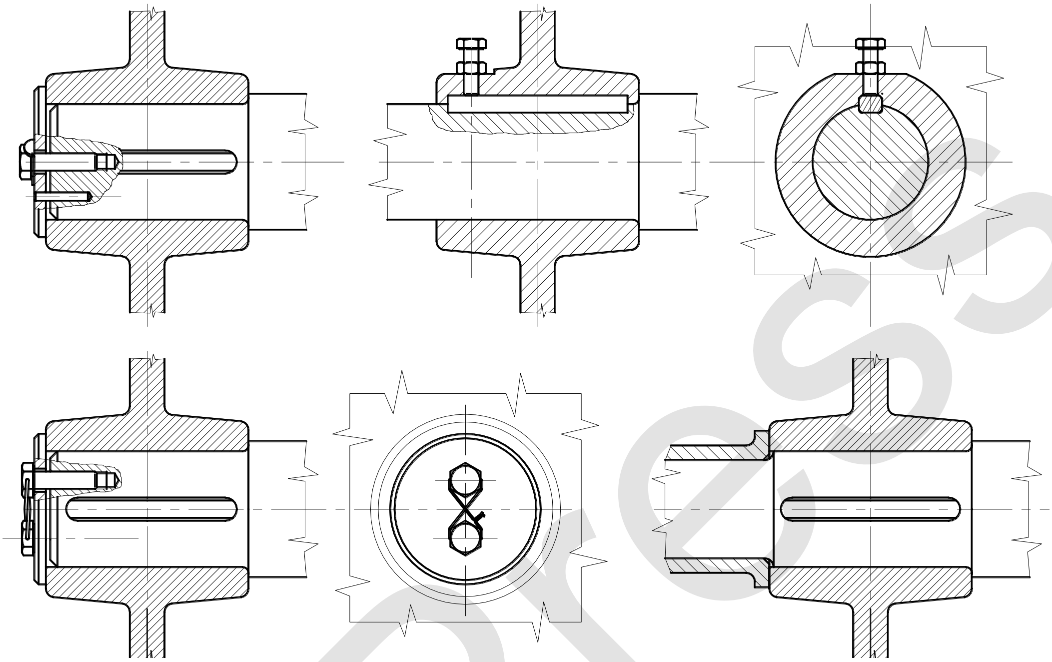


Figure 1.5 Axial positioning of pulley hub

The proper shape of the grooved pulley is vital in terms of belt service life. The dimensions of the groove for different belt sections are standardized. The average surface roughness of the groove side surfaces must be $R_a \le 2.5$ μm.

The datum diameters of the pulleys are standardized because of the series production. In the case of unit production other datum diameter may be applied exceptionally, but the overall shape and machining must conform to the relevant Standards. When designing a belt drive, standard pulley diameters should be applied. If design considerations make it impossible, a standard diameter should, as a minimum requirement, be chosen for the largest pulley in the drive. Fig. 1.1 and 1.2 contain the minimum recommended datum diameter for different belt sections while the big range of other standardized datum diameters may be found in product catalogues. The recommendation is made in the interest of belt service life and overall drive efficiency.

Grooved pulleys are generally balanced in one plane (statically). Balancing in two planes (dynamically) becomes necessary if:

- $v > 30$ m/s or
- the ratio of datum diameter to pulley face width $\frac{d_d}{d_2} < 4$ at $v > 20$ m/s.

## 1.3. Belt drive design

**Abbreviations of physical quantities**

| | | |
|---|---|---|
| $a$ | drive centre distance provisional | (mm) |
| $a_{nom}$ | drive centre distance calculated with a standard belt length | (mm) |
| $b_d$ | datum width | |
| $b_1$ | top width | |
| $c_1$ | arc of contact correction factor | |
| $c_2$ | service factor | |
| $c_3$ | belt length factor | |
| $c_4$ | number of idlers | |
| $d_{dg}$ | datum diameter of large pulley | (mm) |
| $d_{dk}$ | datum diameter of small pulley | (mm) |
| $d_{d1}$ | datum diameter of the driver pulley | (mm) |
| $d_{d2}$ | datum diameter of the driven pulley | (mm) |
| $E$ | belt deflection per 100 mm span length | (mm) |
| $E_a$ | belt deflection for a given span length | (mm) |
| $f$ | load used to set belt tension | (N) |
| $f_B$ | flex rate | (1/sec) |
| $i$ | drive ratio | |
| $k$ | constant for calculating centrifugal force in belt set | |
| $L$ | span length | (mm) |
| $n_g$ | speed of the larger pulley | (rpm) |
| $n_k$ | speed of the smaller pulley | (rpm) |
| $n_1$ | speed of the driver pulley | (rpm) |
| $n_2$ | speed of the driven pulley | (rpm) |
| $P$ | motor or normal running power | (kW) |
| $P_B$ | design power | (kW) |
| $P_N$ | nominal power rating per belt | (kW) |
| $S_a$ | minimum static shaft loading | (N) |
| $T$ | minimum static tension per belt | (N) |
| $v$ | belt speed | (m/s) |
| $x$ | minimum allowance above centre distance $a_{nom}$ for belt stretch and wear (mm) | |
| $y$ | minimum allowance below centre distance $a_{nom}$ for easy belt fitting | (mm) |
| $z$ | number of belts | |
| $\alpha$ | angle of belt drive $\alpha = 90^o - \beta / 2$ | (°) |
| $\beta$ | arc of contact on small pulley | (°) |

***When calculating every physical quantities are in SI unit in Formulas.***

**Design of V-belt drive**

Data available for design

P (kW) normal motor power

n (1/min) speed of the motor (driver pulley)

$$i = \frac{n_{driver}}{n_{driven}} = \frac{n_1}{n_2} \quad \text{drive ratio prescribed} \tag{1.1}$$

Operating condition of the belt drive

## 1.3.1. Nominal power rating, arc of contact correction factor

The nominal power ratings $P_N$ per belt are based upon an internationally recognized basic formula and a theoretical belt life of 25,000 hours under ideal conditions. This formula contains material constants that take the quality of the raw materials used and make allowances for production methods into account. The nominal power ratings $P_N$ are based on the smallest loaded pulley in the drive system.

The belt power rating value $P_N$ is calculated by taking into account:

- the datum diameter of the smaller pulley $d_{dk}$
- the speed of the smaller pulley $n_k$
- the drive ratio i
- an assumed arc of contact at the smaller pulley of $\beta = 180^o$
- a reference belt length for the specific belt section.

The power ratings for different belt sections can be found in product catalogues. In order to account for the true drive data, based on the arc of contact and the belt lengths employed, correction factors for the arc of contact $c_1$ and length $c_3$ have been introduced. Intermediate values for nominal power rating, arc of contact and length correction factors can be found by linear interpolation.

The factor $c_1$corrects the power rating $P_N$ when the arc of contact is smaller than 180°, as the $P_N$ value was calculated on the arc of contact $\beta = 180^o$ on the smaller pulley. Factors $c_1$ are compiled in product catalogues however some values of it are here to show the order of magnitude.

| $\beta = 180^o$ | $\beta = 160^o$ | $\beta = 140^o$ | $\beta = 120^o$ | $\beta = 100^o$ | $\beta = 80^o$ |
|---|---|---|---|---|---|
| 1 | 0.99 | 0.97 | 0.94 | 0.91 | 0.84 |

## 1.3.2. Design power $P_B$ – Service factor $c_2$

$$P_B = P c_2 \tag{1.2}$$

The service factor $c_2$ takes account of the daily operating time and of the type of driver and driven machine. It applies exclusively to two-pulley drives. Other arrangements such as drives with tension and guide idlers have not been taken into consideration.

Table 1.1 Service factors of belt drive

| **Driven machine** | **Prime mover** | |
|---|---|---|
| | mono-phase electric motor with starting torque, tri-phase totally enclosed electric motor (normal starting torque) direct-current motor, internal combustion motor and turbine $n > 600 \frac{1}{\min}$ | mono-phase electric motor with starting torque, tri-phase totally enclosed electric motor (normal starting torque) direct-current motor, internal combustion motor and turbine $n \leq 600 \frac{1}{\min}$ |
| | $c_2$ | $c_2$ |
| **Light operation**<br>Centrifugal pumps, compressors, fans | **1.1** | **1.3** |
| **Medium operation**<br>Sheet cutter, press machines, vibrators, machine tools, | **1.1** | **1.4** |
| **Heavy operation**<br>Piston compressor, press machines, excavators, forging machine, | **1.2** | **1.6** |
| **Extreme operation**<br>grinder, stone-breakers, manglers, woodworking mach. | **1.3** | **1.8** |

Adverse operating conditions (e.g. aggressive dust, particularly high ambient temperatures) are considered by service factors.

In special cases, e.g. increased starting torque (direct on-line starting of fans), on drives with frequent starts and stops, on systems subject to exceptional shock loads, or when significant masses are to be accelerated or braked, the service factor must be increased [19].

## 1.3.3. Selection of belt section

The most efficient power transmission is achieved by selecting as large a pulley diameter as possible for the section in question.

The limits to be observed are the maximum permissible circumferential speeds, namely $v_{\max} = 42$ m/s for high performance wedge belts and $v_{\max} = 30$ m/s for classical V-belts. Experience has shown that the minimum pulley diameters should be avoided. In borderline cases it is recommended that the next smaller section belt be used on the same diameter pulley, as the smaller section will often save both cost and space. Comparing space requirement and costs, the narrow belt normally proves to be significantly superior to the classical V-belt for almost all industrial machinery drives. Fig. 1.6 and Fig. 1.7 show the design charts to select the appropriate belt sections [19].

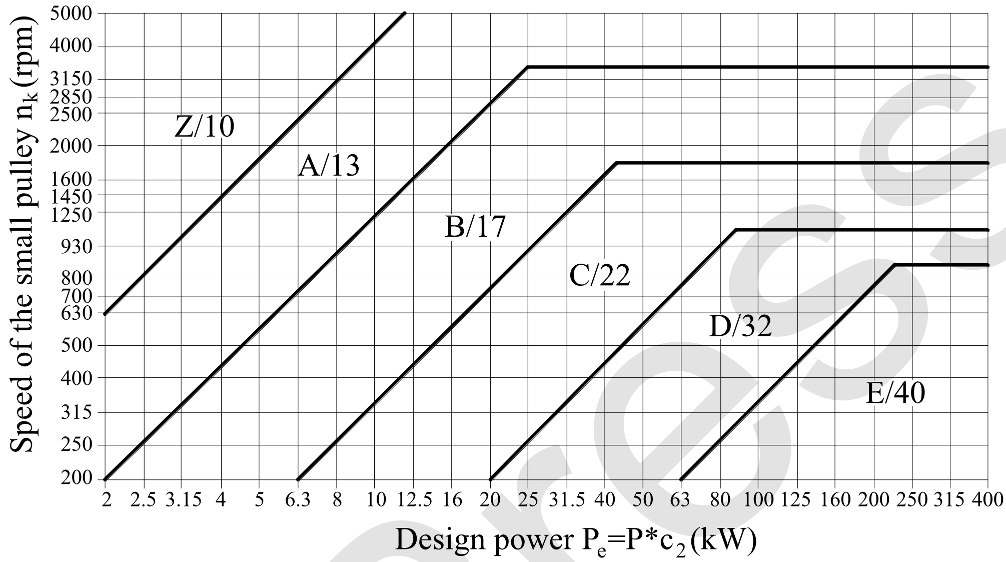



Figure 1.6 Belt section charts of classical belts
Source: [19]

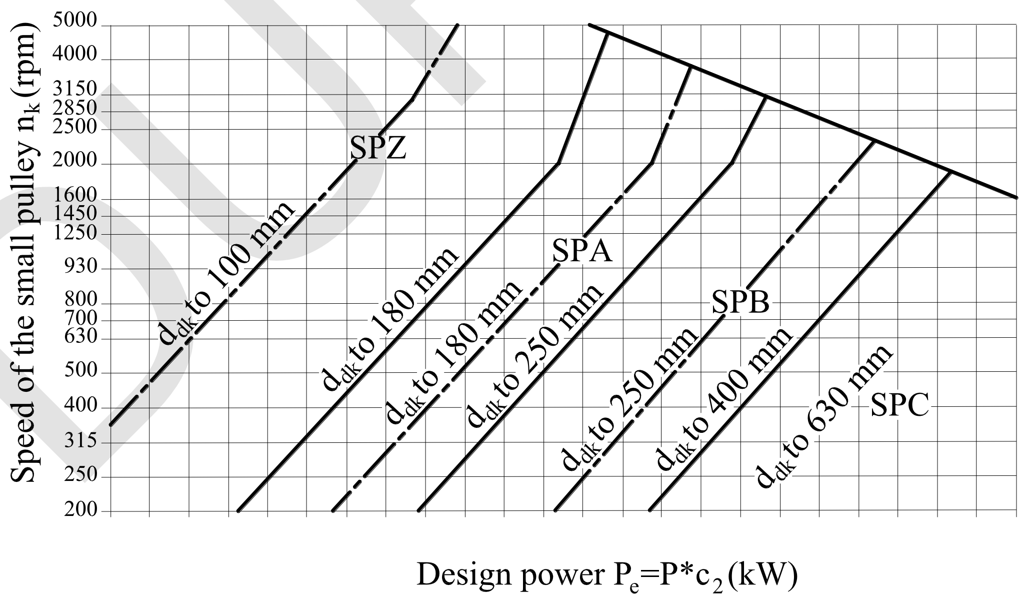



Figure 1.7 Belt section chart of narrow belts
Source: [19]

### 1.3.4. Length factor $c_3$ for V-belts

The length factor $c_3$ takes into account the flex rate of the belt based on the reference length. Its values for different belt sections are compiled in product catalogues however it may be realized if:

belt length > reference length $c_3 > 1.0$

belt length = reference length $c_3 = 1.0$

belt length < reference length $c_3 < 1.0$

### 1.3.5. Number of belts

The number of belts necessary to transmit the power:

$$z = \frac{P_B}{P_N c_1 c_3} = \frac{P c_2}{P_N c_1 c_3} \tag{1.3}$$

It is favorable if $2 \le z \le 5$.

A multiple drive may be carried out by kraftbands. Kraftbands are made up of individual V-belts rigidly connected by a cover plate. This compact drive element with single-belt-characteristics is also known as a joined V-belt. According to application kraftbands are fitted with two, three, four or five ribs. All belts in such a drive stretch at the same rate to keep the load equally divided among them.

### 1.3.6. Datum diameters of pulleys

Determination of the diameters of the pulleys is based on the following criterions:

- the belt speed should be
  - for classical V-belts $5 < v < 25$ [m/s]
  - for narrow V-belts $5 < v < 45$ [m/s]
- standardized diameters should be selected as a datum diameter for both of the pulleys with respect to the belt section, see product catalogues

The calculation must be carried out for the smaller pulley from which the datum diameter can be expressed.

$$v = d_d \pi n \tag{1.4}$$

where: n is the rpm of the particular pulley

If $d_{d_{calc}} = d_{d1_{calc}}$, than $d_{d2_{calc}} = d_{d1_{calc}} i$ (1.5)

If $d_{d_{calc}} = d_{d2_{calc}}$, than $d_{d1_{calc}} = \dfrac{d_{d2_{calc}}}{i}$ (1.6)

After selecting the datum diameters the actual drive ratio must be recalculated and the speed of driven pulley determined.

$$i_{actual} = \frac{d_{d2}}{d_{d1}} \tag{1.7}$$

$$n_{2actual} = \frac{n_1}{i_{actual}} \tag{1.8}$$

### 1.3.7. Drive centre distance (preliminary choice)

The recommended centre distance:

$$0.7(d_{dg} + d_{dk}) < a < 2(d_{dg} + d_{dk}) \quad [a_{opt} = 1.05d_2 \ \ldots \ 1.1d_2] \tag{1.9}$$

Belt datum length:

$$L_{dth} \approx 2a + 1.57(d_{dg} + d_{dk}) + \frac{(d_{dg} - d_{dk})^2}{4a} \tag{1.10}$$

The calculated belt length must be rounded to the nearest standardized datum length, $L_d$ (see product catalogues).
The actual drive centre distance can be calculated from the datum length of the belt selected.

$$a = p + \sqrt{p^2 - q} \tag{1.11}$$

where: $p = 0.25L_d - 0.393\left(d_{dg} - d_{dk}\right)$ (1.12)

$$q = 0.125\left(d_{dg} - d_{dk}\right)^2 \tag{1.13}$$

The arc of contact on small pulley:

$$\beta = 2\arccos\frac{d_{dg} - d_{dk}}{2a} \tag{1.14}$$

### 1.3.8. Belt flex rate

The service life of the belt depends on the number of flexing when running up the pulley. The number of flex/min is the flex rate. For a two pulleys drive it can be calculated as follows:

$$\mathrm{f} = 2\frac{\mathrm{v}}{\mathrm{L_d}} \quad [1/s] \tag{1.15}$$

where: v [m/s] belt speed

$L_d$ [m] datum length of the belt

$f \le f_{allow}$ $\quad f_{allow}$ is given for the belt type

$f_{allow} = 30 \ [1/s]$ classical V-belts

$f_{allow} = 100 \ [1/s]$ narrow V-belts

### 1.3.9. Shaft loading under dynamic conditions

Considering the ”dynamic force” saves the unnecessary expense of:

- premature bearing failure
- shaft failure or
- over designed bearings and shafts

In the case of two pulleys drives, the driver and driven shafts and the bearings are subjected to the same dynamic force, but in opposite directions. When idlers are employed, the magnitude and the direction of the shaft force are almost always different on each pulley. If the magnitude and direction of the dynamic shaft force is to be determined, a graphical solution, using a vector diagram for the dynamic forces in the tight side $S_1$ and the slack side $S_2$, is always recommended. If only the magnitude of the dynamic shaft force is to be determined, this can be achieved using the formula for ” $S_{a\ dyn}$ ”, see Fig. 1.8.

Dynamic tight side tension:

$$S_1 \approx \frac{1.02 P_B}{c_1 v} \qquad (1.16)$$

Dynamic slack side tension:

$$S_2 \approx \frac{(1.02 - c_1) P_B}{c_1 v} \qquad (1.17)$$

Dynamic shaft force:

$$S_{a\ dyn} \approx \sqrt{S_1^{\ 2} + S_2^{\ 2} - 2 S_1 S_2 \cos\beta} \qquad (1.18)$$

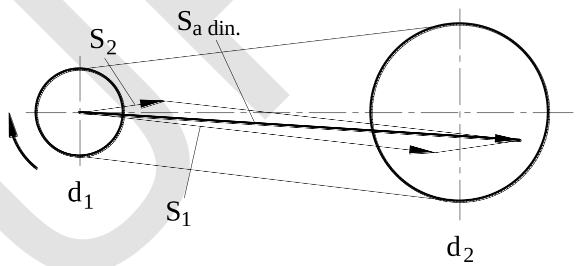



Figure 1.8 Belt forces

As a simplification, the following formula may be applied for calculating the shaft load acting on the common center line of the pulleys:

$$F_{a\ dyn} = \frac{K - c_1}{c_1} \cdot \frac{c_2 P}{v} \qquad (1.19)$$

where: K coefficient

K = 3.0 for classical V-belts

K = 2.04 for narrow V-belts

## 1.3.10. Pretensioning the V-belts

A belt tension which is either too high or too low often results in premature failure. Belts which are over tensioned sometimes cause damage to the bearings on the driver or driven units. The loose belts slide in the pulley grooves and therefore they are wearing fast, as well as cannot transmit the power. This is why the belt tension must be determined and prescribed using one of the following methods:

**Checking the belt tension by span deflection**

This method provides an indirect measurement of the calculated or actual static belt tension. It is applicable for belt sections SPZ, SPA, SPB, SPC, A/13, B/17, C/22, 25, D/32.

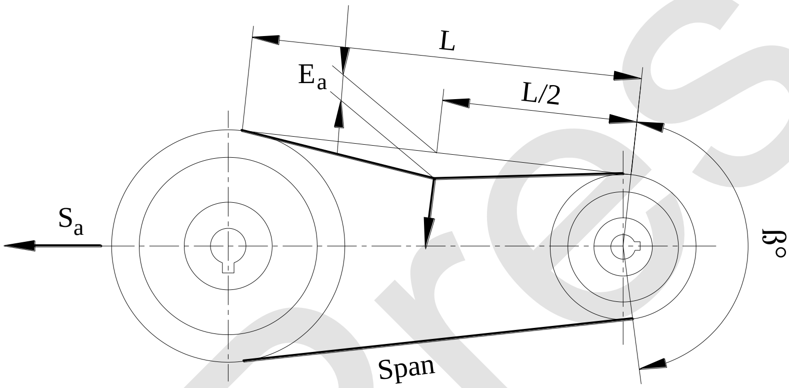



Figure 1.9 Belt span deflection

| | |
|---|---|
| K = 3.0 = belt deflection per 100 mm span length | (mm) |
| $E_a$ = belt deflection for a given span length | (mm) |
| E= load used to set belt tension | (N) |
| L= drive span length | (mm) |
| $S_a$ = minimum static shaft load | (N) |
| T= minimum static tension per belt | (N) |

The static belt tension can be calculated using the following formula:

$$T_0 = \frac{F_{a\ dyn}}{2z} + kv^2 \tag{1.20}$$

$$T \approx \frac{1}{2}\frac{(K - c_1)}{c_1}\frac{P_B}{zv} + kv^2 \tag{1.21}$$

where: k [kg/m] weight per meter of the belt

z number of belts

The belt deflection per 100 mm span length E may be determined from the belt tension/deflection graphs.

The belt deflection for a given span length $E_a$, for the actual drive span length:

$$E_a \approx \frac{EL}{100} \qquad [\mathrm{mm}] \tag{1.22}$$

$$L = a_{nom} \sin\frac{\beta}{2} \tag{1.23}$$

When applying the “f” load perpendicular to the center of the belt to set the belt tension T (taken from Fig. 1.10 for the appropriate belt section), as shown in the illustration (see Fig. 1.9) the $E_a$ deflection may be measured and can be adjusted by modifying the center distances [19]. This way the correct belt tension can be achieved.

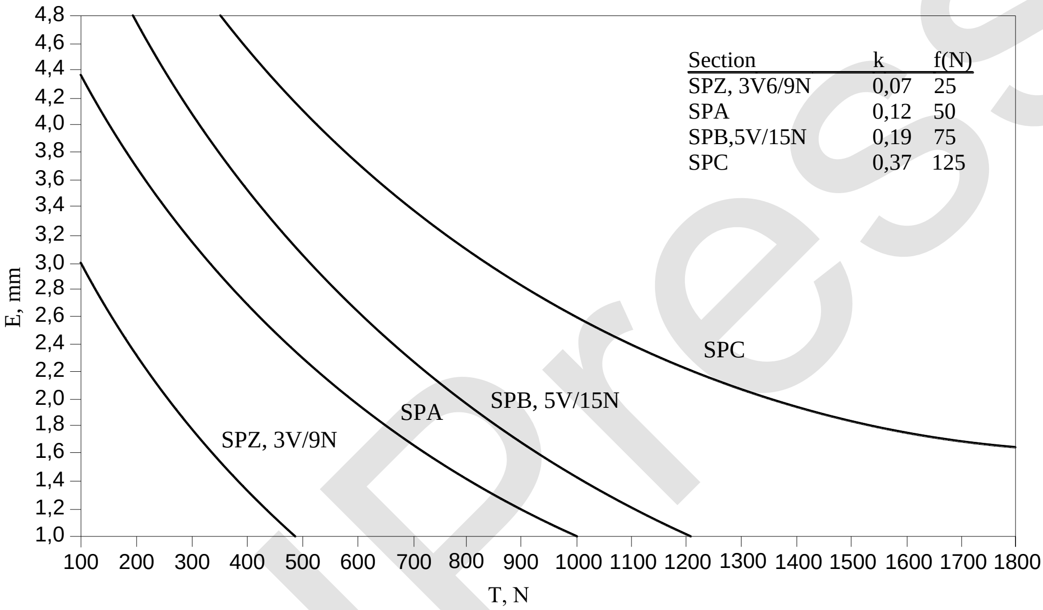



Figure 1.10 Tension force chart for narrow belts
Source: [19]

## 1.3.11. Adjusting the drive center distance

When designing the V-belt drive the facility of changing the belt must be provided. It can be implemented with an adjustable center distance by approaching the pulleys to each other, or in the case of a fix center distance, by applying longer belt with an idler pulley forced to the slack side of the belt. The values of x and y depending on the datum length of the belt for different sections, are compiled in product catalogues and engineering norms, see Fig. 1.11. There are various approaches of the maintaining necessary belt tension. These include using a pivoted-overhung motor drive; increasing the center distance during operation by employing a drive with an adjustable center distance, see Fig. 1.12 and 1.13. Because of the resistance of their interior tension cords to stretch, V-belts do not require frequent adjustment of initial tension.

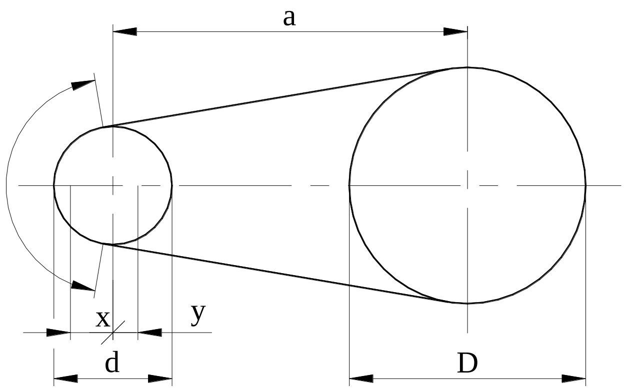
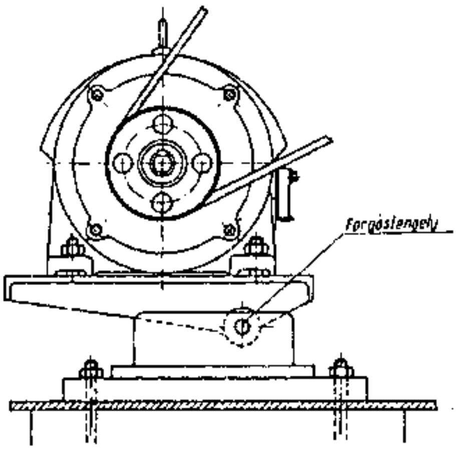



Figure 1.11 Adjusting dimensions of pulleys



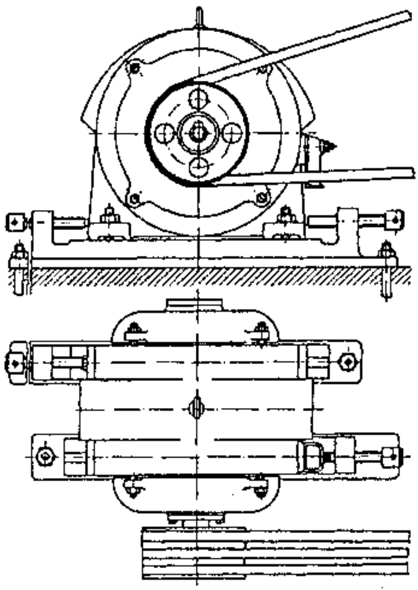


Figure 1.12 Pivoted overhung motor base

Figure 1.13 Adjustable center distance

## 2. Chain drives

Chain drives are normally used for applications below 3000 rpm where accuracy and reliability must be greater than that provided by rubber belts. Chain drives maintain a constant ratio under varying load conditions because the metal chain does not slip or stretch and need only infrequent adjustment. In contrast to belt drives, one strand of a chain drive is always slack. Thus, power is transmitted solely by the tension side. Chain drive transmits power between parallel shafts, but precise location and alignment tolerances are not required, as with gear drives. Chain drives are less sensitive to dust and humidity than belts and are not harmfully affected by sun, oil or grease. They can also operate at much higher temperatures than belts and they do not slip. Chain drives do require frequent lubrication (continual in some applications), they require very close alignment, and since they do not slip they provide no overload protection. There are various type of power transmission chains, however, roller chain and inverted (silent) chain types are the most widely employed. Fig. 2.1 shows the construction of a roller chain and connecting links respectively [20].

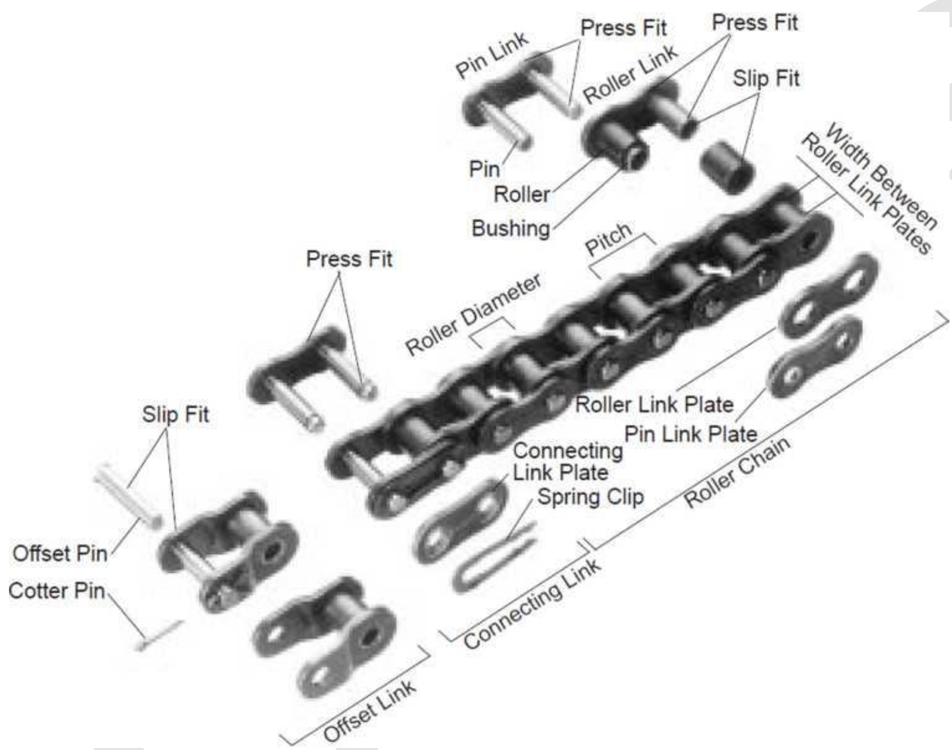



Figure 2.1 Roller chain assembly

Roller chains are of two versions, type A and type B. Type A is involved in the American Standard ANSI and its dimensions are set forth in ANSI 40-1/DIN 8188 standard. Type B is involved in the British Standard BS and its dimensions are set forth in DIN 8187 standard. The designation of the chains according to the ANSI and the DIN standards with the pitch dimensions are compiled as follows [20]:

| **DIN** | **08 B** | **10 B** | **12 B** | **16 B** | **20 B** | **24 B** | **28 B** | **32 B** | **40 B** | **48 B** | **56 B** |
|---|---|---|---|---|---|---|---|---|---|---|---|
| **ANSI** | **40** | **50** | **60** | **80** | **100** | **120** | **140** | **160** | **200** | **240** | **-** |
| | **08 A** | **10 A** | **12 A** | **16 A** | **20 A** | **24 A** | **28 A** | **32 A** | **40 A** | **48 A** | **-** |
| **Pitch inch** | 1/2 | 5/8 | 3/4 | 1 | 1 1/4 | 1 1/2 | 1 3/4 | 2 | 2 1/2 | 3 | 3 ½ |
| **Pitch mm** | 12.7 | 15.88 | 19.05 | 25.4 | 31.75 | 38.1 | 44.45 | 50.8 | 63.5 | 76.2 | 88.9 |

Chains may be single or multiple strands. The typical dimensions of them are standardized and may be found in product catalogues.

## 2.1. Failure modes of chain drives

There are several ways a chain can fail [21]:

- In a tensile failure, the chain is overloaded in tension until it can not function properly, or it is literally pulled apart.
- In a fatigue failure, the chain is loaded repeatedly in tension, at a load below the yield strength, however it may results in the break of link plates.
- In a wear failure, material is removed by sliding, or sliding combined with abrasion or corrosion, until the chain will not fit the sprockets, eventually the chain breaks.
- Galling is rapid wear caused by metal seizure between the chain pin and bushing. This rapid wear is caused by the combination of excessively high speeds and loads.

The key reason causing a chain to jump the sprocket teeth is chain wear elongation. It is the consequence of the link-joint wear occurring between the chain link pin and the bush produces an elongation of the distance between every second chain roller. It is the reason for applying only sprockets with an odd number of teeth. The gradual ride-up of the chain in the teeth that corresponds to its increasing length, cannot be ameliorated by applying an increased tension.

Sprocket wear may be reduced moreover, can be minimized considerably by replacing the chains, as soon as, wear elongation arrives at 1.5...2.0%. In general the allowed extension in length of the chain is of 3 % wear elongation.

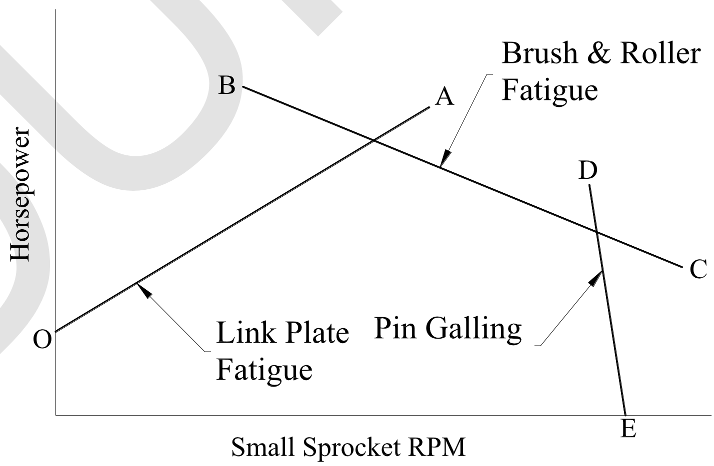



Figure 2.2 Failure modes of chain drive

As the cross-sectional area of the pin is reduced by wear, the strength of the pin decreases and eventually fatigue failure may result. When wear elongation is less

than or equal to 1.5 percent for a transmission chain, or less than or equal to 2 percent for a conveyor chain, there is almost no risk of fatigue failure. Fig. 2.2 shows the failure modes of the chain depending upon the small sprocket rpm and the power transmitted. The power capacity of roller chain at lower speeds is based upon the fatigue strength of the link plate. At higher speeds, the power relies on roller and bushing impact life. The extremely high speeds may result in galling or welding between pin and bushings.
Whereas the chain wear elongation amounts to max. 3% below the teeth number 66 in terms of chain capacity, because of the decreasing take-up capacity in the sprocket gearing with rising number of teeth, the allowed elongation is limited. The interdependence of the take-up capacity of wear elongated chains on the number of sprocket teeth is shown in Fig. 2.3.

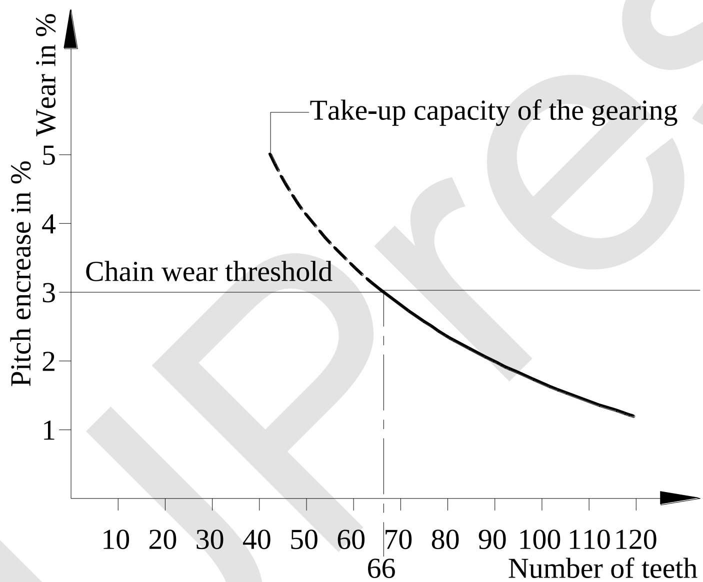



Figure 2.3 Take up capacity of the gearing

## 2.2. Chain drives installation

- An odd number of teeth on the driving sprocket (17, 19, 21, ...) is recommended, typically 17 and 25. Because it causes each small sprocket tooth to contact many or all chain links, minimizing wear. The larger sprocket is ordinarily limited to about 120 teeth.
- Center distance c should be between values that just allow the sprocket to clear.

$$c = 2(r_1 + r_2)\text{, for speed ratio } \frac{n_1}{n_2} < 3$$

$$c = 2(r_2 - r_1)\text{, for speed ratios } \frac{n_1}{n_2} \geq 3$$

- When longer chains are used, idlers may be required on the slack side of the chain.
- The angle of contact for a small sprocket should not be less than $120^{\circ}$.
- An even number of pitch in the chain is preferred to avoid a special link. Chains are manufactured in single, double, triple and quadruple strands.
- The recommended centre distance is (30 - 50)p, or the following centre distances may be applied for different chain pitches:

| | | | | | | |
|---|---|---|---|---|---|---|
| **Chain pitch** | inches | 3/8” | 1/2" | 5/8” | 3/4" | 1” |
| | mm | 9.525 | 12.7 | 15.875 | 19.05 | 25.4 |
| **Centre distance** | mm | 450 | 600 | 750 | 900 | 1000 |

- Shafts must be parallel with each other. For adjusting the shafts’ position, rollers may be used, see Fig. 2.4. The supporting structures of the bearings must be sufficient rigid to maintain true alignment.

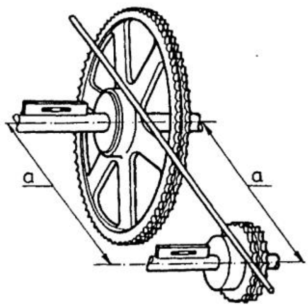



Figure 2.4 Shaft positioning

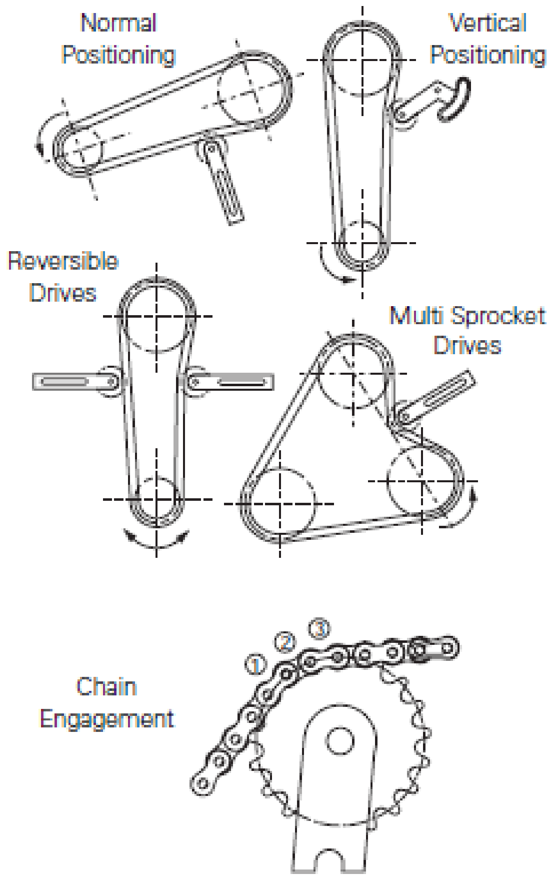



Figure 2.5 Chain tensioning

- Sprockets must be mounted to bearings as close as possible.
- In fixed centre drive an idler sprocket is generally recommended. It should be positioned on the slack side as close to the larger sprocket as feasible. The

tensioning sprocket should have a minimum of three teeth engaged and be a minimum of four links away from the nearest sprocket.

- Roller chain can be used in practically any position (see Fig. 2.5) provided the shafts are parallel. On fixed centre drives the effect on the slack strand must be carefully considered. Where the slack strand is nearly vertical, or where torque variation causes wave or whip in the chain, an idler must be used to take up the excessive slack.
- The idler should preferably be near to the larger sprocket in the drive, located on the outside of the slack strand of the chain. (Where layout makes this impossible it is permissible to locate the idler on the inside of the chain.)
- Chains should be fairly tight at installation with only a small amount of slack. The sag due to the own weight of the chain which can be either new or elongated should not exceed the 2-3% of the centre distance.
- The sprockets' number of teeth influences the wear and running conditions to a considerable extent. If the number of teeth is too small, a greater number of link-joint motions will produce an increased wear and the consequence of polygon effect is significant.

## 2.2.1. The polygonal effect

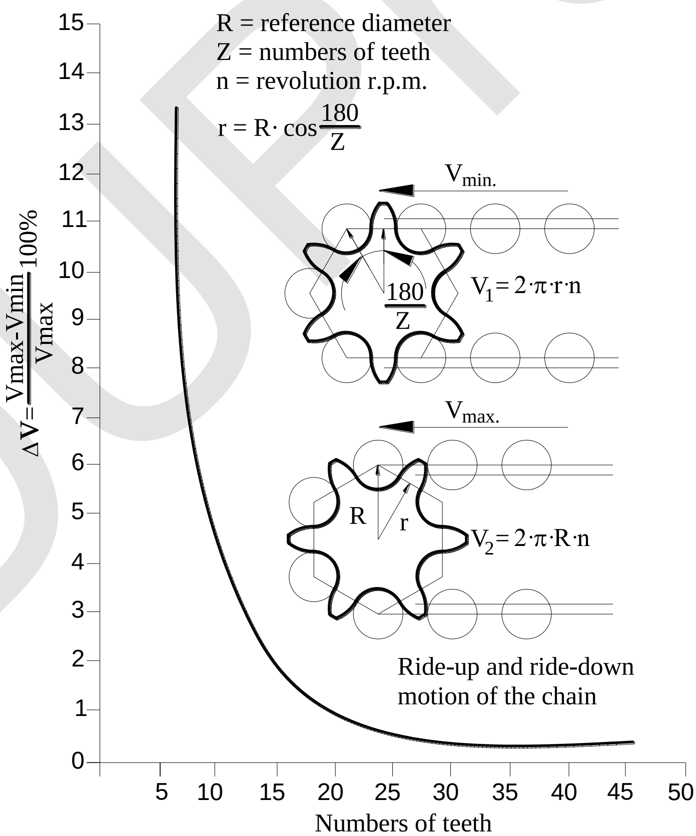



Figure 2.6 Polygon effect

As is shown in the Fig. 2.6, a sprocket is subjected to an acceleration and a deceleration effect when the chain enters and runs off the sprocket and the chain performs a ride-up and ride-down motion. With a smaller number of teeth, uneven running increases on a progressive scale. It generates additional dynamic forces in the chain because of altering tractive forces to be transmitted causing fatigue load of the parts. The uneven load transmission in conjunction with the repeated ride-up and ride-down motions give rise to a jerky chain running.

### 2.2.2. Choice of the number of the sprocket teeth

9 to 10 teeth: should, on principle, be avoided. Suitable merely for adjusting drives with low chain speeds (< 1 m/s). No call for an even and smooth running.

11 to 12 teeth: suitable only for max. 2m/s chain speeds. The specific chain load should be small. No call for a smooth and even chain running.

13 to 14 teeth: suitable for chain speeds of less than 3 m/s, provided the chain load is low and no call for a smooth and quiet running.

15 to 17 teeth: suitable for max. 6 m/s chain speed, provided no special requirements exist for a quiet and vibration-free running.

18 to 21 teeth: suitable for chain speeds up to max. 10 m/s, produce a satisfactory running performance and, smooth running.

22 to 25 teeth: suitable for chain speeds up 15 m/s, produce a satisfactory running performance and, smooth running.

26 to 40 teeth: suitable for chain speeds up 30 m/s. Suitable for highly stressed high-revolution drive sprockets. The polygon effect is negligible. Vibration and noise features meet the highest demands.

45 to 120 teeth: the most appropriate numbers of teeth for driven wheels. They meet all demands for a good running performance. However, due to the reduced take-up capacity of the gearing, the admissible wear elongation is reduced to the following values (see Fig. 2.3):

| | |
|---|---|
| Z = 70 | 2.8% |
| Z = 80 | 2.3% |
| Z = 90 | 2.0% |
| Z = 100 | 1.7% |
| Z = 120 | 1.2% |

125 to 200 teeth: should be avoided. As to their running performance, they offer no improvement compared to the range of 45 to 120 teeth; however, admissible chain wear elongation is reducing.

## 2.3. Chain lubrication

Statistics have shown that 60% of all failures in chains are a direct result of incorrect lubrication [20]. Effective lubrication is essential in order to ensure optimum wear life by forming a film of lubricant between the wearing parts (pin and bush; roller and bush), of the chain. The amount of lubricant required for the chain roller is relatively moderate. Oil will only penetrate into the bearing area of the chain when the chain is slack, therefore oil should be delivered to the slack strand just after the driver sprocket. High speed drives are especially critical. These generally require a continuous stream of lubricant applied across the full chain width in order to act as a coolant as well.

With chain drives having a speed of approx. 0.5 m/s, **lubricating by hand** may be resorted to.

**Drip lubrication**

Drip Feed (for chain speeds from 0.5 up to 1.5 m/s). Oil drops are directed between the side plate gaps with a drip feed lubricator.

**Oil bath lubrication**

Oil Bath or Disc Lubricator (for chain speeds from 1.5 up to 8 m/s). With oil bath lubrication the lower strand of chain runs through an oil sump immerse the chain at its lowest point.

An oil disc may also be used. In this case the disc picks up oil from the sump and deposits it on the chain.

The diameter of the disc should be sufficient to ensure a rim speed between 3 and 15 m/s.

**Forced lubrication**

The pumped oil should be supplied on the inside of the chain loop and at the lower strand, when chain speeds exceed 100 m/s.

## 2.4. Roller chain drive design

Required drive data

| | | |
|---|---|---|
| $P$ | [kW] | motor or nominal running power |
| $n_1$ | [1/min] | speed of the driver shaft |
| $i$ | | drive ratio |
| $a$ | [mm] | drive centre distance provisional |
| Impact type | | (prime mover, driven machine) |
| Drive layout | | |

*When calculating every physical quantities are in SI unit in Formulas.*

**General considerations**

The selection of roller chains relies on the performance tables.

The performance numbers listed in the performance tables relate to the capacity of single-strand roller chains. They apply to a theoretical lifetime of 15,000 working hours at a 3% wear elongation.

The basic conditions on which the performance numbers based:

- the chain drive consists of 2 sprockets and include a tension sprocket in the return strand
- the distance between centers of 30...50 times the chain pitch
- an impact free operation
- chain lubrication as specified in the performance table
- numbers of teeth as specified

## 2.4.1. Operation conditions of the chain drive

The output to be transmitted requires correcting, depending on the types of the driving and driven machines [20].

Table 2.1 Impact coefficient of the chain drive

| | **Driving machine** | | |
|---|---|---|---|
| **Driven machine** | Combustion engine with a hydrostatic transmission | Electric motor | Combustion engine with a mechanical transmission |
| Impact-free operation | 1.0 | 1.0 | 1.2 |
| Average impact load | 1.2 | 1.3 | 1.4 |
| Severe impact load | 1.4 | 1.5 | 1.7 |

| **Impact-free operation** | **Heavy impact load** | **Medium impact load** |
|---|---|---|
| Machines with uniform power draw and without reversing operation | Machines with an uneven power draw and with reversing operation | Machines with a high uneven power draw with reversing operation |
| Continuous mechanical handling equipment; Fans; Centrifugal pumps; Stirring devices | Machine tools; Reciprocating pumps; Textile machines; Wood working machines | Road construction machines; Asphalt cutters; Pulvimixers; Excavator drives |
| Drum drives with a constant power draw and with reversing operation | Elevators, storage and retrieval units for high-bay warehouses; Drum drives with reversing operation | Presses, blanking presses; Drum drives with reversing impacts |

**Design Power $P_B$**

Design Power determined by the impact coefficient is used as the basis for selecting the chain.

$$P_B = YP \tag{2.1}$$

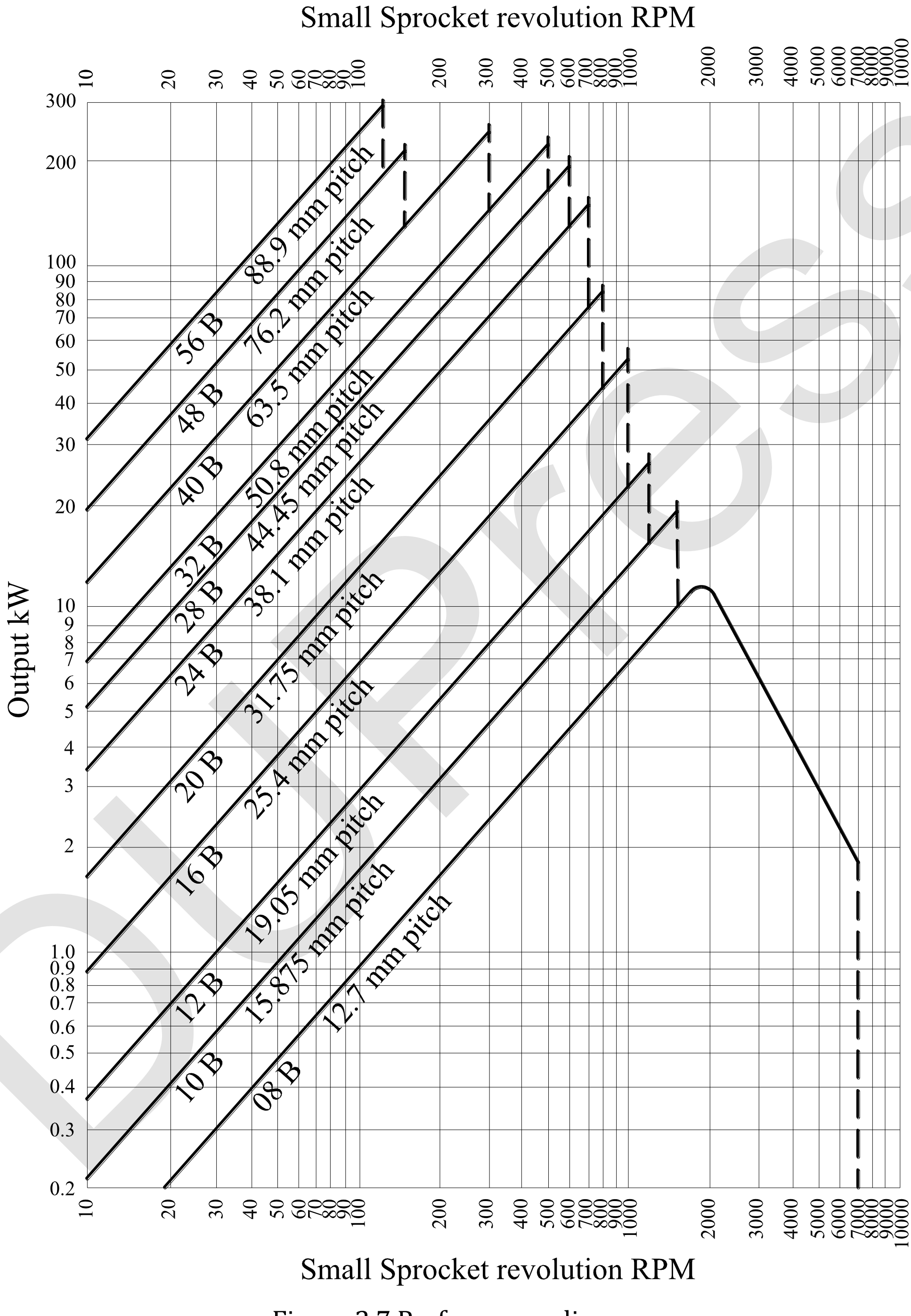



Figure 2.7 Performance diagram

Source: [20]

**Chain Pitch**

Chain pitch can be selected from the performance diagram (see Fig. 2.12), on the basis of the small sprocket revolution and the design power [20].

**Sprocket Sizes**

When fixing the tooth number of the sprockets the general recommendations must be taken into consideration regarding the polygon effect, the take-up capacity of the gearing and the demands for a good running performance, see chapter 2.2.3.

Of course, the number of teeth of the driven sprocket is rounded to whole number of teeth.

$$z_2 = iz_1 \tag{2.2}$$

**Power Rating $P_{output}$**

The power rating of the BS chains for the pitch chosen on the basis of small sprocket rpm and the design power are compiled in catalogue. The power rating tables pertain to single chain and consider the number of teeth of the small sprocket and the lubrication method applied. For satisfying high speed or smooth running requirements or the space is limited, a smaller pitch chain, duplex or triplex drive may be considered. (Single strand chain offers the most economical solution, and should be used where possible.)

**Chain strands factor S**

The distribution of load within a Multiple-strand chains is impaired by an increasing number of strands, hence transmittable output of a multiple chain do not increase on a linear scale with the number of strands. (If matched single chains are applied instead of multiple-strand chains, then the transmittable output will rise on a linear scale with the number of single strands used.)

| Number of Multiple-strand | chain strands factor: S |
|---|---|
| 2 | 1.7 |
| 3 | 2.5 |
| 4 | 3.0 |
| 5 | 3.5 |
| 6 | 4.0 |
| 8 | 4.5 |

The number of chain strands required can be determined after calculating the strand factor:

$$S = \frac{P_B}{P_{output}} \tag{2.3}$$

### 2.4.2. Chain Length

The provisional centre distance is either fixed in the required drive data or may be chosen with respect to the recommendation in chapter 2.2.
If there is no restriction, centre distance may be chosen between 30...50 times the chain pitch.

$$a_{prov} = 40p$$

The length of the chain:

$$L = np \tag{2.4}$$

where: n — number of the chain links
p — pitch of the chain

The number of the chain links:

$$n = \frac{2a_{prov}}{p} + \frac{z_1 + z_2}{2} + \frac{p}{a_{prov}}\left(\frac{z_2 - z_1}{2\pi}\right)^2 \tag{2.5}$$

The calculated number of pitches should be rounded up to an even, whole number of pitches. The exact centre distance pertains to the applied number of pitches must be recalculated.

$$a = \frac{p}{4}\left[\left(n - \frac{z_2 + z_1}{2}\right) + \sqrt{\left(n - \frac{z_2 + z_1}{2}\right)^2 - 2\left(\frac{z_2 - z_1}{\pi}\right)^2}\right] \tag{2.6}$$

### 2.4.3. Checking the chain for tensile strength

The force in the tight chain strand:

$$F_{\max} = \frac{P_B}{v} + qv^2 \quad [N] \tag{2.7}$$

where: $P_B$ — [kW] — design power
v — [m/s] — chain speed
q — [kg/m] — weight per metre of chain

The safety factor against tensile failure

$$n = \frac{Q}{F_{\max}} \geq 5 \tag{2.8}$$

where: Q — [N] — tensile force (see product catalogues)

## 2.5. Roller chain drive constructions

After calculating the chain drive and choosing the chain from the power rating tables or charts, all the data are available (operation circumstances, lubrication mode desired, centre distance, number of teeth of the sprockets, adjustable or fixed centre distance, need of applying idler) for designing and implementing the chain drive.

### 2.5.1. Sprocket constructions

The shape of the sprocket tooth and its groove is standardized.

The service life and the dependability of the chain drive depends greatly on the sprockets applied regarding the tooth profile, its material and heat treatment, the manufacturing process, the manufacturing accuracy with special regard to the pitch and the tooth dimensions.

Figure 2.8 shows different sprocket constructions.

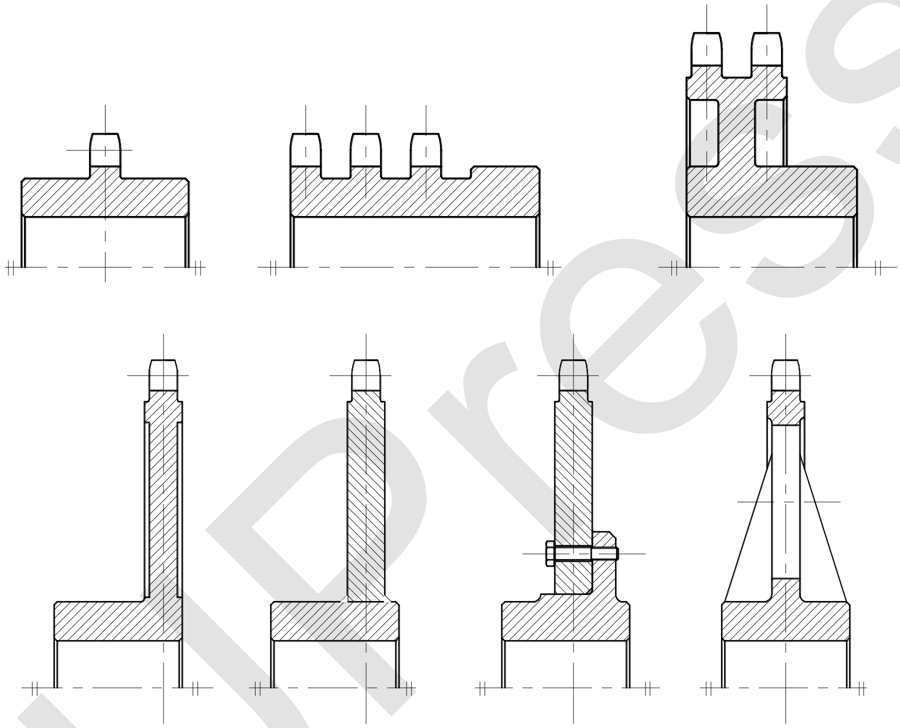


Figure 2.8 Sprocket constructions

In the case of small number of teeth $(9 \le z \le 15)$, the teeth and the hub are manufactured from the same material. In the case of higher number of teeth $(16 \le z \le 30)$, the teeth are manufactured on the crown. The crown, the hub, and the plate between them are manufactured from the same workpiece special in series production. In the case of unit production it is favorable if the hub is manufactured separately and connected to the crown either by welding or bolted joint. The advantage of the fixing by screwed joint is that the cost of replacing the crown is less than replacing the whole sprocket. Sprockets in extremely grate diameter are made in spoked construction.

The materials of sprockets are mostly steel and only in exceptionally cases are cast iron or plastic applied for small load and low speed. The teeth of steel sprockets must be hardened because it is subjected to dynamic load and abrasive wear. Therefore

quenching and tempering steel or case-hardening steel are applied in heat treated condition.
In common operation condition for case-hardening sprockets C15 or C20 carbon steel may be used. After heat treating the surface hardness of this material rages from 55 to 60 HRC and the tensile strength rages from 600 to 900 MPa. The cemented layer-thickness should be about 1 mm but less than 20% of the transverse width of the tooth. These sprockets are abrasion-proof but don't withstand to high impact load.
The teeth surface of sprockets made of quenching and tempering unalloyed steel, C45, C60 are induction hardened or incrusted by flame hardening. The hardness of the case is ranges from 45 to 50 HRC, therefore the tooth is less abrasion-proof but its tensile strength ranges from 1000 to 1300 MPa, so it is appropriate for higher load. The case layer-thickness is the same as for the case-hardening steel.
Quenching and tempering alloyed steels, Cr2, Cr3, Mn2, NCMo4, or case-hardening steels, BCl, BC2 are applied for precision sprockets subjected to very high dynamic load.
The average surface roughness of the teeth up to $v = 8$ [m/s] should be $R_a \leq 6.3 \mu m$ and $R_a \leq 3.2 \mu m$ for higher speed.

## 2.5.2. Casing

The appropriate lubrication of the chain drive can be provided by a chain casing. It may be drip, oil bath, oil bath with slinger disc or forced lubrication. When designing the chain drive with a casing, the distance of shafts is fixed. In this case idler sprocket or sliding shoe may be applied prestressed by a spring on the slack strand of the chain. The casing may be either a welded steel sheet construction or cast iron or plastic. The prestressing mechanism is supported by the casing but this part of it must be stiff enough to bear the reaction force.
The task of the casing is the following:

- prevent contamination from getting into the drive
- store the lubricating oil
- decrease the noise of the drive
- decrease the safety risk by covering the moving parts

Figure 2.9 shows a counter drive shaft with bearing support; with the sprocket of the chain drive and with the pulley of the belt drive.

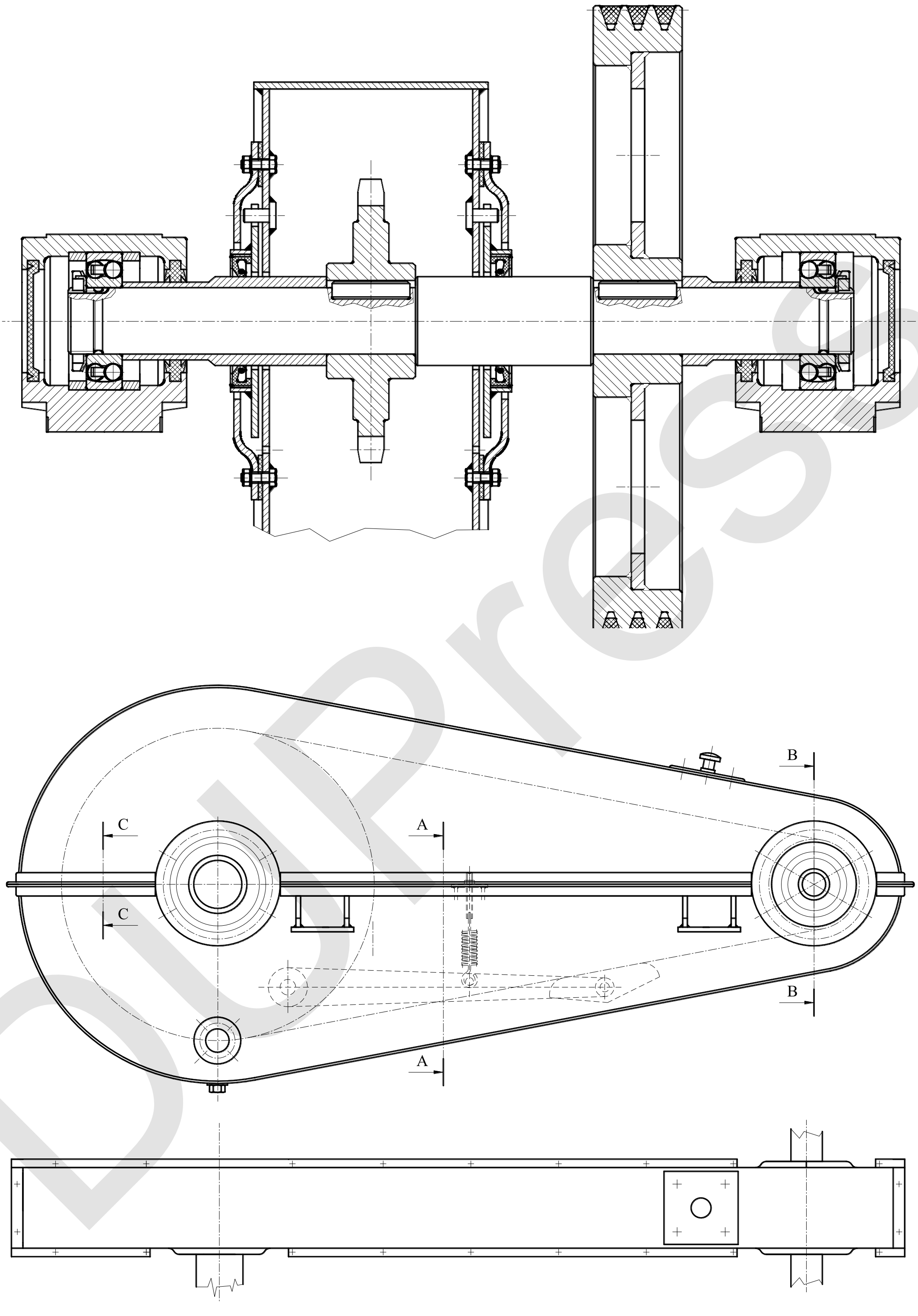



Figure 2.9 Counter drive shaft with casing

Designing the bearing support is detailed in the 3rd chapter. The chain rive is covered by a welded construction casing. The centre distance of the chain drive and hence the casing is fixed therefore a chain tensioner acting in the slack side is applied, see A-A section of Fig. 2.9 in Fig. 2.10.

A-A

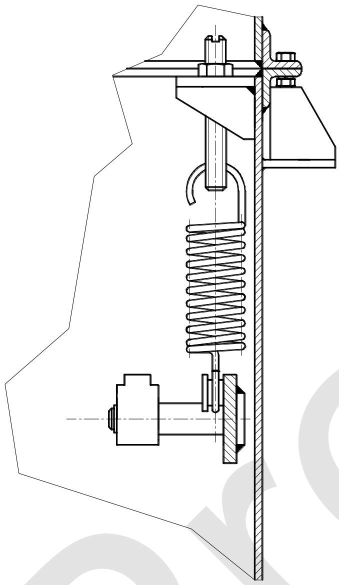


Figure 2.10 Chain tensioner

The recommended radius of the casing round the sprockets according to the empirical formulas are the following:

$$R_{1(2)} = \frac{d_{a1(2)}}{2} + p + 30 \quad \text{[mm]} \tag{2.9}$$

where: $d_a$ [mm] a tip diameter of the sprocket

$p$ [mm] a pitch of the chain

The inner width of the casing should be at least:

$$B = L + 60 \text{ [mm]} \tag{2.10}$$

where dimension of L for the applied chain can be found in product catalogues.

The distance between the casing and the strand should be 10% of the centre distance if the drive has no chain tensioner. When applying chain tensioner, this distance can be smaller.

**Constructions for sealing the casing**

The casing is built up commonly from a bottom part like a house and a top part like a cover. The bottom part accommodates the chain drive and oil is fixed to the

machinery base. The cover is fixed to the bottom part with bolted joints and sealed by packing material, see Fig. 2.9 C-C section in Fig. 2.11.
A part of the counter shaft is covered by the casing filled with oil have to be sealed to avoid leaking. If the casing is made of bent and welded metal sheets and the applied manufacturing processes do not ensure the coaxiality of the rotating shaft and the sealing seat, special solutions are needed.

- Fig. 2.9 D-D section shows a disc holding the seal ring in the seat, see Fig. 2.11. The disc is positioned to the casing by three pins allowing it to move in radial direction [22]. The correct orientation is provided by the sealing ring. The disc must be lightweight in this simple design to decrease the inevitable radial load of the seal lip.

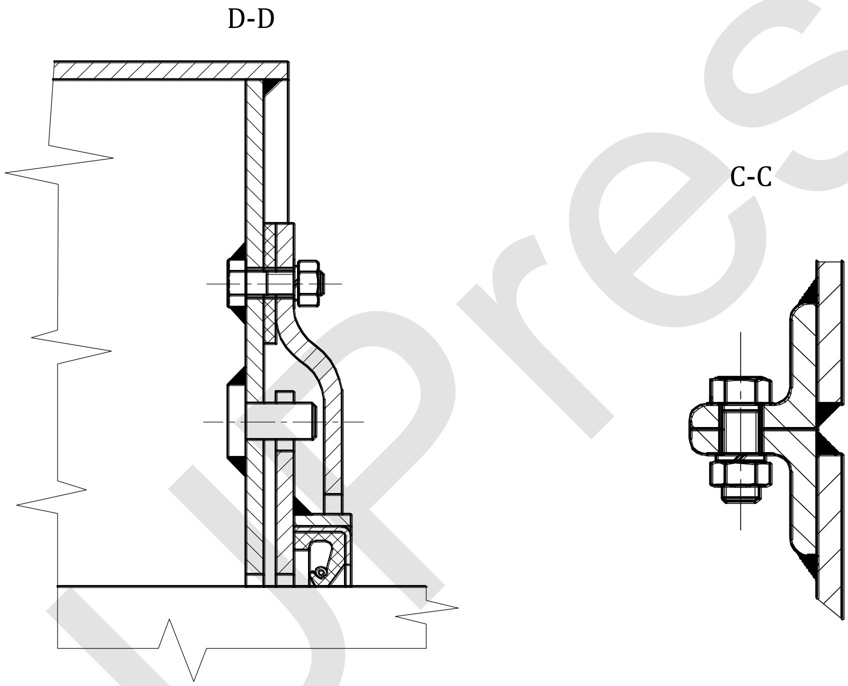



Figure 2.11 Casing seals

- The well experienced construction of the sealing between the transmission and the prop shaft is shown in Fig. 2.12. The inner CV joint’s boot fixed to the transmission holds a plastic bushing in which the prop shaft rotates and floats [23]. Actually, the bushing is stationary bearing accommodate the seal seat with seal ring. The bushing guides the sealing providing the coaxiality between the shaft and the sealing ring. The part of the boot overlapping the bushing acts a scraper ring as well.

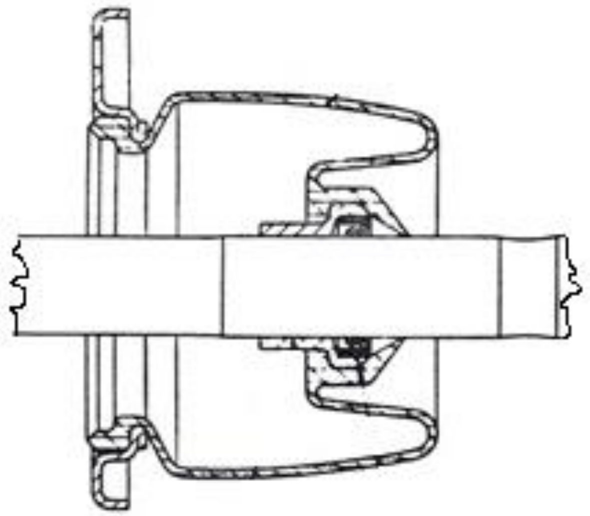


Figure 2.12 Shaft sealing

## 3. Shaft bearings

When designing shaft bearings it is a basic rule to support the shaft radially at least at two places, and to retain the shaft axially in both directions. For instance, the crankshaft bearings of the four-cylinder internal combustion engine, which is supported radially by three or five shaft bearings, is retained axially by sliding bearings in both directions, at one of the shaft bearings. If we carry out axial retainment in both direction with only one bearing, then this bearing will be the locating bearing, the rest of them will be non-locating (floating or dilatation) bearings. This bearing method is usually used for shafts with greater length to diameter ratio. Either the outer ring of the floating bearing can slide in the seat of the bearing housing, or the inner ring can slide along the seat of the shaft, or the axial displacement can occur inside the bearing, itself. Floating bearings are capable of carrying on only radial reaction forces. If axial retainment is ensured by two separate bearings, we talk about cross locating bearings. It is important to allow an adequate axial clearance between the two bearings, and to adjust it correctly during assembly in order to ensure play for dilatation caused by change in temperature. Usually, the axial load of shafts arises only from external loads, however there are bearing types, for instance taper-roller bearings, where the radial load can accommodate axial reaction forces due to the cone angle of the raceway; these reaction forces are supported by the other bearing built in "O" or "X" arrangement. In case of bearings of special design, these reaction forces arising from external or internal loads have to be calculated according to the instructions of the product catalogue [24].

There are several technical solutions for the same shaft bearing implemented with different type of bearings. The optimal construction may be designed considering the bearing stiffness, flexibility and the running accuracy requirements. The price of the rolling bearings depends on its type and dimensions. This is why if there is no special technical requirement then the previous aspect should be considered (most often applied rolling bearing is the single row deep grooving ball bearing).

## 3.1. Shaft bearing options

When designing shaft bearings we have to be familiar with the features of the available bearing types, its angular and radial stiffness. Bearings are grouped according to its line of action determining commonly the feasible supporting and retaining tasks.
The line of action is the line in which direction the rolling elements transmit the load from a ring (disc) to the other one. This way the load gets from the shaft to the bearing housing.
The angle of action is the angle between the line of action and the plane perpendicular to the axis of rotation.

Bearings with feasible line of actions:

1. Bearings having the line of action perpendicular to the axis of rotation: $\alpha = 0°$
   Eligible bearings: needle-roller bearing, cylindrical roller-bearing without supporting discs.
2. Bearings having variable angle of action: $\alpha = \alpha_{(F_a)}$

   The actual angle of action is determined by the ratio of the acting radial and axial bearing forces.
   Eligible bearings: deep grooved ball bearings, self-aligning ball and roller bearings.
3. Angular contact bearings: $0^0 \leq \alpha \leq 90°$
   The race way lays at a particular angle to the axis of rotation.
   Eligible bearings: angular contact ball bearings, taper roller bearings.
4. Bearings having the line of action parallel to the axis of rotation: $\alpha = 90°$
   Eligible bearings: Single and double row ball thrust bearings.

Bearings in the shaft bearing system have different tasks. The given tasks may be achieved by different bearing types with the appropriate supporting of their rings. When dismantling a shaft bearing assembly, we have to record the place and position of every component like: distance disc, distance piece, position of the bearings for the assembly. If the shaft bearing assembly is not reassembled properly, it causes the premature failure either the bearings or the parts assembled on the shaft, e.g. gear drive. It is expedient to make a rough sketch of the bearing construction revealed at dismantling, and we have to understand the operation of the shaft bearing to be able to carry out the reassembly professionally. Fig. 3.1 shows different bearing types with the appropriate ring retaining requirements for radial and axial loads. The lines of action represent the transmission of the bearing forces from the shaft to the bearing housing.

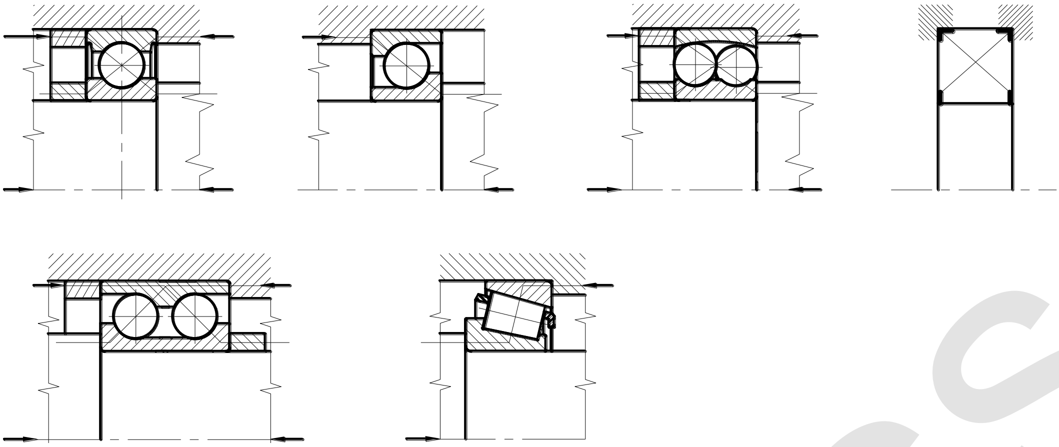
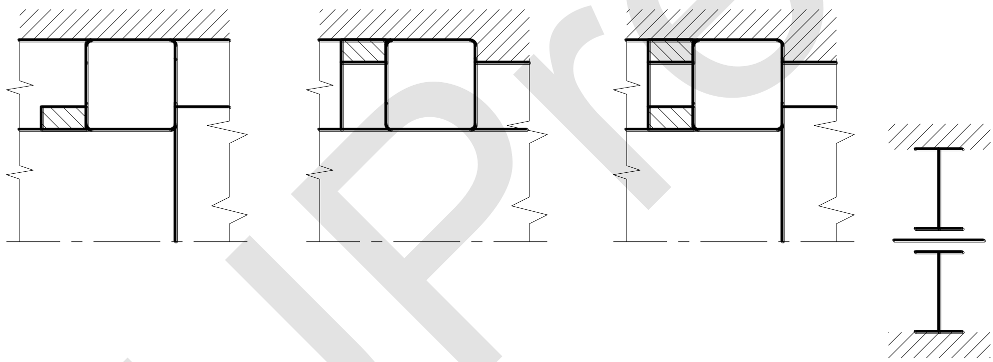


Figure 3.1 Eligible bearings with the lines of action

### 3.1.1. Radial supporting

The radial supporting works as a non-locating bearing.



1-2 bearings 3-4 bearings

Figure 3.2 Radial supporting

Some bearings of the shaft bearing assembly must be constructed to take up only radial forces. It can be implemented either by bearings appropriate only for taking up radial forces (e.g. cylindrical roller-bearings) or by variable angle of action bearings with appropriate retaining of the bearing ring in order to take up only radial forces (floating bearings).

Fig. 3.2 shows four bearings with the necessary bearing ring retaining acting as floating bearing.

1. deep groove ball bearing
2. self-aligning ball and roller bearings
3. cylindrical roller-bearing without supporting discs
4. needle-roller bearing

## 3.1.2. Radial supporting and both axial direction retaining

The bearing works as a locating bearing.
Locating bearings require the retaining of both bearing rings in both axial directions.
Eligible bearings:

- deep groove ball bearings
- self-aligning ball and roller bearings
- double row angular contact ball bearing
- cylindrical roller-bearing without supporting discs

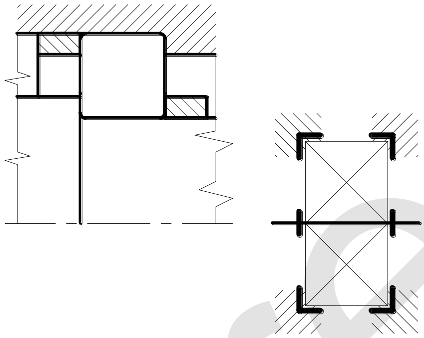


Figure 3.3 Locating bearing

## 3.1.3. Radial supporting and one axial direction retaining

The bearing works as a cross locating bearing.
Eligible bearings with appropriate retaining of bearing rings:

- deep groove ball bearing
- self-aligning ball and roller bearings
- single row angular contact ball bearings
- taper roller bearings

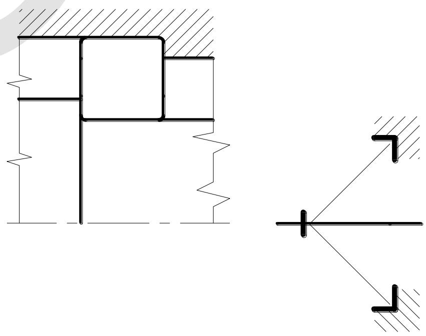


Figure 3.4 Cross locating bearing

## 3.1.4. One axial direction retaining

Pure axial direction retaining can be carried out with thrust ball bearings. When designing the bearing housing the bearing disc can't be supported radially in the housing, this is why the bearing can't transmit radial force to the housing. The other disc has to be fitted on the shaft of course.

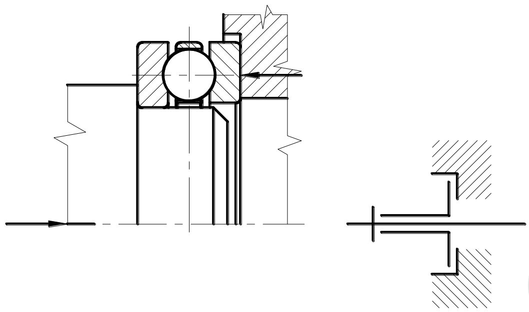


Figure 3.5 Bearing  retaining of a single row thrust ball bearing

## 3.1.5. Both axial directions retaining

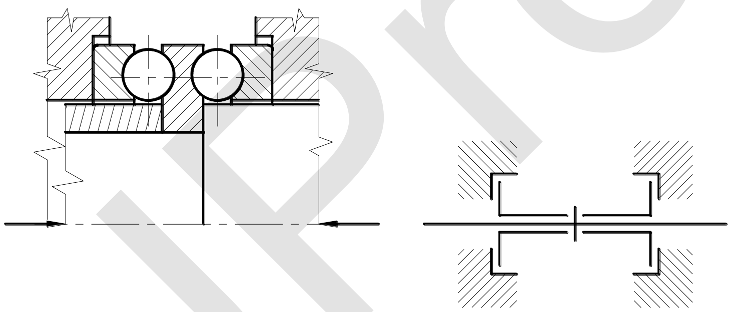


Figure 3.6 Bearing retaining of a double row thrust ball bearing

The central bearing disc has to be fitted and retained on the shaft in both axial directions by a shaft shoulder and e.g. distance piece. The two outer discs mustn't contact neither the bearing housing nor the shaft in radial direction.

## 3.1.6. Large angular stiffness shaft locating bearings

Eligible bearings for large angular stiffness shaft bearings:

- single row angular contact ball bearings in pairs
- taper roller bearings in pairs

The „O” and „X” arrangement designations refer to the position of the line of actions of the bearings, see Fig. 3.7 and Fig. 3.8.. Out of the two arrangements the „O” one has bigger angular stiffness due to the greater retaining distance.

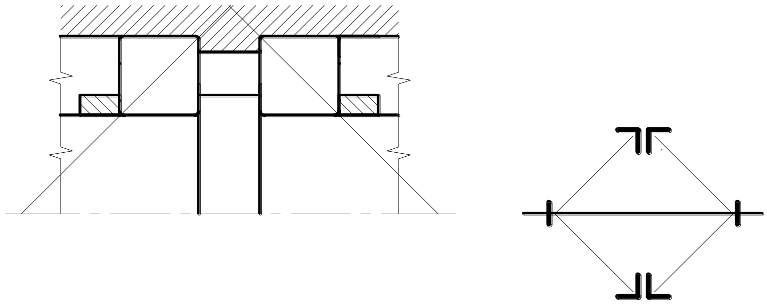


Figure 3.7 O arrangement shaft locating bearing

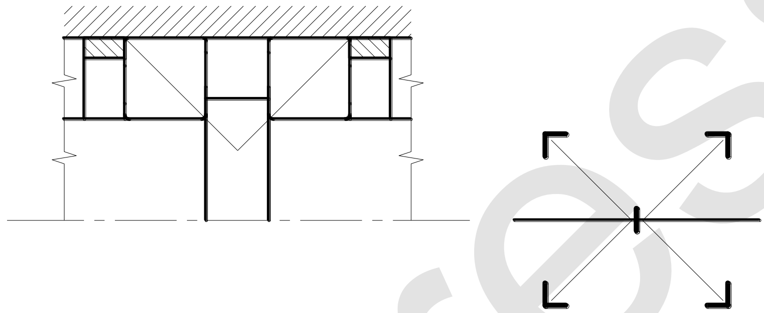


Figure 3.8 X arrangement shaft locating bearing

## 3.2. Shaft bearing constructions

The bearing of the shaft may be implemented by the appropriate application of the above detailed shaft supporting and retaining bearings. When designing the following rules have to be complied with:

- the shaft must be supported radially at least two places,
- the shaft must be retained axially in both directions implemented either by one bearing or in two different bearing places.

### 3.2.1. Locating and non-locating bearing arrangements

The locating and non-locating bearing arrangements are applied for shafts, having a great length in proportion to its diameter. Longitudinal dimensional change because of its dilatation is bigger than the working clearance of the bearings that could cause the shaft to get stuck. The required dimensional change is provided by the non-locating bearing allowing axial displacement. The locating bearing can be implemented either with one bearing (see Fig. 3.9.a) or with large angular stiffness shaft locating bearings (O or X arrangements, see Fig. 3.9.c and d), or the axial and radial loads can be shared with the separated bearings appropriate only for radial supporting and for both axial directions retaining respectively (see Fig. 3.9.b). The possible solutions are demonstrated by the line of actions of the bearings. The optimal solution from the feasible constructions may be chosen considering the order of magnitude of the bearing forces, the angular stiffness and the running accuracy requirements. The corresponding bearing type may be chosen with respect to the bearing rings retaining requirements.

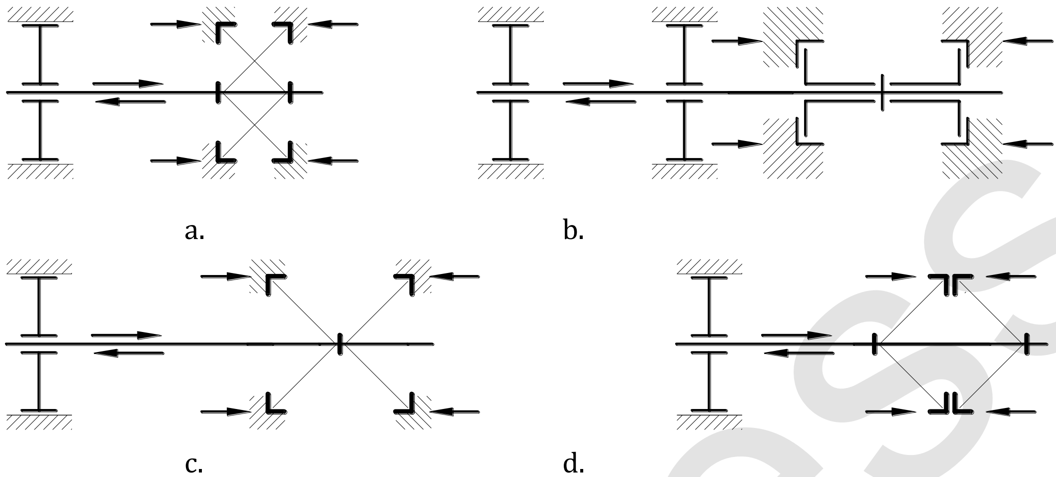



Figure 3.9 Locating and non-locating shaft bearing arrangements

## 3.2.2. Cross located bearing arrangements

These arrangements may be applied for short shaft length having a smaller longitudinal dimensional change due to its dilatation than the axial working bearing clearance. Considering the bearing forces the following arrangements may be applied:

- Cross located, floating bearing arrangements: In the case of moderate axial loads single row deep groove ball bearings or self-aligning ball (roller) bearings may be applied. The axial play of the shaft is adjusted by gaping the floating rings with the bearing housing caps.
- Cross located adjusted bearing arrangements: For greater axial load angular contact ball bearings or taper roller bearings may be applied in O or X arrangements. The axial play of the shaft is the adjusted bearing clearance. The bearing clearance is adjusted with feeler gauge when having easy access to the raceway and the rolling elements, or with dial gauge in the case of smaller bearings. When the construction is simply in design the bearing clearance may be adjusted by loosening the shaft nut with a specific degree in the knowledge of the thread pitch.
- Thrust ball bearings application individually for taking up the axial loads.

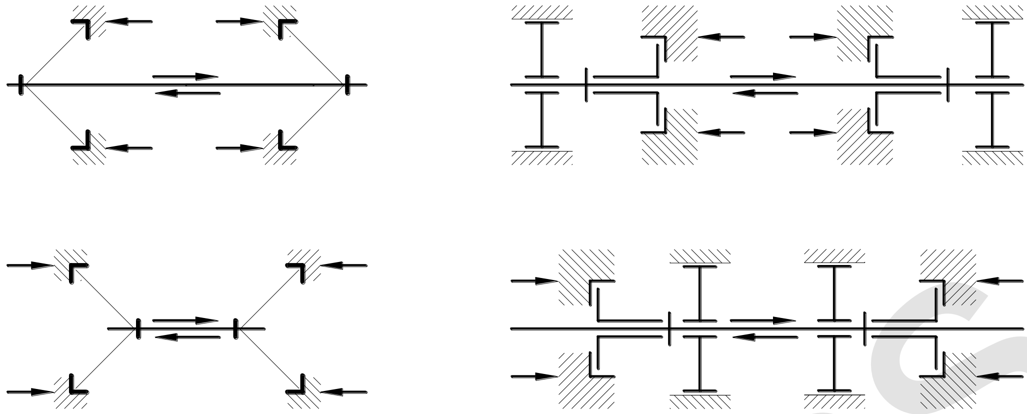
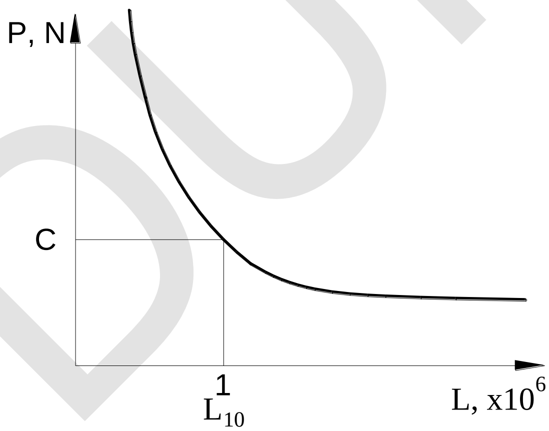


Figure 3.10 Cross located bearing arrangements

## 3.3. Bearing size based on life equations

The selection of rolling bearing is based on the magnitude of the load causing failure. The raceway of the bearing rings or discs are subjected to pulsating load because of revolving of the rolling element on it which may result in the fatigue of the raceway surface.

The selection of the rolling bearings with 90% survival probability is based on the S-N curve of the particular rolling bearing, see Fig. 3.11. If the bearing is well maintained and operates at moderate temperature, metal fatigue is the cause of failure alone. Namely the contact stress occurs on the raceways and on the rolling element are higher than the endurance limit of the material.




$$L_{10} = \left(\frac{C}{P}\right)^p$$

p=3 for ball bearings

$p=\frac{10}{3}$ for roller bearings

Figure 3.11 S-N curve of the rolling bearings

This is why the rolling bearing has limited life. The reason for any other bearing failure may be the not expert mounting, the operation not under the design circumstances which consequents are the overload, the overheating and the unsatisfactory lubrication condition.

The designed service life of the bearing depends on the application field of the machine determining the equivalent dynamic bearing load desired. The S-N curves of the different bearing types with different sizes are identified by two matching points. These are the C basic dynamic load rating and the $L_{10}$=1, the service life expressed in millions of revolutions. Accordingly, the servile life is 1 million revolution if the C is the equivalent dynamic bearing load.
Desired service life of the bearing depending upon the application field:

| | | |
|---|---|---|
| $L_h$ [hours] | 4.000 – 8.000 | intermittent operation |
| | 14.000 – 20.000 | 8 hours operation a day |
| | 50.000 – 60.000 | continuous operation |

The service life:

$$L = L_h\left[h\right]n\left[\frac{1}{min}\right]60\left[\frac{min}{h}\right]\frac{1}{10^6} \tag{3.1}$$

The basic dynamic load rating:

$$C = \sqrt[3]{LP} \quad \text{ball bearings} \tag{3.2}$$

$$C = \sqrt[\frac{10}{3}]{LP} \quad \text{roller bearings} \tag{3.3}$$

where: P [kN] equivalent dynamic bearing load

The method of determining the P equivalent dynamic bearing load for different bearing types based on the bearing forces are detailed in bearing catalogues [24].

# 4. Shaft design

The counter shaft transmits the power of the electric motor to the driven machine. The most important parameters of the drivetrain are the $P_n$ nominal performance of the prime mover and the rpm of it and the machine driven. The method and steps of designing an optimized drivetrain you may see in Drivetrain Optimization book [1].
The determination of the load of the drivetrain is very complex. The load-time pattern depends primarily on the operation of the prime mover and the driven machine, as well as the technological process being carried out. A load altering in time causes fatigue loading, so the machine parts have to be stressed for fatigue, for which the load amplitude needs to be determined.
From $P_n$ we can calculate only the nominal load, which is the mean value of the load-time pattern. Next, we will see how the load amplitude, required for calculating the stress amplitudes, can be determined in general.

Main causes for the development of load amplitude are the following:

- uneven torque supply and demand (operational features of the prime mover and the driven machine)
- dynamic features of the drivetrain elements

- nature of the carried-out technological process
- frequent starting

The above effects act as excitation effects in a flexible system susceptible to oscillation. The prerequisite of an accurate stressing is to know the load-time pattern as precisely as possible.

Methods for determining the loads:

- Measuring: most accurate, but can only be carried out on an operating machine
- Experience: with data measured on machines with similar parameters
- Load model: by creating a load model based on the nominal performance of the prime mover, and the measured data of machines with similar parameters.
- Calculation: with dynamic modelling [25]

In this chapter we are only focusing on the counter shaft stressing, starting with clarifying the load subjected to. If the drivetrain is designed for an operation out of the resonance range, the load-time pattern can be derived from the $P_n$ nominal power.

## 4.1. The load model

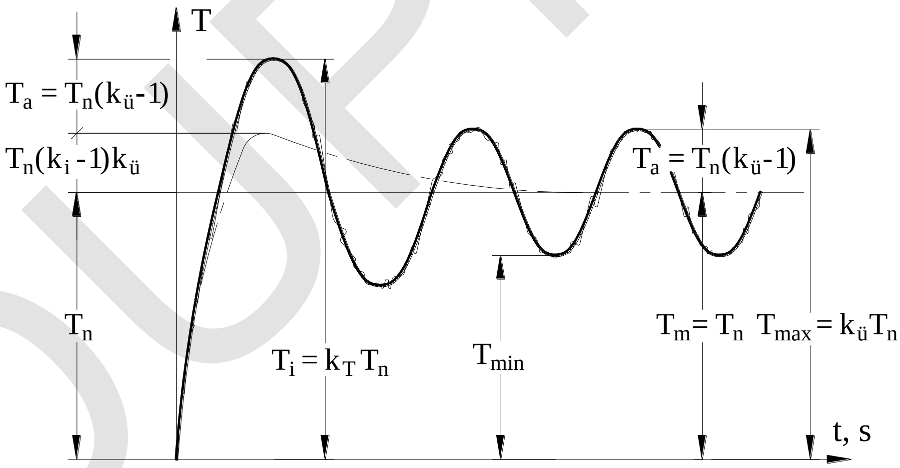



Figure 4.1 The load amplitude and the starting overload

In order to build up the load model we need $k_{\ddot{u}}$ operation factor and $k_i$ starting factor aside from the $P_n$ nominal performance. These values depend on the prime mover and the driven machine, so it takes technological process carried out into consideration. Their values have been measured on different machine groups, and then assembled in engineering directives. When designing a belt and chain drive, these factors are considered by the appropriate chose of $c_2$ factor (for the belt drive, see in chapter 1.3.2), and the Y impact coefficient (for the chain drive, see chapter 2.4.1).

The load at starting:

$$T_i = k_T T_n \tag{4.1}$$

where: $k_T$ overload factor

$$k_T = k_i k_{ü}$$

In non-transient operation the load altering with the time may cause fatigue, this is why we have to check the parts against fatigue. The load amplitude on the basis of the Fig. 3.1:

$$T_a = (k_{ü} - 1) T_n \tag{4.2}$$

The load alternating in time, in the case of a non-transient operation:

$$T_{(t)} = T_n + T_a \sin \omega t \tag{4.3}$$

where: $\omega$ – angular frequency of the excitation effects

By choosing the longitudinal dimensions of the shafts (which later can be modified) and by knowing the loads, the load diagrams can be drawn and the medium $\sigma_m$ and amplitude stresses $\sigma_a$ can be calculated. The most important requirement during stressing is that the shaft works without failure during its service life.

The part is safe to operate if the coefficient of safety is:

$$n = \frac{H_{(t)}}{T_{(t)}} \geq 1 \tag{4.4}$$

where: $H_{(t)}$ load limit

$T_{(t)}$ load

As the load limit (in this case the endurance limit) depends on the material and the load-time pattern, we have to apply a stressing method, considering this relation appropriately (c.f. Goodman and Smith diagram).

### 4.1.1. Load pattern of the rotating shaft

As the load amplitude is superimposing to the mean value calculated from the nominal performance, the load is fluctuating, see fig. 4.1. However, in the case of rotating shafts, cross-sections rotate, so the mean stress is zero and the stress-amplitude $\sigma_a = k_{ü} \sigma_n$, hence the stress is reversed.

### 4.1.2. Loads arising from belt drives

Belt drives are frictional connection drives. Performance transmitting is carried out by peripheral force acting on a given disc diameter. However, the peripheral force provides not only torque, but at the same time results in radial loads, as well. Radial loads may be taken into account with the pretention load of the shaft necessary for tensioning the belt (see Eq. 1.19). This has to be considered when drawing the load diagrams of the shaft, as well. When designing the belt drive we start out with the nominal performance, from which the transmittable performance can be calculated after defining the $c_2$ operation factor. The belt profile and the number of belts depend on the transmittable performance.

### 4.1.3. Loads arising from chain drives

In contrast with belt drives, the transmittable performance and peripheral force does not depend on the tensioning in the case of chain drives. It follows, that chain drives are operable without tensioning, and very often they are used without it. Chain tensioning is usually used in order to avoid chain swinging, to position the driving and driven shafts (cam shaft of internal combustion engines), or to avoid tooth skipping on chain drives with low depth of teeth. It follows from the above, that the nominal shaft load resulting from the chain drive (of course also taking into account the own weight) is calculated from the peripheral force arising in the tight chain strand, which is determined from the torque to be transmitted. After moving the vector of the peripheral force parallel with itself to the centre-point of the sprocket, we resolve it to its components, corresponding to the planes, determined for calculating the reaction forces of the shaft.
When using roller chains we have to take the recommended teeth numbers considering the polygon effect.

## 4.2. Steps of shaft designing

Preliminary design of the shaft:

1. Ascertaining the loads acting on the shaft, considering the belt drive and chain drives.
2. Building up the load model, starting from the nominal performance and assessing the operation factor.
3. Assessing the longitudinal dimensions of the shaft, considering its functions.
4. Drawing the load diagrams of the shaft and calculating the reaction force at the bearings.
5. Determining the critical cross sections and calculating the resultant value of the bending moments and the torque in the critical cross sections (the torque is known since the load diagrams are determined on the basis of it).
6. Material selection. The shaft diameter may be calculated from the material properties and from the allowed stresses.
7. Constructing the shaft (supporting and non-supporting shoulders, end and neck journals, key and splined joints).
8. Checking the stress calculations with the actual dimensions of the shaft. If it is necessary, modifying the dimensions and reconstructing the shaft. The

checking process has to go on with recalculations till the dimensions of the shaft are fixed.

Checking the shaft for:

1. Fatigue
2. Plastic deformation
3. Allowed elastic deflection
4. Critical speed

The object of this book does not cover the introduction of stressing and checking procedures acquired in the mechanics and machine elements courses. However without derivation we give some, often applied formulas for calculating the shaft diameter in the case of static load (preliminary design) and altering load (checking for fatigue).

Combined static load:

$$d_{min} = \sqrt[3]{\frac{\sqrt{M_b^2 + \frac{3}{4} M_t^2}}{0.1\sigma_{allowed}}} \tag{4.5}$$

where: $\sigma_{allowed} = \dfrac{S_y}{n}$, $n = 2.5 - 3$

Fluctuating and shock load:

Applying the maximum energy of distortion theory with modified Goodman fatigue criteria:

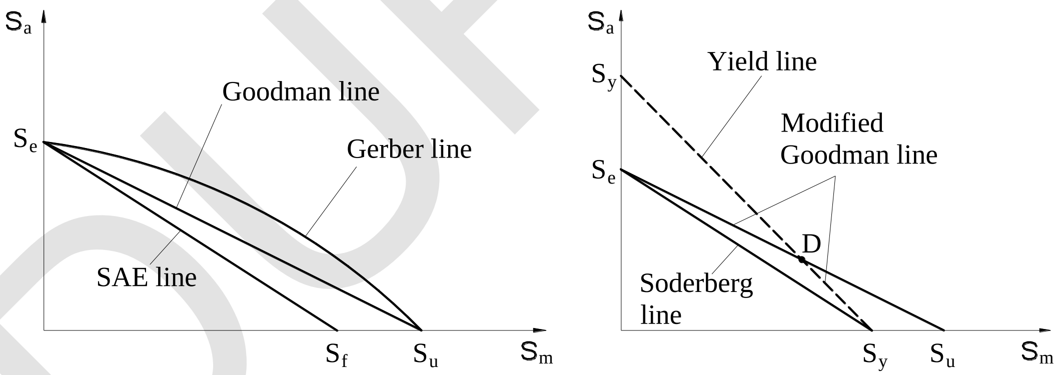



Figure 4.2 Fatigue diagrams

$$\frac{s_y}{n} = \frac{32}{\pi d^3}\left[K_{sb}\left(M_{b_m} + \frac{s_y}{s_e} M_{b_a}\right)^2 + \frac{3}{4} K_{st}\left(M_{t_m} + \frac{s_y}{s_e} M_{t_a}\right)^2\right]^{\frac{1}{2}} \tag{4.6}$$

where: $S_e'$ completely reversed endurance limit of the test specimen

$S_e$ completely reversed endurance limit of the a member

$K_{sb}$ and $K_{st}$ shock factor for bending and torsion

$K_{sb}$ and $K_{st}$ for steady load: 1

minor shock: 1.5

heavy shock: 2.0

For hollow shaft: d is replaced by $D^3\left[1-\left(\frac{d}{D}\right)^4\right]$

In the case of rotating shaft ($M_{h_m}=0$), the shaft diameter:

$$d^3=\frac{32n}{\pi S_y}\left[K_{sb}\left(\frac{S_y}{S_e}M_{b_a}\right)^2+\frac{3}{4}K_{st}\left(M_{t_m}+\frac{S_y}{S_e}M_{t_a}\right)^2\right]^{\frac{1}{2}} \tag{4.7}$$

# 5. Design and stressing steps of the counter drive

- Sketch the drive layout containing the position of the motor, the engine and the counter drive.
- Divide the total drive ratio into the belt and the chain drive considering the recommended ratios for them.
- Determine the drive parameters of the belt and the chain drives.
- Design the belt drive (Eq. 1.1 - Eq.1.23) and the chain drive (Eq. 2.1 – Eq. 2.10).
- Sketch the countershaft specifying the places of the shaft journals, the pulley and the sprocket.
- Draw the load diagram of the countershaft and stress it against fatigue and permanent deformation.
- Design the torque transmitting joints for the pulley and the sprocket hub on the shaft.
- Design the shaft bearing and select the roller bearings from brand catalogue (see chapter 3).
- Design and draw the casing of the chain drive providing the appropriate lubrication and the adjustment of the chain (see chapter 2.5.2).

# References


[1] Zsolt TIBA: Drivetrain Optimization Vol2. Lambert Academic Publishing 2016, ISBN 978-3-659-89543-2

[2] József Tóth Laboncz: Hegesztett alapkeretek Tervezési segédlet YMMF 1990.

[3] István HERCZEG: Szerkesztési Atlasz Műszaki Könyvkiadó, Budapest 1980. ISBN: 963 10 0790 1

[4] Rexroth Hydraulic Cylinder NFPA Industrial Type
http://www.boschrexroth.com/ics/content/allplatforms/zylinder/pdf/ra17041_2013-02.pdf
Downloaded: 07. 05. 2014.

[5] Simrit: Simmering and Rotary Seals Catalog
http://www.google.hu/url?sa=t&rct=j&q=&esrc=s&source=web&cd=1&ved=0ahUKEwjv4725tdfOAhVCJJoKHZnsAHQQFggzMAA&url=http%3A%2F%2Fwww.brammer.com.pl%2FDOWNLOAD%2FSimrit%2520Simmerings00001387.pdf%3Fdownload%3Dc4211bf4015cddaef4d20836151f0fe9&usg=AFQjCNFfiO9B5wgKUvayTtdKNPsv-03gQw
Downloaded: 07. 05. 2014.

[6] SKF: Hydraulic Seals
http://www.skf.com/binary/98-161853/Hydraulic-seals---12393_2-EN.pdf
Downloaded: 07. 05. 2014.

[7] Garlock: Mechanical Seals Technical Manual
https://www.google.hu/?gws_rd=ssl#q=mechanical+seals+technical+manual
Downloaded: 07. 05. 2014.

[8] Parker: Heavy Duty Hydraulic Cylinders
http://www.parker.com/Literature/Industrial%20Cylinder/cylinder/cat/english/HY08-1114-4-2H-3H-Heavy-Duty-Cyl.pdf
Downloaded: 07. 05. 2014.

[9] József Tóth Laboncz: Hidrohenger Tervezési segédlet YMMF 1990.

[10] Shigley J., Charles Mischke, Richard Budynas: Mechanical Engineering Design, 7th Edition Hardcover with access card, 1056 pages©2004, ISBN-13 9780072921939

[11] Ansel Ugural: Mechanical Design: An Integrated Approach, 1st Edition Hardcover with access card, ©2004, ISBN-13 9780072921854

[12] Ulrich ADLER et. al.: Automotive Handbook. Robert Bosch GmbH, 1993, Stuttgart. ISBN: 3-1-419115-X

[13] József Tóth Laboncz: Kétpofás Fék Tervezési segédlet YMMF 1990.

[14] Pintsch Bubenzer: Drum Brakes
http://www.technodyneengg.in/tdec_img/speed%20-o-controls/For%20PDF%201%20(1%20,2).pdf
Downloaded: 03. 07. 2017.

[15] EMG ELDRO: Electro-hydraulic thrusters

http://www.emg-automation.com/fileadmin/content/antriebstechnik/dokumente/Electro-hydraulic-thrusters_EMG_Flyer_EN_Rev00_06-2017.pdf
Downloaded: 03. 07. 2017.

[16] EMG ELDRO: Electro-hydraulic thrusters
http://www.mgh.be/assets/pdf/MGH-EMG-Eldro-Catalogus.pdf
Downloaded: 03. 07. 2017.

[17] József Tóth Laboncz Ékszíj és lánchajtások Tervezési segédlet YMMF 1990.

[18] BANDO: Industrial Power Transmission Products
http://www.bandousa.com/html/pdfs/bu-200_power_trans.pdf
Downloaded: 07. 05. 2014.

[19] Optibelt: Technical Manual V-belt drives
http://www.optibelt.com/fileadmin/content/pdf/Produkte/EN/Optibelt-TH-v-belt-drives.pdf
Downloaded: 07. 05. 2014.

[20] Link-Belt: Roller chain technology
http://www.laudorts.lv/static/pages/images/LinkBelt_GB.pdf
Downloaded: 03. 07. 2017.

[21] Rexnord: Roller Chains
http://www.rexnord.com/ContentItems/TechLibrary/Documents/7010_Rexnord-and-Link-Belt-Roller-Chains_Catalog-p.aspx
Downloaded: 07. 05. 2014.

[22] Simrit: Simmering and Rotary Seals Catalog
http://www.google.hu/url?sa=t&rct=j&q=&esrc=s&source=web&cd=1&ved=0ahUKEwjv4725tdfOAhVCJJoKHZnsAHQQFggzMAA&url=http%3A%2F%2Fwww.brammer.com.pl%2FDOWNLOAD%2FSimrit%2520Simmerings00001387.pdf%3Fdownload%3Dc4211bf4015cddaef4d20836151f0fe9&usg=AFQjCNFfiO9B5wgKUvayTtdKNPsv-03gQw
Downloaded: 07. 05. 2014.

[23] Schmelz, v. Seherr-Thoss, Aucktor: Gelenke und Gelenkwellen
Springer-Verlag Berlin Heidelberg 1988, ISBN 3-540-18322-1

[24] SKF General Catalogue
http://www.skf.com/group/knowledge-centre/subscriptions/displayfactbox.html?itemid=tcm:12-121486
Downloaded: 08. 01.2014.

[25] Zsolt TIBA: Dynamic driveline modelling Debrecen University Press 2010. ISBN 978-963-318-044-0