TECHNICAL UNIVERSITY OF LODZ

DESIGN BASIC OF INDUSTRIAL GEAR BOXES

Calculation and Design Case Example

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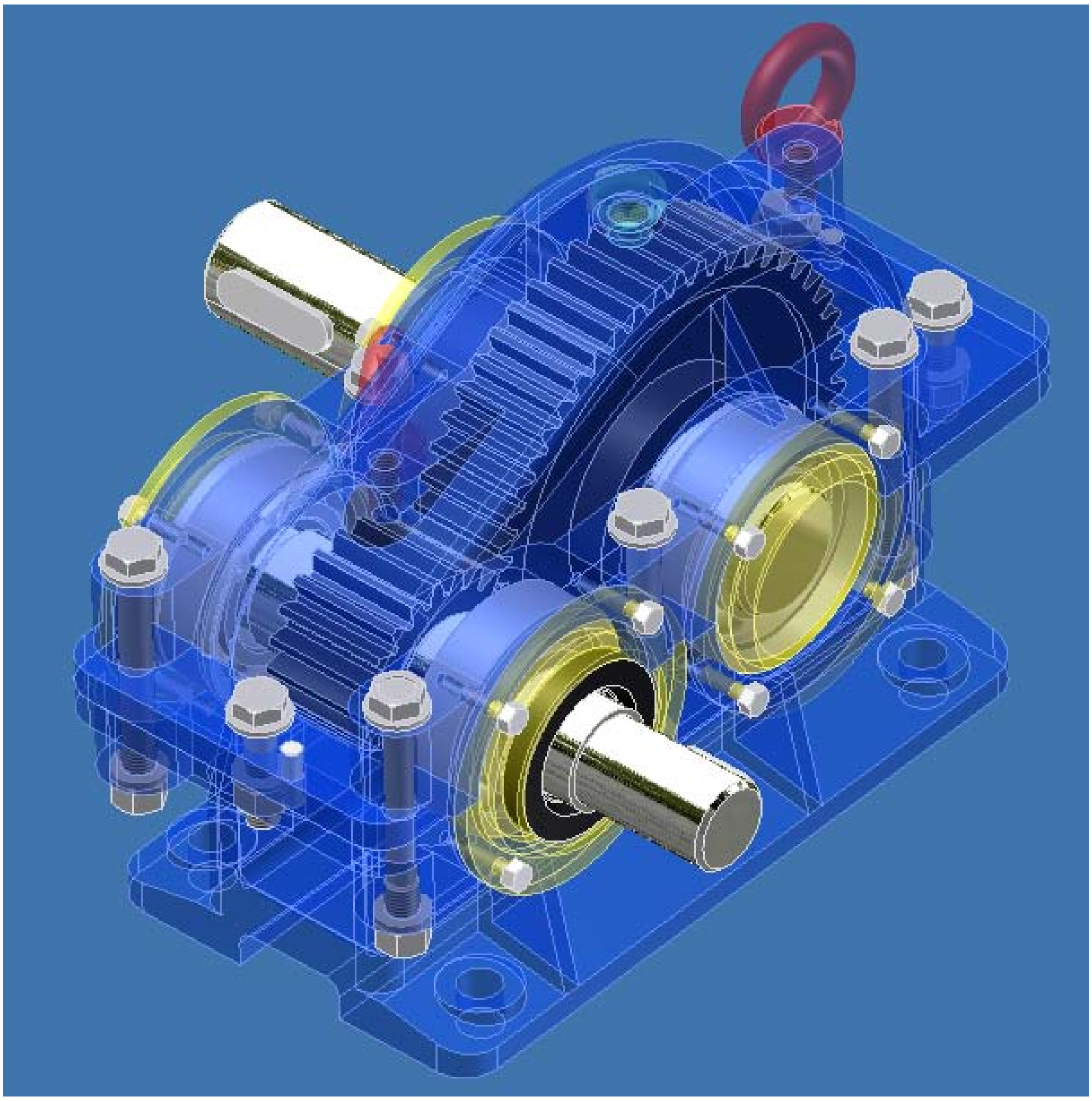

Department of Vehicles and Fundaments of Machine Design

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Chapter 1

BASIC KNOWLEDGE

8.3. Introduction

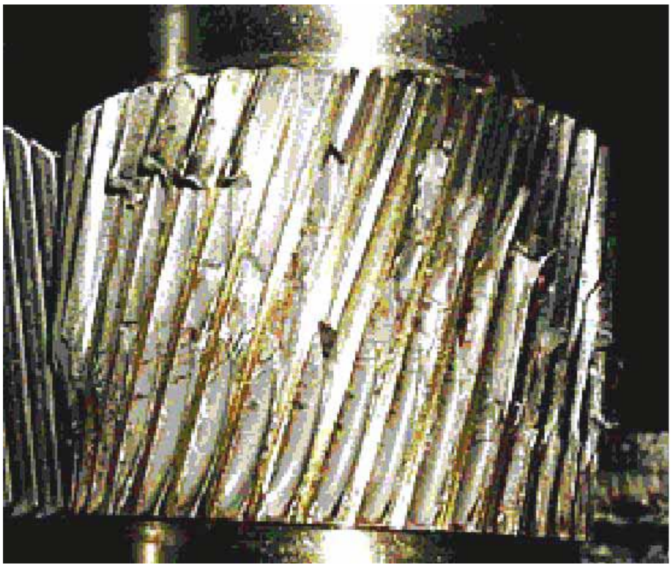
Gear reducers are used in all industries, they reduce speed and increase torque. You will find them between the prime mover (i.e.: electric motor, gas, diesel or steam engine, etc.) and the driven equipment: conveyors, mills, paper machines, elevators, screws, agitators, etc.).

An industrial gearbox is defined as a machine for the majority of drives requiring a reliable life and factor of safety, and with the pitch line velocity of the gears limited to below 25 m/s, as opposed to mass produced gearboxes designed for a specific duty and stressed to the limit, or used for very high speeds etc., e.g. automobile, aerospace, marine gearboxes.

To the competent engineer, the design of a gear unit, like any other machine, may seem a fairly easy task. However without experience in this field the designer cannot be expected to cover all aspects of gearbox design.

The purpose of this booklet is to set out the basic design for an industrial gearbox. It should help students not familiar with gearboxes, lay out a reliable working design. And it is intended for the reader to use his own experience in selecting formulae, stress values etc., for gearbox components.

To avoid the situation presented in the picture below, you should design gearing carefully and correctly.



Damage of helical teeth

1.2 Basic size and selection

The two types of tooth that can be used for both parallel and angled drives are straight or helical (spiral).
Spur gears are easier to manufacture and inspect than helical gears, and they can be rectified more easily at the assembly stage if required. The main disadvantage of a spur gear compared with a helical, is in the tooth engagement process. The whole of the spur tooth enters engagement at the same time, and therefore any pitch (spacing) error will cause interference and noise. Spur gears are generally used for pitch line speeds below 10 m/s in drives that are not loading the teeth to their maximum allowable limits. They are also used where gears are required to slide axially in and out of mesh.
Helical gears can be manufactured on most modern gear cutting machines. They will probably take longer to machine because of the relative wider face, and hence be more expensive than an equivalent size spur gear. However, this is offset by the fact that the helical gear may be capable of carrying up to fifty per cent more load. Conversely, for a given power, helical gears can be made more compact than a spur set. Helical gears are superior to spur gears in most applications, especially where noise must be kept to a minimum, or the pitch line speed is in excess of 10 m/s. These gears are also easier to design to fit given centre distances because there are more parameters that can be re-arranged. The main disadvantage of the helical gear is the axial thrust generated by the gears when working.
Double helical gearing has the same characteristics as the single helical but with the elimination of end thrust, as the two helices producing the thrust are cut with opposite "hands". This type of gearing is also useful where the pinions are of small diameter, as the equivalent face to diameter ratio is only half that of a similar net face single helical gear.
Bevel gears are used for drives requiring the input shaft to be at an angle, usually 90° to the output shaft. They can be cut with either straight teeth, where the same comments as for spur gears apply, or they can be cut spiral which correspond to the helical type of parallel gearing.
Gearboxes can be designed using the same type of gearing throughout, or a combination depending on powers, speeds and application.

TABLE 1.1. SUMMARY OF GEARING – COMMERCIAL GRADE GEARING

	Parallel Axis				Angled Gears			
	Finish cut ground		Finish cut lapped		Finish cut ground		Finish cut lapped	
	Spur Gears	Helical Gears	Spur Gears	Helical Gears	Straight Bevel	Spiral Bevel	Straight Bevel	Spiral Bevel
Max pitch line veloc. [m/s]	7	10	15	25	5	10	10	25
Efficiency per mesh	97%	98-99%	97%	98-99%	97%	98%	97%	98%
Power to weight ratio	Medium	Medium to high	Medium to high	High	Medium	Medium to high	Medium to high	High

1.3 Torque selection

Before starting the preliminary design, the following factors must be known.

- The type, powers and speeds of the prime mover.
- The overall ratio of the gearbox.
- The types of unit required – parallel or angled drive.
- The application.
- Any abnormal operating conditions.
- The disposition of the input to output shaft.
- The direction of rotation of the shafts.
- Any outside loads that could influence the unit, e.g. overhung loads, brakes, outboard bearing etc.
- The type of couplings to be fitted.
- Any space restriction.

To obtain the basic size of gearbox, the nominal torque at the output shaft is calculated, using the absorbed torque at the driven machine, or the prime mover torque multiplied by the gearbox ratio, if the absorbed torque is unknown.

It may be possible to obtain a torque – time diagram of the drive, which will give a comprehensive result of the complete duty cycle.

There are three important points to remember when calculating the nominal torque:

1. That if a brake is positioned anywhere before the gearbox output shaft, the unit should be sized on the brake torque, (assuming this torque is greater than the motor torque). This is because any external loads back driving the gearbox will be sustained by the unit until the brake slips. The above is also true of any form of back stopping (anti-reversing) device. A check should also be made on the kinetic energy that would have to be sustained by the unit if the brake is to be applied in an emergency.
2. That some prime movers, namely electric motors, can develop 2 or more times full load torque (FLT) on start up. If stop/start is a frequent occurrence then the gearbox must be sized accordingly.
3. Those rigid type couplings can transmit shock more easily to the gearbox than can flexible or gear type couplings, and the application factor selected accordingly.

To select the basic size, the nominal torque must be multiplied by a service factor (see Table 2). These are based on field experience and take into account the working conditions for that particular application.

It should also be noted that some motors can run at varying powers and speeds. The maximum torque is used for rating the gears for power based on an equivalent life to suit the duty cycles, while the maximum speed is used to ascertain the pitch line velocities.

Most manufacturers of gearboxes produce excellent free catalogues from which can be gleaned a lot of useful information, including approximate size of units for a given power, thermal ratings, shaft sizes, calculations etc.

TABLE 1.2. APPLICATION FACTORS

Example of Prime Mover	Driven Machine Load Classification		
	Uniform	Moderate Shock	Heavy Shock
Uniform Electric Motor Hydraulic Motor Turbine	1	1.25	1.75
Moderate Shock Multi-cylinder Petrol Engine	1.5	1.75	2.25
Heavy Shock Single-cylinder Petrol engine	1.75	2	2.5

The above figures are based on 10 hrs/day duty. For 3 hrs/day duty multiply above by 0.85. For 24 hrs/day duty, multiply above by 1.25.

NOTE – It is usual to equate a running time of 10 hrs/day to a total life of 22,000 hrs, and 24 hrs/day to 50,000 hrs.

Examples of driven machine classifications
Uniform: Generators, Constant Density Mixer.
Moderate Shock: Bucket Elevators, Concrete Mixers.
Heavy Shock: Stone Crusher, Sugar Mill, Steel Mill Draw Bench.

1.4 Materials and heat treatment

The steel selected for gears must be strong to prevent tooth breakages. It must be hard to resist the contact stresses, and it must be ductile enough to resist shock loads imposed on the gears, due to any outside influence or dynamics built up in the system. The material selected for gears, solid with shaft, must also be capable of resisting any stresses imposed along the shaft.
Through hardened pinions should be made approximately 40 BHN harder than their mating wheel to even out the life of the two parts with respect to fatigue and wear. Bar stock may be used for most industrial applications up to 300 mm dia., above this size forgings are usually used. In cases of high stresses it is advisable to purchase forgings as the structure is far superior to rolled bar. Stepped forgings can also be obtained and may offer a more economic alternative. Cast steel is often used for gear wheels but care must be taken to select a high quality material, devoid of blow holes etc.
Steel for gears is usually treated in one of the following ways:

Through hardened (including annealed or normalised)
The material is heat treated before any machining is carried out. This avoids any heat treatment distortion, but because it has to be machined, there is a limit to the hardness, and therefore strength, to which it is possible to go. Most gear manufacturers dislike machining steel over 350 BHN, as not only does it reduce tool life, it must also have an effect on machine life as well.
The most common steels (to PN-EN 10083-1+A1:1999) in this group is being C40, C45, C50, C55 and C60.

The final selection based on the allowable stress levels and the limiting ruling section involved.

<u>Flame or induction hardened</u>

The gear teeth are first cut into a gear blank, and then surface hardened. This retains the strong ductile core, while giving the tooth flanks a very hard wearing surface. On small teeth, of 4 module and under, the depth of hardening from both sides may converge in the middle and therefore make the whole tooth brittle (see Fig. 1). This is quite acceptable providing a slightly lower allowable bending stress is used for calculating the strength of the tooth, usually 80% of the allowable stress value of steel with hardness equal to that, of the root when in the unhardened condition.
Spin hardening, where the component is spun inside an induction coil, has the same effects as above. See Fig. 1C.

Fig.1A. Full contour hardened

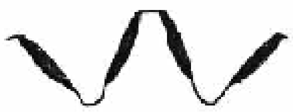


Fig.1B. Flank hardened

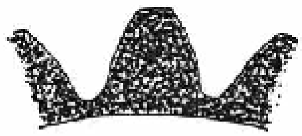


Fig.1C. Spin hardened

Because there is a certain amount of distortion due to the heat treatment, it is usual to leave a grinding allowance on the tooth flanks for grinding after hardening. Hardened gears can be left unground, but because of distortion, a certain amount of hand dressing of the teeth may be required to obtain an acceptable bedding mark when meshed with its mate. As hand dressing is a skilled, laborious job, it is best avoided if at all possible.
Full contour hardening (Fig.1A) hardens the flank and the root of the tooth, and this avoids the abrupt finish of residual stresses in the critical area as in the case of flank hardened teeth (Fig.1B). For flank hardened teeth, use only 70% of the allowable bending stress of steel with the same root hardness in the unhardened condition.
Flame or inductioned, hardened tooth flanks can, depending on the type of steel used, be expected to reach a hardness of 50-55 HRC at the surface and attain case

depths of up to 6 mm. It offers a strong tooth, easily hardened, and wheel rims of suitable steel can, with the proper procedure, be welded to mild steel centres.
Bevel gears are not usually induction hardened because of the tapered teeth, and if flame hardened, care must be taken to ensure that the flame does not damage the thin end (toe) of the teeth.
Suitable steels for flame or induction hardening include 34Cr4, 41Cr4, and 42CrMo4 (to PN-EN 10083-1:1999).

Nitrided
The teeth are finished cut to size in the blanks and are then 11isplac. This is a fairly low-temperature hardening process, and because of this, distortion is kept very low, and there are usually no corrective measures needed. The main disadvantages are,
a) the length of time for the process, which is usually a minimum of 80 hrs, and
b) the case depths obtained after this very long time are only in the region of 0.6 mm maximum, and would not therefore be suitable for heavily loaded large teeth.
Nitriding can give tooth hardness in the region of 68 HRC, which is one of the hardest surfaces available to the gear manufacturer.
Because this process involves subjecting the whole gear to the hardening effects, no further machining, except grinding, can be performed on the gear. Therefore any keyways or holes etc., must be machined into the component before nitriding. It is as well to remember to have threads masked during the process too, or these could become unacceptably brittle. As for any heat treatment process, do not plug holes that could cause expanding air to explode components.
Suitable steel will be 31CrMo12 or 31CrMoV9 (to PN-EN 10085:2003). This should be purchased in the hardened and tempered condition, and then stress relieved after roughing out.

Case carburised and hardened
The steel used is usually a strong, low carbon alloy steel, which after cutting the teeth, is subjected to a carbon rich atmosphere. The carbon is allowed to soak into the skin to a specified depth, and then the gear is hardened, quenched, and tempered. Not only does this hardening affect the case, but it also hardens the core material, giving an extremely strong tooth with a flank hardness of up to 60 HRC and case depth of up to 3 mm.
Because of the high temperatures and long soak times, carburised gears tend to suffer a great deal from distortion unless controlled, and sections should be left “heavy” and symmetrical so as to minimise distortion.
Careful consideration must be given to the manufacturing procedure of carburised gears, as the final hardness prevents any further machining operations except grinding. It is usual to pre-machine pinion shafts from the roughing out stage by turning the outside diameter of the teeth to size and leaving approximately 5mm (depending upon the required case depth) all over elsewhere. The teeth are then cut leaving a grinding allowance. It is then sent for carburising and annealing, and on return, the “unwanted” carbon is machined from the soft shaft. Key ways and holes etc. Can also be machined at this stage. The component is then hardened and tempered.
An alternative to machining the carbon from portions to be left soft is to mask the areas using a copper paint. The disadvantage being that a small scratch can let carbon seep in and maybe cause trouble at the final machining stage.

Threads etc. Should not be carburised, as they would become brittle during hardening and could cause a failure.
Wheels and certain shafts can be pre-machined, leaving just grinding allowance on the sides, teeth, and in the bore. They are then carburised and hardened in one go. The component has just to be ground all over and is then complete. All companies that undertake carburising would be only too happy to offer advice on the best procedure to adopt.
The steel purchased must be fine grain, and in the normalized condition. After any rough machining operation it should be stress relieved.
Common case hardening steels include 18CrMo4, 20MnCr4 and 18CrNiMo7 (to PN-EN 10084:2009).

TABLE 1.3. SUMMARY OF POPULAR GEAR MATERIALS

Through Hardened	Induction or Flame Hardened	Nitrided	Carburised & Hardened
C40 normalised or quenched & tempered	34Cr4	31CrMo12	18CrMo4
C45 normalised or quenched & tempered	41Cr4	31CrMoV9	20MnCr4
C50 normalised or quenched & tempered	42CrMo4		18CrNiMo7
C55 normalised or quenched & tempered			
C60 normalised or quenched & tempered			

1.5 The size of the unit

After calculating the nominal output torque and multiplying by the service factor, the size of the unit is estimated using chart # 1(see fig. 1.1). This chart gives approximate torques only for a given centre distance. The powers should be checked using the required standard e.g. AGMA, ISO, PN-EN, etc., and the gears re-sized if required.
If the input shaft and output shaft protrude on the same side, clearance between the two couplings must be checked to ensure they do not foul one another.
It is now necessary to determine the number of stages (reductions) that will be used to give the overall gearbox ratio. See Table 1.4.
On gear-sets with larger ratios (more than 4/1), the pinions can become slender with respect to their dia., and thus could present problems with bending and twisting. Therefore this should be checked as soon as possible.

TABLE 1.4. APPROXIMATE RATIO RANGES

NUMBER OF REDUCTIONS	OVERALL REDUCTION
Single Reduction	Up to 6/1
Double Reduction	5/1 to 25/1
Triple Reduction	25/1 to 125/1
Quadruple	125/1 to 625/1

Table 1.5 is used for determining the approximate centre distances for the rest of the gearbox and will give a fair distribution of economical gear size throughout the unit.

TABLE 1.5. CENTRE DISTANCES FOR MULTI-STAGE GEAR REDUCERS

Final Centre Dist. Mm obtained from chart # 1	Previous centres mm	Previous centres mm	Previous centres mm
1000	710	500	560
900	630	450	520
800	560	400	280
710	500	360	250
630	450	320	220
560	400	280	200
500	360	250	180
450	320	220	160
400	280	200	140
360	250	180	125
320	220	160	110
280	200	140	100
250	180	125	90
220	160	110	80
200	140	100	70
180	125	90	60
160	110	50	50

Note:
This table, based on BS.R20 progression, is for parallel gears only, or the final stages of a bevel/parallel gear set.

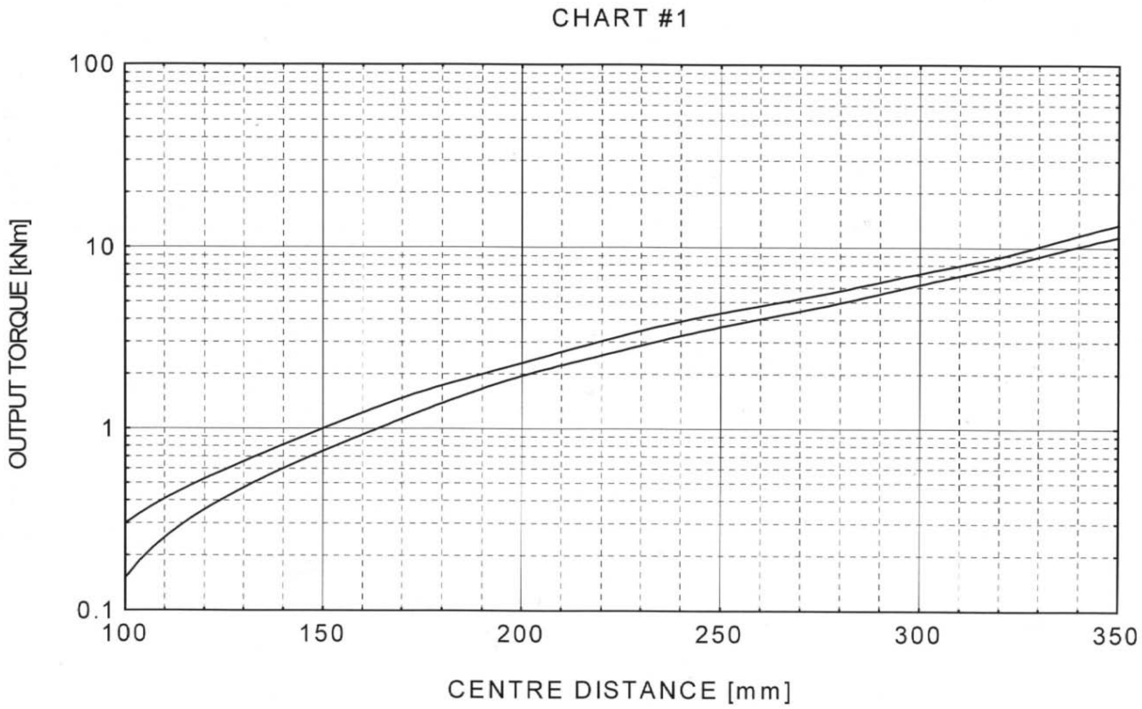


Fig.1.1. Output torque vs. Centre distance for spur gear reducers

1.6 Example

Estimate a center distance between gear axes of a one stage gear reducer with spur gears (see Fig.1.2) knowing:

Electric motor power $P = 22\ kW$,

Motor rotational speed $n_m = 1465\ rpm$,

Belt drive ratio $u_b = 2.4/1$,

Gear ratio $u_g = 3.95/1$,

Gear input operating conditions: *uniform*,

Gear output operating conditions: *moderate shock*,

A running time of the reducer: $t = 3\ hrs/day$.

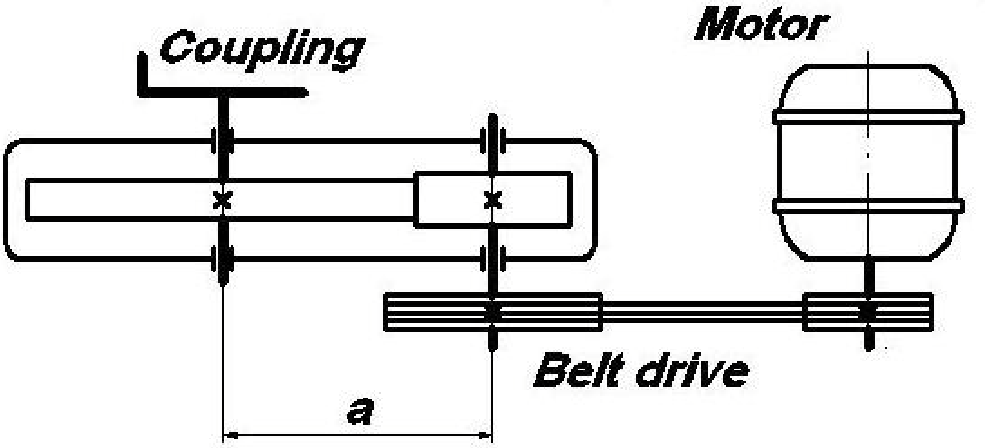


Fig.1.2. Sketch of the reducer

Output torque of the reducer is given by

$$T_{out} = \frac{30 \cdot P}{\pi \cdot n_m} \cdot u_b \cdot u_g \cdot k_s \qquad (1.1)$$

Where k_s is a service factor and

$$k_s = k_a \cdot k_t \qquad (1.2)$$

Where:

k_a is an application factor and in accordance with the Table 2 it equals 1.25 for gear input operating conditions as uniform and output operating conditions as moderate shock.

K_t is a duty factor and for a running time of the reducer of 2 hrs/day equals 0.85.

So

$$T_{out} = \frac{30 \cdot 22}{\pi \cdot 1465} \cdot 2.4 \cdot 3.95 \cdot 1.25 \cdot 0.85 = 1.445 \quad \text{[kN-m]}$$

From the Chart #1 we may evaluate that a centre distance of the reducer should be greater than 166 mm and less than 184 mm, on average – 175 mm.

Chapter 2

GEAR MESH

2.1 Ratios

The type of gearing, the number of stages and the centre distances to be used have already been established. The next step is to determine each single stage ratio to give the overall ratio of the gearbox. It is usual to have a slightly greater ratio at the high speed end and decrease the ratios towards the last reduction. For example a three stage unit of 42/1 may be made up of an input reduction of 4/1, a second stage ratio of 3½/1, and a final reduction of 3/1.

2.2. Tooth-pitch combinations

The tooth combination for each stage is selected and then checked for strength and wear.
The number of teeth for a pinion (the pinion being the member with the least number of teeth) is dependent on a number of factors, including duty, speeds, and hardness. The higher the speed or the smaller the ratio, then a greater number of teeth in the pinion will be required.

TABLE 2.1. APPROXIMATION OF THE NUMBER OF TEETH IN THE PINION

	PINION SPEEDS BELOW 1000 rpm			PINION SPEEDS ABOVE 1500 rpm			USUAL MINIMUM
	Hardness			Hardness			
	600 BHN	500 BHN	400 BHN	600 BHN	500 BHN	400 BHN	
SPUR	17	20	25	18	27	40	25
HELICAL	16	20	25	18	27	40	25
STR. BEVEL	13	18	20	15	25	30	20
SPIRAL BEVEL	12	18	20	15	25	30	20

Table 2.1 gives an approximation of the number of teeth in the pinion. If this is multiplied by the ratio of the reduction being considered, the number of teeth in the wheel will be obtained.

Table 2.2 can also be used to obtain the total number of teeth in the pinion and wheel for a particular ratio. If this is divided by the ratio plus one, then the teeth in the pinion can be found.

TABLE 2.2. TOTAL NUMBER OF TEETH IN PINION AND WHEEL

Ratio	PARALLEL AXIS GEARS		BEVELGEARS	
	Pinion speed below 1000 rpm	Pinion speeds above 1000 rpm	Pinion speeds below 1000 rpm	Pinion speeds above 1000 rpm
1/1	70	80	50	60
3/1	80	100	75	80
5/1	100	120	80	90

Much is written about the advantages and disadvantages of a “hunting tooth”. This is where an extra tooth is added or subtracted usually from the wheel to avoid an exact ratio. One of the advantages of a “hunting tooth” is that it prevents a wheel tooth contacting the same pinion tooth each revolution of the wheel, thus distributing a more even wear pattern. One of the disadvantages is that if a particular tooth on both pinion and wheel has an error, then because they will come together at some time the error will be magnified. Most industrial gearbox designs however, include for hunting teeth.

Prime numbers of teeth over 100 are best avoided, unless it is certain that the machine on which the gear is to be cut can produce that particular number.

Certain industries favour specific sizes of teeth, for example some steel mills or mining applications use low numbers of big teeth, to give a high factor of safety based on strength.

2.3. Pitch and module

To find the pitch or module of parallel axis gearing the following formulae are used. (When using single helical gears, use a helix angle of 12° for the first approximation. Most single helical gears have a helix angle of between 8° and 15°).

For Spur Gears a normal module is expressed as

$$m_n = \frac{2 \cdot a}{z_1 + z_2} \tag{2.1}$$

Where:

a centre distance between gear axes,

z_1, z_2 numbers of teeth respectively for pinion and wheel.

And for Helical Gears

$$m_n = \frac{2 \cdot a \cdot cos\beta}{z_1 + z_2} \tag{2.2}$$

Where β is helix angle of teeth.

TABLE 2.3. PREFERRED METRIC MODULES

1st choice	**1**	**1.25**	**1.5**	**2**	**2.5**	**3**	**4**	**5**	**6**	**8**	**10**	**12**	**16**	**20**	**25**
2nd choice	**1.125**	**1.375**	**1.75**	**2.25**	**2.75**	**3.5**	**4.5**	**5.5**	**7**	**9**	**11**	**14**	**18**	**22**	**28**

The results of the above formulae should give a Module as near as possible to one from table 2.3. Rearrange the formulae using the chosen Module to find the centre distance *"a"*. This should be smaller than the required gearbox centres by anything up to 0.4 Module (the gears being modified at the detail design stage). It may be found that the difference in the calculated centres and the designed centres is too great, in which case either the teeth, module or helix angle will have to be altered to suit. Sometimes this can be achieved by the addition or subtraction of one tooth from the wheel at the expense of a small alteration to the ratio.
For helical ones, both the teeth and the helix angle can be changed, remembering that by increasing the helix, the thrust and overturning moments on the bearings become greater, and the overlap ratio may increase to over the usual max of 1.9. On decreasing the helix check that the overlap ratio does not become smaller than 1.1

$$Overlap = \frac{face\ width \cdot sin\beta}{\pi \cdot m_n} \tag{2.3}$$

Another criteria which should be checked, is the contact ratio, which should be a minimum of 1.2.
For selecting the pitch of bevel gears, it is usual to start with the maximum wheel diameter that can be fitted without fouling any other shaft. From Tables 2.1 & 2.2 the wheel teeth can be estimated and therefore the pitch can also be found. Any alteration to diameter can be made after calculating the allowable loads to the required specification e.g. AGMA, BS.545, etc.
Spiral bevel gears, like helical ones, must have overlap to ensure smoothness of drive. Most industrial spiral bevel gears are manufactured with a spiral angle of 35°**,** to ensure that the face advance ratio (overlap) is greater than 1.2.

2.4 Example

For the reducer from Example of the previous Chapter 1 find a suitable module.
As it is easy to notice, the rotational speed of the pinion is

$$n_1 = \frac{n_m}{u_b} = \frac{1465}{2.4} = 610.4 \quad \text{[rpm]}$$

So this speed is below 1000 rpm and from Table 2.1 we may assume a tooth number of the pinion

$$z_1 = 24$$

Let us notice that a tooth number of the wheel is

$$z_2 = z_1 \cdot u_g = 24 \cdot 3.95 \cong 95$$

Using formula (2.1) we can write

$$m_n = \frac{2 \cdot a}{z_1 + z_2} = \frac{2 \cdot 175}{24 + 95} = 2.94$$

In accordance with Table 2.3 we select the nearest module to estimated value (from recommended first choice series) as

$$m_n = 3 \quad \text{[mm]}$$

For this module the modified centre distance between gear axes is

$$a_{mod} = \frac{m_n \cdot (z_1 + z_2)}{2} = \frac{3 \cdot (24 + 95)}{2} = 178.5 \quad \text{[mm]}$$

NOTE: Hardness of pinion teeth for discussed reducer should be 420 HBN.

2.5. Face-widths

The face-widths of pinions should be kept to a minimum to avoid bending and twisting. On helical gears, it must also be wide enough to give sufficient overlap. For spur and single helical gears, the maximum ratio of face-width to pinion diameter should be around 1 ÷ 1.3, but for double helical gears it may be twice this. Pinions should be slightly wider than their wheels, this ensures that full contact occurs in the event of any axial misalignment at the assembly stage, and also adds support to the ends of the pinion teeth.

For bevel gears, both pinion and gear usually have the same face-widths of up to a maximum of 0.3 times the outer cone distance. Wider face-widths on bevel gears will make the tooth at the toe (inside) unacceptably thin, and may also cause manufacturing difficulties.

The allowable transmittable power should now be checked to the required specification e.g. ISO, AGMA, BS etc.

If the gear-set is found to be grossly under or over-rated, the centres will have to be changed and new gears selected. If the allowable power is only marginally up or down, then the face-width may be altered, but with due consideration to the face/diameter ratio of pinions and overlap of helical gears. If the gears are part of a multi-stage unit, then by changing the ratio of the various stages, but keeping the overall ratio of the gearbox, the same size unit may still be used. For a given centre distance, decreasing the ratio increases the allowable torque and vice versa.

If it is found that there is a great deal of difference between the allowable powers based on strength and wear, then the pitch/tooth combination may be changed.

Increasing the numbers of teeth and making the pitch smaller, lowers the strength rating but may not increase the wear rating significantly.

2.6. Detail of gears

After selecting the teeth and pitch a more detailed look at the gears is required. One of the first subjects, that is usually brought up is correction. Positive correction is applied to gears for many reasons, and the four most important as applied to commercial, speed decreasing gears are as follows:
1. To increase the bending strength of a tooth.
2. To increase the diameter of a gear to enable a bigger bearing or shaft to be used.
3. When applied to pinions of speed decreasing units it increases the beneficial recess action of mating gears.
4. To fit a design centre distance.

Positive correction will make the tooth tip thinner, and this crest width should be greater than 0.3 of the Module for relatively soft material gears and 0.5 of the Module for case hardened or brittle gears. If the correction required gives an undesirably thin crest width, then the outside diameter of the gear can be turned down (topped).
In cases like these, the contact ratio and contact geometry must be closely watched.
It is usual to give positive correction to pinions and negative correction to wheels. The maximum correction per side in industrial gearing is in the region of ± 0.4 m_n.
For speed increasing gears, because it is the wheel that is driving, then to cut down the “harmful” approach action, only enough correction to avoid undercutting should be added to the pinion. In some cases the pinion correction may be negative, here again contact geometry is most important.
Correction is applied to gears by cutting the tooth in a gear blank who’s outside diameter is either smaller (negative correction), or larger (positive correction) than “standard”. It can be applied by any gear cutting operation at no extra cost with standard tools except form milling, in which case the cutter must have the form of the “corrected” tooth.
Backlash must be cut into industrial gears for two reasons. The first one is to cater for any errors in the gears, and the second one is to allow for any dimensional changes due to temperature. The following formula gives a general guide to backlash, but consideration must be given to the application of the gearbox.

$$Minimum\ normal\ backlash = 0.03 \cdot m_n + 0.05\ mm \qquad (2.4)$$

For gears that change rotation or have a torque reversal with the same rotation, then the backlash should be kept to a minimum. Whilst for unidirectional gears under uniform load the backlash may be increased.
It must be noted that the backlash will be increased due to case machining and bearing tolerances, and this must be catered for when tolerancing the gear teeth.
On pinions of small ratios, around 2.5/1 and under, it may be more economic to make the pinion loose, and fit it to an ordinary carbon steel shaft. The bore must be kept below 70% (approximately) of the root diameter of the pinion, to keep the key or shrink fit from affecting the tooth root stresses. If a shrink drive is being used, then the hoop stress should be taken into account when calculating the tooth root stresses. It is usual to reduce the allowable tooth bending stress to 80%, if using interference fit on gears with thin rim sections. Shaft stresses can also present

problems on this type of design, and a rough check is best carried out as soon as possible.
Helical gears in multi-stage gearboxes are always arranged so that the thrust from the pinion on a particular shaft will oppose the thrust of the wheel on the same shaft, assuming one gear is driving and the other being driven. This is done by making both gears the same handing.
Double helical gears can be created by putting two single helical gears back-to-back. As long as there is a minimum gap between the gears of 5 mm, and the keyways are cut when the teeth are lined up, then the centreline will not move too far from the theoretical, and because of the gap, opposing faces of pinion and wheel will not foul. This design of double helical gear will generate end thrust on each element which must be resisted, usually by the bearing retainers, unless the wheel rotates in one direction only, then the thrust from both elements can be arranged to oppose each other.
It must be remembered that on double helical gears, the total face width is made up of two smaller faces and the overlap ratio should be calculated on these, and not on the total face-width. Because of the relatively thinner faces, double helical gearing have higher helix angles, usually 30°, and overlap ratio is of 2 or more are not uncommon.
Solid double helical gears should have a gap for tool clearance if cut by the hobbing method. This also applies to planning if the cutter has to be thrown over to suit the helix angle. A small gap will also allow oil to escape from the mesh.
Bevel gear design present additional problems, and should always be drafted out to supplement the first calculations. The most noticeable problem will be the shaft bending stress behind the pinion, especially on ratios higher than 3/1. Cutter runout must also be checked, and this is quickly done by extending the line that represents the root of the tooth, and keep all bosses or protrusions inside this line. On overhung pinions, the diameter behind the pinion should be greater than the "overhang", that is, the distance between the centre of the bearing and the centre of the face width. The centre distance of the bearings should be at least three times this "overhang".
The resultant thrust on bevel gears should always be arranged to move the pinion out of mesh, this ensures that the gears never close due to axial clearance in bearings.
Bevel gears should have a minimum of two full tooth depths of material under the teeth to give sufficient strength and support. They should also have radial and axial reference surfaces machined into the blanks before cutting, for later inspection purposes.
If gears are required to be hardened, then the depth of hardening should be stipulated. This depth should be twice the depth of the sub-surface stresses. Table 2.4 gives an approximation of total depth based on pitch. For carburised gears this is usually the depth at which the carbon content of the case and core are equal. When stipulating case depths are on drawings, any grinding allowance must also be added to these depths.

TABLE 2.4. TOTAL CASE DEPTHS (Depths at which carbon content in case and core are equal)

m_n	**Depth (mm)**	m_n	**Depth (mm)**
2	0.48	12	2
4	0.96	16	2.3
6	1.27	20	2.5
8	1.52	25	2.7
10	1.65		

The tooth crest must be wider than twice the case depth, otherwise the tip becomes brittle and may not support the load.

Gears that are fitted to shafts, even if keyed, should have a slight interference fit to ensure concentricity when assembled. A minimum of 0.05 mm is quite usual. Because helical gears and bevel ones produce thrust, the gears must always be located on their shafts; this can either be done by spacers or by an interference fit on the shaft.

Gears of below 750 mm diameter are mostly manufactured from solid forgings, while above this size it is usual to weld or shrink a forged rim to a fabricated centre, or use a casting. If fabricated, care must be taken in the welding procedure, as the gear hub is usually mild steel and the rim is an alloy steel. A typical procedure is carried out with low hydrogen welding rods and the components kept at around 250° C until welding is complete. It is then stress relieved at 600° C before the welding temperature falls below 200° C. As can be seen by the temperature, the welding must be carried out by a robotic welding machine. An alternative method is by "buttering" a layer of low carbon weld metal to the low alloy rim and then finally welding the centre to this weld metal at much reduced temperatures.

All gears that have had rough machining operations carried out on them should be stress relieved before finishing.

Some production procedures require "chucking" pieces on the ends of shafts to enable them to be machined. These are removed later, but they must be included when ordering the forgings.

If a pinion shaft is hobbed, then this can allow the gear teeth to be actually cut below the bearing, or other diameters if required.

All gear blanks should be offered to the gear cutting department, clean and concentric, with any reference surface they may require. They should have the edges chamfered or filleted to avoid bumps, and be furnished with some form of handling holes. These are usually tapped in the sides ready to receive eye-bolts. It may be beneficial to lighten the blank by machining a recess in both sides, this not only lightens the gear, but it will slightly reduce its inertia, which in turn reduces the dynamic forces when running.

Large gears, especially cast, should be statically balanced. It must be remembered to cut spiral or helical gears to the opposite hand of their mates.

If gears are to be ground after cutting, then the teeth should ideally be cut with perfect cutters, this leaves a small undercut in the root of the gear, and subsequent grinding of the flanks blends the two together without touching the root. However, there are many occasions when, because of the unavailability of the correct hob, the teeth are pre-cut with standard cutters and then ground. This means the grinding wheel will have to grind into the root and may leave a “step”. The fillet radius will probably also be a lot smaller than a properly cut tooth aid all of this must be taken into account at the “design for loading” stage.

Grinding into the root of a surface hardened gear may release the residue compressive stresses, thus making the bending strength no better than the core material, and also create a serious stress raiser.

TABLE 2.5. SPUR GEAR DESIGN FORMULAE

To Obtain	From Known	Use This Formula[1)]
Pitch Diameter	Module and Number of Teeth	$D = m \cdot z$
Number of Teeth	Module and Pitch Diameter	$z = \frac{D}{m}$
Outside Diameter	Module and Pitch Diameter or Number of Teeth	$D_0 = D + 2 \cdot m = m(z+2)$
Root Diameter	Pitch Diameter and Module	$D_R = D - 2.5 \cdot m$
Base Circle Diameter	Pitch Diameter and Pressure Angle	$D_b = D \cdot cos\alpha$
Base Pitch	Module and Pressure Angle	$p_b = m \cdot \pi \cdot cosa$
Tooth Thickness at Standard Pitch Diameter	Module	$t_{std} = \frac{\pi \cdot m}{2}$
Center Distance	Module and Number of Teeth	$a = \frac{m(z_1 + z_2)}{2}$
Contact Ratio	Outside Diameters, Base Circle Diameters, Center Distance, Pressure Angle	$n_p = \frac{\sqrt{\frac{D_{01} - D_{b1}}{2}} + \sqrt{\frac{D_{02} - D_{b2}}{2}} - a \cdot sin\alpha}{m \cdot \pi \cdot cos\alpha}$
Backlash (linear)	Change in Center Distance	$bl = 2 \cdot D \cdot a \cdot tan\alpha$
Backlash (linear)	Change in Tooth Thickness	$bl = D \cdot t_{std}$
Minimal Number of Teeth for No Undercutting	Pressure Angle	$z_c = \frac{2}{sin\alpha}$

8.. All linear dimensions in millimeters

Chapter 3

SHAFT LOAD CALCULATION

8.3. Design description

Figure 3.1 shows a sketch of a spur gear reducer. A high speed transmission shaft supports a spur pinion I and pulley (A). The shaft is mounted on two ball bearings (B) and (D). The diameters of the pinion and pulley are 72 and 300 mm and their widths are 90 and 110 mm respectively. 22 kW power at 610.4 RPM is transmitted from the pulley to the pinion. Next the power is transmitted at gear ratio of 3.95/1 from the pinion to the gear (F) and the half coupling (H) which they are fixed on a low speed transmission shaft mounted on two ball bearings (E) and (G). F_{max} and F_{min} are the belt tension, while W_t and W_r are the tangential and radial components of the gear-tooth force W. Determine loads of the shafts and their minimal diameters.

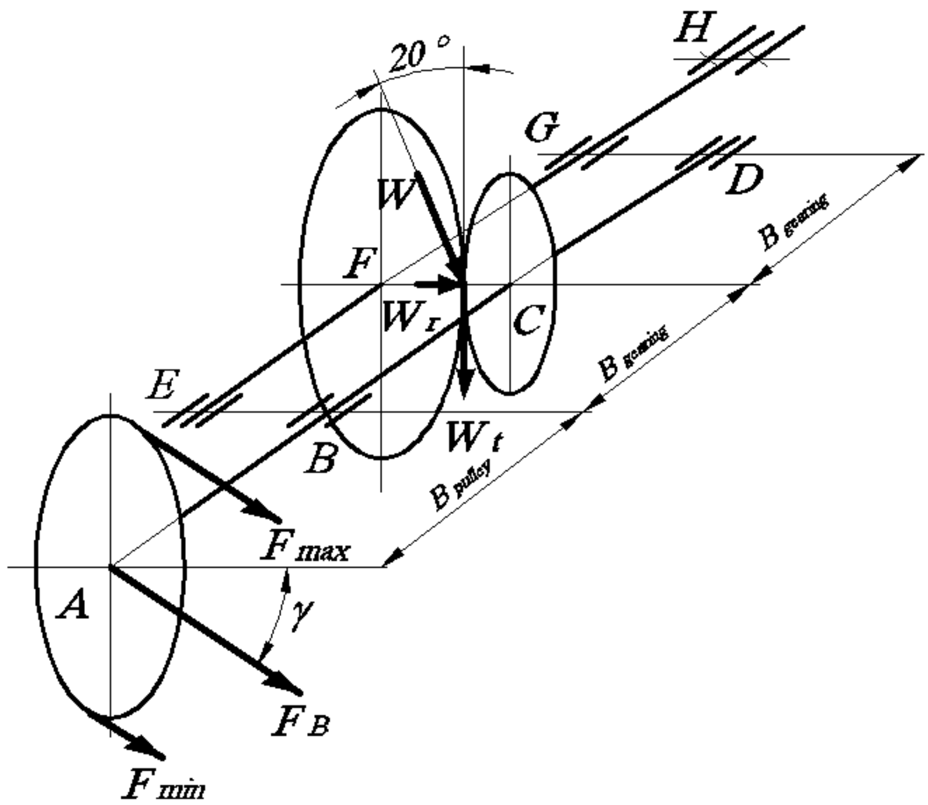


Fig.3.1. Sketch of a spur gear reducer

3.2 Given Data

P = 22 kW; n_{in} = 610.4 RPM, D_{pinion} = 72 mm, D_{pulley} = 300 mm, u_g = 3.95/1

B_{pinion} = 90 mm, B_{pulley} = 110 mm, γ = 30°.

3.3 Transmission torque

The torque transmitted by the high speed shaft is given by:

$$T_{in} = \frac{30}{\pi} \cdot \frac{P}{n_{in}} = \frac{30}{\pi} \cdot \frac{22}{610.4} = 0.344 \quad \text{kN-m}$$

3.4 V- Belt pulley loads

On the other hand

$$T_{in} = (F_{max} - F_{min}) \cdot \frac{D_{pulley}}{2}$$

and

$$(F_{max} - F_{min}) = \frac{2 \cdot T_{in}}{D_{pulley}} = 2.29 \ \text{[kN]} \tag{3.1}$$

It is normally taken to be for V-belt drives

$$F_{max} = 5 \cdot F_{min} \quad \text{(approximately)} \tag{3.2}$$

So, from (3.1) and (3.2)

$$5F_{min} - F_{min} = 2{,}29 \ \text{[kN]}$$

and

$$F_{min} = \frac{2.29}{4} = 0.57 \quad \text{[kN]}$$

Then the total force exerted by the belt(s) on shaft can be found as:

$$F_B = F_{max} + F_{min} = 6 \cdot F_{min} = 6 \cdot 0.57 = 3.42 \text{ [kN]}$$

NOTE: Detailed calculations of V-belt drive are given in Chapter 7.

3.5 Spur pinion loads

Pressure angle equals $20°$. From Fig.3.1

$$T_{in} = W_t \cdot \frac{D_{pinion}}{2}$$

$$W_t = \frac{2 \cdot T_{in}}{D_{pinion}} = \frac{2 \cdot 0.344}{0.072} = 9.556 \text{ [kN]}$$

$$W_r = W_t \cdot tan20° = 9.556 \cdot 0.364 = 3.478 \text{ [kN]}$$

$$W = \frac{W_t}{cos20°} = \frac{9.556}{0.94} = 10.62 \text{ [kN]}$$

3.6 Free body diagram of the high speed shaft

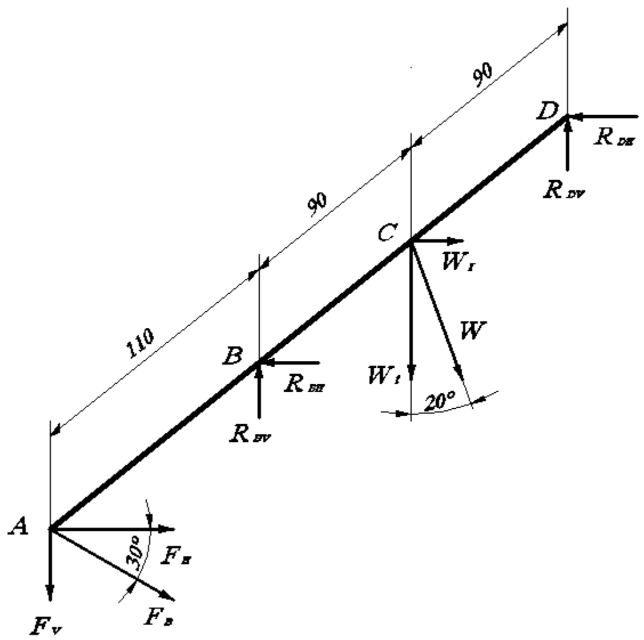


Fig.3.2. Free body diagram of the high speed shaft

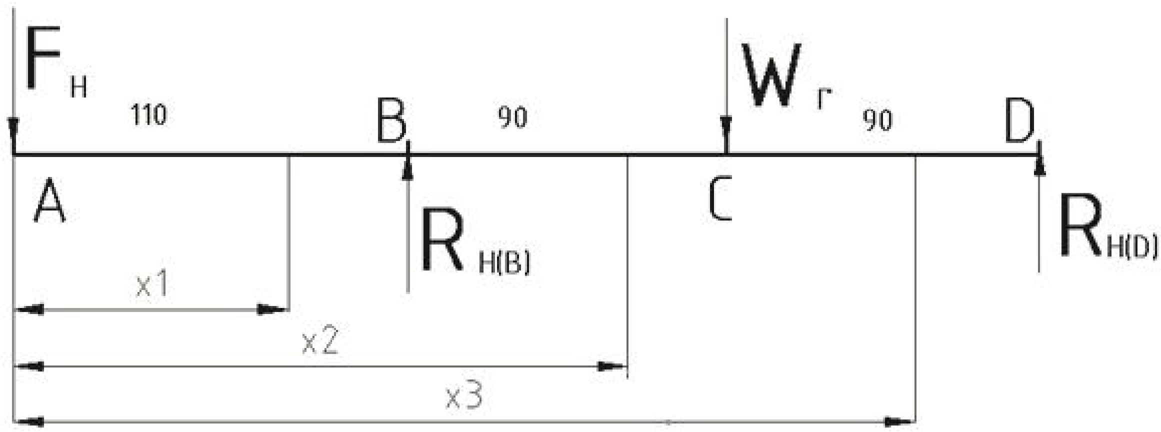
Here

$$F_H = F_B \cdot cos\gamma = 3.42 \cdot cos30° = 2.96 \text{ [kN]}$$

$$F_V = F_B \cdot sin\gamma = 3.42 \cdot sin30° = 1.71 \text{ [kN]}$$

3.7 Calculations and diagrams of bending moment (high speed shaft)

3.7.1 Horizontal Plane



$$\sum F_{iH} = R_{H(B)} + R_{H(D)} - F_H - W_r = 0$$

$$R_{H(B)} + R_{H(D)} = F_H + W_r = 6.44 \text{ [kN]}$$

$$\sum M_{H(D)} = F_H \cdot 0.29 + W_r \cdot 0.09 - R_{H(B)} \cdot 0.18 = 0$$

$$R_{H(B)} = \frac{2.96 \cdot 0.29 + 3.478 \cdot 0.09}{0.18} = 6.51 \text{ [kN]}$$

$$R_{H(D)} = 6.44 - R_{H(B)} = -0.07 \text{ [kN]}$$

Calculations for bending moment diagram:

0< x1 < 110

$$M_H = F_H \cdot x1$$

$M_H(0) = 0$ [kN-m]

$M_H(110) = 2{,}96 \cdot 0.11 = 0.3256$ [kN-m]

110 <x2 < 200

$$M_H = F_H \cdot x2 - R_{H(B)} \cdot (x2 - 0.11)$$

$M_H(110) = 2.96 \cdot 0.11 = 0.3256$ [kN-m]

$M_H(200) = 2.96 \cdot 0.2 - 6.51 \cdot 0.09 = 0.0061$ [kN-M]

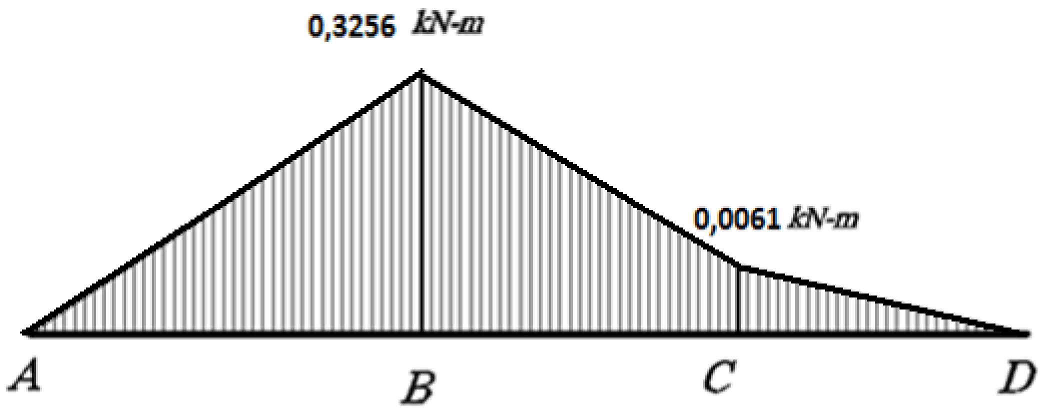
200 < x3 < 290

$$M_H = F_H \cdot x3 - R_{H(B)} \cdot (x3 - 0.11) + W_r \cdot (x3 - 0.2)$$

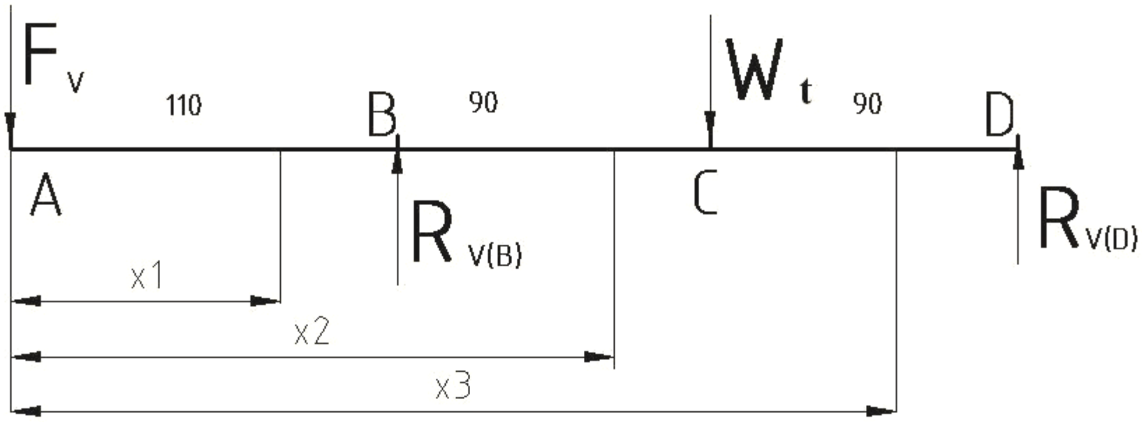
$M_H(200) = 2.96 \cdot 0.2 - 6.51 \cdot 0.09 = 0.0061$ [kN-m]

$M_H(290) = 2.96 \cdot 0.29 - 6.51 \cdot 0.18 + 3.478 \cdot 0.09 \approx 0$ [kN-m]

Bending Moment Diagram for Horizontal Plane



3.7.2 Vertical Plane



$$\sum F_{iV} = R_{V(B)} + R_{V(D)} - F_V - W_t = 0$$

$R_{V(B)} + R_{V(D)} = F_V + W_t = 11.266$ [kN]

$$\sum M_{V(D)} = F_V \cdot 0.29 + W_t \cdot 0.09 - R_{V(B)} \cdot 0.18 = 0$$

$R_{V(B)} = \frac{1.71 \cdot 0.29 + 9.556 \cdot 0.09}{0.18} = 7.53$ [kN]

$R_{V(D)} = 11.28 - R_{V(B)} = 3.733$ [kN]

Calculations For Bending Moment Diagram:

<u>0<x1<110</u>

$$M_V = F_V \cdot x1$$

$M_V(0) = 0$ [kN-m]

$M_V(110) = 1.71 \times 0.11 = 0.1881$ [kN-m]

<u>110<x2<200</u>

$$M_V = F_V \cdot x2 - R_{V(B)} \cdot (x2 - 0.11)$$

$M_V(110) = 1.71 \cdot 0.11 = 0.1811$ [kN-m]

$M_V(200) = 1.71 \cdot 0.2 - 7.533 \cdot 0.09 = -0.336$ [kN-m]

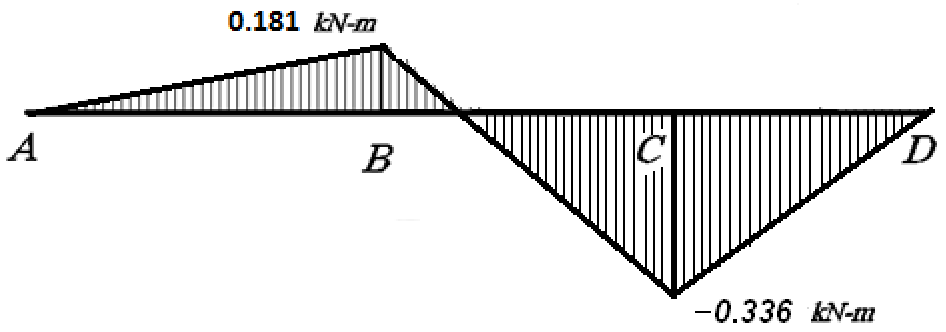
200<x3<290

$$M_V = F_V \cdot x3 - R_{V(B)} \cdot (x3 - 0.11) + W_t \cdot (x3 - 0.2)$$

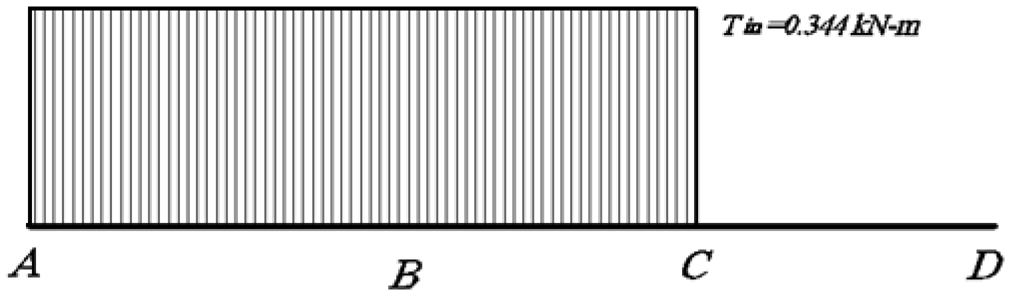
$M_V(200) = 1.71 \cdot 0.2 - 7.533 \cdot 0.09 = -0.336$ [kN-m]

$M_V(290) = 1.71 \cdot 0.29 - 7.533 \cdot 0.18 + 9.556 \cdot 0.09 \approx 0$ [kN-m]

Bending Moment Diagram for Vertical Plane



3.8 Torsion diagram



3.9 Critical section of the high speed shaft

$$M_B = \sqrt{M_{H(B)}^2 + M_{V(B)}^2} = \sqrt{0.326^2 + 0.181^2} = 0.373 \text{ [kN-m]}$$

$$M_C = \sqrt{M_{H(C)}^2 + M_{V(C)}^2} = \sqrt{0.061^2 + 0.336^2} = 0.342 \text{ [kN-m]}$$

Then critical section is B-one.

3.10 Bearing loads of the high speed shaft

$$R_B = \sqrt{R_{H(B)}^2 + R_{V(B)}^2} = \sqrt{6.51^2 + 7.533^2} = 9.956 \text{ [kN]}$$

$$R_D = \sqrt{R_{H(D)}^2 + R_{V(D)}^2} = \sqrt{0.07^2 + 3.733^2} = 3.73 \text{ [kN]}$$

3.11 Minimal shaft diameter (for high speed one)

Minimal shaft diameter of the high speed one in the critical section should be

$$D_B = \left[\frac{32 \cdot SF}{\pi} \cdot 10^6 \cdot \sqrt{\left(\frac{SCF \cdot M_B}{S_n}\right)^2 + \frac{3}{4} \cdot \left(\frac{T_{in}}{S_y}\right)^2}\right]^{\frac{1}{3}} \text{ [mm]}$$

Where:

$\boldsymbol{SF}$ = safety factor; it is recommended as **2** commonly,
$\boldsymbol{SCF}$ = stress concentration factor; it is recommended as **3**,
$\boldsymbol{S_n}$ = endurance strength; it equals **280** Mpa for C45 steel in accordance with EN-10083-2 standard,
$\boldsymbol{S_y}$ = yield strength; it equals **370** Mpa for C45 steel in accordance with EN-10083-2 standard.
$\boldsymbol{M_B}$ and $\boldsymbol{T_{in}}$ are the same as calculated earlier.

$$D_B = \left[\frac{32\cdot 2}{\pi} \cdot 10^6 \cdot \sqrt{\left(\frac{3\cdot 0.373}{280}\right)^2 + \frac{3}{4} \cdot \left(\frac{0.344}{370}\right)^2}\right]^{\frac{1}{3}} \text{ [mm]}$$

For values of *SF* and *SCF* were chosen as high ones, we can take definitely

$$D_B = 43.6 \text{ [mm]}$$

3.12 Simple method of shaft (minimal) diameter calculation

Minimal diameter of the whole shaft may be calculated from the empirical formula:

$$D_{A-D} = \left(\frac{16}{\pi} \cdot 10^6 \cdot \frac{T_{in}}{40 \div 50}\right)^{\frac{1}{3}} \text{ [mm]}$$

$$D_{A-D} = \left(\frac{16}{\pi} \cdot 10^6 \cdot \frac{0.344}{40 \div 50}\right)^{\frac{1}{3}} = (32.7 \div 35.25) \text{ [mm]}$$

3.13 Minimal diameters of high speed shaft ends

Recommended diameters for journals of shaft ends in gear reducers
(PN-M-85000: 1998)

TABLE 2.1. DIAMETERS AND MAX TORQUES FOR JOURNALS OF HIGH SPEED SHAFT ENDS

d [mm]	T_{in} [Nm]	d [mm]	T_{in} [Nm]	d [mm]	T_{in} [Nm]	d [mm]	T_{in} [Nm]
10	8.0	30	200	60	1600	125	16000
12	16.0	32	350	65	2240	130	18000
14	22.4	35	355	70	2800	140	22400
16	31.5	38	400	75	3150	150	25000
18	45.0	40	500	80	4000	160	31500
20	63.0	42	560	85	4500	180	45000
22	90.0	45	710	90	5600	200	63000
25	125.0	50	1000	100	8000		
28	180.0	55	1400	110	11200		

<u>Note:</u> Max transverse force loading the journal in its middle should not exceed

$$250 \cdot \sqrt{T_{in}} \text{ [N]}$$

3.14 Free body diagram of the low speed shaft

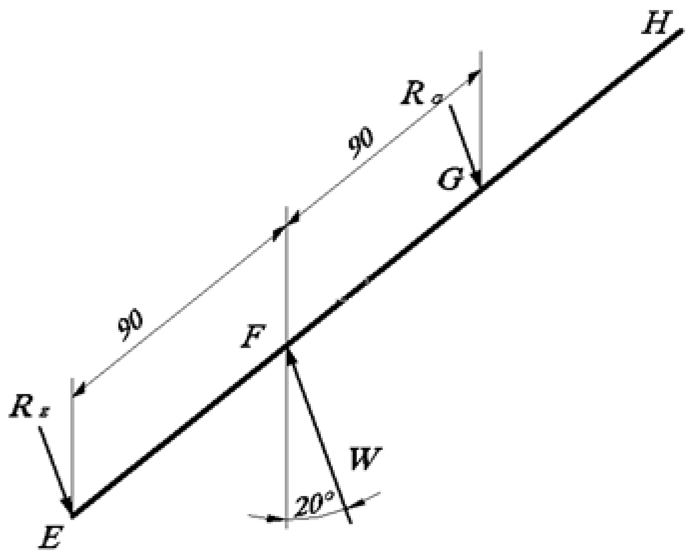


Fig.3.3. Free body diagram of the low speed shaft

3.15 Calculations and diagrams of bending moment (low speed shaft)

Notice that all forces acting on the shaft are in the same plane.

$$\sum F_i = R_E + R_G - W = 0$$

$$R_E + R_G = W$$

$$R_E + R_G = 10.62 \ \text{[kN]}$$

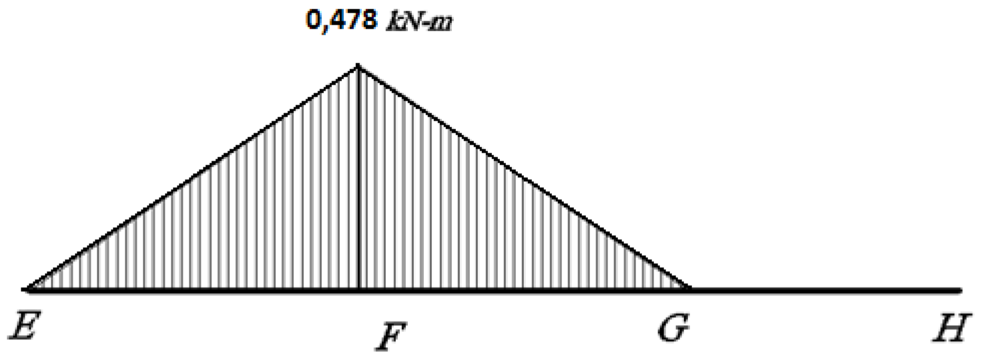
$$\sum M_{(G)} = -W \cdot 0.09 + R_E \cdot 0.18 = 0$$

$$R_E = \frac{10.62 \cdot 0.09}{0.18} = 5.31 \ \text{[kN]}$$

$$R_G = 5.31 \ \text{[kN]}$$

$$M_F = R_E(or\ R_G) \cdot 0.09 = 5.31 \cdot 0.09 = 0.478 \ \text{[kN-m]}$$

Bending Moment Diagram for the Low Speed Shaft



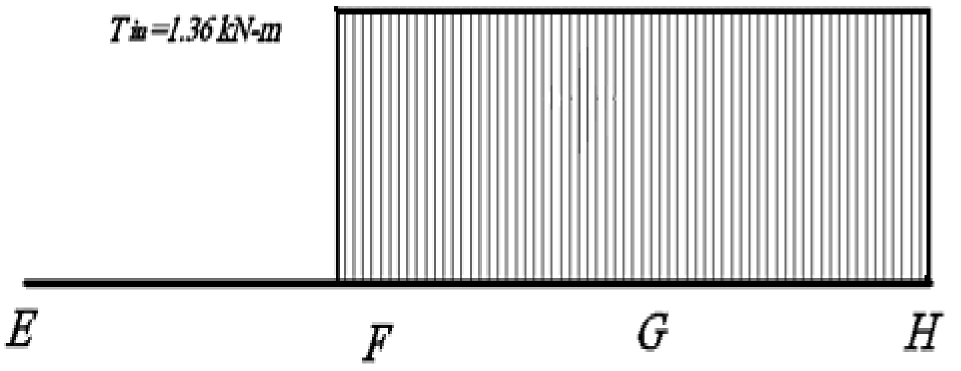
So the critical section for the low speed shaft is F-one.

3.16 Torque acting on the low speed shaft

The torque equals

$$T_{out} = T_{in} \cdot u_g = 0.344 \cdot 3.95 = 1.36 \text{ [kN-m]}$$

3.17 Torsion diagram for the low speed shaft



3.18 Minimal shaft diameter (for low speed one)

Minimal shaft diameter of the low speed one in the critical section should be

$$D_F = \left[\frac{32 \cdot SF}{\pi} \cdot 10^6 \cdot \sqrt{\left(\frac{SCF \cdot M_F}{S_n}\right)^2 + \frac{3}{4}\left(\frac{T_{out}}{S_y}\right)^2}\right]^{\frac{1}{3}} \text{ [mm]}$$

Where:

SF, SCF, S_n, S_y, M_F, and T_{out} are the same as chosen and calculated earlier.

$$D_F = \left[\frac{32 \cdot 2}{\pi} \cdot 10^6 \cdot \sqrt{\left(\frac{3 \cdot 0.478}{280}\right)^2 + \frac{3}{4}\left(\frac{1.36}{370}\right)^2}\right]^{\frac{1}{3}} = 49.7 \text{ [mm]}$$

3.19 Evaluation of minimal diameter for the low speed shaft with empirical method

This diameter is calculated from the empirical formula

$$D_{E-H} = D_{A-D} \cdot \sqrt[3]{u_g}$$

where $D_{A\text{-}E}$ is computed as in §3.12.

$$D_{E-H} = (32.7 \div 35.25) \cdot \sqrt[3]{3.95} = (51.7 \div 55.7) \text{ [mm]}$$

3.20 Minimal diameters of slow speed shaft ends

Recommended diameters for journals of shaft ends in gear reducers **(**PN-M-85000: 1998)

TABLE 2.2. DIAMETERS AND MAX TORQUES FOR JOURNALS OF LOW SPEED SHAFT ENDS

d [mm]	T_{out} [Nm]	d [mm]	T_{out} [Nm]	d [mm]	T_{out} [Nm]	d [mm]	T_{out} [Nm]
18	31.5	35	350	70	2000	140	15000
20	45	40	355	75	2240	160	22400
22	63	45	500	80	2800	180	31500
25	90	48	560	90	4000	200	45000
28	125	50	710	100	5600	220	63000
30	140	55	1000	110	8000	250	90000
32	180	60	1120	125	11200	280	125000

Note: Max transverse force loading the journal in its middle should not exceed:

1. $125 \cdot \sqrt{T_{in}}$ [N] for one-stage gear reducers, and
2. $250 \cdot \sqrt{T_{in}}$ [N] for multi-stages gear reducers.

3.21 Resume

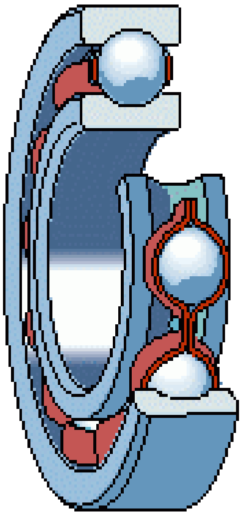
It was taken:

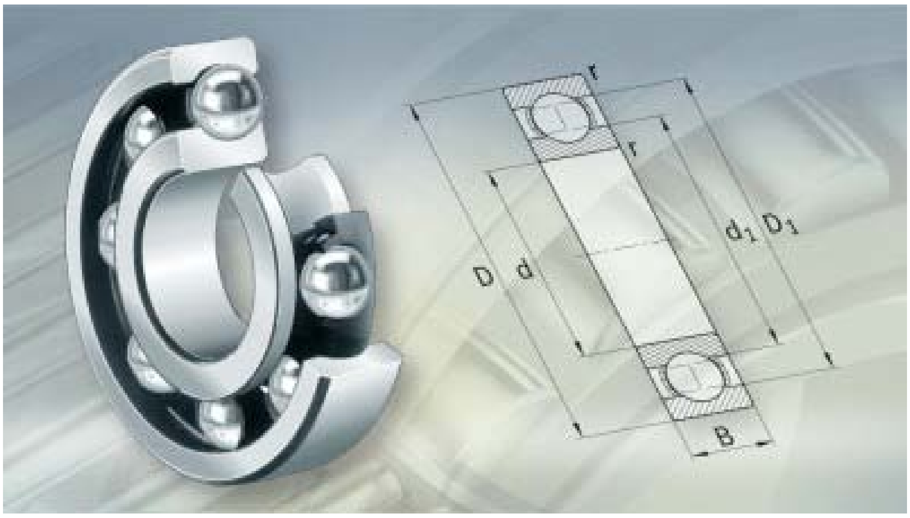
	High speed shaft	Low speed shaft
End shafts diameters (predicted for the pulley mounting – high speed shaft, and the clutch mounting – low speed shaft)	$d_1 = 40$ mm	$d_1 = 55$ mm
For an oil seal ring mounting diameter	$d_2 = 45$ mm	$d_2 = 60$ mm
For bearings mounting diameter	$d_3 = 50$ mm	$d_3 = 65$ mm

Chapter 4

DEEP GROOVE BALL BEARINGS, SINGLE ROW (Basis description and fundamentals calculation)

4.1 View





4.2 Application

Typical of deep groove ball bearings use:

- radial load with possibility of low axial load,
- short rigid shafts,
- high rotation speeds,
- lowest friction among bearings,
- low ability to angle swing,
- with seals unnecessary supervision.

4.3 Ball bearings description

bearing series

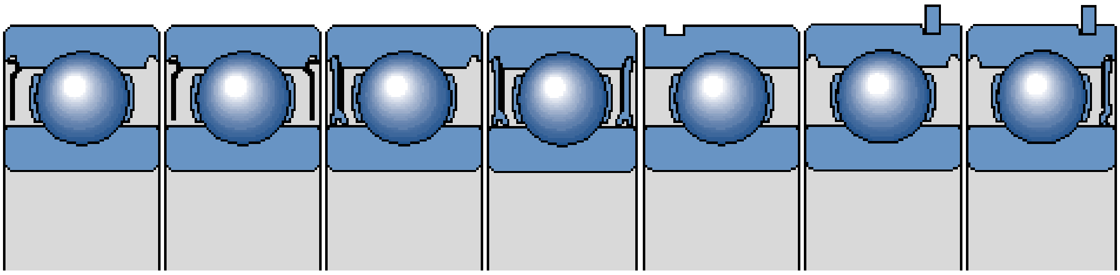
bearing inner diameter – d = 06x5=30 [mm]

6 2 0 6

kind of bearing

diameter kind (2 – light)

4.4 Kinds of constructions



RS - seal on one side

2RS - seal on both sides

Z - shield on one side

ZZ - shield both sides

N - with a snap ring groove, unsealed

NR - with a snap ring groove, unsealed

ZNR - with a snap ring groove, shield on one side

4.5 Theoretical basis

Static basic load rating C_0

Static basic load rating C_0 – static bearing load which work in nominal direction cause stress of Hertz $\boldsymbol{\sigma_H}$ =4200 Mpa (value for ball bearing) between most loaded element and the race.

Dynamic basic load rating C

Dynamic basic load rating C **–** bearing load under which 90% of bearing population reach durability 1 000 000 rotation.

4.6 Life's calculation basis

Life's equation (Lundberg & Palmgren)

$$L = \left(\frac{C}{P}\right)^p \quad \text{[mln revolutions or – mln cycles]}$$

$$L_h = \frac{10^6}{60 \cdot n} \cdot \left(\frac{C}{P}\right)^p \qquad \text{[hrs]}$$

where:

C – dynamic basic load rating [N]

P – equivalent bearing load [N]

p – exponent factor (p = 3 for ball bearings)

n – rotational speed [rpm]

Equivalent bearing load

$$P = XF_r + Y\mathrm{F_a} \quad \text{[N]}$$

where:

F_r – radial load

F_a – axial load

X – radial load factor

Y – axial load factor

Values of X and Y factor

TABLE 4.1. VALUES OF X AND Y FACTORS

	relative axial load	e	if $F_a/F_r \leq e$		if $F_a/F_r > e$	
	F_a/C_0		X	Y	X	Y
Deep groove ball bearing, single row, normal clearance	0,014	0,19	1	0	0,56	2,30
	0,028	0,22				1,99
	0,056	0,26				1,71
	0,084	0,28				1,55
	0,110	0,30				1,45
	0,170	0,34				1,31
	0,280	0,38				1,15
	0,420	0,42				1,04
	0,520	0,44				1,00

Procedure of X and Y factor value calculation

There are following steps to calculate X and Y factors:

1. find value of C (dynamic load) and C_0 (static load) for given bearing (Those data contains bearings catalogue for example http://www.skf.com/portal/skf/home/products?maincatalogue=1&newlink=first&lang=en)
2. calculate value F_a/C_0
3. from table 4.1. read value of *"e"* factor
4. calculate value F_a/F_r
5. read value of X and Y factor from table 4.1.checking if value $F_a/F_r \leq e$ or $F_a/F_r > e$

Effective dynamic load C_e

Effective dynamic load of the bearing is calculated from formula

$$C_e = C \cdot f_t$$

where:

f_t – temperature factor

TABLE 4.2. TEMPERATURE FACTOR f_T

Work temperature [°C]	Factor value f_t
150	1.00
200	0.90
250	0.75
300	0.60

Effective equivalent load

Effective equivalent load is calculated from formula

$$P_e = P * f_d$$

where:

f_d – dynamic load factor

TABLE 4.3. DYNAMIC LOAD FACTOR f_d

Machine work condition	f_d
Smooth, without strokes	1
Smooth with possibility overloading to 25%, light shocks	1 ÷ 1,2
Normal work condition, possibility overloading to 50%, shocks and strokes	1,2 ÷ 1,8
Work with big load	1,8 ÷ 2,5
Hard work, big shocks and strokes	2,5 ÷ 3,5

Effective life

Effective life is calculated from formula

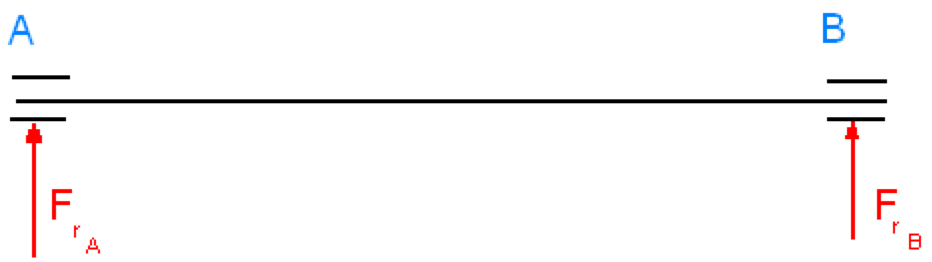
$$L_e = \left(\frac{C_e}{P_e}\right)^p \qquad \text{[mln. revolutions or Mc]}$$

or

$$L_{he} = \frac{10^6}{60*n}\left(\frac{C_e}{P_e}\right)^p \quad \text{[h]}$$

4.7 Example # 1

Shaft load sketch



Given:

A and B bearings – 6406

F_{rA} = 5000 N

F_{rB} = 3000 N

F_a = 0

n = 1500 rpm

Bearing temperature 150^0 C

Possible overload 25%

<u>Calculation</u>

From bearings catalogue (http://www.skf.com/skf/productcatalogue/jsp/viewers/productTableViewer.jsp?&lang=en&newlink=1&tableName=1_1_1&presentationType=3&startnum=15)

Deep groove ball bearings, single row

SKF

Principal dimensions			Basic load ratings dynamic	static	Fatigue load limit	Speed ratings Reference speed	Limiting speed	Mass	Designation	
d	D	B	C	C_0	P_u					
mm			kN		kN	r/min		kg	-	
30	62	16	20,3	11,2	0,475	-	7500	0,20	6206-RS1 *	
30	62	16	20,3	11,2	0,475	24000	15000	0,20	6206-RZ *	
30	62	16	20,3	11,2	0,475	24000	15000	0,20	6206-Z *	
30	62	20	19,5	11,2	0,475	-	7500	0,24	62206-2RS1	
30	72	19	29,6	16	0,67	20000	13000	0,35	6306 *	
30	72	19	32,5	17,3	0,735	22000	14000	0,33	6306 ETN9	
30	72	19	29,6	16	0,67	-	6300	0,35	6306-2RS1 *	
30	72	19	29,6	16	0,67	20000	11000	0,35	6306-2RZ *	
30	72	19	29,6	16	0,67	20000	11000	0,35	6306-2Z *	
30	72	19	29,6	16	0,67	-	6300	0,35	6306-RS1 *	
30	72	19	29,6	16	0,67	20000	13000	0,35	6306-RZ *	
30	72	19	29,6	16	0,67	20000	13000	0,35	6306-Z *	
30	72	27	28,1	16	0,67	-	6300	0,48	62306-2RS1	
30	90	23	43,6	23,6	1	18000	11000	0,74	6406	
31,75	69,85	17,462	22,5	13,2	0,55	20000	14000	0,30	RLS 10	Aftermarket only
31,75	69,85	17,462	22,5	13,2	0,55	-	7000	0,30	RLS 10-2RS1	Aftermarket only
31,75	69,85	17,462	22,5	13,2	0,55	20000	10000	0,30	RLS 10-2Z	Aftermarket only
31,75	79,375	22,225	33,2	19	0,815	17000	12000	0,50	RMS 10	Aftermarket only
34,925	76,2	17,462	27	15,3	0,655	18000	13000	0,35	RLS 11	Aftermarket only
34,925	88,9	22,225	41	24	1,02	15000	11000	0,63	RMS 11	Aftermarket only
35	47	7	4,75	3,2	0,166	28000	18000	0,030	61807	
35	47	7	4,75	3,2	0,166	-	8000	0,030	61807-2RS1	
35	47	7	4,75	3,2	0,166	28000	14000	0,030	61807-2RZ	
35	55	10	9,56	6,8	0,29	26000	16000	0,080	61907	
35	55	10	9,56	6,8	0,29	-	7500	0,080	61907-2RS1	

is obtained:

C = 43600 N, C_0 = 23600 N

Because $F_a = 0$

A bearing equivalent load equals

$P_A = F_{rA}$ = 5000 N

B bearing equivalent load equals

$P_B = F_{rB}$ = 3000 N

From table 4.3

f_d = 1,2

Effective equivalent load of A bearing

$P_{eA} = P_A \cdot f_d$ = 5000·1,2 = 6000 N

Effective equivalent load of B bearing

$P_{eB} = P_B \cdot f_d$ = 3000·1,2 = 3600 N

From table 4.2.

f_t =1

Effective dynamic load

$C_{eA} = C_{eB} = C_e = C \cdot f_t = 43600 \cdot 1 = 43600$ N

Bearing A effective life

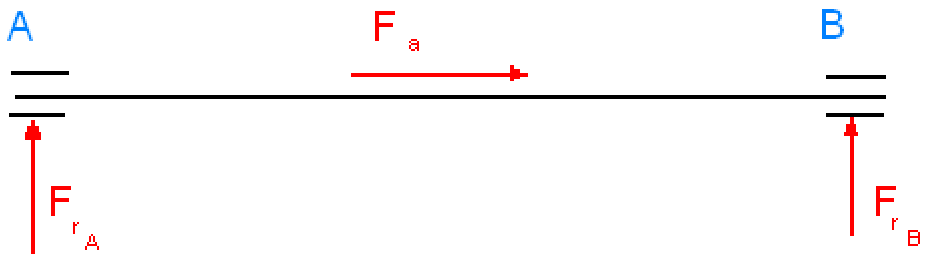
$$L_{ehA} = \frac{10^6}{60*n}\left(\frac{C_{eA}}{P_{eA}}\right)^p = \frac{10^6}{60*1500}\left(\frac{43600}{5000}\right)^3 = 7623{,}3 \text{ [h]}$$

Bearing B effective life

$$L_{ehB} = \frac{10^6}{60*n}\left(\frac{C_{eB}}{P_{eB}}\right)^p = \frac{10^6}{60*1500}\left(\frac{43600}{3600}\right)^3 = 19738{,}3 \text{ [h]}$$

4.8 Example # 2

Shaft load sketch



<u>Given:</u>

A and B Bearings – 6406

$F_{rA} = 5000$ N

$F_{rB} = 3000$ N

$F_a = 1000$ N

$n = 1500$ rev/min.

Bearing temperature 150^0 C

Possible overload 25%

<u>Calculation</u>

From bearings catalogue

C=43600 N, C_0 =23600 N

Note: Axial load is carried by only one of bearings

A Bearing

$F_{aA} = 0$

Relative axial load F_a/C_0

$$\frac{F_{aA}}{C_{0A}} = \frac{0}{23600} = 0 \Longrightarrow e = 0$$

$X_A = 1$, $Y_A = 0$

Bearing A equivalent load equals

$P_A = F_{rA} = 5000$ N

Bearing B

$F_{aB} = F_a = 1000$ N

Relative axial load F_a/C_0

$$\frac{F_{aB}}{C_{0B}} = \frac{1000}{23600} = 0{,}042$$

From table 4.1.

$e = 0{,}24$

$$\frac{F_{aB}}{F_{rB}} = \frac{1000}{3000} = 0{,}3$$

$$\frac{F_{aB}}{F_{rB}} = \frac{1000}{3000} = 0{,}3 > e = 0{,}24$$

$X_B = 0{,}56$, $Y_B = 1{,}85$

Bearing B equivalent load

$$P_B = X_B F_{rB} + Y_B \mathrm{F_{aB}} = 1 \cdot 3000 + 0{,}56 \cdot 1000 = 3560 \text{ N}$$

From table 4.3

$f_d = 1{,}2$

Effective equivalent load of A bearing

$$P_{eA} = P_A \cdot f_d = 5000 \cdot 1{,}2 = 6000 \text{ N}$$

Effective equivalent load of B bearing

$$P_{eB} = P_B \cdot f_d = 3560 \cdot 1{,}2 = 4272 \text{ N}$$

From table 4.2

f_t =1

Effective dynamic basic load rating

$$C_{eA} = C_{eB} = C_e = C \cdot f_d = 43600 \cdot 1 = 43600 \text{ N}$$

A Bearing effective life

$$L_{ehA} = \frac{10^6}{60*n}\left(\frac{C_{eA}}{P_{eA}}\right)^p = \frac{10^6}{60*1500}\left(\frac{43600}{5000}\right)^3 = 7623{,}3 \text{ [hrs]}$$

B Bearing effective life

$$L_{ehB} = \frac{10^6}{60*n}\left(\frac{C_{eB}}{P_{eB}}\right)^p = \frac{10^6}{60*1500}\left(\frac{43600}{4272}\right)^3 = 11812 \text{ [hrs]}$$

Chapter 5

NUMERICAL EXAMPLE OF BALL BEARING SELECTION

Bearings supporting the high speed shaft (see Chapter 3 – SHAFT LOAD CALCULATION) in B and D points are loaded by R_B = 9.956 kN and R_D = 3.73 kN respectively (see fig. 5.1 below).

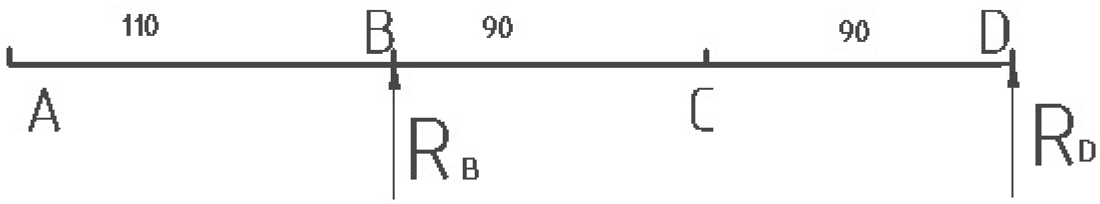


Fig.5.1. Free body diagram of the high speed shaft

The shaft rotates at n_{in} = 610.4 RPM and running time of the gear reducer is 6 hrs/day. What the bearings should be selected?

First let us notice that the running time of 6 hrs/day is usual to equate to a total life of 13.2 khrs, so L = 13.2 khrs.

To select the bearings we must calculate their needed basic load ratings (a dynamic one). For this purpose we calculate a life time for the high speed shaft in million of cycles:

$$L_{C(in)} = \frac{60 \cdot n_{in} \cdot L}{1000} \quad \text{[Mc]}$$

$$L_{C(in)} = \frac{60 \cdot 610.4 \cdot 13.3}{1000} = 483.4 \quad \text{[Mc]}$$

Now we can calculate a needed basic load rating for the bearing (in point B):

$$C_B = R_B \cdot \sqrt[3]{L_{C(in)}} \quad \text{[kN]}$$

$$C_B = 9.956 \cdot \sqrt[3]{483.4} = 78.1 \text{ [kN]}$$

To select a bearing, apart its needed basic load rating, we must know its kind and bore. As to the kind we take deep groove ball bearings single row. They best fits to our reducer.

Note: Deep groove ball bearings are used in a particularly wide variety of applications. They are simple in design, non-separable, capable of operating at high and even very high speeds, and require little attention or maintenance in service. These characteristics coupled with a price advantage make deep groove ball bearings the most popular of all rolling bearings.

As to a bore of the bearing we take a one of 50 mm
(a diameter of 40 mm we predict for the pulley mounting, a diameter of 45 mm – for an oil seal ring mounting, and at last a diameter of 50 mm – for bearings).

From SKF catalogue we have selected a bearing of 6410.

Deep groove ball bearings, single row
Product information

Tolerances , see also text
Radial internal clearance , see also text
Recommended fits
Shaft and housing tolerances

Principal dimensions			Basic load ratings		Fatigue load limit	Speed ratings		Mass	Designation	
			dynamic	static		Reference speed	Limiting speed			
d	D	B	C	C_0	P_u				* - SKF Explorer bearing	
mm			kN		kN	r/min		kg	-	
50	80	16	22,9	16	0,71	-	5000	0,26	6010-RS1 *	
50	80	16	22,9	16	0,71	18000	11000	0,26	6010-RZ *	
50	80	16	22,9	16	0,71	18000	11000	0,26	6010-Z *	
50	80	23	21,6	16	0,71	-	5000	0,37	63010-2RS1	
50	90	20	37,1	23,2	0,98	15000	10000	0,46	6210 *	
50	90	20	37,1	23,2	0,98	-	4800	0,46	6210-2RS1 *	
50	90	20	37,1	23,2	0,98	15000	8000	0,46	6210-2RZ *	
50	90	20	37,1	23,2	0,98	15000	8000	0,46	6210-2Z *	
50	90	20	37,1	23,2	0,98	-	4800	0,46	6210-RS1 *	
50	90	20	37,1	23,2	0,98	15000	10000	0,46	6210-RZ *	
50	90	20	37,1	23,2	0,98	15000	10000	0,46	6210-Z *	
50	90	23	35,1	23,2	0,98	-	4800	0,52	62210-2RS1	
50	110	27	65	38	1,6	13000	8500	1,05	6310 *	
50	110	27	65	38	1,6	-	4300	1,05	6310-2RS1 *	
50	110	27	65	38	1,6	13000	6700	1,05	6310-2Z *	
50	110	27	65	38	1,6	-	4300	1,05	6310-RS1 *	
50	110	27	65	38	1,6	13000	8500	1,05	6310-Z *	
50	110	40	61,8	38	1,6	-	4300	1,55	62310-2RS1	
50	130	31	87,1	52	2,2	12000	7500	1,90	6410	
50,8	101,6	20,637	35,1	23,2	0,98	14000	10000	0,70	RLS 16	Aftermarket only
55	72	9	9,04	8,8	0,375	19000	12000	0,083	61811	
55	72	9	9,04	8,8	0,375	-	5300	0,083	61811-2RS1	
55	72	9	9,04	8,8	0,375	19000	9500	0,083	61811-2RZ	
55	80	13	16,5	14	0,6	17000	11000	0,19	61911	
55	80	13	16,5	14	0,6	-	5000	0,19	61911-2RS1	

Its basic load rating (dynamic) is 87.1 kN.

Dimensions of the bearing are as follow (see fig.5.2 and fig.5. 3):

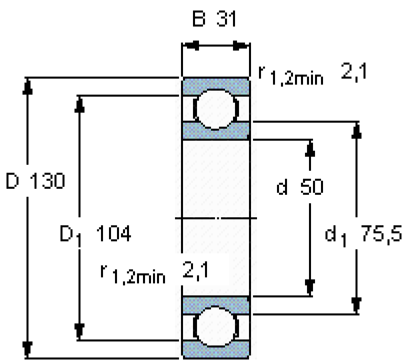


Fig.5.2. Dimensions of the selected bearing for the high speed shaft

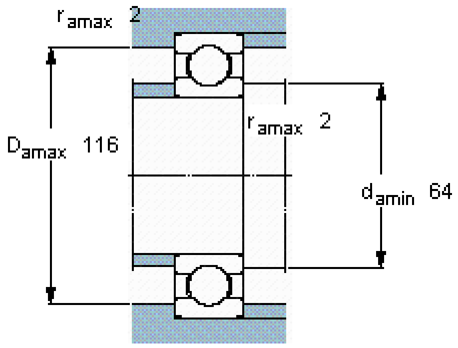


Fig.5.3. Dimension of mounting places for the selected bearing

As a bearing in D point of the high speed shaft supporting we take the same one like in B point, i.e. 6410 although it is loaded much less than the bearing in B point of the supporting. Such a way causes a gear case machining to be done much easier.

The low speed shaft rotates at

$$n_{out} = \frac{n_{in}}{u_g} = \frac{610.4}{3.95} = 154.5 \text{ [RPM]}$$

and the both bearings are loaded by the same forces of 5.1 kN; $R_E = R_G = 5.1$ kN. So, bearings for the shaft may be the same as well (see fig.5.4)

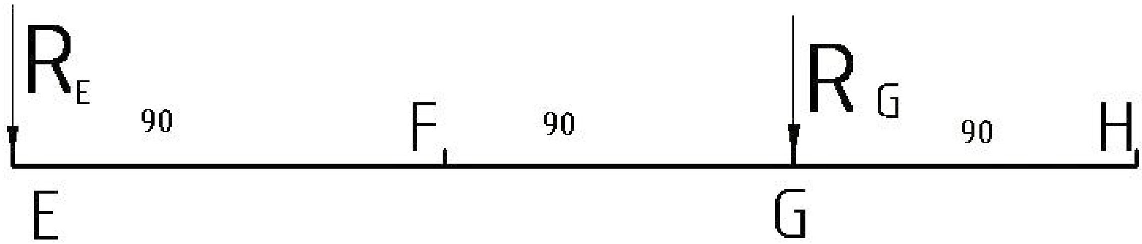


Fig 5.4. Free body diagram of the high speed shaft

A life time for the low speed shaft in million of cycles is

$$L_{C(out)} = \frac{60 \cdot n_{out} \cdot L}{1000} \text{ [Mc]}$$

$$L_{C(out)} = \frac{60 \cdot 154.5 \cdot 13.2}{1000} = 123.3 \text{ [Mc]}$$

A needed basic load rating for the bearing is (points E,G):

$$C_{E,G} = R_E \cdot \sqrt[3]{L_{C(out)}} \text{ [kN]}$$

$$C_B = 5.1 \cdot \sqrt[3]{123.3} = 24.4 \ \ [\text{kN}]$$

As to bores of the bearings for the low speed shaft we take ones of 65 mm (a diameter of 55 mm we predict for a half-coupling mounting, a diameter of 60 mm – for an oil seal ring mounting, and at last a diameter of 65 mm – for bearings).

From SKF catalogue we have selected the bearings of 6213.

Deep groove ball bearings, single row
Product information

Tolerances , see also text
Radial internal clearance , see also text
Recommended fits
Shaft and housing tolerances

Principal dimensions			Basic load ratings		Fatigue load limit	Speed ratings		Mass	Designation	
			dynamic	static		Reference speed	Limiting speed			
d	D	B	C	C_0	P_u				* - SKF Explorer bearing	
mm			kN		kN	r/min		kg	-	
63,5	139,7	31,75	92,3	60	2,5	9500	6700	2,05	RMS 20	Aftermarket only
65	85	10	12,4	12,7	0,54	16000	10000	0,13	61813	
65	85	10	12,4	12,7	0,54	-	4500	0,13	61813-2RS1	
65	85	10	12,4	12,7	0,54	16000	8000	0,13	61813-2RZ	
65	90	13	17,4	16	0,68	15000	9500	0,22	61913	
65	90	13	17,4	16	0,68	-	4300	0,22	61913-2RS1	
65	90	13	17,4	16	0,68	15000	7500	0,22	61913-2RZ	
65	100	11	22,5	16,6	0,83	14000	9000	0,30	16013 *	
65	100	18	31,9	25	1,06	14000	9000	0,44	6013 *	
65	100	18	31,9	25	1,06	-	4000	0,44	6013-2RS1 *	
65	100	18	31,9	25	1,06	14000	7000	0,44	6013-2Z *	
65	100	18	31,9	25	1,06	-	4000	0,44	6013-RS1 *	
65	100	18	31,9	25	1,06	14000	9000	0,44	6013-Z *	
65	120	23	58,5	40,5	1,73	12000	7500	0,99	6213 *	
65	120	23	58,5	40,5	1,73	-	3600	0,99	6213-2RS1 *	
65	120	23	58,5	40,5	1,73	12000	6000	0,99	6213-2Z *	
65	120	23	58,5	40,5	1,73	-	3600	0,99	6213-RS1 *	
65	120	23	58,5	40,5	1,73	12000	7500	0,99	6213-Z *	
65	120	31	55,9	40,5	1,73	-	3600	1,25	62213-2RS1	
65	140	33	97,5	60	2,5	10000	6700	2,10	6313 *	
65	140	33	97,5	60	2,5	-	3200	2,10	6313-2RS1 *	
65	140	33	97,5	60	2,5	10000	5300	2,10	6313-2Z *	
65	140	33	97,5	60	2,5	-	3200	2,10	6313-RS1 *	
65	140	33	97,5	60	2,5	10000	6700	2,10	6313-Z *	
65	140	48	92,3	60	2,5	-	3200	3,00	62313-2RS1	

Their basic load rating (dynamic) is 55.9 kN.

Dimensions of the bearings are as follow (see fig.5.5 and fig. 5.6):

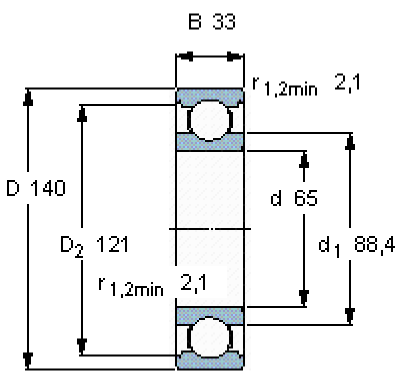


Fig.5.5. Dimensions of the selected bearing for the low speed shaft

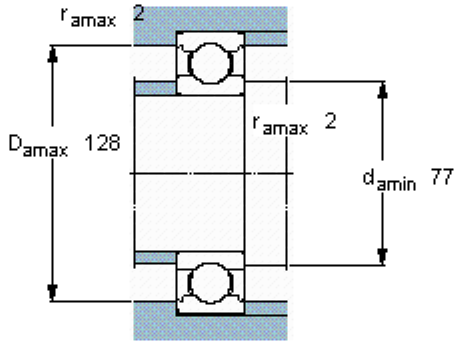
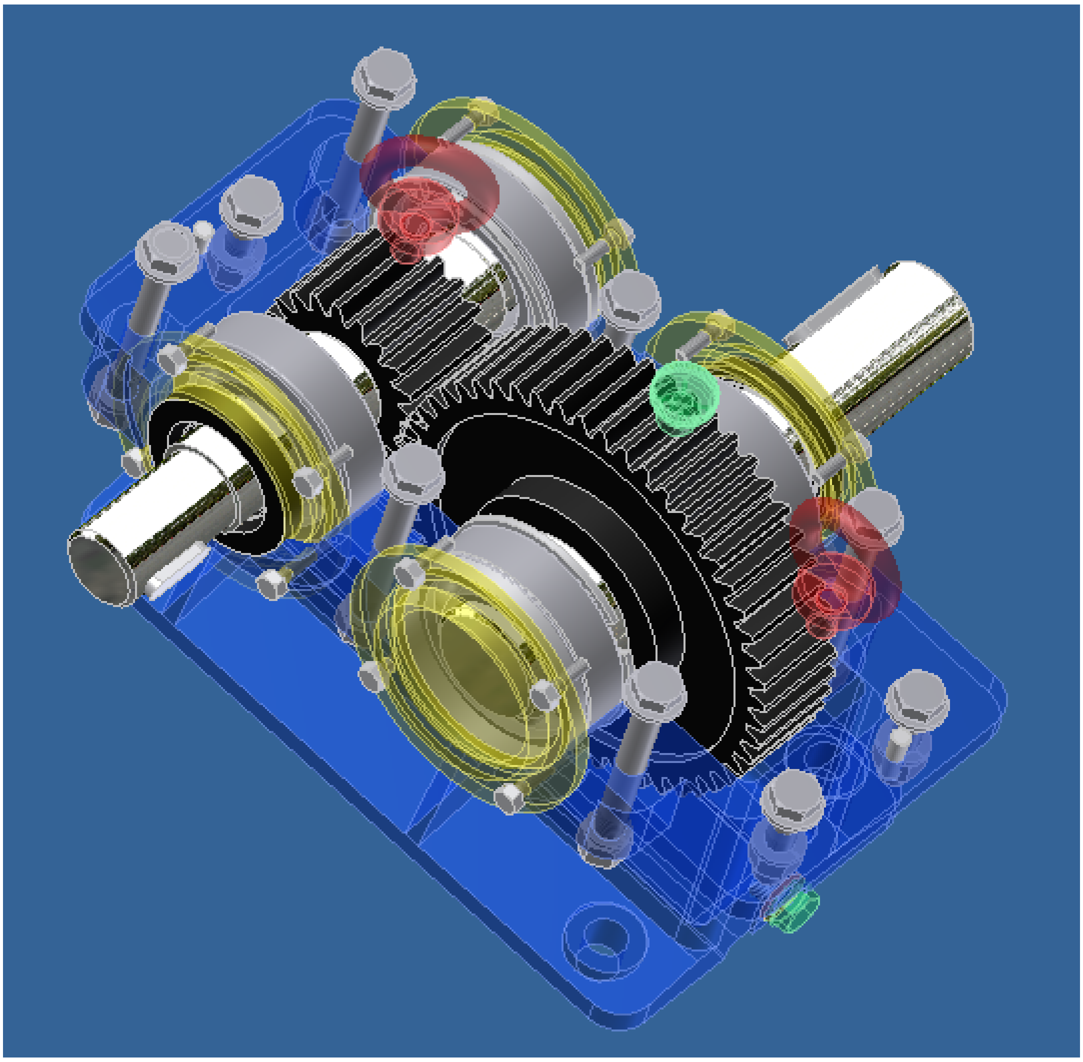


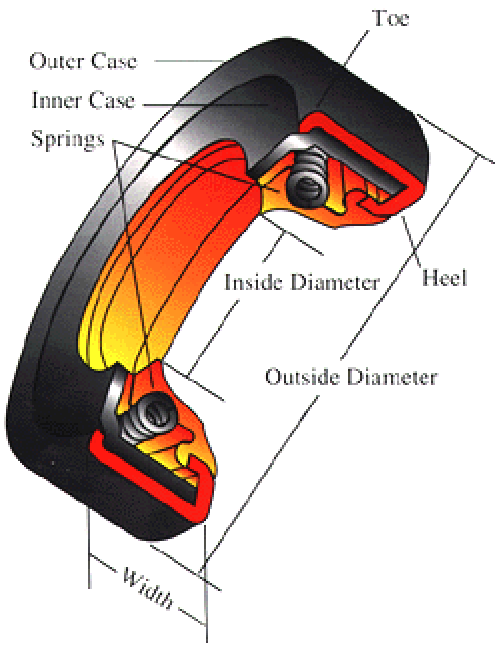
Fig.5.6. Dimension of mounting places for the selected bearing – the low speed shaft

Chapter 6

RADIAL SHAFT SEALS



6.1 Seals design



The radial shaft seals are produced as standard according to ISO-6194. The seals are different in the form and material of the shell or outside. Using materials that is metal, completely elastomer or elastomer reinforced with steel.
The sealing lip design may be of the traditional straight type or the highly efficient design, for example sinusoidal and made of several different materials.

6.2 Type and destinations of materials

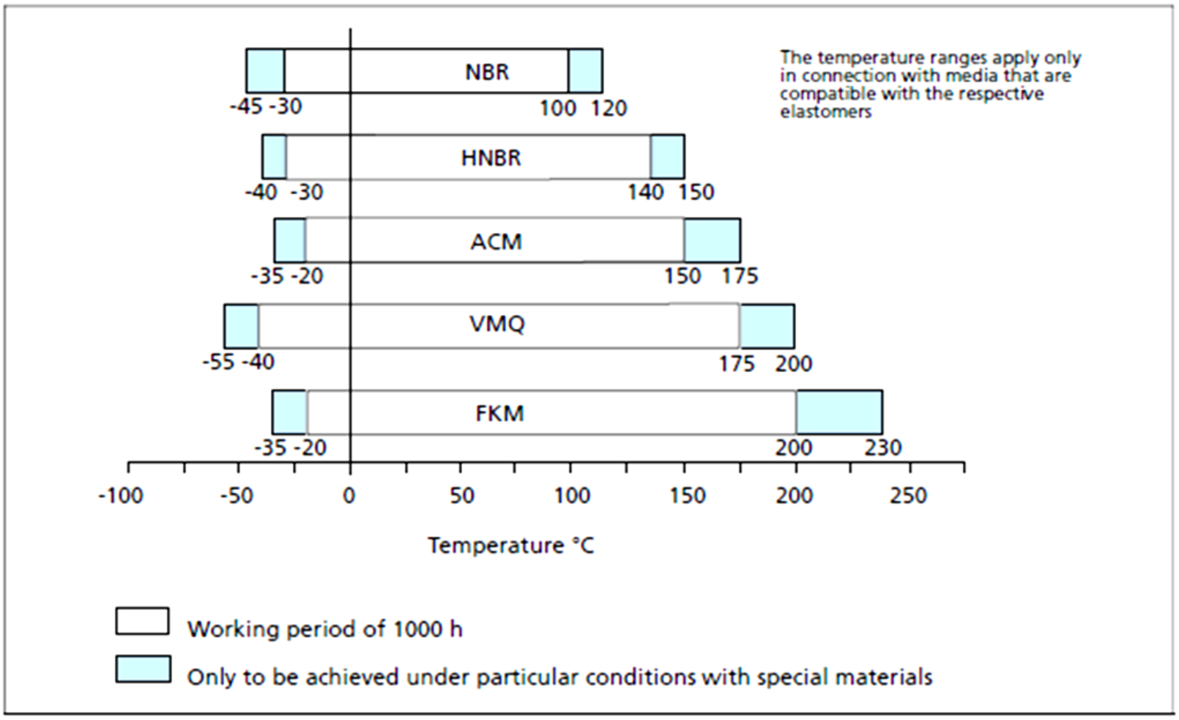
It is very important to take into account the environment in which the seal will operate when you are selecting the sealing element material. The most important factors are temperature, medium being sealed, pressure, and shaft speed. The seals are made from:

- Nitrile rubber (NBR)
- Acrylic rubber (ACM)
- Silicone rubber (VMQ)
- Fluorinated rubber (FKM)
- Hydrogenated Nitrile rubber (HNBR)

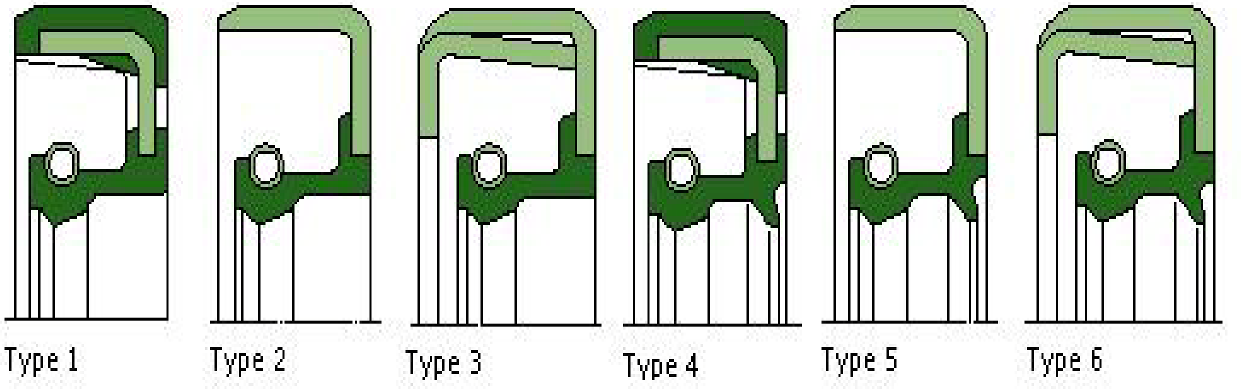
6.3 Materials recommendation

Materials for sealing common media		Acrylonitrile Butadiene Rubber NBR	Fluoro carbon Rubber FKM	Polyacrylate Rubber ACM	Silicone Rubber VMQ	Hydrogenated Acrylonitrile Butadien Rubber HNBR
		Material designation				
		Material Abbreviation				
		N	V	A	S	H
		Max. permissible constant temperature (°C)				
Mineral fluids	Engine oils	100	170	125	150	130
	Transmission oils	80	150	125	130	110
	Hypoid transmission oils	80	150	125	- -	110
	ATF oils	100	170	125	- -	130
	Hydraulic fluids (DIN 51524)	90	150	120	- -	130
	Greases	90	- -	- -	- -	100
Flame retardant hydraulic fluids (VDMA 24317) (VDMA 24320)	Oil-water emulsion	70	- -	- -	60	70
	Water-oil emulsion	70	- -	- -	60	70
	Aqueous solutions	70	- -	- -	- -	70
	Water-free fluids	- -	150	- -	- -	- -
Other media	Fuel oils	90	- -	- -	- -	100
	Water	90	100	- -	- -	100
	Lyes	90	100	- -	- -	100
	Air	100	200	150	200	130

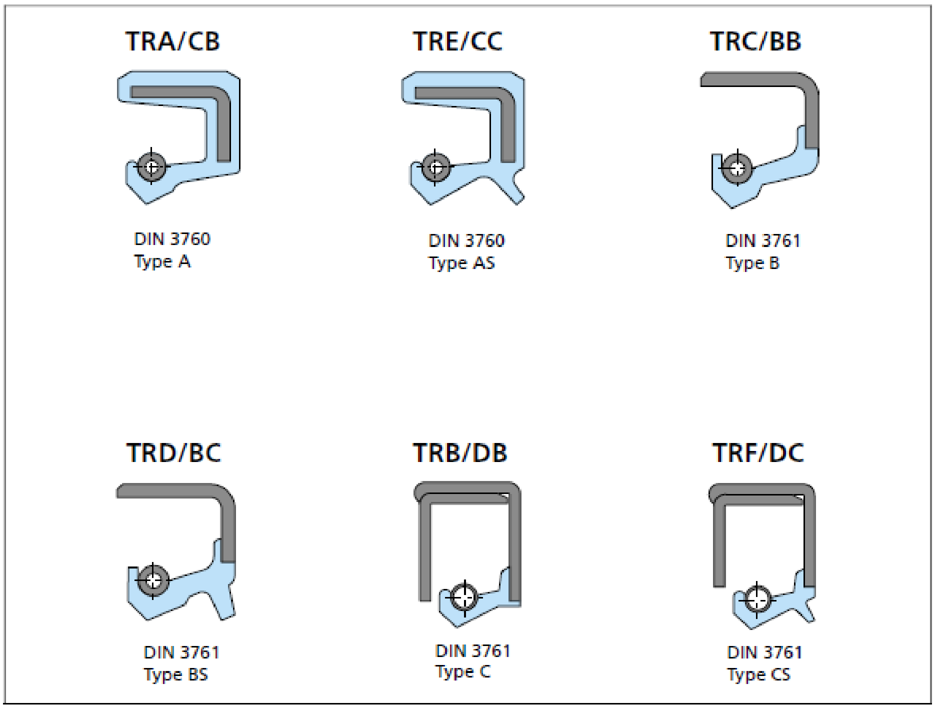
6.4 Temperature limits according to material types



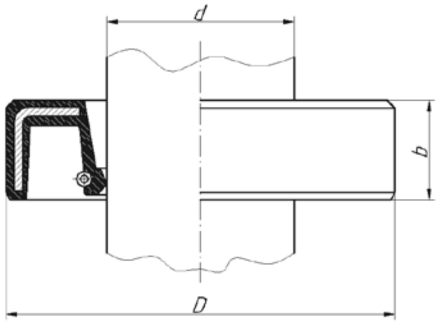
6.5 Design types of radial seals



Standard types designed according to DIN 3760 (3761)



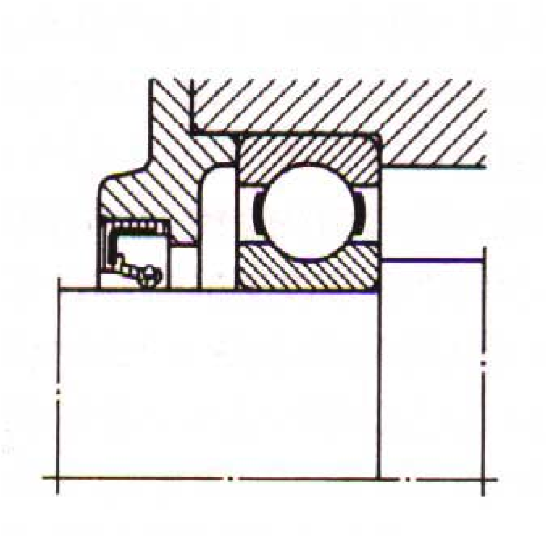
6.6 Radial shaft seals diameters in accordance with ISO – 6194



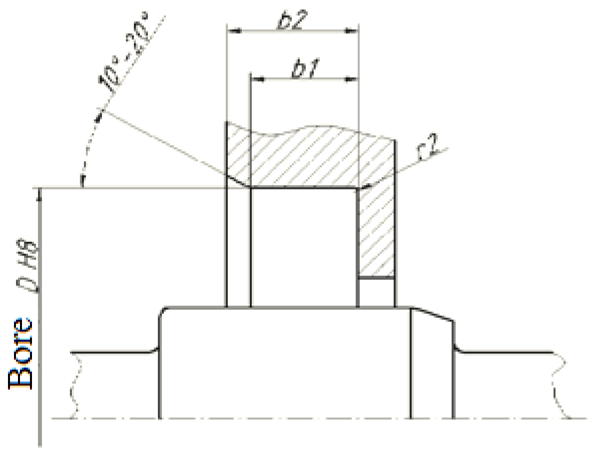
Designation of a radial shaft seal that is A kind and possesses: inside diameter (diameter of the shaft) *d* , outside diameter *D*, height *b* :

d	*D*	*b*	*d*	*D*	*b*	*d*	*D*	*b*
10	19,22,24,26,28	7	26	40	7	48	62	8
	22	8		42,45,47	10		65,70,72, 80	10
	30	10	28	40,47,50,52	7		72	12
11	22,26	7		47,50,52	10	50	65,72,80	8
	30	10	30	40,42,47,50,52, 62,72	7		68,70,72,75,80	10
12	22,24,26,28,30	7					72,80	12
	28,30,32	10		47,50,52,55,62	10	52	68,72	8
13	28	7		55,62	12		72,75,80	10
14	24,28,30	7	32	45,47,50,52	7	55	70,72,75,80	8
	30,32,35	10		44,50,52	10		72,75,80,85,90	10
15	24,26,30,32,35	7		52,55	12		80	12
	30	8	35	47,50,52,55,62, 72	7		80	13
	30,32,35,40	10				56	70,80	8
16	28,30	7		52,55,58,62,72	10	58	80,90	10
	32,35,40	10		52,55,56	12		80	12
17	28,30,32,35,40	7		80	13		80	13
	35,40	10	36	50,52,54	7	60	75,80,85	8
18	30,32,35,40	7		62	10		80,85,90	10
	35,40	10	38	50,52,55	7		85	12
19	35	7		55,58,62	10		80,85,90	13
20	30,32,35,40,47	7	40	52,55,60,62,80	7	62	80,85,90	10
	37,40,42,47	10		55,60,62,65,68, 72,80	10	63	85,90	10
21	40	10				65	85,90,95,100	10
22	32,35,40,42	7		62,65	12		90	13
	40,42,47	10	42	55	7	68	90,100	10
23	40	7		55,68	8	70	90,95,100	10
24	35,40,47	7		62,65,68,72	10		90,110	12
	40,47	10		68	12		90,95	13
25	35,37,40,42,45, 47,50,52,62,72	7	45	60,62,65,72,80	8	72	100	10
				62,65,68,72,80	10		110	12
	42,45,47,50,52, 62	10		62	12	75	95,100,105	10
							100,110	12
							100	13

If the main purpose of the seal is to prevent lubricant from leaving the housing, the seal should be fitted with the lip facing inwards (see figure below).



6.7 Mounting of radial shaft seal in housing



Seal width b[mm]	*b_1 = 0,85 x b [mm]*	*b_2 = b + 0,3 [mm]*	*r_{2max}*
7	5,95	7,3	0,5
8	6,80	8,3	
10	8,50	10,3	
12	10,30	12,3	0,7

Necessary tightness between the hole of housing and outside seals surface is guaranteed by mounting press. Recommended tolerance of hole is H8.

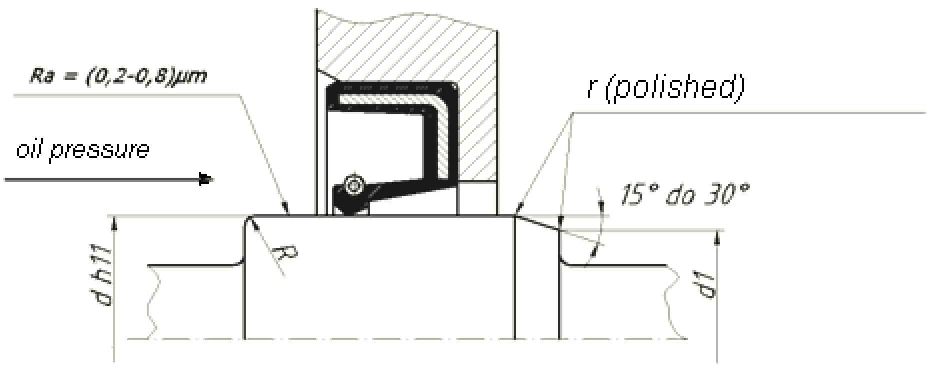
Surface qualities of housing nest by ISO 6194-1 are:

Ra =1,6 ÷ 6,3 µm

Rz = 10 ÷ 20 µm

Rmax = 16 ÷ 25 µm

6.8 Mounting of radial shaft seal on shaft



Recommended values of diameter d_1 and radius R

d	d_1	R
< 10	d– 1,5	2
over 10 to 20	d – 2,0	2
over 20 to 30	d – 2,5	3
over 30 to 40	d – 3,0	3
over 40 to 50	d – 3,5	4
over 50 to 70	d – 4,0	4
over 70 to 95	d – 4,5	5

6.9 Radial shaft seals under the pressure

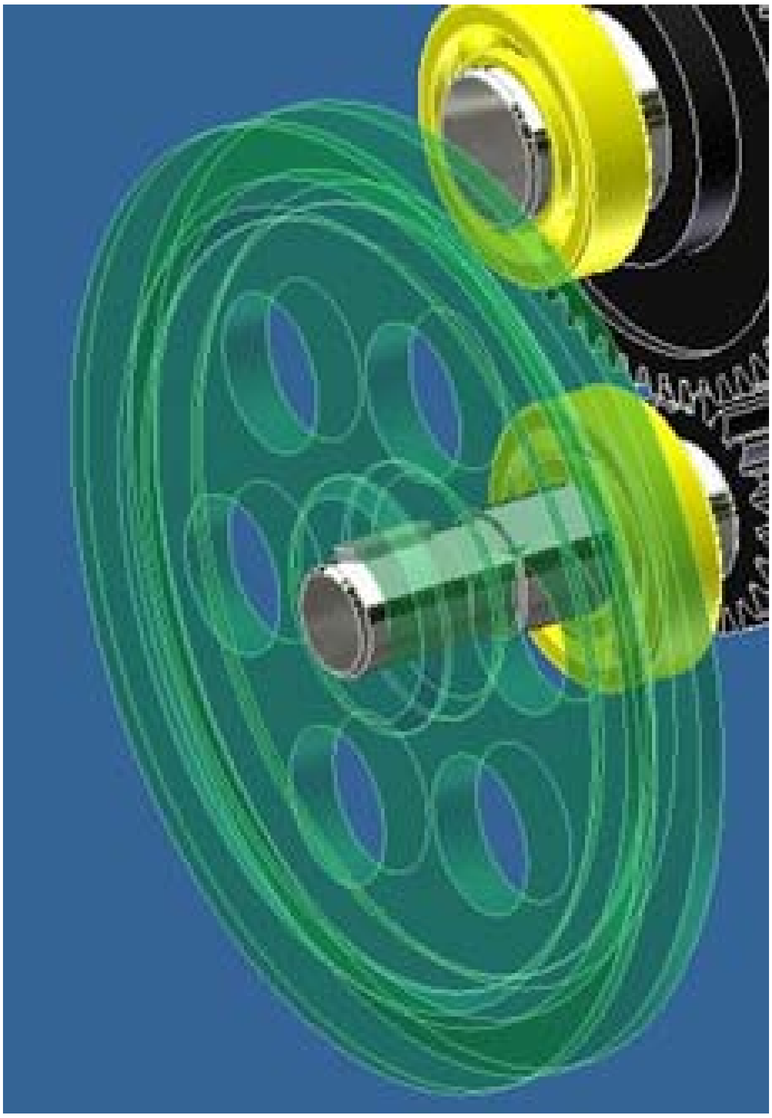
When shaft seals works under the pressure, the contact area between seal lip and shaft surface increases. The friction increases and generate a heat. As a result required shaft speed can’t be maintained. It must be reduced in relation to the magnitude of pressure.

6.10 Frictional loss

Frictional loss is a very important problem, when low powers are transmitted.
The value of frictional loss, measured in Watts, is 25÷200 for seal diameters to 50 mm and 200-350 or higher for seal diameters above 50 mm.

Chapter 7

V-BELT DRIVES
(Basis data and calculation in accordance with PN-M-85203: 1967)



7.1 V- belt power capacity

Power capacity of the drive is calculated from the formula:

$$P = P_0 z \frac{K_L K_\varphi}{K_T} \text{ [kW]}$$

where:

P – duty power

P_0 – duty power per belt (power transmitted with one belt)

z – number of belts

K_L – length factor

K_φ – contact factor

K_T – service factor

7.2 Small pulley equivalent diameter

It is calculated from

$$D_e = k_i \cdot D_1$$

where:

k_i – transmission ratio factor

D_1 – outside diameter of small pulley

7.3 Transmission ratio factor k_i

i	≤ 0,55 > 1,8	0,56 …0,83 1,21…1,8	0,84…0,95 1,05…1,2	095…1,05
k_i	1,15	1,10	1,05	1,00

Where

$$i = \frac{D_2}{D_1} \text{ V- belt drive ratio}$$

7.4 V – belt length

It is calculated from

$$L = \pi\frac{D_2 - D_1}{2} + \pi\frac{\gamma}{180}(D_2 - D_1) + 2A\cos(\gamma)$$

where:

$sin(\gamma) = \frac{D_2 - D_1}{2A}$

D_1 – outside small pulley diameter

D_2 – outside big pulley diameter

A – axis distance

7.5 Axis distance (recommended)

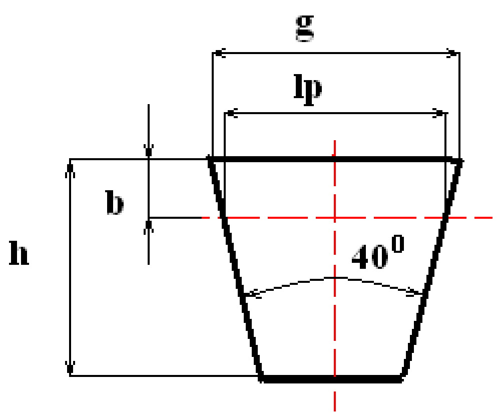
It is evaluated from

$$\frac{D_2 + D_1}{2} + c \leq A \leq 2(D_2 + D_1)$$

where:

c = 50 mm

7.6 V – belt dimensions (in accordance with PN-ISO 4184: 2000)



Tolerance of the angle $40^0 \pm 1^0$

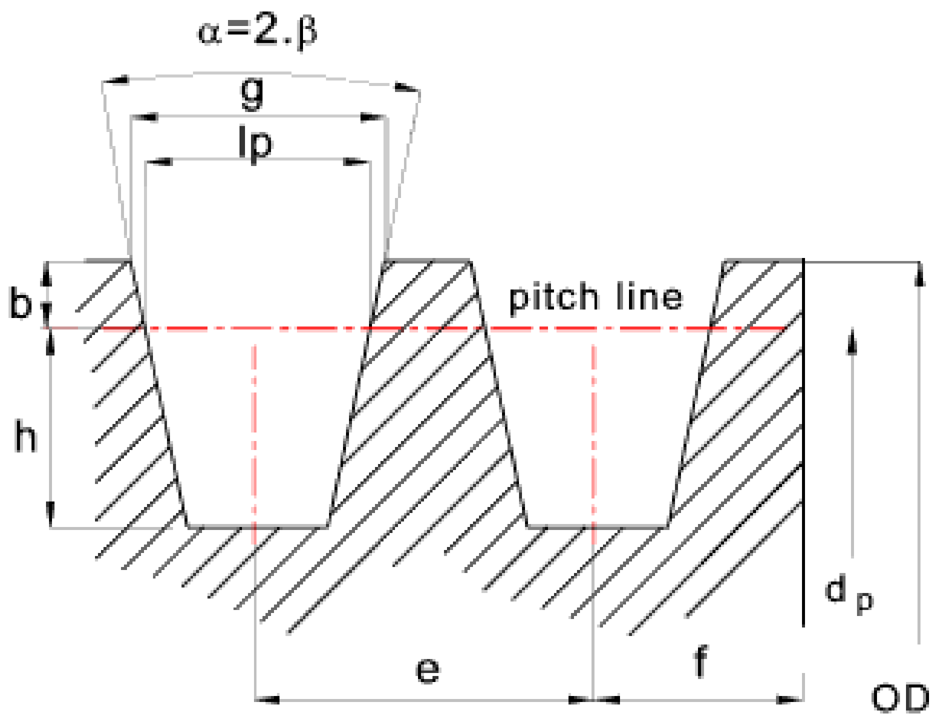
Belt section	g [mm]	lp [mm]	h [mm]	b [mm]
SPZ	10	8,5	6	2
SPA	13	11	8	3
SPB	17	14	11	3,5
SPC	22	19	14	4,5
SPD	32	27	19	7
SPE	38	32	23	8

7.7 V-belt length

V-belt length L		Belt section						V-belt length L		Belt section					
Recom-mended	Not recom-mended	Z	A	B	C	D	E	Recom mended	Not recom-mended	Z	A	B	C	D	E
400									2650						
	425							2800							
450									3000						
	475							3150							
500									3350						
	530							3550							
560									3750						
	600							4000							
630									4250						
	670							4500							
710									4750						
	750							5000							
800									5300						
	850							5600							
900									6000						
	950							6300							
1000									6700						
	1050							7100							
1120									7500						
	1180							8000							
1250									8500						
	1320							9000							
1400									9500						
	1500							10000							
1600									10600						
	1700							11200							
1800									11800						
	1900							12500							
2000									13200						
	2120							14000							
2240									15000						
	2360							16000							
2500									17000						
	2650							18000							

7.8 Pulley Groove dimensions

Dimensions of the pulley groove are presented in below sketch. Their values are gathered in below table.

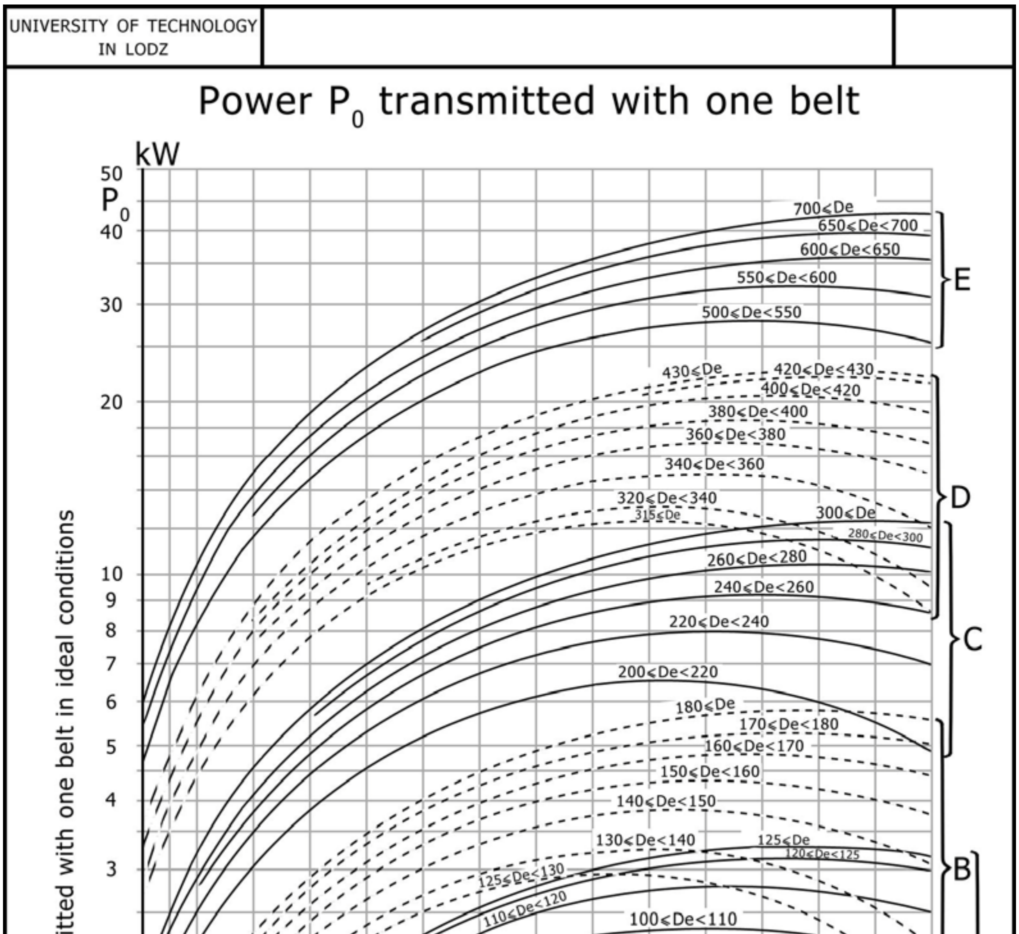


Belt Section	lp mm	b mm	h mm	e mm	f mm
SPZ	8,5	2,5	9,0	12,0±0,3	8,0±1
SPA	11,0	3,3	11,0	15,0±0,3	10,0+2/-1
SPB	14,0	4,2	14,0	19,0±0,4	12,5+2/-1
SPC	19,0	5,7	19,0	25,5±0,5	17,0+2/-1
SPD	27,0	8,1	27,0	37±0,6	24,0+3/-1
SPE	32,0	9,5	32	44,5±0,8	29,0+4/-1

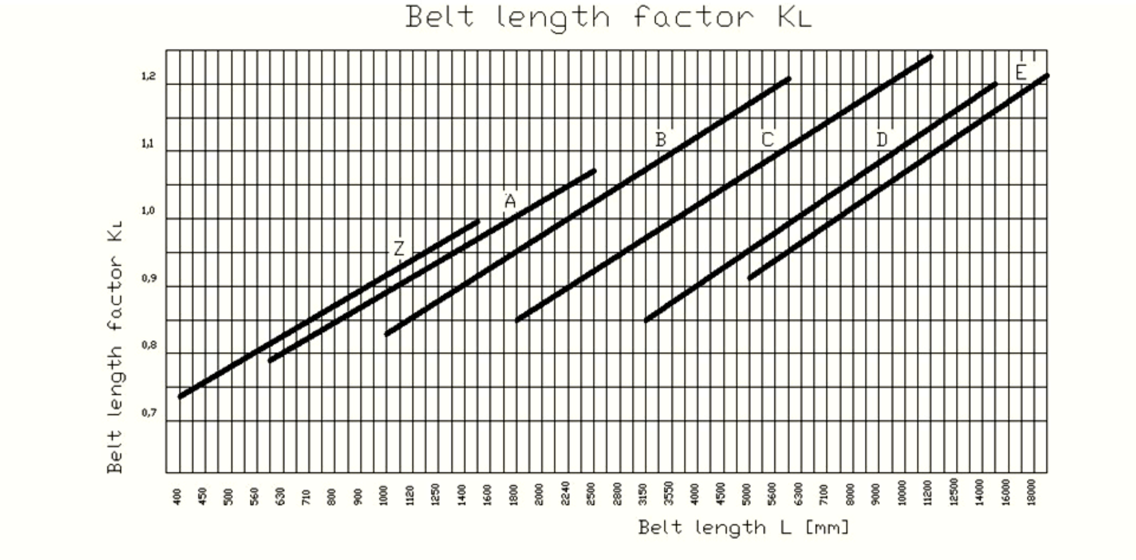
7.9 Pulley diameters d_p

<table>
<tr><th colspan="2">Pulley diam. D_p</th><th colspan="6">v-belt cross section angle α</th><th colspan="2">Pulley diam. D_p</th><th colspan="6">v-belt cross section angle α</th></tr>
<tr><th>Recom-mended</th><th>Not recom-mended</th><th>Z</th><th>A</th><th>B</th><th>C</th><th>D</th><th>E</th><th>Recom-mended</th><th>Not recom-mended</th><th>Z</th><th>A</th><th>B</th><th>C</th><th>D</th><th>E</th></tr>
<tr><td>63</td><td></td><td rowspan="4">34^0 $\pm1^0$</td><td colspan="5" rowspan="6"></td><td>400</td><td></td><td rowspan="13"></td><td rowspan="13">38^0 $\pm1^0$</td><td rowspan="19">38^0 $\pm1^0$</td><td rowspan="25">38^0 $\pm30'$</td><td rowspan="4">36^0 $\pm30'$</td><td rowspan="4"></td></tr>
<tr><td></td><td>67</td><td></td><td>425</td></tr>
<tr><td>71</td><td></td><td>450</td><td></td></tr>
<tr><td></td><td>75</td><td></td><td>475</td></tr>
<tr><td>80</td><td></td><td rowspan="6">35^0 $\pm1^0$</td><td>500</td><td></td><td rowspan="25">38^0 $\pm30'$</td><td rowspan="4">36^0 $\pm30'$</td></tr>
<tr><td></td><td>85</td><td></td><td>530</td></tr>
<tr><td>90</td><td></td><td rowspan="6">34^0 $\pm1^0$</td><td colspan="4" rowspan="6"></td><td>560</td><td></td></tr>
<tr><td></td><td>95</td><td></td><td>600</td></tr>
<tr><td>100</td><td></td><td>630</td><td></td><td rowspan="25">38^0 $\pm30'$</td></tr>
<tr><td></td><td>106</td><td></td><td>670</td></tr>
<tr><td>112</td><td></td><td rowspan="15">38^0 $\pm1^0$</td><td>710</td><td></td></tr>
<tr><td></td><td>118</td><td></td><td>750</td></tr>
<tr><td>125</td><td></td><td rowspan="8">36^0 $\pm1^0$</td><td rowspan="6">34^0 $\pm1^0$</td><td colspan="3" rowspan="8"></td><td>800</td><td></td></tr>
<tr><td></td><td>132</td><td></td><td>850</td><td colspan="2" rowspan="6"></td></tr>
<tr><td>140</td><td></td><td>900</td><td></td></tr>
<tr><td></td><td>150</td><td></td><td>950</td></tr>
<tr><td>160</td><td></td><td>1000</td><td></td></tr>
<tr><td></td><td>170</td><td></td><td>1050</td></tr>
<tr><td>180</td><td></td><td rowspan="8">36^0 $\pm1^0$</td><td>1120</td><td></td></tr>
<tr><td></td><td>190</td><td></td><td>1180</td><td colspan="3" rowspan="6"></td></tr>
<tr><td>200</td><td></td><td rowspan="13">38^0 $\pm1^0$</td><td rowspan="10">36^0 $\pm30'$</td><td colspan="2" rowspan="8"></td><td>1250</td><td></td></tr>
<tr><td></td><td>212</td><td></td><td>1320</td></tr>
<tr><td>224</td><td></td><td>1400</td><td></td></tr>
<tr><td></td><td>236</td><td></td><td>1500</td></tr>
<tr><td>250</td><td></td><td>1600</td><td></td></tr>
<tr><td></td><td>265</td><td rowspan="8"></td><td></td><td>1700</td><td colspan="4" rowspan="4"></td></tr>
<tr><td>280</td><td></td><td rowspan="7">38^0 $\pm1^0$</td><td>1800</td><td></td></tr>
<tr><td></td><td>300</td><td></td><td>1900</td></tr>
<tr><td>315</td><td></td><td rowspan="5">36^0 $\pm30'$</td><td rowspan="5"></td><td>2000</td><td></td></tr>
<tr><td></td><td>335</td><td></td><td>2120</td><td colspan="5" rowspan="4"></td></tr>
<tr><td>355</td><td></td><td rowspan="3">38^0 $\pm30'$</td><td>2240</td><td></td></tr>
<tr><td></td><td>375</td><td></td><td>2360</td></tr>
<tr><td>400</td><td></td><td>2500</td><td></td></tr>
</table>

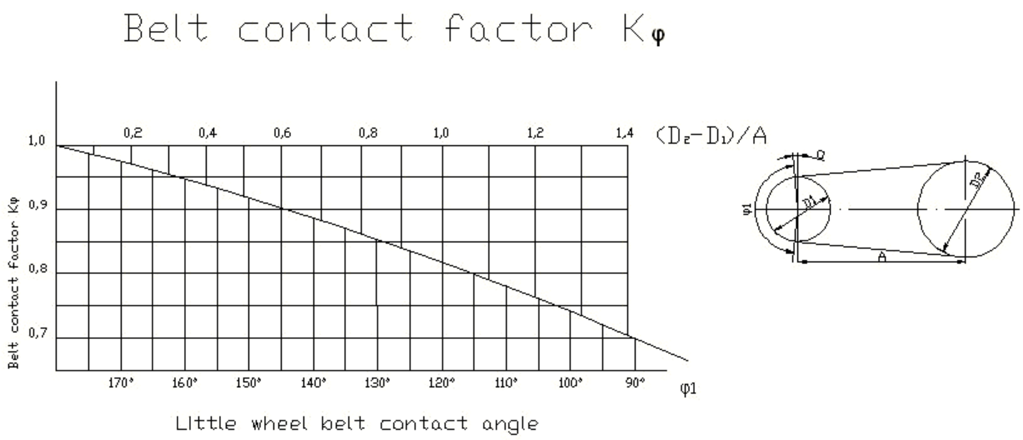
7.10 Duty power per belt (power transmitted with one belt) P_0



7.11 Belt length factor K_L



7.12 Belt contact factor $K\varphi$



7.13 Service factors K_T (time and work conditions factor)

Service Factors for Synchronous Drives

Driven Equipment; | v	Operating Period /day								
	< 8 h	< 16 h	> 16 h	< 8 h	< 16 h	> 16 h	< 8 h	< 16 h	>16 h
	Driving Machine:Smooth Starting torque 100% - 150% FL			Driving :Moderate Vibration Starting torque 150% - 250% FL			Driving Machine: Heavy Vibration Starting torque 250% - 400% FL		
	Electic Motors –Star Delta DC motors-shunt wound IC engines over 8 cyl'rs Water Turbines Steam Turbines			AC motors starting torque IC engines with 4-6 cylinders			AC Motors (DOL) Two stroke Engines IC engines under 600 rpm,		
Instrumentation Office display equipment Medical Equipment	1	1.2	1.4	1.1	1.3	1.5	1.2	1.4	1.6
Domestic Appliances Oven Screens Woodworking Drills Woodworking Drills Lathes	1.1	1.3	1.5	1.2	1.4	1.6	1.3	1.5	1.7
Liquid agitators, Heavy Woodworking Tools, Belt conveyors	1.2	1.4	1.6	1.4	1.6	1.8	1.6	1.8	2
Sludge Agitators, Conveyors, Milling Machines Shaper Machines Grinding Machines	1.3	1.5	1.7	1,5	1,7	1,9	1.6	1.8	2
Brick Machinery Rubber Calendar Mills Rubber Extruders Centrifugal Blowers Elevators	1.4	1.6	1.8	1.6	1.8	2	1.8	2	2.2
Centrifugers, Paper Pulpers, Hammer Mills,	1.5	1.7	1.9	1.7	1.9	2.1	1.9	2.1	2.3
Blowers-positive 68isplacement Pulverisers. Mine Fans	1.6	1.8	2	1.8	2	2.2	2	2.2	2.4
Reciprocating Compressors Crushers Steelmaking machines Reciprocating pumps	1.7	1.9	2.1	1.9	2.1	2.3	2.1	2.3	2.5

Service Factors of V–Belt Drives

Driven Equipment; | v	Operating Period /day					
	< 8 h	< 16 h	> 16 h	< 8 h	< 16 h	>16 h
	Driving Machine: Reasonable Smooth Starting torque 100% - 200% FL			Driving Machine: Coarse Starting torque 200%-400%		
	Electic Motors –Star Delta DC motors-shunt wound IC engines over 8 cyl'rs Turbines			DC Motors with series/mixed excitation AC Motors (DOL) Two stroke Engines IC engines under 600 rpm,		
Smooth Machine Tools-Lathes drills etc Liquid agitators, Blowers, Exhausters Light Duty conveyors Centrifugal Pumps	1,0	1,1	1,2	1,1	1,2	1,3
Uneven Machine Tools-Milling /Gearcutting;etc Printing machines Laundry Machinery Generators Piston pumps and compressors(4+ cylinder) Fans and blowers Chain conveyors Elevators Circular saws for wood Transmissions Printing ,Paper-making machines Food-industry machines-dough mixers Heavy screens Rotary furnaces High-speed grinders Positive Displacement Rotary Pumps Revolving Vibrating screens Punches/Presses	1,1	1,2	1,3	1,2	1,3	1,4
Planing machines Vertical shapers and wood-processing machines Piston pumps and compressors with one or two cylinders Bucket elevators Fans and blowers of heavy types Exciters Screw and drag conveyors Crushers Piston Pumps Presses heavy flywheels Weaving machines Machines for cleaning cotton Machines for pressing and pelletising fodder Positive Displacement Blowers	1,2	1,3	1,4	1,3	1,4	1,6
Hoists Excavators dredgers Heavy presses shears Mechanical hammers Mills(ball, rod, tube) Stone crushers Hammer mills Crushers (gyratory, jaw, roll) Sawmill machines Rubber Calendars, extruders, Mills	1,3	1,4	1,5	1,5	1,6	1,8

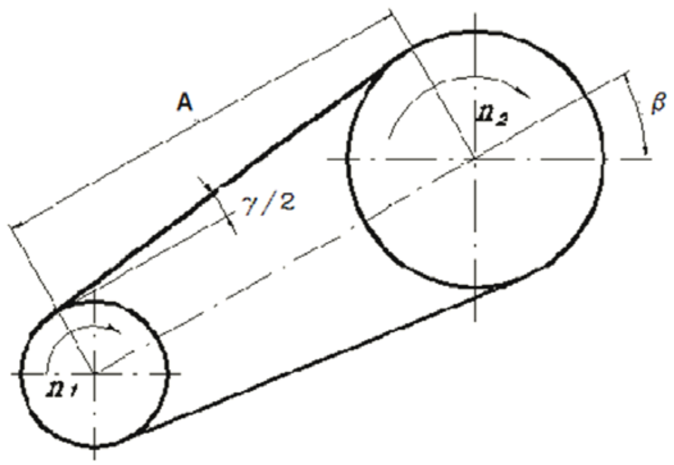
7.14 Example of V-belt drive calculation

Let us consider the V-belt drive that has following parameters:

- AC electric motor power P = 3 kW
- Motor rotational speed n_1 = 950 rpm
- Output rotational speed n_2 = 320 rpm
- Distance between axes of pulleys A = 480 mm
- Slope of the distance line between axes of pulley $\beta = 55^0$
- Acceptable error of belt drive ratio Δ = 3%

Driven is a fan less than 8 hrs a day. Find a size of the drive, a number of belts, and a force acting on the low-speed shaft.

A sketch of the drive is presented in the below figure.



Sketch of the calculated drive

1. <u>Limitation of the belt drive ratio</u>

Belt drive ratio

$$i = \frac{n_1}{n_2} = 2.969$$

$$i_{max} = i \cdot \left(1 + \frac{\Delta}{2 \cdot 100}\right) = 3.013$$

$$i_{min} = i \cdot \left(1 + \frac{\Delta}{2 \cdot 100}\right) = 2.924$$

2. <u>The belt of A-shape is chosen (look at “Duty power per belt” diagram)</u>

Formula for a (recommended) distance of axis

$$\frac{D_1 + D_2}{2} + 50 \leq A \leq 2 \cdot (D_1 + D_2)$$

But $i = \frac{D_2}{D_1}$

and $D_{1max} = 2 \cdot \frac{A-50}{1+i} = 216.69$ mm

$$D_{1min} = \frac{A}{2 \cdot (1+i)} = 60.47 \text{ mm}$$

It is chosen (look at “Pulley Diameters d_p” table)

$$D_1 = 125 \text{ mm}$$

So $D_2 = i \cdot D_1 = 371.09$ mm

After correction (in accordance with “Pulley Diameters d_p” table)

$$D_{2cor} = 375 \text{ mm}$$

And a real ratio of the drive is as follow

$$i_{real} = \frac{D_{2cor}}{D_1} = 3$$

The belt drive ratio is acceptable.

3. <u>Selection of a length of the belt</u>

Angle of a slope of the belt is as follow

$$sin\gamma = \frac{D_{2cor} - D_1}{2 \cdot A}$$

And $\gamma = asin\frac{D_{2cor} - D_1}{2 \cdot A} = 15.1°$

Calculation of a length of the belt

$$L = \pi \cdot \frac{D_1 + D_{2cor}}{2} + \pi \cdot \frac{\gamma}{180} \cdot (D_{2cor} - D_1) + 2 \cdot A \cdot cos\gamma = 1778 \text{ mm}$$

It is chosen (look at “V-belt length” table)

$$L_r = 1800 \text{ mm}$$

Estimation of the distance between pulley axis

$$A_{est} = \frac{1}{2 \cdot cos\gamma} \cdot \left[L_r - \frac{1}{2} \cdot \pi(D_1 + D_{2cor}) - \frac{\gamma \cdot \pi}{180} \cdot (D_{2cor} - D_1)\right] = 507.3 \text{ mm}$$

Angle corrected of a slope of the belt

$$\gamma_c = asin\frac{D_{2cor} - D_1}{2 \cdot A_{est}} = 14.27°$$

Real value of the distance between pulley axis

$$A_r = \frac{1}{2 \cdot cos\gamma}\left[L_r - \frac{\pi}{2} \cdot (D_1 + D_{2cor}) - \frac{\gamma \cdot \pi}{180} \cdot (D_{2cor} - D_1)\right] = 491.32 \text{ mm}$$

$$\Delta A = \frac{A_r - A}{A} \cdot 100 = 2.36\ \%$$

4. <u>Selection of a belt number</u>

Transmission ratio factor

for $i = 2.969$ from “Transmission ratio factor” table

$$k_i = 1.15$$

Belt length factor

for A-shape belt and L_r = 1800 mm from “Belt length factor” diagram

$$k_L = 1$$

Belt contact factor

for $\frac{D_{2cor} - D_1}{A_r} = 0.51$ in accordance with “ Belt contact factor” diagram

$$k_\varphi = 0.93$$

Belt velocity

$$v = \frac{\pi \cdot D_1 \cdot n_1}{60 \cdot 10^3} = 6.22\ \text{m/s}$$

Effective diameter

$$D_e = k_i \cdot D_1 = 143.8\ \text{mm}$$

For D_e = 143.8 mm and v = 6.22 m/s from “Duty power per belt” diagram there is found that acceptable power of one belt is

$$P_0 = 1.4\ \text{kW}$$

Driven is a fan less than 8 hrs a day. So service factor from “ Service Factors of V-belt Drives” table is

$$k_T = 1.2$$

So a number of belt calculated

$$z_C = \frac{P}{P_0} \cdot \frac{k_T}{k_L \cdot k_\varphi} = 2.77$$

And the number of belts

$$z = 3$$

Angle of a groove of the pulleys is

$$\alpha = 36° \pm 1°$$

The drive will have 3 V-belts of A-shape and 1800 mm of length.

5. <u>Force acted on the shaft</u>

Efficiency of the drive

$$\eta = 0.95$$

Coefficient of friction between the pulley and the belt

$$\mu = 0.25$$

Apparent coefficient of friction

$$\mu_a = \frac{\mu}{sin\alpha} = 0.96$$

Useful tension of the belt

$$F_u = \frac{P \cdot \eta}{v} = 0.458 \text{ kN}$$

Angle of contact

$$\varphi_L = (180 - 2 \cdot \gamma) \cdot \frac{\pi}{180} = 2.615 \text{ rd}$$

Euler's formula factor

$$m = e^{\mu_a \cdot \varphi_L} = 12.3$$

Initial tension of the belt

$$F_0 = \frac{1}{2} \cdot F_u \cdot \frac{m+1}{m-1} = 0.27 \text{ kN}$$

Tension in an active part of the belt

$$F_1 = F_0 + \frac{1}{2} \cdot F_u = 0.5 \text{ kN}$$

Tension in a passive part of the belt

$$F_2 = F_0 - \frac{1}{2} \cdot F_u = 0.04 \text{ kN}$$

Equivalent load of shafts (electric motor and machine)

$$F_e = \sqrt{F_1^2 + F_2^2 + 2 \cdot F_1 \cdot F_2 \cdot cos2\gamma} = 0.53 \text{ kN}$$

Angle of slope of the load from the drive axis

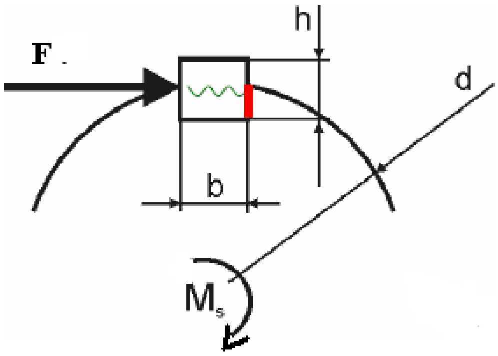
$$\Theta = atan\left(\frac{F_1 - F_2}{F_1 + F_2} \cdot tan\gamma\right) = 12.9°$$

Angle of total slope of the load

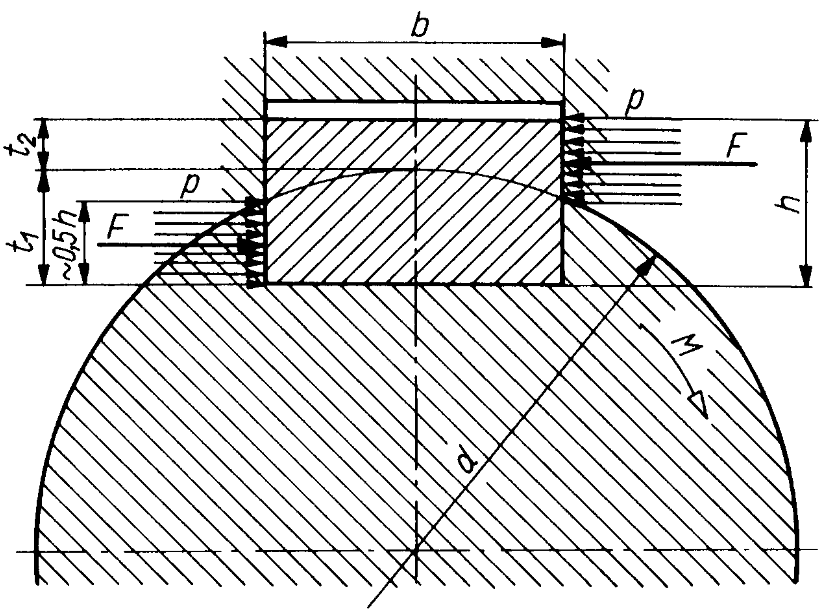
$$\psi = \beta + \Theta = 67.9°$$

KEY JOINT
Keyway and key dimensions

8.1. Key load sketch



8.2. Tension distribution



8.3. Durability calculation

<u>Important:</u>

Pressure area determine the durability of key joint (red line on load sketch)
Calculation of key cutting are passed over (green line on load sketch)

Acceptable stress in a key is calculated from the formula

$$\sigma_d = k\frac{F}{z \cdot A_d} \le k_d$$

where:

$$F = \frac{M_s}{d}$$

here:

k_d – permissible tension

z – number of keys (max. 2)

k – factor (for key joint $k = 1$)

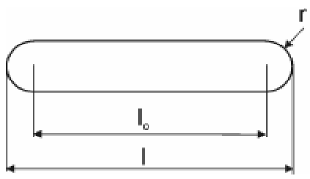
A_d – area of pressure

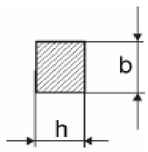
- for key form B (square at both ends)

$$A_d = b \cdot \frac{h}{2}$$

- for keys form A (radius at both ends)

$$l_0 = l - 2 \cdot r = l - b$$





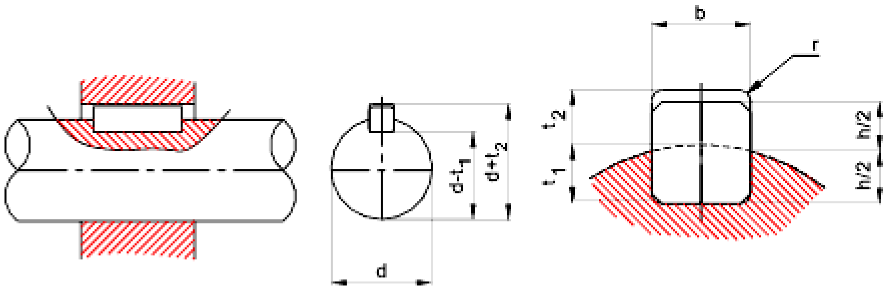
$$A_d = (l - b) \cdot \frac{h}{2}$$

Kind of material		Permissible tension k_d [MPa]	
Key	**Shaft/Hub**	**Static joint**	**Moving joint**
E295 E360	Cast iron	30-50	20-40
E295 E360 C45	Steel	60-90	20-40
E360 C45	Hardened steel	200-300	120-200

8.4. Specification for metric rectangular keys and keyways

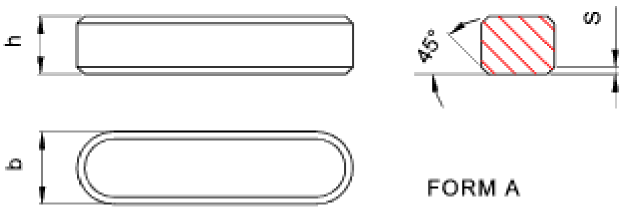
(Source: http://www.roymech.co.uk/Useful_Tables/Keyways/keyways.htm}

Keyway and key dimensions - form B (square at both ends)



Nominal Shaft Diameter d		Key	KeyWay											
		b x h width x thck	Width b						Depth				Radius r	
			Nom	Tolerance Class					Shaft t_1		Hub t_2			
				Free		Normal		Close/Int	-	-	-	-		
Over	To			Shaft H9	Hub D10	Shaft N9	Hub Js9	Shaft/Hub P9	Nom	Tol	Nom	Tol	Max	min
6	8	2x2	2	+,025 0	+0,06 +0,02	-0,004	+0,012	-0,006	1,2	+0,1 0	1,0	+0,1 0	0,16	0,08
8	10	3x3	3			-0,029	-0,012	-0,031	1,8		1,4		0,16	0,08
10	12	4x4	4	+0,03 0	+0,078 +0,030	0 -0,030	+0,015 -0,015	-0,012 -0,042	2,5		1,8		0,16	0,008
12	17	5x5	5						3,0		2,3		0,25	0,16
17	22	6x6	6						3,5		2,8		0,25	0,16
22	30	8x7	8	+0,036 0	+0,098 +0,040	0 -0,036	+0,018 -0,018	-0,015 -0,051	4,0	+0,2 0	3,3	+0,2 0	0,25	0,16
30	38	10x8	10						5,0		3,3		0,40	0,25
38	44	12x8	12	+,0430	+0,12 +0,050	0 -0,043	+0,021 -0,021	-0,018 -0,061	5,0	+0,2 0	3,3	+0,2 0	0,40	0,25
44	50	14x9	14						5,5		3,8		0,40	0,25
50	58	16x10	16						6,0		4,3		0,40	0,25
58	65	18x11	18						7,0		4,4		0,40	0,25
65	75	20x12	20	+0,052 0	+0,149 +0,065	0 -0,052	+0,026 -0,026	-0,022 -0,074	7,5		4,9		0,60	0,40
75	85	22x14	22						9,0		5,4		0,60	0,40
85	95	25x14	25						9,0		5,4		0,60	0,40
95	110	28x16	28						10,0		6,4		0,60	0,40
110	130	32x18	32	+0,062 0	+0,18 +0,080	0 -0,062	+0,031 -0,031	-0,026 -0,088	11,0		7,4		0,6	0,4
130	150	36x20	36						12,0	+0,3 0	8,4	+0,3 0	1,0	0,7
150	170	40x22	40						13,0		9,4		1,0	0,7
170	200	45x25	45						15,0		10,4		1,0	0,7
200	230	50x28	50						17,0		11,4		1,0	0,7
230	260	56x32	56	+0,074 0	+0,220 0,100	0 -0,074	+0,037 -0,037	-0,032 -0,106	20,0		12,4		1,6	1,2
260	290	63x32	63						20,0		12,4		1,6	1,2
290	330	70x36	70						22,0		14,4		1,6	1,2
330	380	80x40	80						25,0		15,4		2,5	2,0
380	440	90x45	90	+0,087 0	+0,260 0,120	0 -0,087	+0,043 -0,043	-0,037 -0,124	28,0		17,4		2,5	2,0
440	500	100x50	100						31,0		19.5		2,5	2,0

Key dimensions – form A (radius at both ends)



Width b		Thickness h		Chamfer S		Range Of Lengths	
Nom	Tol(h9)	Nom	Tol(h9)	Min	Max	From	Inc
2	0 -0,025	2	0 -0,025	0,16	0,25	6	20
3		3		0,16	0,25	6	36
4	0 -0,030	4	0 -0,030	0,16	0,25	8	45
5		5		0,25	0,40	10	56
6		6		0,25	0,40	14	70
8	0 -0,036	7	Tol (h11) .... 0 -0,090	0,25	0,40	18	90
10		8		0,40	0,60	22	110
12	0 -0,043	8		0,40	0,60	28	140
14		9		0,40	0,60	36	160
16		10		0,40	0,60	45	180
18		11	0 -0,110	0,40	0,60	50	200
20	0 -0,052	12		0,60	0,80	56	220
22		14		0,60	0,80	63	250
25		14		0,60	0,80	70	280
28		16		0,60	0,80	80	320
32	0 -0,062	18		0,60	0,80	90	360
36		20	0 -0,130	1,00	1,20	100	400
40		22		1,00	1,20	-	-
45		25		1,00	1,20	-	-
50		28		1,00	1,20	-	-
56	0 -0,074	32	0 -0,160	1,60	2,00	-	-
63		32		1,60	2,00	-	-
70		36		1,60	2,00	-	-
80		40		2,50	3,00	-	-
90	0 -0,087	45		2,50	3,00	-	-
100		50		2,50	3,00	-	-

(Source: http://www.tasmanindustries.co.uk/images/stories/productbrochures/keys.pdf)

Key lengths

(Source: http://www.tasmanindustries.co.uk/images/stories/productbrochures/keys.pdf)

Width b	2	3	4	5	5	6	6	8	8	10	10	12	12	14	14	16	16
Height h	2	3	4	3	5	4	6	5	7	6	8	6	8	6	9	7	10
tolerance h9 +0.00	-0.025	-0.025	-0.030	-0.030	-0.030	-0.030	-0.030	-0.036	-0.036	-0.036	-0.036	-0.043	-0.043	-0.043	-0.043	-0.043	-0.043
tolerance h11 +0.00	-0.025	-0.025	-0.030	-0.060	-0.030	-0.075	-0.030	-0.030	-0.090	-0.075	-0.090	-0.075	-0.090	-0.075	-0.090	-0.090	-0.090
Length L	Preferred Stock Length																
6				-	-	-	-	-	-	-	-	-	-	-	-	-	-
8				-	-	-	-	-	-	-	-	-	-	-	-	-	-
10										-	-	-	-	-	-	-	-
12										-	-	-	-	-	-	-	-
14												-	-	-	-	-	-
15												-	-	-	-	-	-
16												-	-	-	-	-	-
18												-	-	-	-	-	-
20														-	-	-	-
22														-	-	-	-
25																-	-
28																-	-
30																	
32																	
35																	
36																	
40	-	-															
45	-	-															
50	-	-															
55	-	-	-														
56	-	-	-														
60	-	-	-														
63	-	-	-														
65	-	-	-														
70	-	-	-	-	-												
75	-	-	-	-	-												
80	-	-	-	-	-	-	-										
85	-	-	-	-	-	-	-										
90	-	-	-	-	-	-	-										
95	-	-	-	-	-	-	-	-									
100	-	-	-	-	-	-	-	-									
110	-	-	-	-	-	-	-	-									
120	-	-	-	-	-	-	-	-		-							
125	-	-	-	-	-	-	-	-		-							
130	-	-	-	-	-	-	-	-		-							
140	-	-	-	-	-	-	-	-		-							
150	-	-	-	-	-	-	-	-		-		-		-			
160	-	-	-	-	-	-	-	-		-		-		-			
170	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-		
180	-	-	-	-	-	-	-	-		-	-	-		-			
190	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
200	-	-	-	-	-	-	-	-		-		-	-	-		-	
220	-	-	-	-	-	-	-	-	-	-		-	-	-		-	-

Width b	18		20		22			24	25						28		
Height h	7	11	8	12	9	14	22	24	9	14	16	20	22	25	10	16	28
tolerance h9 +0.00	-0.043	-0.043	-0.052	-0.052	-0.052	-0.052	-0.052	-0.052	-0.052	-0.052	-0.052	-0.052	-0.052	-0.052	-0.052	-0.052	-0.052
tolerance h11 +0.00	-0.090	-0.110	-0.090	-0.110	-0.110	-0.110	-0.130	-0.052	-0.090	-0.110	-0.110	-0.130	-0.130	-0.052	-0.090	-0.110	-0.052
Length L	Preferred Stock Length																
32	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
35	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
36	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
40	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
45			-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
50			-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
55					-	-	-	-	-	-	-	-	-	-	-	-	-
56					-	-	-	-	-	-	-	-	-	-	-	-	-
60							-	-	-	-	-	-	-	-	-	-	-
63							-	-	-	-	-	-	-	-	-	-	-
65							-	-			-	-	-	-	-	-	-
70							-	-			-	-	-	-	-	-	-
75							-	-			-	-	-	-			-
80							-	-			-	-	-	-			-
85							-	-			-	-	-	-			-
90							-	-			-	-	-	-			-
95							-	-			-	-	-	-			-
100							-	-			-	-	-	-			-
110							-	-			-	-	-	-			-
120							-	-			-	-	-	-			-
125							-	-			-	-	-	-			-
130							-	-			-	-	-	-			-
140							-	-			-	-	-	-			-
150							-	-			-	-	-	-			-

Chapter 9

GEAR – CASE DESIGN

Presented below picture should be helpful for a gear-case design.

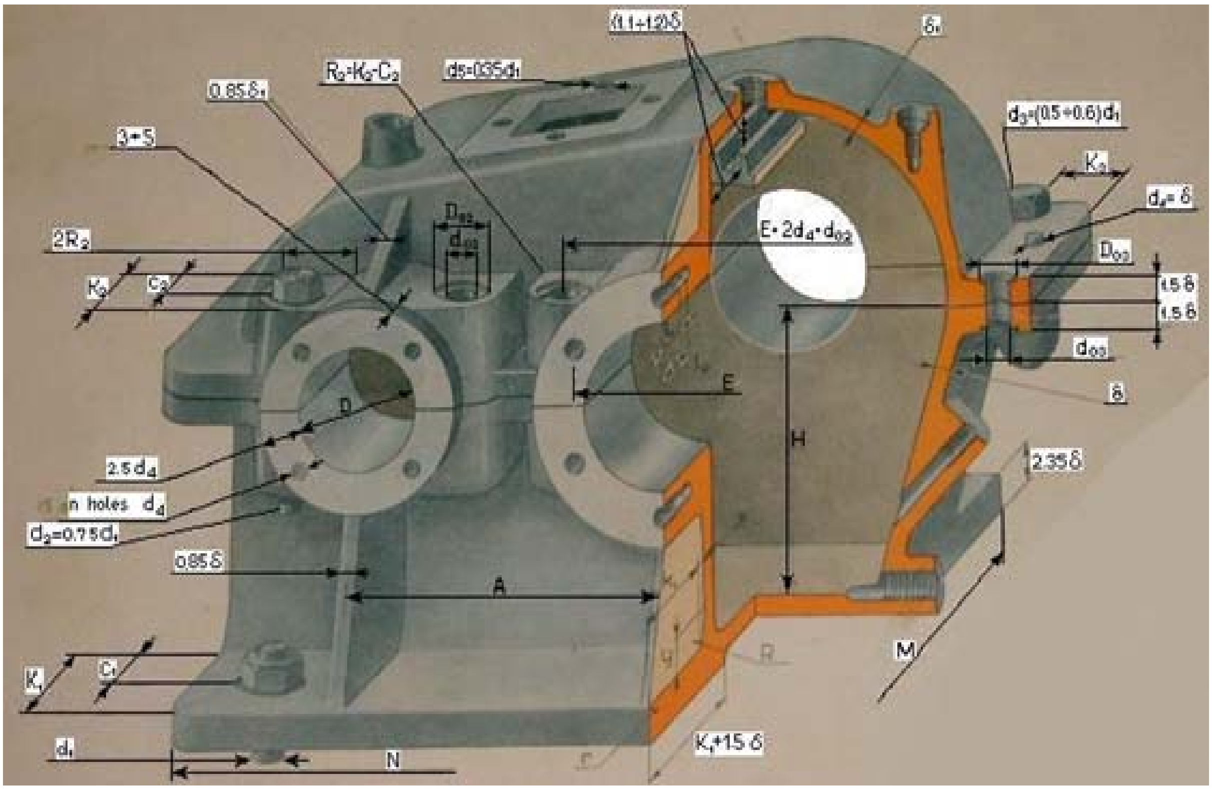


Fig.9.1. Gear-case for a one stage reducer

Where:

$$H = A$$

$$E = 1{,}2 \cdot D$$

$$\delta = 0.025 \cdot A + 1 \geq 7.5$$

$$\delta_1 = 0.02 \cdot A + 1 \geq 7.5$$

$$d_1 = 12 + 0.036 \cdot A$$

$$d_4 = 0.08 \cdot D$$

To design (draw) a gear-case we should have given:

- Centre distance (A),
- Bores of housings (D) for selected ball bearings of shafts; their nominal diameters are the same like outside diameters of the bearings,
- Face – width of the pinion.

First step of our design is to evaluate a side wall thickness of the gear-case. We use the formula:

$$\delta = 0.025 \cdot A + 1 \geq 7.5 \text{ mm (for a case),}$$

and

$$\delta_1 = 0.02 \cdot A + 1 \geq 7.5 \text{ mm (for a top of the case).}$$

<u>Note:</u> All formulae used here possess an approximate character only.

<u>*And so, for example for $A = 178.5$ mm we have got*</u>

$$\delta = 0.025 \cdot 178.5 + 1 = 5.46 \text{ mm}$$

Supporting with it we accept

$$\delta = \delta_1 = 8 \text{ mm}$$

Next we choose a size of foundation bolts. We do it using the formula:

$$d_1 = 12 + 0.036 \cdot A$$

$$d_1 = 12 + 0.036 \cdot 178.5 = 18.4 \text{ mm}$$

TABLE 9.1. DIMENSIONS OF A FLANGE (K), A DISTANCE FROM THE WALL TO THE BOLT AXIS (C), A HOLE FOR THE BOLT (D_0), AND A SPOTFACE (D_0)

d	**K**	**C**	**d_0**	**D_0**
M6	22	12	7	14
M8	24	13	9	17
M10	28	15	11	20
M12	33	18	13	36
M14	33	18	15	28
M16	40	21	17	32
M18	46	25	20	34
M20	48	25	22	38
M24	54	27	26	45
(M27)	60	30	29	50
M30	66	33	32	60
(M33)	72	36	36	65
M36	78	39	40	70
(M40)	84	41	44	76
M42	90	44	46	82
(M45)	95	46	50	88
M48	101	49	52	95

In accordance with Table 9.1 it was chosen

$$d_1 = M18$$

$$K_1 = 46 \text{ mm}$$

$$C_1 = 25 \text{ mm}$$

$$d_{01} = 20 \text{ mm}$$

$$D_{01} = 34 \text{ mm}$$

Notice that the thickness of this flange is

$$thickness = 2.35 \cdot \delta = 2.35 \cdot 8 = 18.8 \approx 20 \text{ mm}$$

and its bottom width is

$$width = K_1 + 1.5 \cdot \delta = 46 + 1.5 \cdot 8 = 58 \text{ mm}$$

As to bolts of the housing, their size is evaluated with

$$d_2 = 0.75 \cdot d_1 = 0.75 \cdot 18 = 13.5 \text{ mm}$$

In accordance with Table 9.1 it was chosen

$$d_2 = M14$$

$$K_2 = 33 \text{ mm}$$

$$C_2 = 18 \text{ mm}$$

$$d_{02} = 15 \text{ mm}$$

$$D_{02} = 28 \text{ mm}$$

Caps for bearing housings
(Blind and for lip seal ones)

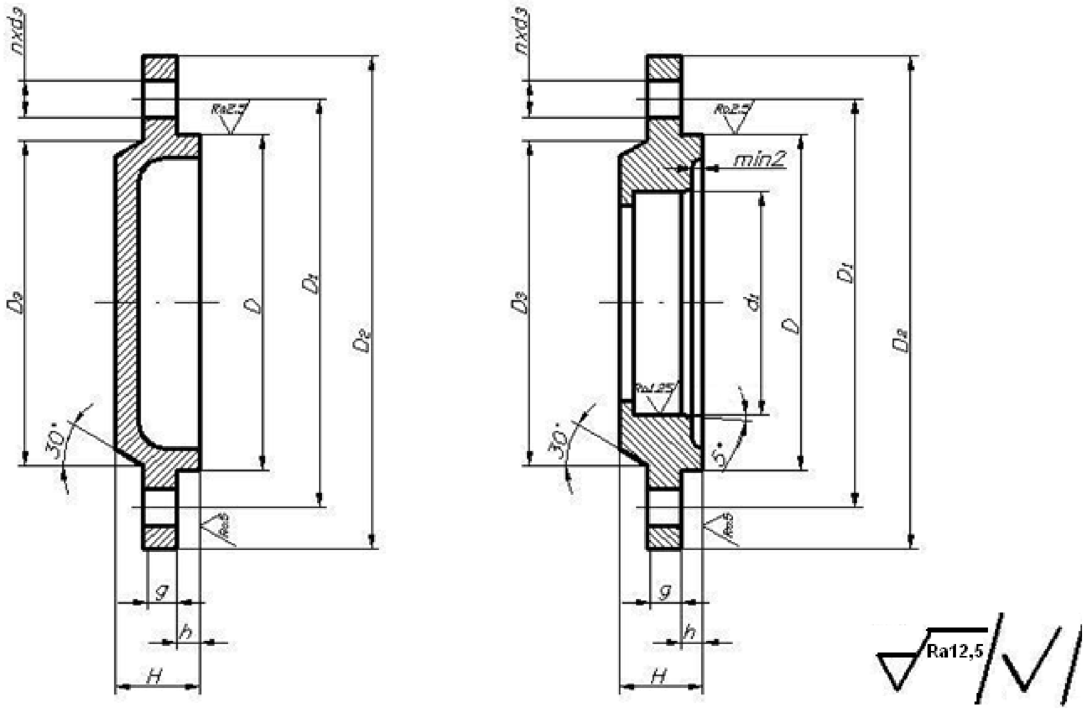


TABLE 9.2. DIMENSIONS OF CAPS FOR BEARING HOUSINGS

<table>
<tr><th>D
h11, d11</th><th>D_1
±0.1</th><th>D_2</th><th>D_3</th><th>n</th><th>d_3</th><th>h
-0.1</th><th>g</th><th>H
min</th><th>d_1
H8</th></tr>
<tr><td>47</td><td>58</td><td>72</td><td>45</td><td rowspan="2">3</td><td rowspan="9">6.6</td><td rowspan="2">3</td><td rowspan="2">5</td><td rowspan="2">13</td><td>35</td></tr>
<tr><td>50</td><td>63</td><td>78</td><td>50</td><td>40</td></tr>
<tr><td>60</td><td>73</td><td>88</td><td>58</td><td rowspan="6">4</td><td rowspan="8">4</td><td rowspan="7">6</td><td rowspan="4">15</td><td>40</td></tr>
<tr><td>(72)</td><td>83</td><td>98</td><td>70</td><td>55</td></tr>
<tr><td>80</td><td>91</td><td>106</td><td>78</td><td>65</td></tr>
<tr><td>(85)</td><td>96</td><td>111</td><td>82</td><td>72</td></tr>
<tr><td>90</td><td>101</td><td>116</td><td>87</td><td rowspan="3">17</td><td>75</td></tr>
<tr><td>100</td><td>111</td><td>126</td><td>97</td><td>85</td></tr>
<tr><td>110</td><td>121</td><td>136</td><td>106</td><td rowspan="6">6</td><td>90</td></tr>
<tr><td>120</td><td>133</td><td>153</td><td>116</td><td rowspan="3">9</td><td rowspan="5">7</td><td rowspan="3">20</td><td>100</td></tr>
<tr><td>125</td><td>138</td><td>158</td><td>120</td><td rowspan="4">5</td><td>110</td></tr>
<tr><td>140</td><td>153</td><td>173</td><td>136</td><td>110</td></tr>
<tr><td>160</td><td>175</td><td>197</td><td>153</td><td rowspan="2">11</td><td rowspan="2">22</td><td>120</td></tr>
<tr><td>180</td><td>195</td><td>217</td><td>173</td><td>—</td></tr>
</table>

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