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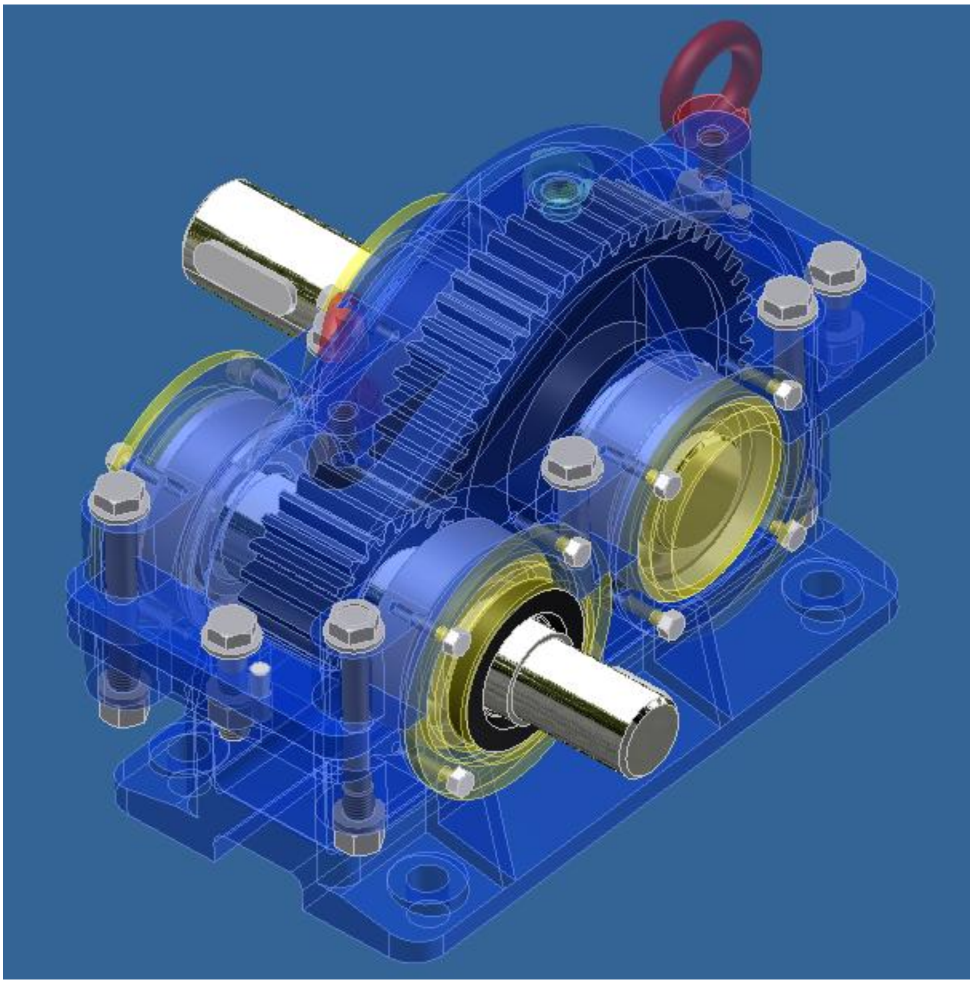
DESIGN BASIC OF INDUSTRIAL GEAR BOXES

Calculation and Design Case Example



Part 0

BASIC KNOWLEDGE

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.

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. 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

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 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.

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.

Flame or induction hardened

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

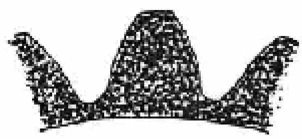


Fig.1C. Spin hardened

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 nitrided. 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 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			

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). 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 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 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 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 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.

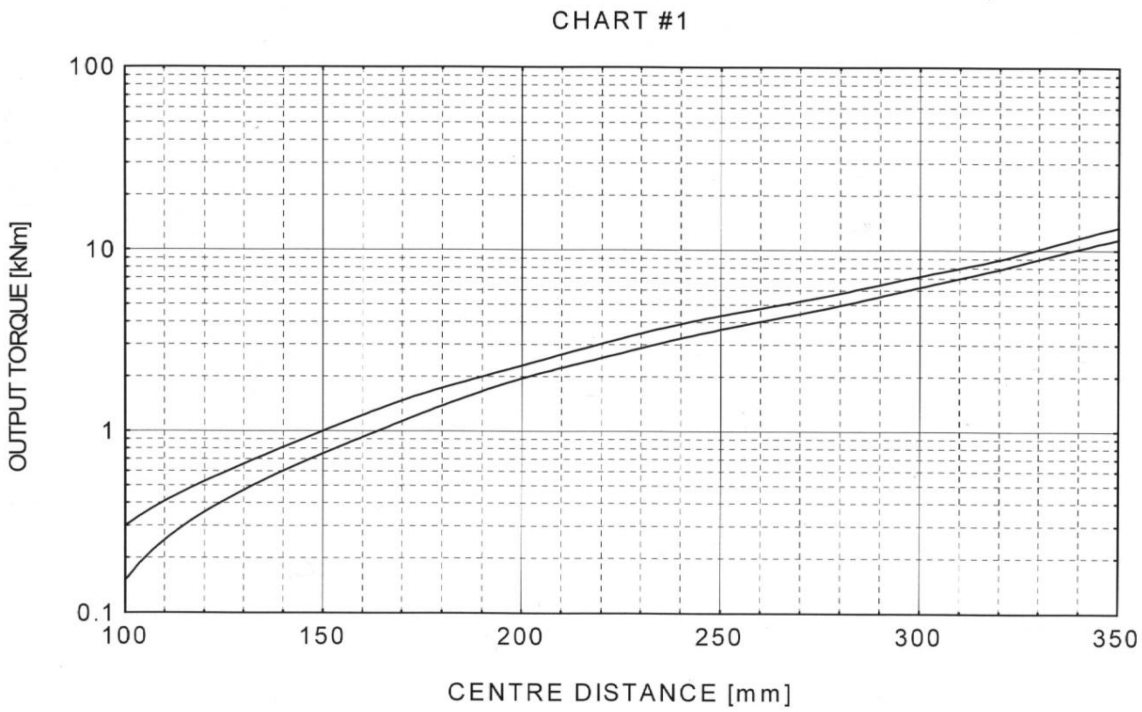


Fig.1. Output torque vs. centre distance for spur gear reducers

EXAMPLE 1

Estimate a center distance between gear axes of a one stage gear reducer with spur gears (see Fig.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.*

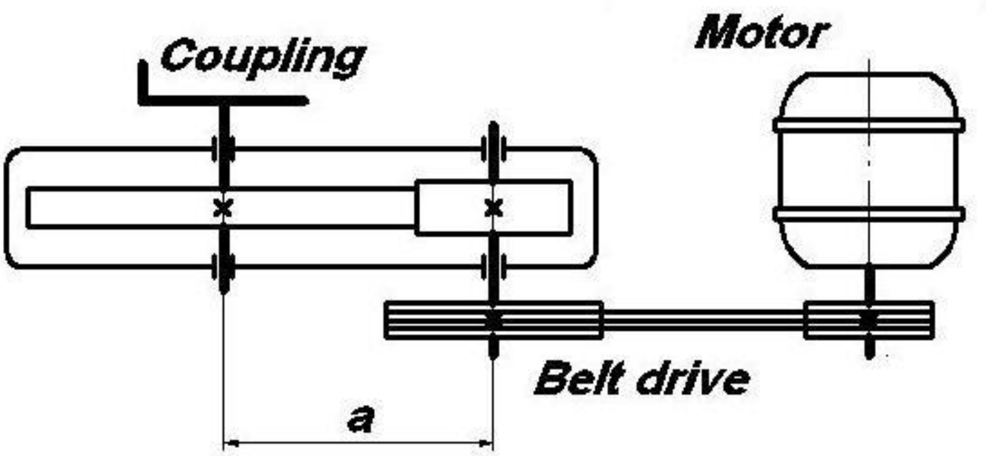


Fig.2. Sketch of the reducer

Output torque of the reducer is given by

$$T_{out} = \frac{30 \cdot P}{\pi \cdot n_m} \cdot u_b \cdot u_g \cdot k_s \qquad (1)$$

Where k_s is a service factor and

$$k_s = k_a \cdot k_t \qquad (2)$$

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

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

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.