

HIGH EFFICIENCY HIGH TORQUE GEARBOX FOR MULTI MEGAWATT WIND TURBINES

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Abstract: The gearbox in a wind turbine is essential for a availability and reliability. In order to develop its optimum design features new way of gearbox designs are proposed. Epicyclic gear drives with multiple planet gears would be technically more correct to solve this torque transmission problem. This technically demanding objective is sensibly solved by the proposed gearbox solution. The following article presents the patented new design [WO 2008/104258 A1] and determines it from a scientific standpoint.

KEYWORDS: GEARBOX, MULTI MEGAWATT, TORQUE SPLIT

1. Introduction

Wind Turbines have a giant rotor, in some cases as large in diameter as a football field, generating today up to 10 million Nm of torque. The gearboxes commonly have 40:1 to 135:1 step-up ratios, taking wind energy from the rotors at about 10 to 16 rpm up to 400, 1'200 or 1'800 rpm at the generator. The trouble is, when wind speed suddenly changes, a relatively small amount of acceleration and angular movement at the gearbox input gets multiplied max 135 times at the output — building up massive amounts of torsional windup and strain energy in the gears [22]. Therefore, leaving the gearbox out has recently become a matter of discussion. First signs of wear already appeared after a few months of operation and not after 20 years as theoretically calculated and certified. This led to enormous additional costs for the wind plant owner. Wind turbines with gearboxes must regain the highest possible reliability as fast as possible. This can occur with improved gearbox designs.

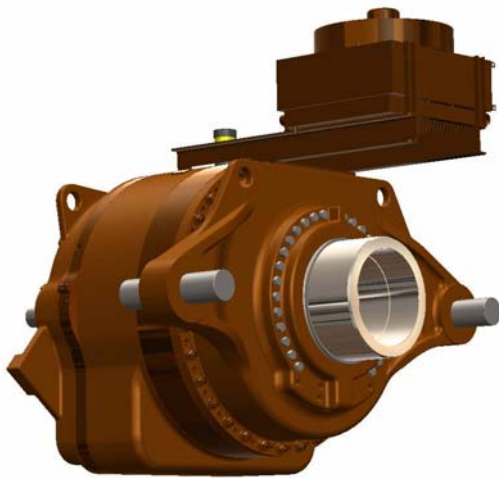


Fig. 1 Wind Turbine Gearbox. Source: GGS GmbH

2. The prior art

Reliability problems stem from the fact that the wind doesn't blow in a nice, steady stream. It's turbulent, changes speed and direction in an instant, and creates loading conditions that play havoc with mechanical systems [10]. Therefore operation and loading of a wind turbine speed increasing gearbox is unlike most other gear applications. Downtime and unreliability are two of the main reasons electricity from wind is expensive. Generally the gearbox still has a low rate of damage compared to other components of a wind turbine, but it leads to long downtime of the wind turbine with correspondingly high costs [21]. The current average turbine size up to 2MW uses normally a 3-stage planetary/helical gearbox with one planetary stage and two single helical gear stages which are not really optimal exploited. Larger size gearbox design uses coupled epicyclic trains. The calculation of ratios, tooth loads, and efficiencies becomes quite complicated for the coupled epicyclic train [1,14]. The different possible designs of these wind turbine gearboxes may present more or less theoretical

and calculation difficulties for all design engineers. There is still a great need for practical gearbox designs.

3. The purpose

One of the most important dimensioning values for the main gearbox is the drive torque which is induced via the rotor blades and the main shaft. Coaxial epicyclic stages with multiple planet gears would be technically correct to solve this force transmission problem. Epicyclic gear divides torque into more power paths to lower the forces and stresses on each gear. A maximal compactness and lightweight construction is thus achieved [13]. In order to also meet with the requirements regarding the increase of the rotational speed to a correspondingly efficient value of the generator, several coupled planetary gear trains are necessary. This technically demanding objective is sensibly solved by the proposed three stage epicyclic drive train arrangement followed by a multiple power path single increasing stage with two output pinions. The following article presents the patented new design [19] and determines it from a scientific standpoint.

4. Layout of the new gear



Fig. 2 Outlet view 6MW gearbox, weight 50t. Source: GDC GmbH

The schematic layout of the gearbox is shown in Figure 2 and a cross section is shown in Figure 3.

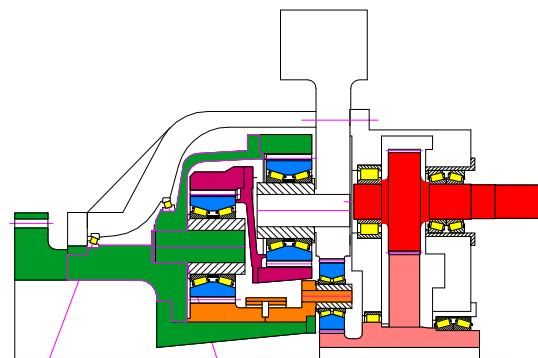


Fig. 3 Cross section 5MW gearbox. Source: GDC GmbH

The introduction of the torque thereby is on the left hand side. As will be disclosed in the following explanations, the internal power split is the main reason for the high degree of efficiency of the gearbox solution. The type of this epicyclic gear is coupled epicyclic trains with up to 8 planets in each train. Approx. 40% of the income power is powering the 1st stage and around 60 % by the 2 stage. The planet carrier of the first stage is rotating while planet carrier of 2nd stage is mounted rigid and act as reaction member of the coupled tree trains. In this way the large amount of torque is distribute ideal to the third epicyclic train, the rotational speed is massively increased and the power is transferred by the output spur gear (forth stage) as a transfer gear to two generators. Working with a differential (coupled epicyclic trains) causes a size reduction of the gearbox layout. This gives an ideal high torque low speed gearbox system for large wind turbine systems. The total pitch cylinder volumes of all components are only 40% of that of a conventional planetary gear box.

The well-balanced gear design in the individual epicyclic trains with a very high number of planets gear is the result of a long graphic and mathematical iteration process. The desired factors of safety were agreed between the manufacturer and the customer. Only ISO 6336 part 1,2,3 and 5 should be used as a recalculation method for the bending stress on the tooth root and the stress on the tooth flank [11]. The application factor K_A calculated on the basis of ISO 6336 part 6 takes the variable wind loads into account. The stresses on the surface of gear teeth are usually determined by formulas derived from the work of H. Hertz of Germany [24]. In wind turbine gearboxes the contact occurs at a very high pressure and gliding, frictional heat is generated in the contact surface. With slowly rotating high-output planet gears, these meshing conditions must also be determined via the flash temperature criteria deduced by Prof. Blok [6].

5. Determination of the ideal torque and transfer ratio

Calculating the transfer ratio occurs with the help of the "torque-method" described in [2,3,4,5]. We assume for this method working with the ideal torque standard ratio efficiency $\eta_0 = 1$. And the following nomenclature is used:

number of teeth in sun gear = z_1

number of teeth in ring gear = z_3

torque at sun gear = T_1

torque at ring gear = T_3

torque at carrier S = T_S

torque ratio $t = |z_3/z_1| > 1$,

and the size of torque is in following order: $T_1 < T_3 < |T_S|$.

Accordingly the torque is defined as:

$$T_1 : T_3 : T_S = +1 : t : -(1+t) \tag{1}$$

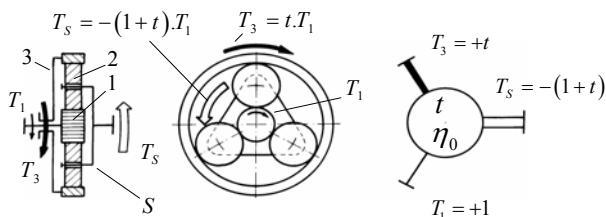


Fig. 4 Elementary (one-carrier) Planetary Gear

The procedure is:

1. The known and very useful Wolf's symbol [18] is used, however with a few additions in order to achieve the greatest possible clearness as shown in figure 4. The shafts with the aligned torques, i.e. the sun gear shaft 1 and the annulus gear shaft 3 are marked with single lines. Their thickness is based on value of the torques, where $T_1 < T_3$. The planet carrier S has the greatest, however opposite torque, which equals that of the other two torques in absolute terms: The planet carrier S is marked with double lines.

$$T_S = -(T_1 + T_3) \tag{2}$$

2. A torque ratio t is defined (formed) based on the aligned ideal torques T_1 and T_3 (without gear loss):

$$t = \frac{T_3}{T_1} = \left| \frac{Z_3}{Z_1} \right| > +1 \tag{3}$$

which proves to be very advantageous for the analysis of coupled planetary gear trains.

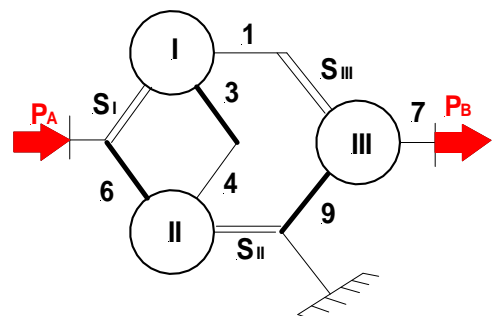
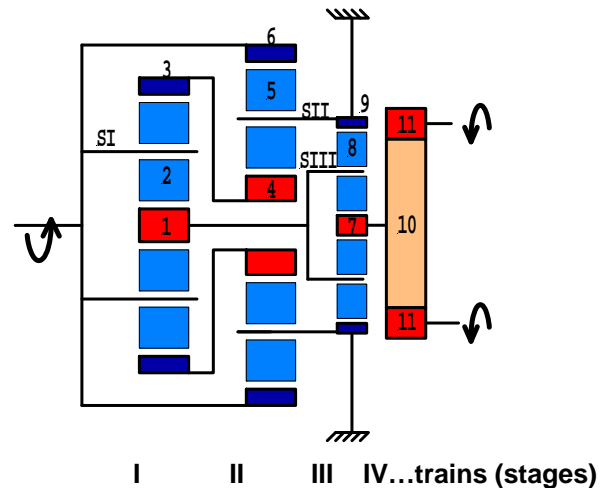


Fig.6 Structural Wolf 's schema of the coupled three planet carrier train

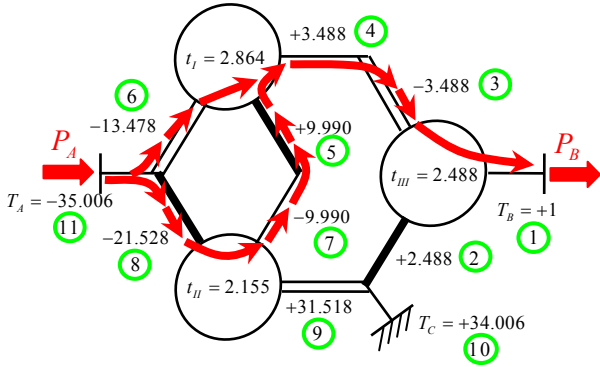


Fig. 7 Verification of the ideal torques of the coupled three-carrier planetary gear train.

It is advantageous to start with a torque +1 (unity) of one of the sun shafts, which however is not a condition. Any positive or negative value can be used. In the present case, one starts off with the output torque T_B of the sun shaft 7 of the third planetary gear train. with:

$$T_B \equiv T_7 = +1 \quad (4)$$

One proceeds step by step by using the individual torque ratios t_I , t_{II} , t_{III} of the three gear trains I, II and III. The order of the torque calculations is marked by the green circled numbers in figure 7. The following must be taken into account in this operation:

Both free linkage shafts (floating members) – i.e. the annulus gear shaft 3 with sun gear shaft 4 and sun gear shaft 1 with carrier shaft SIII, do not have an outward connection. Two equal but opposite torques, i.e. with different algebraic signs, therefore act on their ends.

The drive input torque T_A and the reaction torque T_C of the connected linkage shafts – i.e. the carrier shaft S_I with annulus gear shaft 6 and the carrier shaft S_{II} with annulus gear shaft 9 result from the sum of the torques of the corresponding gear train shafts.

If the ideal torques of the assembled gear is at hand, it is easy to calculate the transfer ratio i with the following formula using the drive torque T_A and the output torque T_B :

$$i = -\frac{T_B}{T_A} = -\frac{+1}{-35.006} = +\frac{1}{35.006} \approx \frac{1}{35} \quad (5)$$

The easily handled torque method makes the considerable calculation work of the “Willis Formula” superfluous, possible calculation errors moreover not being excluded with said method. The “Kutzbach method” is discarded from the start, especially with the elaborately assembled multi-carrier planetary gear trains, on the one hand as a matter of principle because of the inaccuracy of the graphical method, and on the other hand because of the impossible accumulations of lines. Another advantage of the torque method is the possibility to control the correctness of the calculation based on the balance conditions of the ideal torques. In the present case, the control confirms the correctness of the calculations:

$$\sum T_i = T_A + T_B + T_C = -35.006 + 1 + 34.006 = 0 \quad (6)$$

6. Determination of the real torque and of the efficiency

The real torque of all three single-carrier planetary gear trains, with which the inner gear losses are factored into the determination is needed in order to calculate the efficiency of the assembled gear. The three stationary efficiency values η_{oI} , η_{oII} , η_{oIII} of the three single-carrier planetary gear trains on the one hand and the flow direction of the three rolling powers P_{WI} , P_{WII} and P_{WIII} also from the three trains that are responsible for the dominating meshing losses are needed for this. There are 2 possibilities for the flow direction of the rolling powers P_{WI} , P_{WII} and P_{WIII} . In the example of the trainI, these possibilities are the following refer to Figure 8:

- 1) Flow direction from the sun wheel 1 to the rotating annulus 3.
- 2) Flow direction from the rotating annulus 3 to the sun wheel 1.

The rolling power P_{WI} is transferred from the sun wheel 1 to the annular gear 3 when the directions of the torque T_1 and the angular speed ω_1 coincide and vice versa. The 3 following terms ensue for the real torque T_3 of the annulus gear:

$$T_3 = \eta_{oI} \cdot t_I \cdot T_1 \quad (7)$$

$$T_3 = \frac{t_I \cdot T_1}{\eta_{oI}} \quad (8)$$

Calculating the flow direction of the rolling power P_W is in principle not always clear and easy to determine. In such cases, the very useful trying method by Seeliger [15] is used in such cases. In the present case, the flow directions of the three rolling powers P_{WI} , P_{WII} and P_{WIII} do not cause difficulties.

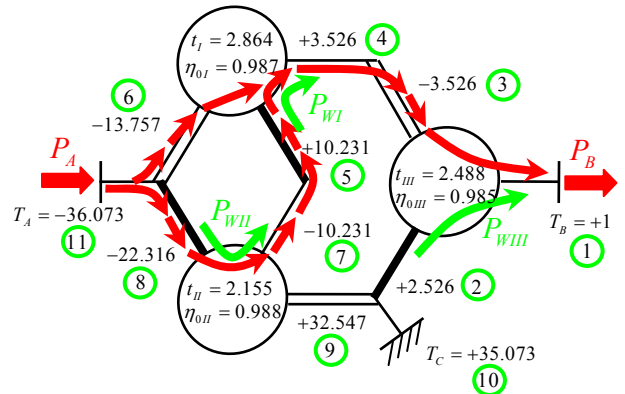


Fig. 8 Verification of the efficiency using real torque with the power flow directions

The most reliable determinations of the efficiency of gear drives are doubtlessly the experimental one. With the enormous output power of the gears of wind power plants, this is however easier said than done. One must thus resort to a theoretical determination of the efficiency for a first approximation. To determine the stationary efficiency η_{oI} , η_{oII} , η_{oIII} of the three single-carrier planetary gear trains, the very simple formula by Förster [8], which primarily takes the dominant impact of the number of teeth into account, is used here for lack of space. With respect to the first single-carrier planetary gear train, the formula reads as follows:

$$\eta_{oI} = 1 - \left[0.15 \cdot \left(\frac{1}{z_1} + \frac{1}{z_2} \right) + 0.2 \cdot \left(\frac{1}{z_2} - \frac{1}{|z_3|} \right) \right] \quad (9)$$

Further partial gear losses are usually taken into account across-the-board with a certain percentage.

In the present case, the calculation with the concrete number of teeth yields the following numerical value for the individual stationary efficiency of the three single-carrier planetary gear trains:

$$\eta_{oI} = 98,7\%; \eta_{oII} = 98,8\%; \eta_{oIII} = 98,5\% \quad (10)$$

Figure 7 illustrates the procedure for determining the efficiency η . The flow direction of the shaft powers is marked red and the flow direction of the rolling powers P_{wI} , P_{wII} and P_{wIII} is marked green. The order of the determination of the real torque is the same as with the determination of the ideal torque. With the marked flow directions of the three rolling powers, the following connection exists between the torques and the sun gear shaft 1, 4 and 7 and the corresponding annulus gear shafts 3, 6 and 9:

$$\begin{aligned} T_3 &= t_I \cdot \frac{T_1}{\eta_{oI}} \\ T_6 &= t_{II} \cdot \frac{T_4}{\eta_{oII}} \\ T_9 &= t_{III} \cdot \frac{T_7}{\eta_{oIII}} \end{aligned} \quad (11)$$

The real torques is used in order to first determine torque conversion μ according to the following formula:

$$\mu = \frac{T'_B}{T'_A} = \frac{+1}{-36.073} = -\frac{1}{36.073} \quad (12)$$

The efficiency η is then calculated as follows:

$$\eta = -\frac{\mu}{i} = -\frac{-\frac{1}{36.073}}{+1} = +\frac{35.006}{36.073} = 0.9704 \approx 97\% \quad (13)$$

When determining the efficiency η , a control of the correctness of the calculation is possible in a similar manner than with the ideal torques:

$$\sum T'_i = T'_A + T'_B + T'_C = -36.073 + 1 + 35.073 = 0 \quad (14)$$

7. The problem of load distribution and its mastery

In planetary gears with high numbers of planet wheels, the tangential force is not distributed in a very even manner onto the individual power paths. The inevitable manufacturing anomalies are responsible for this. This is taken into account during calculation with the distribution factor K_γ . A great number of patented

distribution devices exist. Part of these distribution devices function on the basis of a kinematic principle. The other part of the distribution devices uses the flexibility of certain gear elements for load distribution. It is of fundamental importance to create a flexibility which makes it possible for the planet wheel to shift parallel and in a geometrically precise manner. Invented by British engineer Raymond Hicks in the mid-1960s, he has developed a product called “flexpin” (Figure 9) [20]. The flexpin equalizes loads on planets by anchoring them onto a single wall planetary carrier in a torsionally compliant manner. Instead of fixing the angular position of the planet gears, as is the case with two wall planetary carrier conventional systems, flexible pins deflect circumferentially and independently along the carrier pitch circle — which ultimately equalizes forces on the planets. There is a force F and a moment M by the end of the cantilever sleeve/pin as shown in figure 10. They ensure an optimum load balance between the meshed teeth and even load distribution across the entire width of the tooth face at both full and partial load. Flexpins have been used in many industrial applications, including wind turbines, with notable success [23].

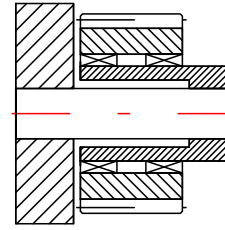


Fig. 9 Flexpin no loaded

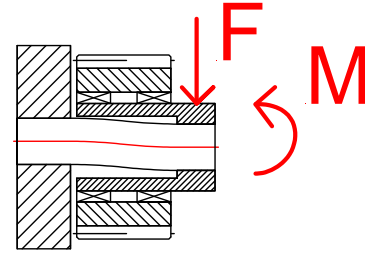


Fig. 10 Flexpin conter reaction loaded.



Fig. 11 Flexpin made by GDC GmbH

This flexibility occurs in the present design by flexpins made by GDC [9]. This is commonly referred to a “asymmetrical flex-pin” shown in Figure 11. This design can solve the objective if the rigidity ratios are correctly laid out. If the geometrical ratios are incorrectly chosen, the pin-sleeve combination tilts into a non-advantageous tilt. This inclination influences the load distribution factor $K_{H\beta}$ of the planetary wheels.

8. The measurement of the load distribution

The load distribution on the individual planetary wheels can take place in different manners:

- 1) Via the deformation (bending) of individual teeth of the annulus gear [17].
- 2) Via the deformation (bending) of the axes of the planetary wheels.

A positive aspect of the first method is that the tooth load, in which one is actually interested, is directly measured. This method however requires the annulus gear to be accessible, which is not always given. In addition, there is the great complexity of the test arrangement and of its analysis. The most unpleasant aspect is however that the load on the planetary wheels cannot be determined as a continuous function of the attitude (rotation angle).

The mentioned disadvantages do not apply to the second measuring method. The deformation (bending) of the axes of the planetary wheels is taken as a basis for their load, strain gauges being advantageously used. The most beneficial aspect of this method is that the load of the individual planetary wheels can be determined as a continuous function of the attitude, which is more useful. This method also has some disadvantages, for instance that the tooth load is practically not determined, but rather the axis load. In a strict sense, the axis load is not interesting for the load distribution, but rather the tooth load. It is not excluded with planet gears with great masses and considerable torque of inertia of the gears, though only at very high rotational speeds (turbo-gears, that the difference between the axis load and the tooth load becomes considerable, possibly between both teeth mesh contacts – internal and external – of a planetary wheel.

By definition, $K\gamma$ is the ratio of the greatest deflection or strain of the sleeve/pin assemble of a planet wheel, divided by the average value of all deflections/strains of the sleeve/pin assemble per load level.

In the following table measured values are exemplarily indicated for a single-carrier planetary gear train with the number of planetary wheels of 7 as shown in figure 12. This great number of planetary wheels as expected brings the greatest uncertainty with respect to the load sharing. This is why this single-carrier planetary gear train was chosen for the measurement.

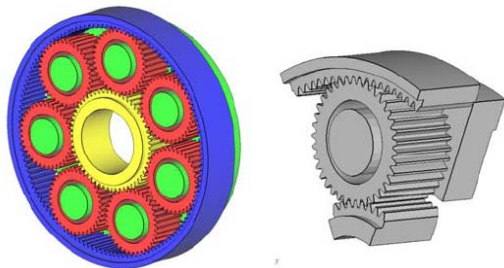


Fig. 12 View of planetary stage with 7 planets.

Table 1: Measurement of $K\gamma$ on different load levels:

Load in [%]	20	50	75	88	100
$K\gamma$ [-]	1.5 0	1.2 3	1.1 5	1.1 5	1.1 4

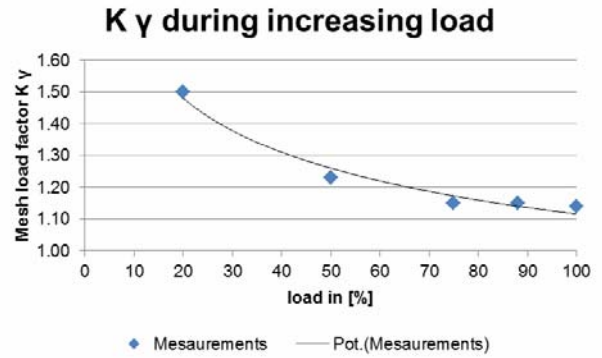


Fig. 13 $K\gamma$ on different load levels.

9. Advantages of the solution

The design of this 5MW wind turbine planetary gearbox has the following advantages:

- 1) It is 20% shorter, more compact, and lighter than a comparable existing gearbox design because of the use of multiple planets per train (up to 8).
- 2) Higher safety factors SH (pitting) and SF (bending) of the gears are obtained.
- 3) A maximum face width of the gears of only 250 mm, instead of 500 mm is achieved at the input epicyclic train stage.
- 4) A maximum external diameter of only 600 mm of the planet gears, instead of 1000mm is achieved.
- 5) A distribution of wind rotor input torque is achieved between the coupled epicyclic trains at a 4:6 ratio. This feature allows for the use of identical planetary wheels in both trains, which is an important manufacturing advantage.
- 6) Big sun wheels are used. Large radii of curvature thus ensure at all points of contact of the pair sun wheel and planetary wheel, which leads to very deep and well balanced flash temperatures according to Blok.
- 7) Only one torque arm is used as a reaction member and constructional element, which makes the supply of lubricant and cooling oil for the indentations and roller bearings possible in a simple manner.

10. Summary

As a consequence of the consistent use of multiplicity coupled planetary arrangement with single wall planet carriers, with a maximum number of planetary wheels in one train and thanks to the mastery of the problem of load distribution we have succeeded in engineering a compact and light increasing gearbox for a 5MW wind turbine. The solution is lighter and shorter than solutions used so far. This type of compact gearbox will remain a groundbreaking achievement for further wind turbines with a rotor torque of up to 25'000'000 Nm.

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