ABSTRACT

In the past two decades the wind turbine industry has witnessed a considerable number of catastrophic accidents, many of which were due to gearbox failure. Ever increasing power ratings at decreased rotor speeds result in rotor torques of some million Nm. This imposes tooth loads and planet/pinion bearing loads on the order of a hundred tons within the first step-up stage. Such heavily loaded gearboxes, correctly (or rather innocently) designed according to the relevant codes, can be self-destructive. Due consideration should be given to the elastic environment in which the gears exist. Otherwise, appreciable, unsymmetrical/unequal elastic deformations in unwanted directions lead to gear tooth edge loading, in addition to overloading the bearing(s) near that edge. Designers of wind turbine gearing have in recent years identified several concepts and measures to be taken for counteracting the asymmetry of elastic deformations or mitigating their effects. In addition to giving a brief survey of such new design concepts, this paper suggests the use of selected types of curved-tooth cylindrical gears (so-called C-gears), primarily for their self-aligning capability; they allow four degrees of freedom (4-DOF), in contrast to the 3-DOF spur and helical gears and the 2-DOF double-helical gears. In addition, these gears offer a unique set of further advantages. When used in at least the most heavily loaded, first step-up stage, the design will be rendered quasi-exactly constrained; largely tolerant of misalignment due to elastic deformations, and the gearbox reliability should be improved, by design.

1. INTRODUCTION

Unless and until direct-drive wind turbine generators become competitive, regarding weight and cost, energy harvesting from the wind will continue to depend on there being step-up gearboxes. In modern utility-scale wind turbine generators, the gearbox is one of the most expensive yet also crucial pieces of equipment in the nacelle. To win the renewable energy challenge, gearboxes should simply not unexpectedly or prematurely fail. In the past two decades the wind turbine industry has witnessed a considerable number of catastrophic accidents, in many of which gearbox damage was the culprit. Spinato et al. [1] conclude that gearbox failure is responsible for the highest downtime as compared to other wind turbine subsystem failures, hence for the highest cost associated therewith. The gearbox can thus be considered the “Achilles’ heel” of a wind turbine.

According to Musial et al. [2] “the majority of wind turbine gearbox failures appear to initiate in the bearings.” This statement becomes more accurate when complemented by “due to the arduous loading conditions of the gears.” This means that the bearings themselves should not necessarily be bad, but it would reflect a damage scenario that develops as follows: Appreciable, unsymmetrical/unequal elastic deformations in ‘unwanted’ directions lead to gear tooth edge loading, in addition to overloading the bearing(s) nearer to that edge. (Roller bearing life is known to diminish to one-tenth its value when load is doubled). Pitting initiates in the gear tooth flanks and/or the bearing rolling surfaces. The bearings start becoming a little slack, further deteriorating the gear mesh conditions. Tooth flank pitting proceeds destructively, until a substantial piece of the gear tooth fractures, and it is needless to describe the consequences. Power ratings of 3 MW and more are typically obtained with a rotor diameter of at least 100 m that turns at a rated speed of 14 rpm and slower, in sites of average wind speeds. This gives a rotor torque of 2 MNm and more, which imposes planet bearing loads in the order of magnitude of a hundred tons in the first step-up stage. Such heavily loaded gearboxes could be self-destructive if ‘innocently’ designed according to the relevant codes. The now available highly
purified, highly stressable gear materials are rated at an AGMA allowable contact stress number of up to 1900 MPa (275 ksi).

As design engineers are tempted to exploit these characteristics for heavily loaded gearing, some might overlook the hazard associated with the meshing teeth acting as pincers rather than evenly loaded interfaces. As noted by Crowther et al. [3], current wind turbine gearbox design codes do not explicitly recommend the consideration of gearbox housing flexibility or planet carrier flexibility in the design process. Walford [4] even raised concerns about the applicability of the existing standards used for ‘component’ design (viz. gears and bearings), suggesting that they should be re-evaluated. But it should be remembered that these codes are intended for designing gears that operate as intended; in full-face contact with an estimable, reasonable load-distribution factor. Hence, there is no evidence of an imminent need to revise these codes.

It was recently concluded [1] that wind turbine gearboxes are of a mature technology, such that substantial improvements in the designed reliability of existing gearboxes are unlikely. However, the present paper stresses the notion that successful wind turbine gearboxes that endure 20 years of surprise-free operation should be more professionally conceived and detailed according to the concepts of reliability by design, whenever applicable. In the present context, the more important of these concepts are those which (a) eliminate or mitigate appreciable unsymmetrical/unequal elastic deformations and, even better, (b) render the design quasi-exactly constrained by applying self-aligning principles. In this paper, a brief overview of some of those novel design concepts is given, and the use of curved-tooth cylindrical gears (so-called C-gears) is proposed to satisfy the requirements under (b).

2. THE FIRST-STAGE RATIO

The first step-up stage is the highest loaded subsystem in wind turbine gearboxes. It is where the designer endeavors to safely transform the extremely low-speed and high-torque combination of the rotor hub into an easier one that could be handled by further stages. When a single generator is being driven, which is the case in the majority of present wind turbines, the first stage may be a planetary or a star gear set. In some cases multiple generators are driven, where the first step-up stage would consist of a bull gear driving a number of pinions on fixed axes.

**Planetary Versus Star Gear Sets.** The tendency to use a first star stage to dump the rotor torque directly on the ring gear reflects the designers’ aim to transform the high rotor torque into planet tooth and bearing load at as large a radius as possible. However, as Table 1 shows, these loads in a star set are higher than in a planetary set of the same size by the factor of planetary-to-star ratio; typically around 1.25, which is why many more designs are seen with a planetary first stage than with a star one. This factor is independent of the number of planets. Planetary gear experts also know that, based on pitting resistance calculations, the maximum carrier torque per planet is obtained in the vicinity of the first case in Table 1, with a star ratio of two. Multiplied by 7 or 8; the largest number of planets in cage-type carriers and in single-flange carriers, respectively, this configuration could handle the maximum possible rotor torque, but at a step-up ratio of only three and at the expense of the bearings being small.

<table>
<thead>
<tr>
<th>Table 1. Comparative attributes of planetary and star gear sets for various ratios</th>
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<tbody>
<tr>
<td>Planet/sun pitch-diameter ratio</td>
</tr>
<tr>
<td>Ring/sun ratio (star ratio)</td>
</tr>
<tr>
<td>Planetary ratio (with fixed ring)</td>
</tr>
<tr>
<td>Carrier/ring radius ratio, C/R</td>
</tr>
<tr>
<td>Ratio of bearing loads in star and planetary sets = 2C/R</td>
</tr>
</tbody>
</table>

3. COUNTERACTING THE ASYMMETRY OF ELASTIC DEFORMATIONS

In either case of planetary or star first step-up stage, the gearbox design will include a ‘planet’ carrier, which supports the pins of planet bearings or the bearings of planet pins. The carrier is an arduously loaded component in wind turbine gearboxes which, however, does not normally break, it deforms, and so do the pins under the heavy loads that act in circumferential direction. A conventional cage-type carrier is an unsymmetrical component, which is torqued at one end face.
The torque is split up among that face and an opposite one, between which the planet bearing pins are straddle mounted. The two faces will elastically phase shift, the pins will skew, the load distribution across the face of the gears will concentrate more towards the drive end, and the bearings at that end will be more heavily loaded than the ones in the free flange. The gears will be edge-loaded and the bearings at the drive end will be overloaded. In a single-sided or open carrier, the pins are cantilevers and similar conditions will prevail, but these carriers offer the advantage of using as many planets as possible; there are no bridges passing between them to attach another flange. Over the years, designers of wind turbine and similar heavily loaded gearing have developed several concepts for countering the asymmetry of elastic deformations or mitigating their effects. Implementing these concepts does make a heavily loaded pinion much better conform to “one” mating gear, but the doubly meshing planets with multiply meshing sun and ring gears make these designs only partly effective. This raises concerns about the planetary system of gear being the most appropriate choice for wind turbines. Nevertheless, a brief account of some of these concepts is given in the following.

3.1 Adopting Self-Aligning Principles. The technical literature abounds in designs of planetary gear sets in which the deformation problem is apparently completely solved by applying self-aligning principles to the planet bearings; each planet being supported on one spherical roller bearing. The bearing outer race is sometimes directly machined into the planet, and the inner race is mounted on a cantilever pin in a single-sided carrier. Particularly in heavily loaded gearing, this arrangement suffers, however, two ‘additional’ shortcomings: (a) It would seldom be possible to accommodate a correct size bearing inside a planet (of appreciable face-width-to-diameter ratio) to carry twice the tangential load in the input stage and endure a billion revolutions. Planets are usually rather seen supported on packs of two or four bearings, non-self-aligning. (b) When the outer race rotates while the fixed inner race is skew, the roller track wobbles; the rollers skid on the inside of the outer race in axial direction through a sine wave per revolution. As pointed out by Arafa [5], the correct strategy would have been to use so-called spherical concave-roller bearings, with the spherical surface on the inner race. These bearings allow the inner races/pins to be tilted while the outer races with planets roll in-plane, without skidding. But bearings of appropriate dimensions are yet to be made available by bearing manufacturers, and planetary gearboxes using them are yet to be seen.

3.2 Supporting the Planet Pins in Their Middle. This configuration aims to make the planet teeth conform better to the ring and sun gear teeth by distributing the load equally among two identical halves of each planet and evenly across the (smaller) face width. The planet halves are supported in bearings on the two cantilevers of a pin that is fitted in its middle in one central carrier-flange, which extends from the carrier driving-end to that location within the planetary gear set. This design was recently suggested by Saenz de Ugarte et al. [6] and by Willie [7].

3.3 Making the Planet Carrier Symmetrical. Mostafi [8] suggested to use an ‘independent’ symmetrical carrier that straddle mounts the planets, and to support and drive that carrier through spherical heads/sockets located nearly in its central transverse plane. The spherical heads are the ends of cantilevers that protrude from the drive flange on the turbine rotor shaft, whereby a least-constraint support is realized.

3.4 Applying the Principle of Matched deformations. This design principle has originally been introduced by Pahl and Beitz [9]. It aims to combat the detrimental effects of uneven load/stress distribution in mechanical components and assemblies. Without mentioning the name of that design principle as such, it is reported by Hicks et al. [10] to have been successfully applied to planet supports. A single-sided planet carrier receives the pins, press fitted or flange-bolted, to act as cantilevers, and to bend under load. A quill is fitted onto the free end of the pin, clearing the annular space, and carries the planet bearings at another shorter, backward cantilever length; in series with the first one. All the elastic deformations of the inflected pin, the quill, and the carrier flange precisely matched, the planets under load could be maintained un-skewed. A further advantage of providing the system with such elastic supports is that of equally sharing the load among the planets. This design concept is recently seen suggested once more by Fox [11].

It should finally be noted that in almost all first-stage planetary or star gear sets in wind turbine gearboxes only straight-tooth spur gearing is used. This is because helical planets are subjected to a tilting moment – in the plane of planet and sun axes – that would worsen the already critical loading and alignment conditions. (Double-helical planetary sets are not often found in these applications). Consequence is that planetary gearing is deprived of the privilege of being helical, for strength, better running-in, and noiselessness. A solution was recently given by Culiffe and Hicks [12] in addition to the idea of using flexible pins. Narrow rolling contacts can be provided at both ends of the planetary set by fitting two rings on each planet beyond the teeth, which will roll on plain cylindrical tracks in the ring and sun gears, a little deeper than their root circles. The rolling contacts thus established react the tilting moments on the helical planets.

4. Connectivity of Spur or Helical Gear Pairs

A pair of involute tooth flanks of spur or helical gears is in straight-line contact when the gear axes are parallel. The straight line is a common generatrix of the involute helicoidal tooth surfaces of two helical gears of the same lead. The lead
becomes of infinite length in the case of spur gearing. This pair features two so-called positive constraints; the one against penetration and the one against skewing in the plane of action (which contains the loci of the centers of curvature of the transverse flank sections along the instantaneous contact line; the tangent lines of the plane of action with the two base cylinders). This skewing, if enforced under the effect of misalignment or unequal elastic deformations under load on both sides of the gear(s), leads to edge loading. In addition to the positive constraints the pair features one virtual or non-positive constraint against skewing normal to the plane of action; in a sense of not “expecting” mobility in that direction or else the contact geometry departs from the postulated straight-line case. With external–internal gears, where the flanks resemble a cylindrical inside a hollow cylindrical surface, this skewing, if exploited, leads to edge loading the flanks at the two side planes under a high mechanical advantage. With external–external gears the skewing degenerates the full line contact to point contact in the middle of the face width.

Based upon the foregoing discussion, a spur or helical gear pair is rated as to have a connectivity of three, i.e. three degrees of freedom (3-DOF). These are the mobilities in relative rolling, axial sliding, and the insensitivity to center-distance changes, which is typical of involute gearing. But this pair remains highly sensitive to misalignment, in the sense of both shaft skewing and in-plane out-of-parallelism. The 3-DOF spur and helical gear pairs have one of their constraints acting in an anti-self-aligning sense, another constraint as a virtual one, and the only useful constraint is thus the one against penetration; that the gears do transmit power.

5. C-GEARS

Gears for heavy power transmission must not be confined to the limitations imposed by spur and helical gearing. A parallel-axis gear pair with connectivity of more than three and without having any of its constraints in either of the above described ways of action would be very advantageous. Curved-tooth gears, C-gears for short, are later shown to satisfy this need. These are cylindrical gears with their teeth curved in the lengthwise direction, featuring convex and concave flanks, as the sample in Fig. 1 depicts. However, C-gears have been much more often portrayed than produced, and some of the disclosed drawings even reflect an awareness of the suitable proportions. But these gears are not of one type; a glance at Fig. 1 may indicate any one out of a dozen C-gear types. The literature abounds in suggestions of how the kinematics and/or machines for cutting and/or grinding C-gears could be configured. In the enumerative study by Arafa [13] the suggested geometries of C-gears were classified into 11 types, while Bedewy [14] made a comparative study of them, including one further possibility. Some of those types are of subordinate relevance, two of them being of ‘approximate’ geometry, and will be omitted from the following discussion.

5.1 A Brief History. The first kinematically sound conception of “gears with circular-arc-shaped teeth” is attributed to Böttcher [15], who applied for that patent in 1909. He introduced three of those dozen types which could now be identified. The kinematic geometry of generating involute gearing by straight-edge, V-shaped blades, face-mounted on a cutter head was clearly presented, in addition to two generating machine constructions. One machine was for ‘face milling’ C-gears in single-indexing, using either one cutter head with full space-width blades to generate one tooth space after another; the convex and the concave flanks simultaneously, or two separate cutter heads for the two flanks. The first type of gears, shown in A in Table 2, will have a compulsory curvature mismatch \( R_{cv} - R_{vc} \) of half a pitch between the mating convex and concave flanks, and point contact in the mid-plane is obtained, while the second type, shown in C in Table 2, could be in full-face contact when the two radii of curvature in the generating rack pitch plane are equal \( R \).

![Figure 1. A typical C-gear: 24-tooth pinion with a face advance of one pitch, face width of 20 x module, and tooth inclination angle of about 32° at the side planes](image)

The other machine was for generating a third type of C-gears, shown in D in Table 2, now in continuous indexing (face hobbing); it featured a differential mechanism. The cutter head had two finishing blades at 180° apart, one for the convex flank with its inner straight edge and the other for the concave flank with its outer straight edge; both edges being adjusted at the same radius. The tooth traces will be trochoidal rather than circular, but full-face contact between meshing gears is obtained with opposite-hand trochoids.
Table 2. Enumeration of the curved-tooth gear types to be generated in single or continuous indexing by V-cutters* (CV-gears)

<p>| | | |</p>
<table>
<thead>
<tr>
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<tbody>
<tr>
<td>A.</td>
<td>Single indexing (Böttcher 1919)</td>
<td>B.</td>
</tr>
<tr>
<td></td>
<td>One and the same multi-blade cutter head for both flanks of both gears</td>
<td></td>
</tr>
<tr>
<td>C.</td>
<td>Single indexing (Böttcher 1919)</td>
<td>D.</td>
</tr>
<tr>
<td></td>
<td>For both gears: inside cutter for the convex flanks; outside cutter for the concave flanks, with several blades each</td>
<td></td>
</tr>
<tr>
<td>E.</td>
<td>Single indexing (Gleason 1920)</td>
<td>F.</td>
</tr>
<tr>
<td></td>
<td>Male cutter for both flanks of the gear tooth space, female cutter for both flanks of the pinion tooth, with several blades each</td>
<td></td>
</tr>
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</table>

* The V-blades are shown in the gear mid-plane, instantaneously centered in a tooth space or over a tooth.
Soon afterwards, Böttcher [16] recognized that the transverse tooth profile in the generating-rack side planes deviates slightly from the straight towards the conic section “hyperbola” without affecting the conjugacy of mating gears though, that the pressure angle increases from the mid-plane towards the side planes (only in his drawings), and the possibility of making asymmetric teeth, with the driving flanks of a smaller pressure angle than the coast flanks. The varying pressure angle makes the plane of action warped and the base surfaces barrel-shaped.

A third possibility of generating C-gears in line contact is to make the (equal) concave and convex curvatures in one driving direction different than the (equal) curvatures in the opposite direction. This method was suggested by Gleason [17], using a male cutter for generating the tooth spaces of one gear, such as in type A, and a female cutter that straddles the tooth of the other gear for generating both its flanks. This is type E in Table 2. It should be noted that the female cutter should be used with the pinion, for its teeth to retain their normal thickness in the side planes. (The male cutter makes the gear teeth a little thinner there). The continuous-indexing counterparts of types A and E have recently been added by Bedewy [14] and Dai et al. [18], respectively, which complement the entries in Table 2. The gear types in Table 2 are referred to as CV-gears; by virtue of the variable pressure angle as well as the V-shaped blades used in their generation.

Since their conception, more than a hundred patents have been issued on C-gears, which soon covered the dozen types. But these gears continue(d) to be re-invented, and some regard them today as still being “novel.” Since the majority of these patents are older than 20 years, newer patents will not secure intellectual property rights and anyone could manufacture C-gears and their machines according to the old established principles or embodiments. But no one did. Nevertheless, several machine tool builders have, over the years, been granted or assigned patents on gear generator designs suitable for C-gears. Examples are given in chronological order in the documents by Gleason Works [19], Ingranaggi Mammano [20], Boor [21] Gebrüder Bühler [22], Gleason Works [23], Kotthaus [24], and Charles [25]. But none of these manufacturers seems to have built the machine or produced C-gears, with only one exception [22] (for a limited duration of time), as documented by Wittmann [26].

5.2 Recent Research. The past century did not see published research on C-gears, except for one paper in 1966 by Ishibashi [27], reporting that gears have been generated according to Böttcher’s types A and C, without reference to such. The author identified that the generating rack tooth profile progressively became hyperbolic and the pressure angle increased towards the side planes. Because the experimental gears were chosen to have rather small tooth curvatures, interest was focused on assessing the self-centering aptitude of the sets. Conclusion was that the larger the curvature the better the self-centering action will be.

Interest in investigating C-gears was awakened only in the past ten years. In addition to reference [18], research results were published by Tseng and Tsay [28,29] on Böttcher’s type A gears, another Tseng and Tsay [30,31] on C-gears that are necessarily of very limited curvature, to be cut with customary hobs, and by Wu et al. [32] on what was meant to be an application of the Wildhaber-Novikov circular-arc gearing principle in C-gears, in an obvious misinterpretation of the theory and practice of that principle though.

It could be noted that the research on C-gears, as well as the earlier suggestions by the machine tool builders give an impression that all types would be bound to have a variable pressure angle, viz. CV-gears.

6. CC-GEARs

The underlying concept of generating all CV-gears was to use a basic-rack tooth in cutting rotation at a suitable radius about an axis perpendicular to the rack pitch plane. This method makes the cutting edge(s) sweep a conical curtain, with the consequence of obtaining teeth with variable pressure angle. It could be shown [13] that the tangents of the pressure angles in the mid-plane and the side plane (represented by the slope of the tangent to the hyperbola at the pitch circle) relate by cosine the tooth inclination at the gear sides. Thus, a 20° pressure angle in the middle will increase to 23° at the sides when the tooth inclination there was 31°. The variable pressure angle is a major drawback of CV-gears, since it makes them sensitive to center distance variation; a common phenomenon in heavily loaded drive trains, such as those of rotorcraft and wind turbine generators. It is also concluded that CV-gears do not lend themselves to profile shifting because the varying pressure angle changes the center distance differently [13].

CC-gears, those with a pressure angle that remains constant along the gear face, can be generated using cutter heads whose blades have their cutting edges trace a cylindrical curtain, with its generatrix remaining parallel to the axis of rotation; the involute profile remaining identical in all transverse sections. Using such cutters (male and female) to generate C-gears has been mentioned [25] (with some modification of the machine spindle heads), but without any description of the product or the kinematics. Much earlier, Lewis [33] suggested the same cutters for generating the concave and convex flanks in rolling over the base circle, just as will be seen in Table 3. The important advantage of a constant pressure angle is the insensitivity of gearing to center distance variation, resulting in a 4-DOF pair. But since two gears contact along portions of two and three tooth pairs alternately, at a relatively large radius of curvature though, CC-gears could be said to be of quasi-4-DOF. Slight crowning or mismatch between the convex and concave flank radii would ease the mobility in the shaft skewing sense and bring the gear pair closer to being of 4-DOF. The two constraints are those against penetration and axial floating; none of the constraints being against edge loading, which the gear pair will not be prone to. CC-gears and spur (or helical) gears are seen to
switch among each other the constraint and mobility pertinent to edge loading and axial motion. Therefore, CC-gears are machine elements that are closest to being in full self-alignment, this feature being inherent in them rather than imparted to the bearings and/or their supports as previously explained in section 3. In addition, slender pinions, when twisted under torque, should still conform to the mating gear teeth, because it is not the tooth curvature that primarily changes upon elastic windup. Like in all other types of gearing, the pinions should not be made too slender, but even then, bowing can be counteracted by crowning.

Two basic types of CC-gears (generated in single indexing) could be identified, in which the cylindrical blade-edge path is circular or slightly elliptical, respectively. These types are detailed and compared in Table 3, citing the relevant references, and their basic generating kinematics is shown in Figs. 2 and 3.

### Table 3. The two basic types of CC-gears

<table>
<thead>
<tr>
<th>Type designation</th>
<th>CCA-Gears</th>
<th>CCB-Gears</th>
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<tbody>
<tr>
<td>Drawing Figure</td>
<td>Fig. 2</td>
<td>Fig. 3</td>
</tr>
<tr>
<td>Concise description</td>
<td>C-gears in contact along a circular arc that moves along the plane of action, wrapping around the base cylinders</td>
<td>C-gears having circular-arc tooth traces of the same radius in all sections parallel to the pitch plane of their basic rack</td>
</tr>
<tr>
<td>Curtain swept out by the blade edge(s); geometry of the rack tooth flanks</td>
<td>Circular cylinder</td>
<td>Elliptic (oblique) cylinder; tilted at the pressure angle to the normal to tangential feed</td>
</tr>
<tr>
<td>Tooth trace, on the rack pitch plane</td>
<td>Elliptic, identical for both flanks</td>
<td>Circular, identical for both flanks</td>
</tr>
<tr>
<td>Generating-rolling motion</td>
<td>Over the base cylinder</td>
<td>Over the pitch cylinder</td>
</tr>
<tr>
<td>References on the kinematics of gearing and cutting</td>
<td>Inoue [34], Arafa [13]</td>
<td>Cantrell [36], Melloy [37], Sidorenko et al. [38]</td>
</tr>
<tr>
<td>References on grinding</td>
<td>Inoue [34], Beljaev et al. [35]</td>
<td>Not Available</td>
</tr>
</tbody>
</table>

Figure 2. Basic generating kinematics of CCA-gears
6.1 CC-Gear Cutting Machines. Generating the tooth flanks of CCA-gears using cylindrical cutters as shown in Fig. 2 requires a generating-rolling motion that feeds the plane containing the end points of the straight edges tangentially to the base cylinder, according to Table 3. This further implies that the whole cutter head over-travels the point of tangency in order to cut the flanks down to the same depth across the face. This condition poses a stringent limitation on the number of teeth that could be cut; the base circle must lie outside the root circle by an amount that accommodates blade nose rounding, approximately to working depth. (The limitation becomes even tighter with positive profile shift). This attribute is identified in the grinding machine of Beljaev et al. [35], which is shown to grind a pinion with a rather small number of teeth. To overcome this limitation, Inoue [34] devised a CCA-gear generator that uses a flaring-cup milling cutter – and also a grinding wheel – to generate the flanks in an up/down serpentine to the same depth. This machine is considered too elaborate, and it seems to never have been built. Reconsidering the concept would mean giving up on the major advantage of high-speed hard cutting, only made possible by the rotating cutter heads that so far have been considered for all C-gear types.

Machines for generating CCB-gears of the geometry shown in Fig. 3 have been suggested according to two principles. Cantrell [36] devised a cutter head in which full-space-width V-blades were made to rotate about their own axes at a rate equal and opposite to the cutter rotation, to maintain the blade faces in the gear transverse plane, and to make the blade edges trace out the two oblique cylinders simultaneously. Melloy [37] and a three-quarter century later Sidorenko et al. [38] suggested using a curved-tooth rack cutter in reciprocating, parallel circular-arc motion to act as a gear planer. However, both methods would result in very arduous metal-cutting conditions; the rake angles at both edges vary between too much positive to negative values. The virtue of high-speed hard cutting is again lost.

It is concluded that, on the one hand, none of the earlier suggestions is good enough for manufacturing CC-gears, not by today’s standards anyway and, on the other hand, CC-gears could represent a breakthrough in the design of heavily loaded systems, because of the extensive set of advantages enumerated in the following.

6.2 Advantages of CC-Gears. Both types of CC-gears have advantages inherent in them; not shared by spur, helical and double-helical gearing, in addition to common advantages. The uniquely inherent advantages are:

1. Being a self-aligning, quasi-4-DOF pair, which would not be subjected to edge loading in an elastically deforming environment, nor impose the consequences of edge loading on the adjacent bearings.
2. Not requiring ‘lead’ correction for the windup of slender pinions.
3. Ease of crowning in the tooth lengthwise direction by a small curvature mismatch between the concave and the convex flanks, for accommodating pinion bowing.
4. Possibility of making different pressure angles on the drive and coast flanks.
5. Lending themselves to high-speed hard-cutting by CBN-coated blades, possibly not requiring finish grinding, by virtue of the face-milling with cutter heads of substantial diameters.

In addition, the common advantages shared with one or other gearing types are:

1. Running quietness.
2. Not generating axial thrust loads, and not having apex-mismatch problems.
3. Not having a center recess like double-helical gearing, hence being lighter and less prone to rim “cupping.”
4. Accepting profile shift (usually the pinion) for higher durability.

7. DISCUSSION AND CONCLUSION

Some researchers argue that present wind turbine gearboxes are of a mature technology and that substantial improvements in the designed reliability are unlikely [1]. However, the novel design concept of using CC-gears in wind turbine gearboxes offers the potential of achieving such an improvement. CC-gears are also being envisaged by Arafa and Bedewy [39] and Bedewy [40] for application in helicopter transmissions owing to the unique set of advantages they could offer.

The first step-up stage of a wind turbine gear box would consist of a bull CC-gear driving an even number of pinions, at a ratio on the order of 1:10, which pinions are straddle-mounted in pairs of axially floating bearings, all in a housing of their own. Pairs of suitably selected couplings, shafts, and anti-friction plunging joints would go to half that number of individual second-stage gearboxes, each containing two axially floating gears to drive one axially fixed output pinion. Each of the locked trains thus formed should include means for finely phasing one gear relative to its shaft for equal power splitting. Propeller shafts that carry a brake disc would then drive the multiple generators at medium speed. Such a configuration promises to satisfy the criteria of quasi-exact-constraint design, self-alignment, separately allocating functional requirements, modularity, in-situ accessibility, and maintainability. Single component weights and sizes would facilitate handling through the tower or the hatch on top of the nacelle, whenever necessary, greatly reducing the downtime and cost of maintenance.

This paper is an introduction to a novel conceptual design and a motivation to value the advantages CC-gears could offer – in particular to wind turbine systems. The paper should trigger the innovation of ‘serious,’ patent-worthy kinematic embodiments for machining either type of these gears according to its basic kinematic principle yet without the limitations outlined earlier.

REFERENCES