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The Effect of Gearbox Architecture on Wind Turbine Enclosure Size

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[The statements and opinions contained herein are those of the author and should not be construed as an official action or opinion of the American Gear Manufacturers Association.]

Abstract

Gearbox architecture – the type of gearing used, the overall gear ratio, the number of increaser stages, the number of meshes, the ratio combinations, and the gear proportions– can have a profound effect on the “package” size of a wind turbine. In this paper the author applies a common set of requirements to a variety of potential gearbox designs for a 2.0 mW wind turbine and compares the resulting “geared component” weights, gearbox envelope sizes, generator sizes, and generator weights. Each design option is also evaluated for manufacturing difficulty via a relative cost estimate.

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The importance of macro geometry

Much has been written in recent years on optimizing the “micro” geometry of gears, *i.e.*, determining the best profile or lead modifications. With this paper we propose to take a step back and consider the “macro” geometry instead. By “macro geometry” we mean the number of stages in the gear train, the type of gears used, and the amount of gear ratio used in each stage. This basic architecture of a gearbox, its “macro geometry”, is a fundamental factor in meeting the overall design objectives. Enhanced micro geometry can improve performance in the field but cannot make up for poor decision making on the basic design. Through the design exercise described in this paper we will also illustrate the interaction of “architecture” with the overall size of the drive package. One of the issues we have with the recent emphasis on micro geometry is that the modifications can only be optimized for a specific load condition. For many applications, such as wind turbines, the gearbox will be subjected to a very wide range of conditions, for most of which it will not be “optimized.” If the basic gear train design is well thought out it will be less dependent upon “optimization” for its success.

Design conditions

The design conditions selected represent a simplified specification for a 2.0 mW wind turbine gearbox, see Table 1. They do not reflect any actual design project and the results presented in this paper are not intended to be applied to any future project. The typical wind turbine design specification will include a much more detailed load spectrum, for example, along with requirements for intensive gear rating analysis. The conditions used for this paper provide a “level playing field” by which preliminary designs could be rapidly developed. The objective is to compare preliminary designs in such a way as to identify those which merit further consideration on actual projects.

Table 1. Design conditions

Design inputs	Transmitted power: 2.0 mW x 1.5 application factor = 3.0 mW [4,023 HP] Required life = 85,000 hours at full load Input speed: 15 rpm Output speeds: 150, 300, 600, 900, 1200, 1500, 1800 rpm Corresponding increaser ratios: 10, 20, 40, 60, 80, 100, and 120:1
Design constraints	Minimum number of pinion teeth: 18 Maximum face width/pinion pitch diameter ratio: 1.25 [per helix] Minimum face contact ratio [m_f] = 1.00 per helix Number of planets - 5 for ratios up to 4:1 - 4 for ratios between 4.05:1 & 6:1 - 3 for ratios between 6.05:1 & 13:1 Maximum individual mesh ratio: 6.5:1 [exception made for 10:1 single reduction] No divided power path arrangements which require radially timed sub-assemblies Compliance with AGMA rating standards for load sharing between planets Compliance with AGMA rating standards for load distribution factor Gear quality set at AGMA Q-11 per AGMA 2000 All external gears carburized and hardened
Gear arrangements considered	Single, double, triple and quadruple reduction external helical One planetary stage with zero, one, or two external helical stages Two planetary stages with zero, one, or two external helical stages
Design evaluation criteria	Number of components Estimated weight of gears and non-housing components Approximate envelope dimensions Relative manufacturing costs

Design constraints

An experienced gearbox designer has usually developed a set of guiding principles to speed his or her work. The author has spent much of his career designing special, one-off gearboxes where a conservative design philosophy is required out of respect for a lack of qualification testing and development time. The constraints adopted for this paper are reflective of that experience and the author recognizes that other designers may disagree with the limits he has established. The reasons for each of these constraints is discussed in the following paragraphs.

Minimum number of pinion teeth

The choice of 18 for a minimum number of pinion teeth was made based upon maximizing the tooth strength, achieving a minimum profile contact ratio of 1.30, and reducing the grind cycle time. [Form grinding cycle times are a function of the number of teeth, stock allowance, and face width.] Having designed parallel axis gear sets with as few as 3 pinion teeth and as many as 42 pinion teeth, 18 is a good minimum to avoid hobbing issues [undercutting, problems with start of active profile overlapping the top of the fillet] while still providing an acceptable profile contact ratio.

Maximum face width/pinion pitch diameter ratio

As gear capacity and cost tend to follow a volume function, pay careful attention to the “FD squared” principle [where F is the face width and D is the pinion pitch diameter]. It was not unheard of, back in the 1960s and 1970s, to have a face diameter ratio of up to 2.00 in through hardened industrial gearboxes. As the service hours accumulated on these long thin pinions it became apparent that torsional deflection adversely effected the life of these drives. In later design work we have had the opportunity to see the beneficial effects of reducing the F/D ratio to the 1.00/1.25 range and have avoided using a higher value ever since.

Minimum face contact ratio, M_f

If helical geometry is to be fully effective, a minimum face contact ratio of 1.00 per helix is needed. The

adjustments in the gear rating formulas to account for M_f values of less than 1.00 have limited testing behind them so they should be avoided. Once the complications of thrust and overturning moment are introduced to the bearing evaluation process, it seems prudent to insure that the gears will enjoy the full benefits of helical load sharing.

Number of planets

Figure 1 shows the geometry behind my limits on the number of planets. We recognize that non-standard geometries can allow some adjustment to these ratio limits but find them to be good guidelines for general design. As ratings are all about “power per mesh” we have chosen to use the maximum number of planets wherever possible.

Maximum individual mesh ratio

The “FD squared” principle referenced earlier plays a big part in the decision to limit individual mesh ratios to less than 6.5:1 except in the case of a single stage 10:1 double helical gear set. That exception serves as an excellent illustration of how rotating mass increases very rapidly as set ratio goes up, see Figure 2, case A.

Radial timing

As mentioned above, rating calculations are based upon power per mesh. When multiple meshes are used to share the load it becomes incumbent upon the designer to insure that load sharing is uniform or that the drive train can accommodate the anticipated degree of inequality. Our experience with industrial divided power path drives makes us very skeptical that uniform distribution ever occurs and the highly variable nature of the loads in wind turbines further increases my discomfort. For this reason we have limited the designs in this paper to those which do not require radial timing or load sharing adjustment outside the planetary stages.

Planet load sharing

Load sharing within planetary stages is widely understood within the gear design community. We are aware of the creative approaches used to reduce the variation in load between planets but decided it was best to comply with AGMA standard adjustment factors for this exercise.

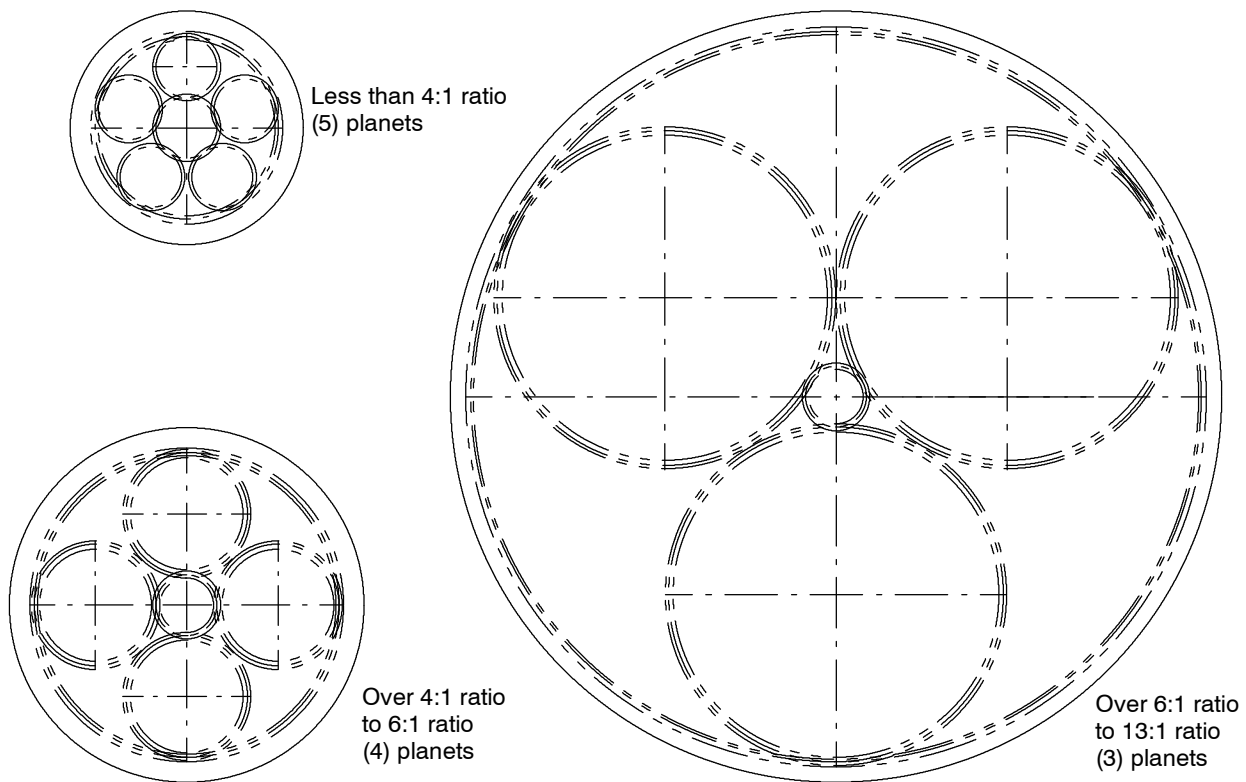


Figure 1. Number of planets vs. stage ratio

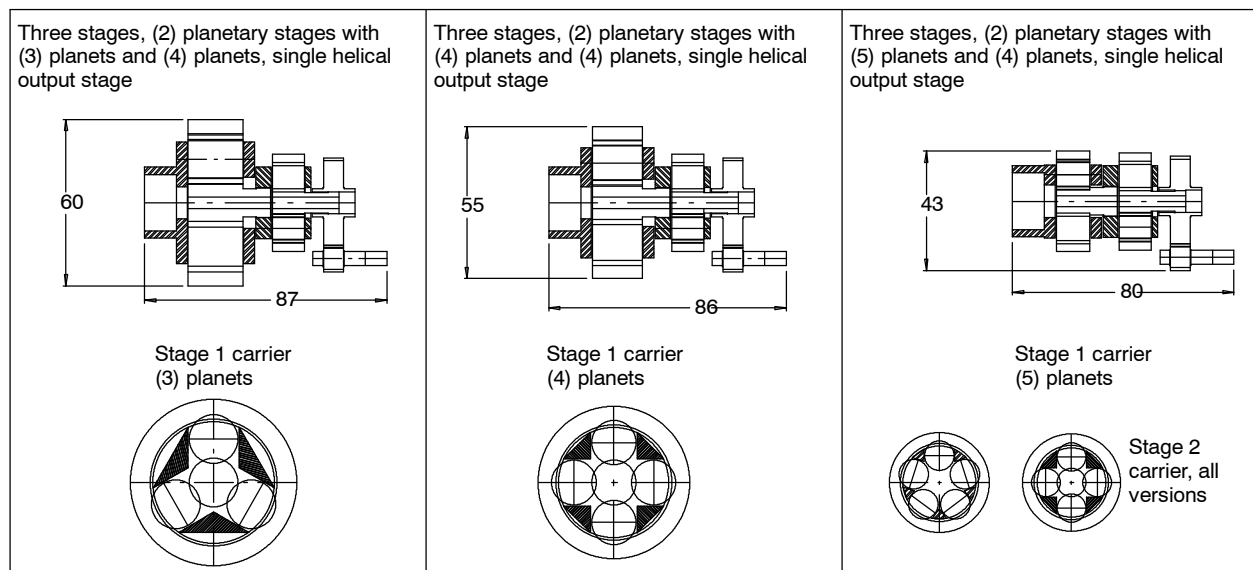


Figure 2 The effect of increasing the number of planets

Load distribution factor, C_m

While recognizing the advanced methodology being widely used to modify tooth geometry to improve operating load distribution, we have elected to comply with the C_m calculations in AGMA 2001. The purpose of this exercise is to demonstrate the effect of macro geometry on overall drive size and the potential improvement available through additional effort on C_m was not significant.

Gear set quality

Modern computer controlled gear grinding equipment is capable of consistently producing AGMA Q13 (AGMA 200–A88) parts. Considering the accuracy and loaded deflections of the mountings, however, we have reduced the gear quality to AGMA Q11 levels for this exercise. The highly variable nature of the wind turbine duty cycle along with the complexity of the assemblies contributed to our decision. The effect of improved mounted quality would not change the relative size of one design solution compared to another.

Heat treat

All external gearing in this study is Grade 2 carburized and hardened. As the durability rating of the internal gears was not a limiting factor they are calculated as through hardened [285 BHN minimum]. The alloy selection on the carburized parts and the addition of surface hardening to internal gears does not effect the final envelope size.

Evaluation of gear arrangements

As with most widely studied applications, current wind turbine gearboxes have coalesced around a narrow range of designs, typically one or two planetary stages with one or two helical stages at the high speed end. Many other arrangements are possible and the purpose of this paper is to evaluate competing designs for this demanding service. Comparison of the overall size, weight, and relative

cost of each arrangement will determine whether alternate designs are worthy of further study. The size of a drive system and its weight are major factors in the design of a tower. The number and size of the geared components have a major influence on the cost of a gearbox. If only the geared components are considered, planetary arrangements have an obvious advantage in terms of physical size and weight. When the planet carriers enter the discussion, however, the weight advantage begins to diminish.

Methodology

Using the guidelines described above, the first step in this exercise was to design the anticipated gear sets in 1 NDP. As gear ratings are parametric in nature, the approximate tooth size needed to carry a specific load can be found by taking the cube root of the ratio between the 1 NDP rating and the target rating. All other dimensions for the set can be found by dividing the 1 NDP dimensions by the final NDP selected. As the dynamic factor decreases as size decreases, the rating summary charts show ratings slightly higher (<10%) than the minimum acceptable values.

Once the required gear sets were designed they were arranged into typical gear trains in a CAD program. Bearing journals, shaft extensions, planet carriers, and output hubs were sized using conservative stress levels. ***No attempt has been made to execute detailed design on the (28) gear trains studied. The preliminary layouts could be developed further but met the purposes of this exercise in the condition presented.*** Each design was then evaluated for approximate enclosed volume, estimated weight, and relative cost to manufacture. Figures 3 through 9 show the CAD layouts for each increaser ratio in the same scale. Table 2 shows the relative cost, estimated weight, and approximate volume comparison for the (28) designs. Tables 3 through 10 provides the gear geometry for the gear sets used in the designs.