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A DESIGN SOFTWARE TOOL FOR CONCEPTUAL DESIGN OF WIND TURBINE GEARBOXES

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Abstract

The paper reports the development of a design software tool for wind turbine gearboxes. It facilitates the conceptual design of wind turbine gearboxes supporting designs with different combinations of epicyclic and parallel gear stages. Analyses of gear bending strength and pitting resistance are in accordance with the AGMA2001 standard. The calculations of the AGMA geometry factors I and J are verified in accordance with the AGMA 908 information sheets. A case study of a 2MW and three phase asynchronous generator with a nominal rotational speed around 1600 rpm has been tested to demonstrate the capabilities of the design software tool.

Key words: Wind Turbine, Gearbox, Conceptual Design, Design Tool.

1. Introduction

Worldwide, there has been a continual increase in the use of wind power since 1996, with the power generation capacity of new installations increasing year on year. The biggest consumers are the USA, Germany and Spain who account for over half of the world market. Statistics released by the European Wind Energy Association (EWEA) show that 43% of all new electricity generating capacity built in the European Union in 2008 was wind energy, exceeding all other technologies including gas, coal and nuclear power [1]. In the UK, wind energy overtook hydropower to become the largest renewable generation source, contributing 2.2% of the electricity supply with a great potential to further increase its capacity by developing offshore wind farms.

A wind turbine (WT) is capital-intensive with considerable operational and maintenance costs compared to conventional fossil fuel fired technologies. The gearbox is one of the important subsystems in an indirect drive wind turbine and the gearbox alone counts for about 13% of the overall cost of a WT [2]. The field data has shown that the gearbox has a low availability due to its high downtime per failure and gearbox failure incurs high costs for repair [3].

The major sources of failure in gears of conventional applications are bending stress causing tooth fracture and contact stress causing surface damage including pitting, scoring and abrasive wear. Methods for analysing gear loading capacity used by the industry are mostly based on the American Gear Manufacturers Association standard ANSI/AGMA 2001 [4] and its ISO counterpart, BS ISO 6336 [5]. Historically, early wind turbines were subject to very high failure rates due to fundamental gearbox design errors and consistent under-estimation of the operating loads. As understanding has grown these problems have been identified and reduced, particularly with the introduction of internationally recognised standards [6-7].

Over the past 15 years data on wind turbine operation has been collected from many different countries by organisations. A survey of three regions by Tavner et al. [8] shows a gearbox failure rate of around 0.1 failures per turbine per year, based on manual records of operators. However, the literature shows the gearbox to have the highest mean downtime of any subassembly. Publicly available data on the breakdown of gearbox failures by component is hard to find. The literature suggests that the gearbox failures are a result of inadequate bearings. One source which refers to specific components is the E.ON UK report on the first year of operation of the Scroby Sands Wind farm, located off the coast of Norfolk [9]. Forty-three incidences of major unplanned work are reported, of which 39 are caused by failures within the gearbox. All of these are failures of the bearings, 27 were down to the intermediate speed shaft and the remaining 12 down to the high speed shaft. In a field of just 30 turbines this is clearly a systematic problem exposed in the first year of operation but the turbine manufacturers, Vestas, are among the market leaders and this indicates a design deficiency in not being able to rate the bearings correctly, in agreement with the NREL observations [10]. Despite the belief that gear design is maturing, tooth failure is still contributing to eventual gearbox failure. In August 2007 Clipper Windpower discovered a gear tooth failure problem with their 2.5 MW Liberty series turbines which resulted in the need to repair and refit over 50 turbines on several sites. A design problem combined with quality control issues were blamed (Ragheb, 2009) [11].

Development of new gearbox concepts with design evaluations in the conceptual design phase is essential to eliminate gearbox design faults and to improve the operational performance. A WT design generally involves multi discipline design activities; an easy to use design software tool for WT gearboxes would allow the design team to explore various possible design concepts in the early phase of the design. Although commercial software is available, it can be expensive, unsuitable for wind turbine applications or too complicated to be used by engineers without specialist gear design knowledge.

This work has developed a design software tool to address these issues and to facilitate the conceptual design of wind turbine gearboxes. It supports designs with different combinations of epicyclic and parallel gear stages and allows arbitrary transmission ratios at individual stages to accommodate required input/output speeds of the gearbox. It is specifically designed for easy to use with a user friendly graphical interface, providing defaults and options based on wind turbine gear technology. It allows quick, accurate analysis and refinement of conceptual gear design by providing instant results for proposed design changes. Specifically, the software tool produces analyses of bending strength and pitting resistance of gear teeth in accordance with the AGMA2001 standard. This is because research shows that the majority of gearbox failures are attributable to bending fatigue of the teeth or abrasive wear of the bearings - with pitting fatigue damage to the tooth surfaces providing a source of abrasive material. Using the software tool, results for the AGMA geometry factors I and J are calculated and verified in accordance with the AGMA 908 information sheets. A case study illustrating the use of the software and its capabilities is presented. The tool also links with subsequent stages of the design by allowing further analysis via exporting of gear geometry to Finite Element Analysis software. This software could be used among the members of a wind turbine design team to aid concept design integration and improvement.

2. Review of Current Gear Design Software Tools

An evaluation of existing software solutions for the conceptual design of gearboxes was undertaken. The evaluation informed decisions on how the program being developed for this project should present itself to the user, and what features should be included. Two main software providers and their products are summarised as following.

RomaxWIND

Romax Technology is a specialist transmission software developer whose flagship product, RomaxDesigner, is an industry leading transmission CAE package. It has an edition dedicated to wind turbine applications, RomaxWIND. The software uses the ISO 6336 standard to calculate the safety factors for each gear in terms of contact stress and bending stress. The 20 year duty cycle is input as a histogram of cycles against torque and applied using Miner's rule. RomaxWIND provides an export function to allow more detailed analysis of the components using an independent Finite Element analysis package.

GearCALC/KISSsys

GearCALC is the specialist gear design module of the mechanical design package KISSsys. It provides a design environment into which components created in the specialist modules can be deployed. This system is not constrained to designing transmission systems but, like RomaxWIND, can be used to put together gears, shafts and bearings. GearCALC is used to design one gear stage at a time, and the results display the factors of safety for bending stress and pitting resistance in accordance with the AGMA 2001 standard. Again there is the option of applying Miner's rule for fatigue analysis. Once the user is satisfied with the design and results obtained the gear stage can be exported to CAD software or into the KISSsys environment.

There are constraints of using the above commercial software and the cost of license is high. For example, *RomaxDesigner* can provide a bespoke consultancy package the cost of using the software is unknown but likely to be high. *GearCALC* by developer KISSsoft as a non-specialised (without epicyclic design) gear systems package can be downloaded at a cost of $4,000 for an individual license. The American Gear Manufacturers Association (AGMA) provides a *Gear Rating Suite* which can be downloaded at a cost of $1,500. It is based on ANSI/AGMA 2001 standard and the ISO6336 standard but not specifically designed for wind turbine gearbox applications.

3. Design of the Software Tool

The program developed in this project aims for conceptual design of wind turbine gears. Its purpose is to help researchers and designers gain an understanding of the effect of changing the design parameters, compare one system/arrangement with another and permit the exchange of this information between design groups. The project has decided to adopt the AGMA 2001 standard [4] and it is the authors' option that it presents a more complete method. The user may not have studied the AGMA standard and the software should reduce the need for this time consuming exercise and make the gear design more accessible. The analysis should include the bending strength and pitting resistance safety factor calculations, the maximum transmissible power, and a fatigue analysis for each stage. Wind turbines typically have between one and three stages in a combination of epicyclic and parallel gear transmissions. The program, named as 'PinWheel', was written using Microsoft Visual Studio 2005 in the Visual C++ .NET programming language.

3.1 Implementing the AGMA 2001 Standard

The key calculations performed by the program are the AGMA 2001 gear load calculations. A particular challenge faced was in evaluating the Bending Strength Geometry Factor (J-factor) and Pitting Resistance Geometry Factor (I-factor) – which are not defined in the AGMA 2001 standard. Several methods are found in the literature for evaluating both and this is due to the complexity of AGMA's suggested method – presented in information sheet AGMA 908 [12]. For the J-factor in particular the iterative calculation method is considered tedious and when working through the calculations manually, designers are often directed to a series of charts of J-factor against the number of teeth on the gears [13]. Each chart is only valid under certain conditions and invalid otherwise. Interpolation between these charts is not recommended, so another method was needed to evaluate J-factor given any set of input parameters. To maintain the calculation accuracy, this project implemented the full AGMA 908 calculations for both I and J factors. These calculations were tested in spreadsheets before being implemented into the program. Verification of the results generated by PinWheel was performed by comparison with tabulated data produced by AGMA 908.

3.2 Structure of the Program

The program is written using the Object Oriented Programming (OOP) paradigm. The major classes in PinWheel fall into two categories - those which represent a user interface and those which contain data on a particular gear stage or the system as a whole. Figure 1 illustrates how the user can navigate through the program to perform the tasks required to edit, analyse and produce a report on the system.

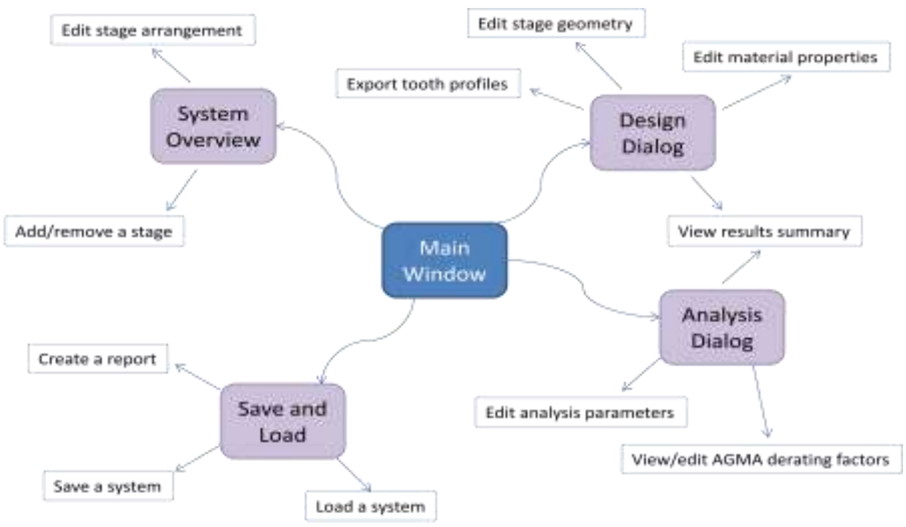

Figure 1 User navigation through PinWheel

The MainWindow class contains the functionality to save, load or create a new gear system and provides a gateway to access the design dialog and analysis dialog of each gear stage.

The Design Dialog allows the user to view and edit the geometric and material properties of a gear stage. A panel displaying the meshing teeth of the gear and pinion is shown in the dialog window, and updates the tooth profile if changes are made to the geometry. The option to export the tooth profiles for further Finite Element analysis is presented here. To prevent the user from creating a gear geometry which is not possible in reality, only certain combinations of user-specified values are permitted.

The Analysis Dialog shows the results of the analysis of the stage and allows the user to alter some of the factors and design criteria (e.g. design life, manufacturing accuracy) which will affect the results. This class contains the calculation functions to evaluate the AGMA 2001 rating equations and updates the results whenever a change is made to the geometry, material properties or design criteria. The analysis can be affected more directly by overriding some of the AGMA 2001 derating factors, which is useful to users who are familiar with using the standard.

The System Overview displays the gear transmission type (epicyclic or parallel), ratio and schematic of each of the gear stages in the system, as well as the rotational speed at the input to the system and after each successive stage. The user can change the type and ratio of each stage, and the system input speed. The options for adding or removing stages (between a minimum of 1 and a maximum of three) are also presented.

3.3 Ease of use

As well as providing a graphical user interface which will be familiar to users of other Windows programs, PinWheel provides a facility to automatically create a ‘ready-made’ gear system from the minimum of user inputs. When creating a new system the user needs to specify only the input rotational speed, output rotational speed and the power to be transmitted through the gears. From this information the program creates a gear system using the set of rules listed below. These rules have been devised as ‘rules of thumb’ based upon observations of typical wind turbine gear design and a consideration of technical and economic factors. The rules are presented in the order they are applied:

Rule 1: The overall gear ratio to be achieved is the ratio of the specified input speed to the specified output speed.

Rule 2: The gear system can consist of between 1 and 3 stages, each of which will be either parallel or epicyclic.

Rule 3: Parallel stages can have an individual stage ratio between 1 and 6, epicyclics between 4 and 12. To simplify the process, only whole number or half number (1.5, 2.5 etc.) ratios are permitted. If the overall gear ratio cannot be achieved exactly using these ratios, the nearest approximation to the overall ratio which can be achieved is used and the user informed.

Rule 4: A combination which achieves the overall gear ratio in fewer stages is preferred as this is likely to be a cheaper solution.

Rule 5: A combination which achieves the overall gear ratio whilst using fewer epicyclic stages is preferred as this is again likely to be cheaper.

Rule 6: The pinion is set to a default diameter of 500 mm and the other gear diameters are calculated from this.

Rule 7: The following default values are used: face width is 250 mm; stage module 25 mm; addendum 1; dedendum 1.25; pressure angle 20°; helix angle 0°.

If the user already knows the number of stages they wish the system to contain and the type (epicyclic or parallel) for each stage this information can be supplied and rules 4 and 5 are therefore bypassed. The resulting gear system is not optimised and will require alteration by the user for their particular requirements. However, these alterations are easily made in the software.

4. Results and Discussion

4.1 Verification of the Program

In order to verify the results calculated by the program, a comparison was made with a report generated using GearCALC, a commercial software program evaluated in section 3. The GearCALC report is taken from a tutorial paper by Fish and Halter [14] and provides a complete set of calculation results for the derating factors, geometrical properties, pitting resistance and bending stress in accordance with the AGMA 2001 standard. The example used is a 2:1 reduction gear stage with an input speed of 1750 rpm and transmitted power of 60 horsepower (44.74kW). Table 1 presents the results obtained using GearCALC and those obtained using PinWheel. Imperial units are used in the GearCALC report but have been converted to metric in the table, and all values are quoted to the same number of significant figures as in the GearCALC report. Where there is a discrepancy between the results the difference has been quoted as a percentage. A negative difference indicates PinWheel produced a lower answer than GearCALC.

Table 1 Comparison of GearCALC and PinWheel results

Calculated Parameters by AGMA Standard		GearCALC Tutorial		PinWheel		Difference	
		Pinion	Gear	Pinion	Gear	(%)	
Stress Numbers							
Contact Stress Number (MPa)	S_c	1108.2		1108.0		-0.02	
Bending Stress Number (MPa)	S_t	331.4	297.4	330.7	298.3	-0.20	0.29
Maximum Contact Stress Number (MPa)	S_{ac}	1060.3	969.2	1060.3	969.2	0.00	0.00
Maximum Bending Stress Number (MPa)	S_{at}	364.9	336.5	364.9	336.5	0.00	0.00
Safety Factors							
Safety Factor for Pitting	S_H	0.96	0.87	0.96	0.87	0.00	0.00
Safety Factor for Bending	S_F	1.10	1.13	1.10	1.13	0.00	0.00
Power Ratings							
Pitting Resistance Power Rating (kW)	P_{ac}	40.95	34.23	40.97	34.21	0.05	-0.06
Bending Strength Power Rating (kW)	P_{at}	49.27	50.62	49.36	50.46	0.18	-0.31

4.2 Case Study – Designing a Reliawind R80 Gearbox

In order to demonstrate using the program a case study has been prepared illustrating three stages in the design of a proposed new gearbox. The gearbox is proposed by EU Reliawind project as one of the generic wind turbine configurations, R80 [15]. It has a nominal power of 1.5-2.0 MW and a gearbox design of one epicyclic stage followed by two parallel stages with a transmission ratio of approximately 100.

The R80 is to use a 50 Hz, three phase asynchronous generator with a nominal rotation speed around 1600 rpm. Under good conditions the turbine blades are designed to rotate at around 16 rpm. Using just this information a new gear system can be created in PinWheel. Figure 2 shows this information entered into the New System Dialog.

Figure 2 Input data for R80 gearbox

The System Overview is displayed as shown in Figure 3 and changes can be made to the arrangement of different gear stages and transmission ratios. In this specific case study, gear ratios are 6, 4 and 4.2 respectively, for one epicyclic stage and two parallel stages. By clicking 'Apply Changes', the System Overview will apply the changes made and display scaled diagrams of the three stages.

The 'Gear Geometry' tab in the stage dialog shows that each stage has a 500 mm diameter pinion with 50 teeth and a face width of 250 mm as is used by PinWheel by default, as shown in Figure 3. The largest stage is the epicyclic, which has a pitch diameter of 2.5 meters for the ring gear. Through-hardened steel is the default material used in PinWheel and this will be used for the R80 gearbox design. The 'Material Properties' tab presents options to alter values but no editing is required in this example. In the analysis dialog 'Analysis' tab some of the options presented are changed in accordance with the known manufacturing procedures to be used. The fields edited are:

- The checkbox indicating that lead correction will be used is checked
- The checkbox indicating that gear compatibility will be improved at assembly is checked.
- The *AGMA manufacturing accuracy grade*, A_v, is changed from 8 to 7.

The default design life of 120,000 hours, default reliability of 99.99% and default shock loadings are not changed. When this information is entered into the program the bending strength and pitting resistance calculations are updated automatically, as shown in Figure 4. Read from the 'Results Summary' at the bottom of the current window the safety factor results for the three stages are presented in Table 2.

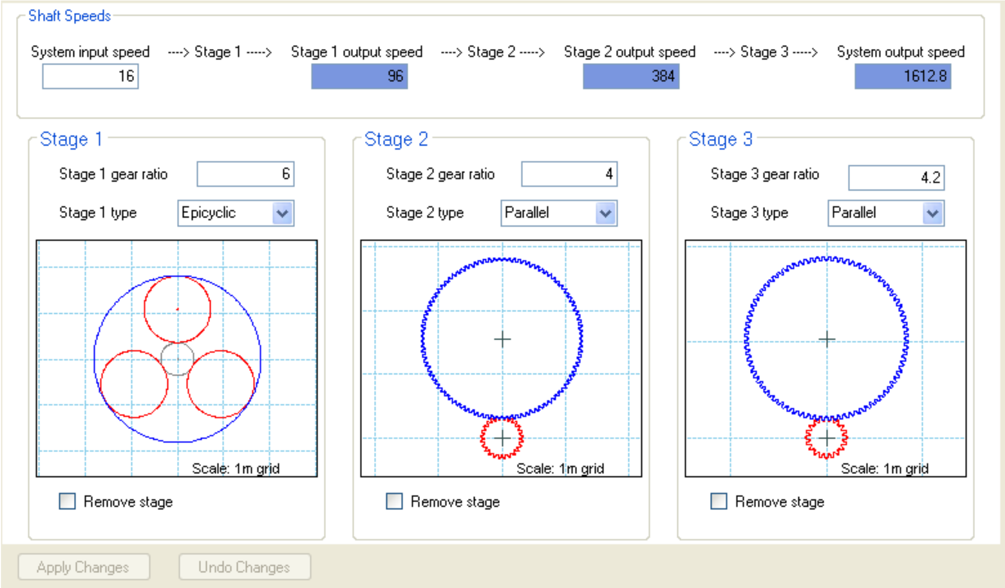

Figure 3 The system overview with 3 gear stages

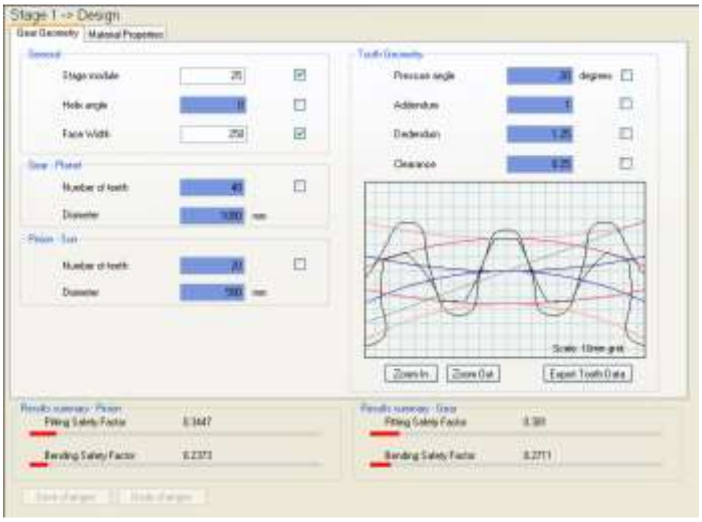

Figure 4 The initial design dialog for determine the geometry of the stage

Table 2 – The initial design safety factors for each gear stage

Gear Stage		Pitting Resistance Safety Factor	Bending Strength Safety Factor
Stage 1	Pinion	0.34	0.24
	Gear	0.38	0.27
Stage 2	Pinion	1.29	0.78
	Gear	1.39	0.87
Stage 3	Pinion	2.13	2.42
	Gear	2.29	2.68

Comparing the pitting resistance safety factors with the minimum recommended value from the AGMA 6006 standard of 1.25 it can be seen that the designs for stages 2 and 3 would be acceptable under this standard but the stage 1 design would not. Similarly the AGMA 6006 recommended minimum bending strength safety factor is 1.56 – and only stage 3 is acceptable in this case.

Working from the stage dialog PinWheel updates the results instantly each time a change is made to the design. In this way even an inexperienced user can quickly develop an appreciation for the relative impact different changes make. Several combinations of geometrical alterations are tested in order to find a conceptual stage design which meets the safety factor criteria. Table 3 shows one possible solution for three stages. In the first epicyclic stage, it uses a 25 mm module combined with a 20° helix angle and an increased face width of 750 mm. 29 teeth on the sun gear and 58 on the planet gear are used and this yields a ring gear diameter of 3.625 meters, as shown in Figure 5. The safety factors for this proposed three stage design are shown in Table 3 and meet the AGMA 6006 recommendations. Figure 6 shows the Design Analysis dialog which calculating derating factors for updated stage 1 design. An experienced user who is familiar with AGMA 2001 standard can make changes to the parameters in this dialog depending on the operational condition of the intended application. Reports including the detailed calculations of geometrical data, material data, load data, derating factors, I-factor and J-factor, stress numbers and power ratings can be generated for each stage. Gear tooth profile can be exported for detailed stress analysis using Finite Element analysis software.

Table 3 – The updated design parameters for each gear stage

Gear Stage		Module (mm)	Helix Angle (°)	Face Width (mm)	Number of Tooth	Pitting Resistance Safety Factor	Bending Strength Safety Factor
Stage 1 (epicyclic)	Sun	25	20	750	29	1.27	2.14
	Planet Gear				58	1.41	2.32
Stage 2 (parallel)	Pinion	16	10	500	27	1.31	2.32
	Gear				108	1.42	2.71
Stage 3 (parallel)	Pinion	12	20	220	25	1.28	2.45
	Gear				105	1.39	2.85

In calculating the transmitted tangential load it is assumed that the load on the gears is constant over the number of load cycles. As this load will be determined by the wind speed it will, in reality, show significant variations. Presently, the software uses the average rotational speed of the input shaft (supplied by the user) and power transmitted to determine the average load. Design evaluation under cyclic loading requires applying a spectrum of load cases and the use of a cumulative fatigue model such as the Miner's Rule. The load spectrum is input as the number cycles spent in ranges of input shaft speeds. The safety factors are calculated for each range and the total safety factor is the cumulative total. This provides an improvement over the assumption of constant loading, but the average load case is still a valid estimation. Inclusion of Miner's Rule would be a way of developing the program in the future.

Figure 5 The design dialog of updated stage 1 design

Figure 6 Calculation of derating factors of updated stage 1 design

5. Conclusions

The software tool developed, PinWheel, provides effective means to design wind turbine gearboxes at the conceptual design phase. It produces analyses of bending strength and pitting resistance in accordance with the AGMA2001 standard. PinWheel results for the AGMA geometry factors I and J were carried out in accordance with the AGMA 908 information sheet. A case study of a 2 MW turbine with a gearbox of three stages has been carried out to demonstrate the capabilities of the tool. This software could be used among the members of a wind turbine design team to aid concept design integration, and ultimately contribute to increasing the reliability of wind turbines.

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