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GYROTORQUE^{TM} TRANSMISSION SYSTEM FOR WIND TURBINES

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IMPROVEMENTS IN DRIVE TRAIN RELATED TECHNOLOGY

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ABSTRACT

The GyroTorque^{TM} transmission system employs gyroscopic torque reaction to transmit power offering an alternative to the gearbox and electrical variable speed drive of a conventional wind turbine. The power transmission is fundamentally oscillatory and is rectified by mechanical elements.

A precessing gyro maps speed to torque and, since the wind turbine rotor inertia strongly filters rotor speed variation, output power is insensitive to wind turbulence because it reflects wind turbine rotor speed variability rather than rotor torque variability. The GyroTorque^{TM} system has only bearing losses and potentially a high efficiency. Mechanical control of the input to the GyroTorque^{TM} system enables wide range variable speed operation of the wind turbine rotor using a conventional synchronous generator.

At present, a 6 gyro system driven by an axial cam and connected to a conventional synchronous generator is the preferred system. Loads and power quality have been addressed with computer simulation models of the GyroTorque^{TM} system.

Outline assessment of system mass and cost gives encouragement that it may be less than for conventional transmission systems.

KEY WORDS
GyroTorque^{TM}, transmission system, wind turbine, power quality, synchronous generator, turbulence insensitivity.

INTRODUCTION

The basic GyroTorque^{TM} arrangement (Figure 1) is as proposed by the inventor, Mr M Jegatheeson [1].

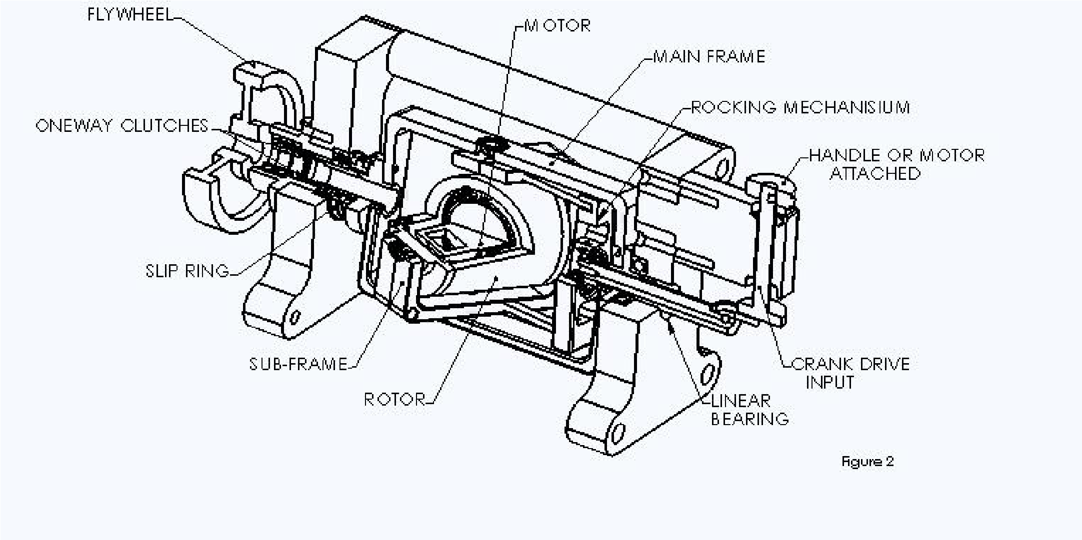

Figure 1 A GyroTorque^{TM} unit based on a demonstration model

The GyroTorque^{TM} system input is the rotary motion of the crank shaft. This causes an oscillatory translation through the linear bearing which, via the rocking mechanism, rocks the gyro axis contained in the sub-frame. Altering the orientation of the gyro spin transmits torque to the main frame. This torque is oscillatory and using a one-way clutch system and flywheel beyond the clutch, the output motion is made rotary and uni-directional. Note that Figure 1 represents a concept demonstration model only and the GyroTorque^{TM} unit in a wind turbine system whilst fundamentally similar may differ in many details.

The GyroTorque^{TM} unit as illustrated in Figure 1 has input and output shafts at right angles which is not usual (but also not unknown) in wind turbine systems. A preliminary evaluation of the GyroTorque^{TM} system for a wind turbine application was conducted by Garrad Hassan & Partners Limited (GH) on behalf of Gyro Energy Limited (GEL). Some key conclusions were:

- with a single gyro, it is impractical to regulate wind turbine rotor reaction torque by variation of gyro speed. The gyro may typically have more stored energy than the wind turbine rotor and consequently torque control based on gyro speed will involve long time constants and be ineffective.

- gyro bearing loads are critical. The gyro bearings experience radial forces associated with a transverse torque which is greater than the input shaft torque by a factor which depends on the linkage geometry and output wave form.

- control of rotor torque reaction via variable input link geometry is considered a promising and essential feature of GyroTorque^{TM} system design.

- the half-rectified output power waveform of a single GyroTorque^{TM} unit leads to unacceptable power quality. Thus a system with multiple gyros is considered to be essential.

Following the preliminary evaluation a further stage of design development has addressed all of the above issues. Discussion of this follows after some clarification of the basic power transmission characteristics of the GyroTorque^{TM} system.

Some of the most important findings are that, with multiple gyros:

- adequate control response can be achieved to allow the use of gyro speed variation as a means of operating the wind turbine rotor in variable speed in wind speeds below rated wind speed.

- the intrinsically variable gyro output waveforms are shifted to higher frequency through the gearing up effect of the input cam drive and the output of the multiple gyro system can be filtered effectively by a synchronous generator to produce very smooth output power.

FUNDAMENTALS OF GYROTORQUE^{TM} POWER TRANSMISSION

Analysis of the GyroTorque^{TM} system reveals that the average power transmitted is proportional to the;

- angular momentum of the gyro,
- input speed of each GyroTorque^{TM} unit
- output speed of the GyroTorque^{TM} unit within limits related to main frame inertia
- mean position and amplitude of rotation of the gyro axis in its sub-frame.

For any given wind turbine design, the input speed range will be prescribed by choice of start up speed and maximum tip speed. The output speed (range) is set by the generator. This means that the transmitted power and input and output speeds of the GyroTorque^{TM} units are also prescribed. These speeds may differ from rotor speed or generator speed of the wind turbine if there is a gearbox somewhere in the transmission path in series with a GyroTorque^{TM} unit but they are otherwise fixed by the system arrangement. Thus the angular momentum of the gyro set is uniquely determined by the rated power demand and there is only freedom to optimise the relative contributions from gyro inertia and gyro angular speed.

Change of mean position of the gyro axis angle and or gyro speed variation are possible bases of torque regulation in operation below rated wind speed. Above rated wind speed the gyro geometry or speed need not be altered as pitch control is used to regulate rotor speed.

A GyroTorque^{TM} system with a rated electrical power output of 1 MW has been developed and simulations conducted to derive design loads, appraise performance and investigate system dynamics.

SYSTEM CONCEPT

Preliminary work suggested that multiple GyroTorque^{TM} units are required to smooth the output power which is essentially a half rectified waveform with *the frequency of the wind turbine input shaft* although it is carried at output speed. Gyro bearing loads have a critical impact on GyroTorque^{TM} system design and, for a single GyroTorque^{TM} unit transmitting all the power, the radial loads on the bearings were excessive and out of the range of standard rolling element bearings.

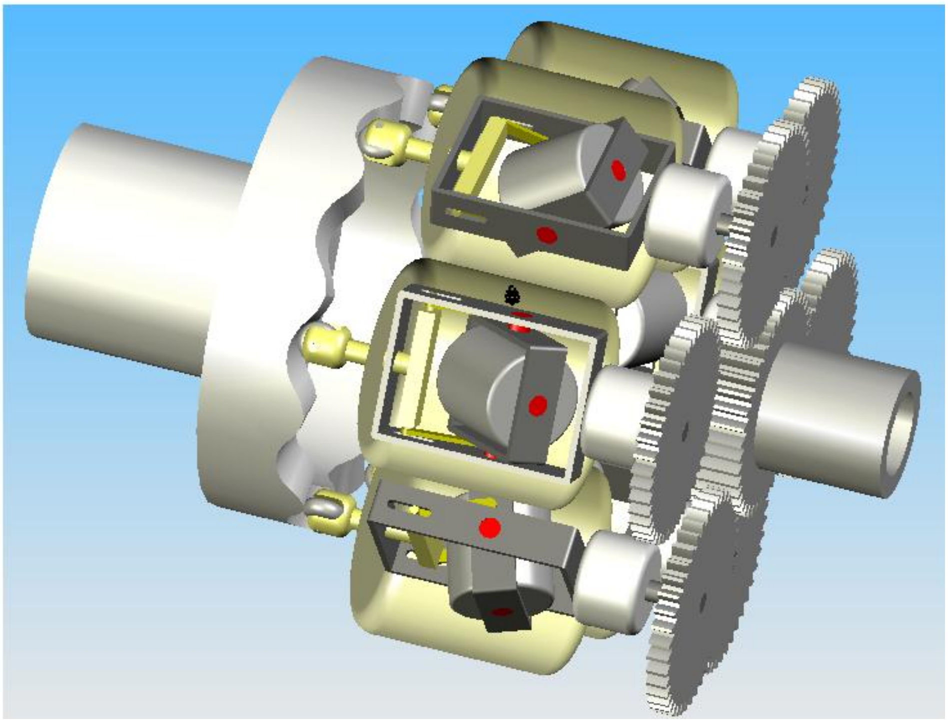

Figure 2 GyroTorque^{TM} System with Axial Input Cam and 6 GyroTorque^{TM} Units

The system (Figure 2) has evolved as a solution to both these problems. Rotation of the input cam causes oscillation of the gyro axes. Six GyroTorque^{TM} units each at **π/3** relative phase are summed to provide a mechanical output to a synchronous generator (not shown). Although this mechanical output (Figure 3) is still highly irregular, the variations are at high enough frequencies to be very effectively filtered by the synchronous generator (Figure 4). The axial cam system effectively gears up the input frequency by **m**, the number of cam lobes (presently 11), and hence the input torque per GyroTorque^{TM} unit is reduced by a factor of **6m.** This considerably alleviates the gyro bearing loads and for gyro speeds up to about 2400 rpm, use of standard rolling element bearings is feasible.

The clutches (non-friction type subject to a separate patent of M Jegatheeson) are represented in the schematic of Figure 2 as cylinders adjacent to the output gear train. The mechanical outputs of the 6 GyroTorque^{TM} units are collected in a 1:1 gearbox. Although relatively large, this gearbox being on the output stage (running at 1000 rpm in the megawatt scale system evaluated by simulation) is lightly loaded.

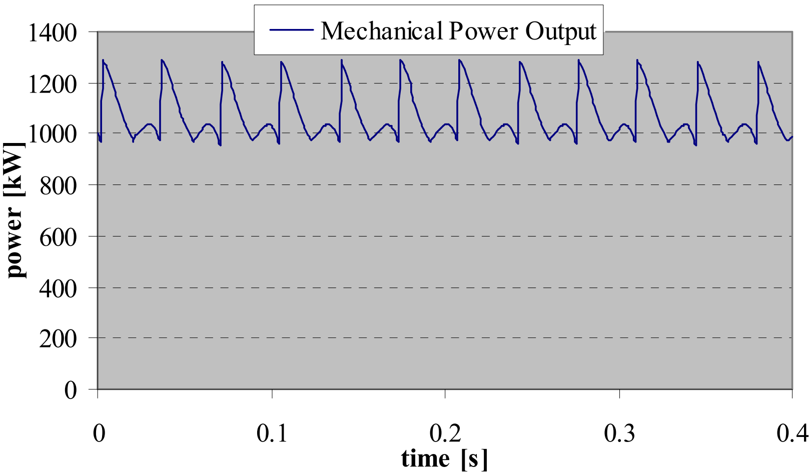

Figure 3 Mechanical output power of 6 GyroTorque^{TM} units

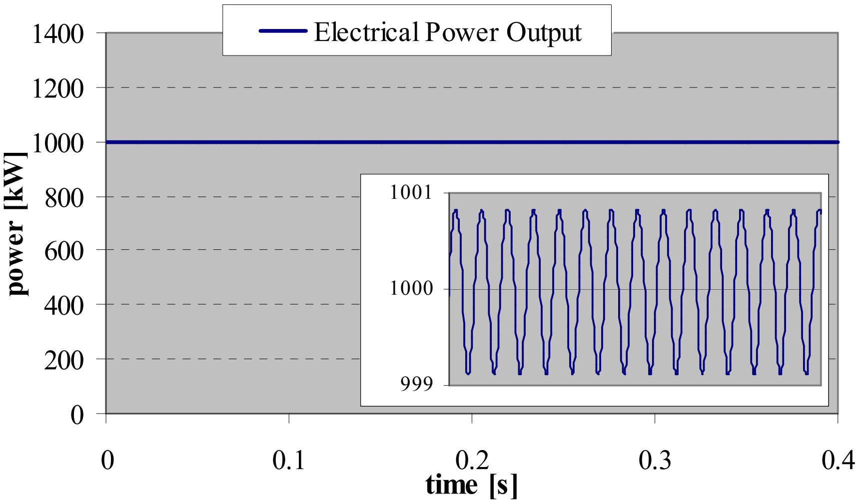

Figure 4 Electrical output power from the synchronous generator

SIMULATION MODEL

The 6 GyroTorque^{TM} system was analysed by W Leithead (Strathclyde University) and a computer simulation model was developed. This model is presently restricted to operation above rated wind speed at full rated power and without any dynamic change of the GyroTorque^{TM} linkage geometry.

The simulation allows a representation of turbulent wind as it affects the input aerodynamic torque with a closed loop control system to regulate rotor speed. The control system operates by varying blade pitch angle to balance the reaction torque from the input cam system with the input aerodynamic torque in order to hold rotor speed constant.

At the top level of the simulation system, the 6 GyroTorque™ units are represented as a unit interacting with the input cam and output generator and then, at a deeper level as individuals each similar but at **π/3** relative phase. The clutches of each GyroTorque™ unit are represented by appropriate logic and the mechanical outputs summed and then connected to a third order synchronous generator model.

SIMULATION RESULTS AND ENGINEERING IMPACTS

Gyro speed

Although no torque (other than a small torque from the gyro motor to make up bearing losses) is applied on the axis of rotation of the gyro and its speed is constant in its own frame of reference, the gyro speed is varying relative to the sub-frame (Figure 5). The stator of the gyro motor is fixed in the sub frame. The gyro comprises a hollow steel cylinder providing the required rotational inertia and carrying some internal copper bars to act as the rotor of an electric motor which makes up gyro bearing losses.

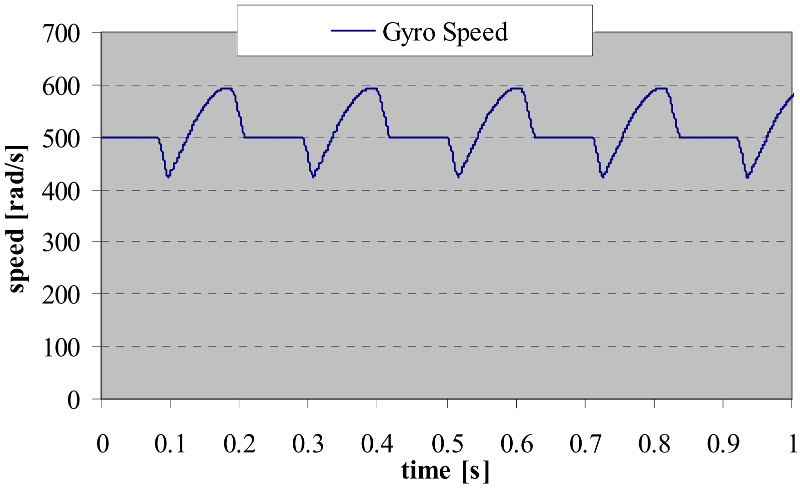

Figure 5 Gyro speed relative to the sub frame which contains its bearings

It is important that the gyro motor (rated at around 3 kW for one of 6 gyros in a system with total output of 1 MW electrical power) is activated only when the main frame is stationary and the gyro speed is constant (see waveform of Figure 5) relative to the sub frame.

Gyro bearings

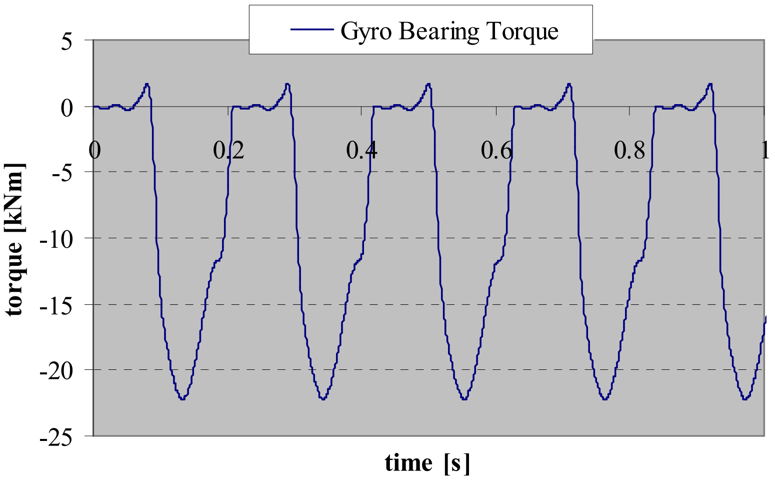

Figure 6 Tgx – largest reaction torque on the gyro bearings

The gyro is directly in the transmission path and the transverse torque on the gyro axis as a vector parallel to the transmission axis cannot be less than the nominal torque commensurate with the shaft speed and power transmitted. However the torque about an axis at right angles to the transmission axis is typically much higher than this. This is in part due to the irregularity of the gyro output waveform and the need to have a peak transmission torque appreciably higher than the average which determines the power transmitted.

As the gyro axis (principal axis of rotation of the gyro defined in its sub frame as the **y** axis, refer to Figure 1) rocks about another axis (through the sub frame bearings) arbitrarily defined as the x axis, it experiences torques about both the **x** and **z** axes which introduce high radial loads in the gyro bearings. By far the largest torque component is the torque about the x axis, **Tgx** (Figure 6).

$$\mathbf{Tgx(t) := (Igz - Igy)\cdot\left[\sin(\theta(t))\cdot\cos(\theta(t))\cdot(\phi d(t))^{2}\right] - \left(Igy\cdot\cos(\theta(t))\cdot\phi d(t)\cdot Nr(t) + Igx\cdot\theta dd(t)\right)}$$

The bearing reaction torque **Tgx(t)** is approximately represented by the above equation where. The first term can be made zero by a design in which **Igz** (inertia of gyro about the z axis) and **Igy** (inertia of gyro about the y axis its principal axis of rotation) are made equal. Clearly the gyro axis angle **θ**, is changing at the same frequency as the input speed (product of wind turbine rotor speed and number of cam lobes). The middle term of **Tgx(t)** consists of the product of gyro angular momentum **Igy.Nr(t)** (where **Nr(t)** is the gyro speed) and the output speed, **Φd(t)** (time derivative of output angle, **Φ)** with a further factor **cos(θ(t))** which will be maximum at unity and remain close to unity for comparatively small angular movements of the gyro axis. This middle term is the main term and the last term involving **θdd(t)**, the second derivative of **θ** is minor. A major challenge in the GyroTorque^{TM} design is in dealing with the bearing loads associated with the torque, **Tgx**.

GyroTorque^{TM} output speed

The main frame (Figure 1) of the GyroTorque^{TM} unit is accelerated by the gyro torque to synchronous speed (1000 rpm or 105 rad/s in Figure 7) at which point a clutch engages, the synchronous generator holds output speed and the gyro torque provides power to the generator. As the gyro axis reverses rotation and the applied torque also reverses, the speed of the main frame falls below synchronous and the main frame comes to rest being prevented from reverse rotation by a second clutch.

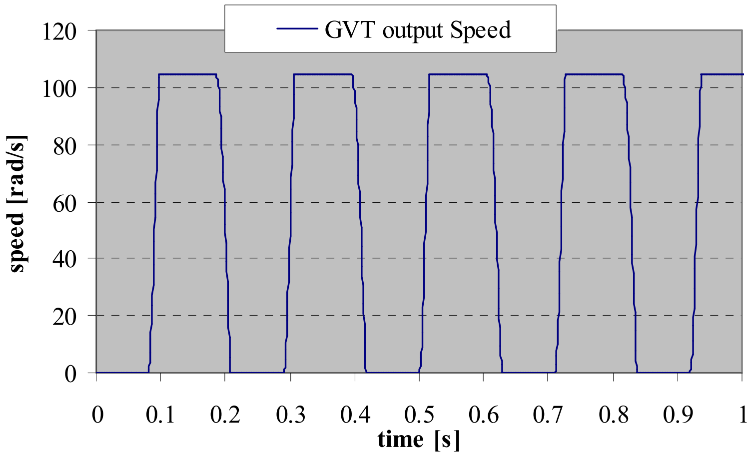

Figure 7 Output speed of a GyroTorque^{TM} unit (before the clutch system)

The GyroTorque^{TM} unit experiences these quite severe reversals of speed and loading associated with the oscillatory nature of the power transmission. However the system only experiences such well defined cyclic loading and the impact of wind turbulence (minor modulations on amplitude and frequency of the characteristic waveforms of Figures 6 and 8) is almost negligible.

GyroTorque^{TM} system loads

As may be expected, all the critical loads in a GyroTorque^{TM} unit are associated with the comparatively low speed input system. The cam imposes a high thrust on the input slide mechanism. This thrust (Figure 8) must be reacted at a bearing which also allows relative rotation of the main frame at output speed (1000 rpm).

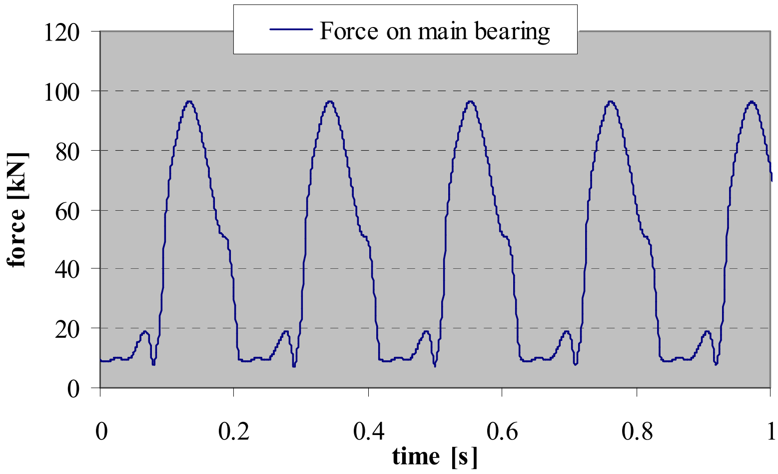

Figure 8 Force on main thrust bearing in GyroTorque^{TM} unit

There is no basic feasibility problem with the thrust bearing but the design must avoid significant friction losses. The GyroTorque^{TM} system has no significant losses except bearing losses and, at full load operation, can afford losses amounting to about 6% of rated power to remain competitive with a gearbox and electronic power converter. There would appear to be good prospects of keeping within this budget for losses although a more detailed level of engineering will be needed to confirm this.

Output power quality

The output power quality is well illustrated in Figure 4. A key to satisfactory performance is to have the irregularities in the mechanical output power at a high enough frequency to be satisfactorily filtered by the synchronous generator.

Insensitivity to turbulence

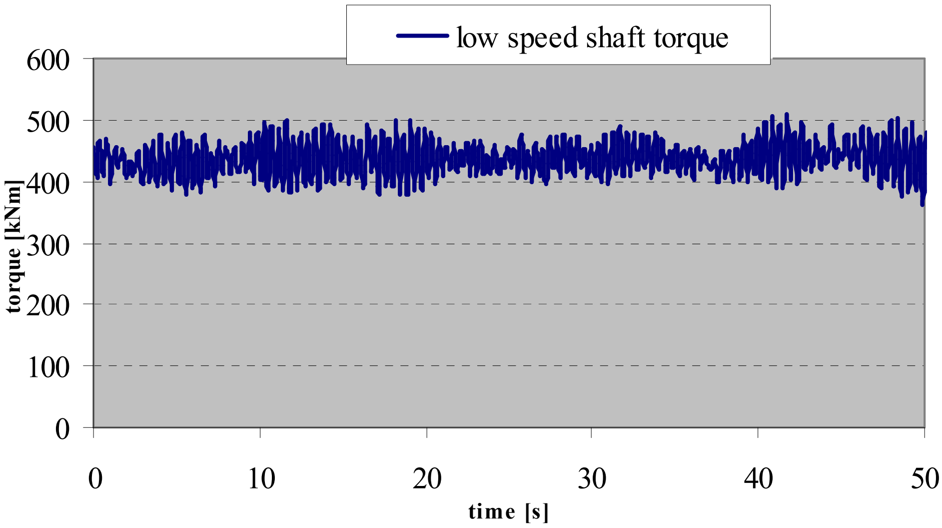

Figure 9 Typical history of low speed shaft torque of a 1 MW wind turbine

A typical history for the low speed shaft torque of a 1 MW rated wind turbine is illustrated in Figure 9. The data is based on a run of the Garrad Hassan design software, *Bladed for Windows,* with a turbulent wind file (mean wind speed 18 m/s, turbulence intensity 10%) as input to a representation of a conventional 1 MW wind turbine system. On account of the high inertia of the wind turbine rotor, the associated shaft speed (Figure 10) shows much reduced variability.

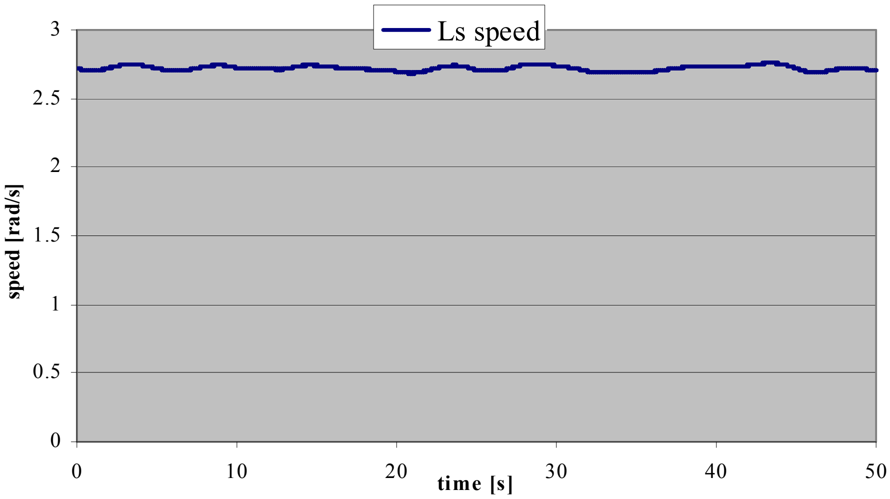

Figure 10 Typical history of low speed shaft speed for a variable speed 1 MW wind turbine

As has been mentioned, the gyros of the GyroTorque^{TM} system in effect map torque to speed and the (synchronous) generator of the GyroTorque^{TM} system experiences torque variations (Figure 11) similar to the wind turbine rotor speed variations (Figure 10).

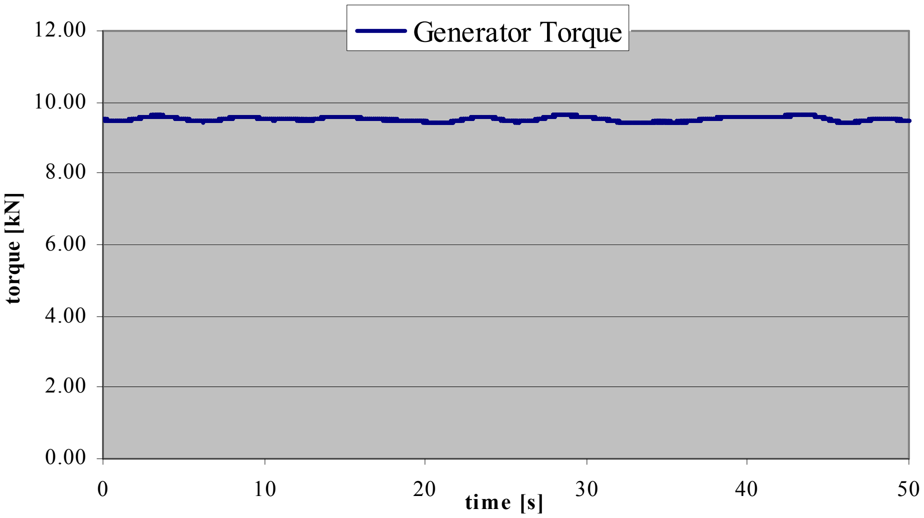

Figure 11 Generator torque from GyroTorque^{TM} system at 1 MW rated output

Thus while the GyroTorque^{TM} must be designed for quite demanding cyclic load variations (Figures 6 and 8, for example) the duty is insensitive to turbulence. Unlike a gearbox, where the exact duty (in terms of extreme loads reflected in time at level data and the related service factor) has sometimes been contentious, site wind conditions, in particular severe wind turbulence, will have little impact on the design loading of a GyroTorque^{TM} system.

CONTROL

A previous study considered an arrangement with three gyros without the present input cam arrangement. It was concluded that control by variation of gyro speed was not feasible. The rotational inertia stored in the gyros was commensurate with the energy stored in the main wind turbine rotor. Thus the torques involved in changing gyro speed would have been extremely large and the response extremely slow.

In a system with 17 cam lobes, there is a reduction in time constant of the gyros compared to the previous arrangement of more than two orders of magnitude. This would allow the option of control by gyro speed variation to be considered.

In Figure 12, a theoretical curve is presented of the steady state variation in gyro speed required to regulate power according to the demands of ideal torque tracking, maintaining a maximum rotor performance co-efficient in operation below rated wind speed. In the theoretical calculation a simple approach is adopted where the geometric and inertial effects are assumed to be invariant and the speed is scaled to reduce the gyro angular momentum to match the required power level.

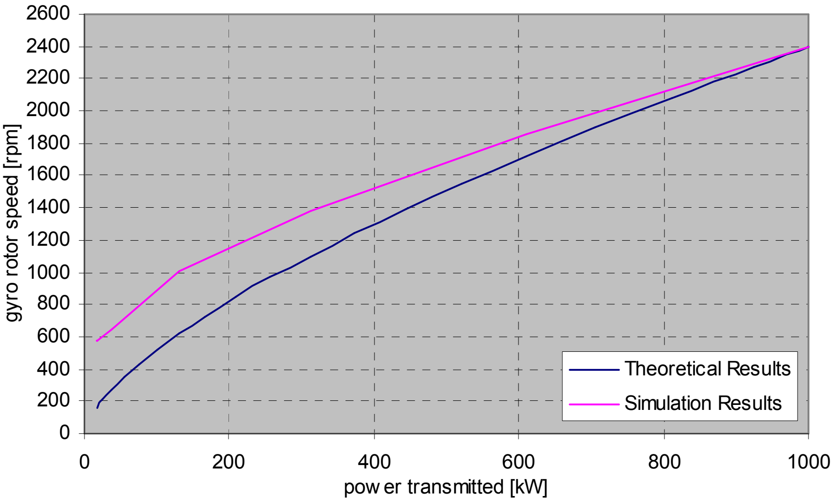

Figure 12 Rotor speed control

However the simulation results show that, probably due to the time involved in acceleration and deceleration of the main frame inertia, the system becomes less efficient at low gyro speeds. This is helpful as it reduces the variation in gyro speed required to regulate power.

Some exploration was conducted of limits on the effect of input gearing which is associated with the number of lobes on the cam. In varying the number of cam lobes, simulation results revealed an additional important factor in the dynamic response of each gyro torque unit. There is a force "spike" on the actuating linkage at the start of the power stroke. For example, in Figure 13, time series of the magnitude of the resultant force[1] on the Gyro Torque^{TM} input linkage show an initial spike during the acceleration of the main frame before it reaches synchronous speed. At synchronous speed this bearing thrust loading peaks (excepting circumstances when the peak of the preceding spike is higher) and remains relatively high. Power is transmitted to the output system and generator during the part of the cycle corresponding to the thrust force being at a high level.

The spike becomes relatively worse with reduction in gyro speed (implying higher gyro inertia for a given power level and with increase of input frequency). In the present configuration with gyros running around 2400 rpm, the spike restricts the ratio of Gyro Torque^{TM} input frequency to turbine rotor frequency to a factor of about 17. As shown in Figure 13, with a 23 lobe cam, the spike would triple the design load on the main thrust bearing relative to the base level (around 150 kN for the 23 lobe, 2400 rpm case) essential for power transmission.

More extensive analysis shows that the spike on the thrust force is fundamentally related to the total rotational inertia about the sub-frame axis including linkage elements.

The important clarifications of system dynamics that now emerge are:

- As has always been understood, the main frame must be accelerated from rest to synchronous speed and decelerated to rest in each input cycle of the Gyro Torque^{TM} system. The total inertia of the Gyro Torque^{TM} components that rotate about the main frame axis will therefore affect the time that can be spent in useful power production in each cycle and tend to limit input cycle frequency.

- As evidenced from the present work, the total rotational inertia about the sub frame axis, including linkage elements, considered in the context of the actuating linkage geometry, will affect reaction forces in linkage system. These forces may become design drivers tending to limit cycle input frequency.

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[1] This force is the maximum resultant force on either of the bearings that connect to the link arm which applies a moment about the sub frame.

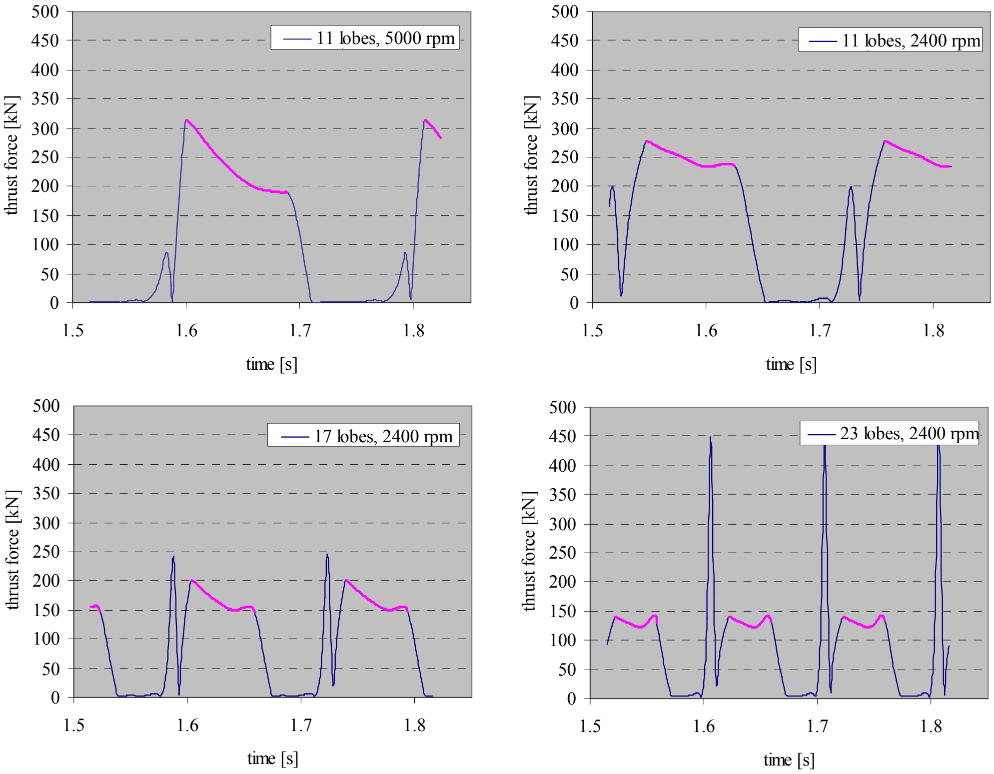

Figure 13 Effects of system configuration on the input linkage thrust force

Detailed dynamic modelling and further optimisation of input linkage design are required to establish at what point sub-frame inertial reaction forces may restrict input frequency. At present about 17 cam lobes may be feasible but it is highly desirable to have 23 as the extra gearing will generally reduce system mass and cost.

SYSTEM MASS

Power transmitted by the system is directly proportional to the net angular momentum in all the gyros. Clearly this is conserved if gyro speed is increased and gyro inertia appropriately decreased. A high gyro speed has two main penalties:

- Increased bearing loss varying as the square of speed.
- Increased difficulty in finding bearing solutions as there is no significant alleviation of the design bearing loads with speed increase.

Figure 14 shows the variation of bearing loss with gyro speed for rolling element bearings that are considered sufficient for the application.

Although the mass and rotational inertia of the gyro will always decrease with increasing gyro speed, there is a definite minimum in the mass of the gyro system comprising gyro motor, stator, gyro rotor and bearings. This minimum exists because, as speed is increased, friction losses in the bearings increase. Thus, on account of the friction losses, although the mass of the gyro rotor is decreasing with increasing gyro speed, the rating of the motor and its total size and weight increase.

Figure 15 shows that, in the present system, minimum system mass is obtained at a gyro speed around 2500 rpm. It is interesting to note that when design loads from the simulation are taken into account, this speed of 2500 rpm is around the limit in terms of feasibility of using standard rolling element bearings. Hydrostatic bearings may be considered and would facilitate very low bearing losses but not necessarily achieve overall benefit as power must be provided to pressurise the working fluid.

Working outward from the gyros as the fundamental units in the system associated with the required power rating, loads on frames and bearings were determined allowing estimation of stress on the structural components and estimates of size and mass of all system components. It was noted that the output gears were relatively massive considering the high operating speed and low torque involved. The size of gears is not related to loading but instead is determined by the spatial arrangement and the size of the gyro torque units. A gear of large diameter tends also to be of increased thickness in order to provide adequate out-of-plane bending stiffness in the gear. These effects jointly contribute to the gears being unduly massive.

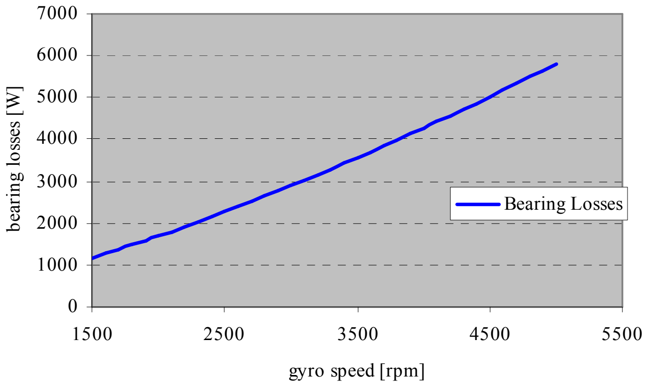

Figure 14 Gyro bearing losses

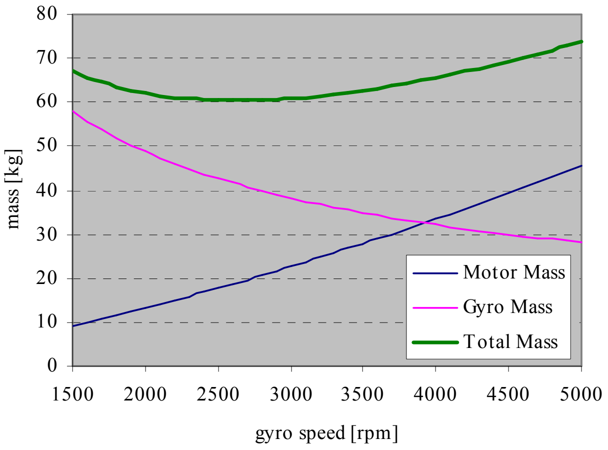

Figure 15 Effects of gyro speed on gyro system mass

An alternative gear arrangement was considered as in Figure 16. The new system has 10 smaller gears, six outer gears connected to the gyro torque output shafts, three intermediate idling gears and a central output gear on the generator shaft without any axial stepping required to maintain compatible rotations. Note that this is not the case in the basic arrangement (left-hand diagram in Figure 16) where the alternate gyro torque output gears must be stepped axially to avoid clashing.

Provisional estimates suggest about 25% mass reduction of the 10 gear system compared to the 7 gear system. These results are indicative only and it is expected that these and other options would be evaluated more fully in a detailed design.

Considering all components in the system, gyro sub-frames, main frames, link arm sets, output gear sets, main cam, energy pairs, roller pistons, housings, and including electric motors, the gyro torque system mass was estimated as about 5.5 tonnes for a system with 11 cam lobes and 6 gyros, decreasing to under 4.5 tonnes for a system with 23 lobes and 6 gyros.

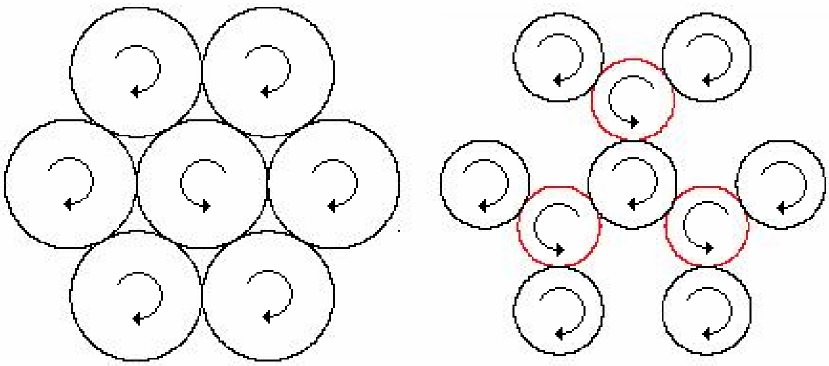

Figure 16 (left) 7 gear axially staggered arrangement and (right) 10 gear with 3 idler arrangement

At 1 MW rating it was estimated that in a conventional system, the gearbox mass would be around 6.5 tonnes and there would be mass associated with the variable speed drive system above 0.5 tonne. Thus the gyro torque system is displacing conventional components with a total mass around 7 tonne and this preliminary evaluation of system mass is quite favourable for the Gyro Torque^{TM} concept.

COST

GH experience of gearboxes in wind turbine applications suggest that they can be costed at around £8,000 per tonne with a direct relationship between torque, mass and cost. The overall impression of the Gyro Torque^{TM} system is that it will have some components with more parts within the Gyro Torque^{TM} units in terms of number of bearings which may be more expensive per unit mass than a gearbox and may be costed at around say £12,000 per tonne. On the other hand the complex components do not account for a very large proportion of total mass and the total mass may also be less than for an equivalent gearbox and variable speed drive.

Component	11 Lobes, 6 Gyros Mass [kg]	11 Lobes, 6 Gyros Cost [£]	23 Lobes, 6 Gyros Mass [kg]	23 Lobes, 6 Gyros Cost [£]
6 gyros including bearings and motor	580.46	6,966	434.51	5,214
6 sub frame	113.47	1,362	101.98	1,224
6 main frame	110.36	1,324	101.50	1,218
6 link-arm sets	15.34	184	13.16	158
Output gear set	1821.02	18,210	1497.49	14,975
Cam	888.63	3,555	680.80	2,723
6 roller pistons	7.05	85	6.91	83
Gyro Torque^{TM} system housing	1964.18	7,857	1609.30	6,437
System total	**5500.51**	**39,543**	**4445.62**	**32,032**

Table 1 Gyro Torque^{TM} system mass and cost summary

The costs in Table 1 are based on estimated cost per tonne of Gyro Torque^{TM} components at £12,000, housing structure and cam at £4,000, gears at £10,000.

A cost base for conventional equipment at megawatt scale and comparison with an equivalent Gyro Torque^{TM} system was established in previous work . That work is reproduced here and then extended and evaluated further in the context of the present Gyro Torque^{TM} system cost estimates.

Some significant distinction must be made between the actual manufacturing cost of a component i.e. the cost to the component supplier and the cost to the wind turbine manufacturer of the component which is in effect the price of the component. The wind turbine manufacturer will then sell on the complete wind turbine system with a further mark up constituting the price of the wind turbine.

The data of Table 1 is based on a price split of bought-in components typical of large commercial (land based) wind turbines employing active pitch control in combination with an electrical variable speed system. It can be seen that the combined price of gearbox and variable speed system amounts to almost 27% of total turbine price. In the present context price includes delivery and installation.

Conventional wind turbine system			
	Price fraction	**Price [£/kW]**	**Price for 1 MW [£]**
Blades	0.212	80.56	80560
Hub	0.028	10.64	10640
Gearbox	**0.162**	**61.56**	**61560**
Rotor bearings	0.050	19.00	19000
Generator	0.106	40.28	40280
Nacelle	0.088	33.44	33440
Yaw	0.027	10.26	10260
Variable speed system	**0.106**	**40.28**	**40280**
Pitch system	0.088	33.44	33440
Tower	0.133	50.54	50540
Total turbine	**1.000**	**380.0**	**380000**

Table 2 Price make up of a conventional wind turbine system

System with Gyro Torque^{TM} transmission			
	Price fraction	**Price [£/kW]**	**Price for 1 MW [£]**
Blades	0.212	80.56	80560
Hub	0.028	10.64	10640
Gyro Torque^{TM}	**0.268**	**101.84**	**101840**
Rotor bearings	0.050	19.00	19000
Generator	0.106	40.28	40280
Nacelle	0.088	33.44	33440
Yaw	0.027	10.26	10260
Pitch system	0.088	33.44	33440
Tower	0.133	50.54	50540
Total turbine	**1.000**	**380.000**	**380000**

Table 3 Gyro Torque^{TM} cost allowance for parity

Table 3 reproduces Table 2 with the cost fraction and costs of gearbox and variable speed drive combined as a cost limit for viability of the Gyro Torque^{TM} system.

Referring to Table 1 where the cost of the Gyro Torque^{TM} system is estimated in the region of £30,000 to £40,000, this is very well within the limits of around £100,000 for parity with a conventional system. The clutch designs are special, based on innovative ideas of the Gyro Torque^{TM} unit inventor, M Jegatheeson and as yet, mass and cost has not been estimated. A few other sundry component costs are missing and while that in itself is unlikely to consume much of the margin between £40,000 and £100,000, the main factor to consider is that the concept engineering is incomplete, little detailed engineering has been done and the costs estimates are consequently quite approximate.

Thus while it would be too optimistic to conclude that the Gyro Torque^{TM} system is definitely cost effective and advantageous compared to the conventional power train solutions, there is every encouragement to continue the work to a further level of investigation finalising concept design and preparing for the engineering of a test prototype.

CONCLUSIONS

The Gyro Torque^{TM} concept is being developed by GEL (Gyro Energy Limited) for the renewable energy market and by HHL (Hybrid Holdings Limited, USA Inc.) for automotive and other applications.

- The Gyro Torque^{TM} system for wind turbines provides a transmission system with complete insensitivity to wind turbulence and wide range variable speed without power electronics.
- The system concept with a synchronous generator provides good output power as confirmed by simulation.
- Bearing systems and control mechanisms are subject to further detailed engineering but the simulation results suggest that the load specifications can be met with commercially available components.
- Preliminary mass and cost estimates suggest that the Gyro Torque^{TM} system could be advantageous compared to the conventional geared transmission with electrical variable speed drive.

REFERENCES

1. M. Jegatheeson. “Gyroscopic Variable Transmission (GVT) a New Invention.” Proc. 2002 Global Windpower Conference, Paris, April 2002.