THE ULTRA LOW NOISE GEARBOX

BY

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ABSTRACT

As a result of intensive research over the last 10 years it is now possible to achieve very significant reductions in gear noise. This article outlines the development of the calculation procedures for low noise gears, and describes the research carried out in the Royal Navy's 8 MW Marine Gear Research Rig. The results of extensive noise and vibration studies on single and double helical gears are presented, as well as measurements on 8 MW gears design optimised for low noise. It is shown that optimizing gear design using these new design techniques can reduce gear noise significantly.

Introduction

Warships must be quiet at patrol speed to avoid detection by sonar. Particularly significant is the noise resulting from rotating machinery that generates distinct 'tones' as a result of excitation at shaft rotational frequency and its harmonics. In gearing, noise and vibration is generated at both gear rotational frequency due to out of balance, eccentricity and swash and at Tooth Contact Frequency (TCF) and its harmonics.

The design rules for naval gearing have, in the past, considered only gear strength, which is the fatigue strength against pitting and tooth breakage. Until very recently no techniques were available for optimizing the design of gears for low noise and vibration. The significant improvements in gear noise which were nevertheless achieved, were primarily the result of improvements in gear accuracy, initially due to a change from gear shaving to grinding and latterly the result of significant improvements in grinding accuracy. Although gear accuracy will continue to improve, the benefit to gear noise reduction is unlikely to increase.

In the past, the design of gears for low noise has been dominated by the search for even higher accuracy, with the design of involute and lead correction based on empirical rules and, above all, experience and intuition. In effect, gears could not be 'designed' for low noise.

This article shows that this situation has changed radically. It is now possible to design the detailed geometry of gears to achieve not only high strength but also low noise.

Background

Work in the 60's by researchers such as MUNRO[1] showed clearly that gear noise is related to the very small change of velocity ratio which occurs between mating gears as individual teeth come into and go out of mesh. At the time, the analytical techniques were not available to calculate accurately these changes of velocity ratio or Transmission Error (TE).

The development of powerful computers has introduced techniques such as Finite Element Analysis (FEA), which has been developed into a comprehensive tool for analysing complex structures. Its application to the analysis of the elastic deflection of meshing gear teeth has not been easy, primarily due to the complex three dimensional model which is required to accurately represent mating gears.

Based on earlier work by A.J. PENNELL and his research workers at the University of Newcastle upon Tyne[2, 3] the Design Unit has developed a comprehensive gear analysis procedure (DU-GATES – **G**ear **A**nalysis for **TE** and **S**tress) for the design of high performance, low noise gearing.

Since 1992 the RN has supported this research into design techniques for low noise gears, and funded fundamental research into gear noise and vibration. To meet the RN's special requirements, DU-GATES was conceived at the outset as a design tool for optimizing the detailed geometry of high contact ratio marine propulsion gears.[4, 5] The validity of the concept and of the calculation procedure has been validated exhaustively against measured data from the RN's Marine Gear Research Rig (MGRR).[6]

Gear noise and vibration

Gear noise and vibration is not caused by gear teeth 'clashing' as they engage, but is excited by inertia forces resulting from small changes of velocity ratio between pinion and wheel. The change of velocity ratio, expressed as the kinematic error in relative displacement between pinion and wheel at the pitch diameter is usually termed TE. It is the result of both geometric error and variable elastic mesh deflection.

When two perfect (i.e. error free) involute gears are meshed together under no-load conditions, rotation of the driving gear results in uniform rotation of the driven gear in proportion to the number of gear teeth in the two gears. When two real gears* are meshed together at zero load, with particular shaft misalignment, and the rotation is traced against time (or phase of mesh), it is observed that there is a deviation from the uniform motion defined by the simple gear ratio. This is due to geometric deviations, which are termed the 'unloaded' or 'geometric TE'.

When torque is then applied to the geometrically perfect gears mentioned above, there is elastic deflection of the gear teeth at the point of contact due to bending, sheer and contact (Hertzian) deflection. The total elastic deflection of pinion and wheel teeth, will vary depending on the local stiffness of pinion and wheel teeth, and how this sums along the contact lines. When load is then applied, the relative rotation of pinion to wheel will vary by a small amount over time (or angle of rotation) depending on the variability of mesh stiffness. This is the 'elastic' TE.

The sum of geometric TE and elastic TE is the total of kinematic TE. Under dynamic conditions, that is when gear inertia is considered, the acceleration and deceleration of pinion and wheel in response to Kinematic Excitation (KE) results in dynamic forces in the gear mesh. In wide faced gears with relatively small bearing span, a further source of excitation is the fluctuation of bearing loads with phase of mesh due to variation in axial position of the resultant of the distributed load along the contact lines between mating gear teeth. This results in quasi-static variation of bearing load at tooth contact frequency, which also excites gearbody and shaft and thus gearcase vibration.

The excitation responsible for gear noise and vibration is thus T.E. and quasi-static bearing load variation. The level of noise and vibration that results from this excitation depends on the inertia and the stiffness of:

- Gears.
- Shafts.
- Bearings.

and also the:

- Dynamic characteristics of the gearcase.
- Stiffness and dynamic behaviour of the mounting between gearcase and hull.

This is shown schematically in (FIG.1). Until the dynamic response to KE of the total 'system' from gear teeth to the hull can be fully modelled it is impossible to predict the level of vibration. However, in a linear system, the response is proportional to excitation, and thus reducing excitation, i.e. TE, and quasi-static bearing force variation, will reduce gear noise and vibration.

* That is gears with pitch, profile and lead deviation and mounting errors within specified tolerances

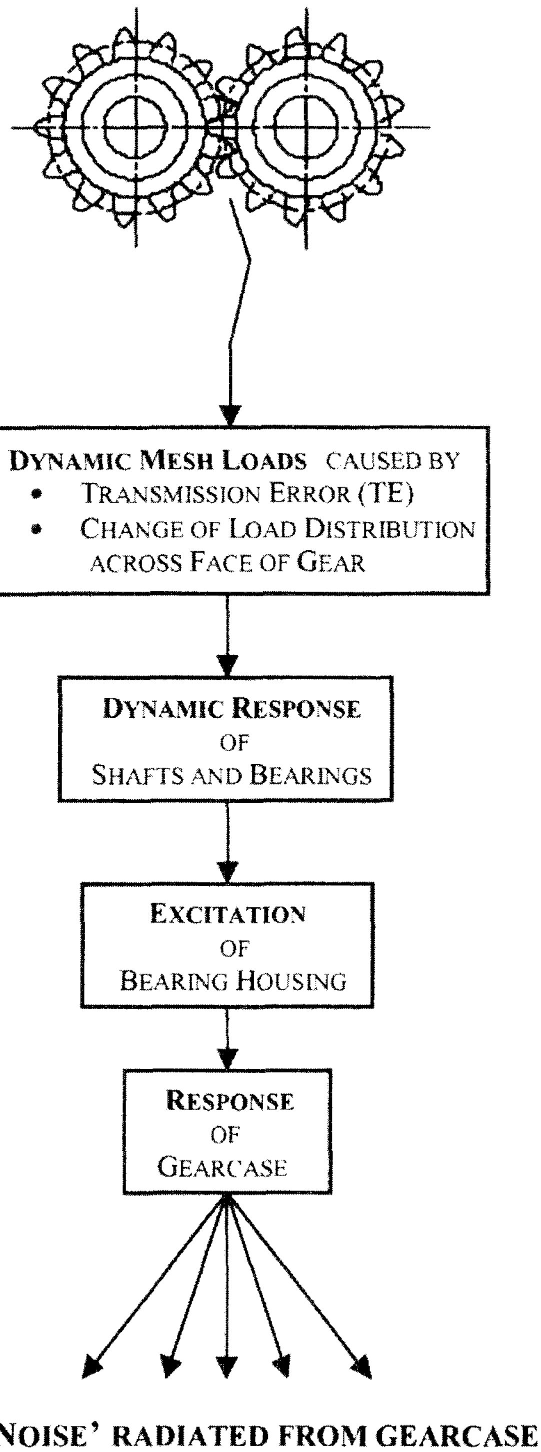

FIG.1. – TRANSMISSION OF GEAR NOISE AND VIBRATION

The MGRR

To study the noise and vibration of typical naval gearing, the RN has funded the setting-up of a MGRR at the University of Newcastle upon Tyne. Initially rated at 4 MW, the MGRR is now operated at up to 8 MW. The rig is designed to test typical primary and secondary mesh naval gears, that is helical gears with a face width to diameter ratio of about one, at pinion speeds up to 6,000 rev/min. The size of the test rig was governed by:

- Cost.

- The electric power available in the Gear Laboratory.
- The requirement to measure pinions and wheels in the National Gear Metrology Laboratory at the University of Newcastle.

These constraints resulted in the following overall specification:

Gear centres:	400 mm
Gear ratio:	3:1
Wheel ref diameter:	600 mm
Pinion ref diameter:	200 mm
Face width:	200 mm
Maximum speed:	6,000 rev/min
Maximum torque (pinion):	15,000 Nm
Maximum power:	8 MW

The large power to be transmitted by the gearbox can only be generated economically in a back to back test rig configuration. The general arrangement, as shown in (FIG.2), is conventional, with a test gearbox and a similar slave gearbox joined by torsionally compliant shafts and axially compliant membrane couplings. Both gearcases are very massive, to reduce gearcase resonance and minimize gearcase vibration amplitudes. The test gearcase is 'floated' on self-levelling pneumatic vibration isolators (Barry-Mounts) which give a gearcase natural frequency of 2.4 Hz.

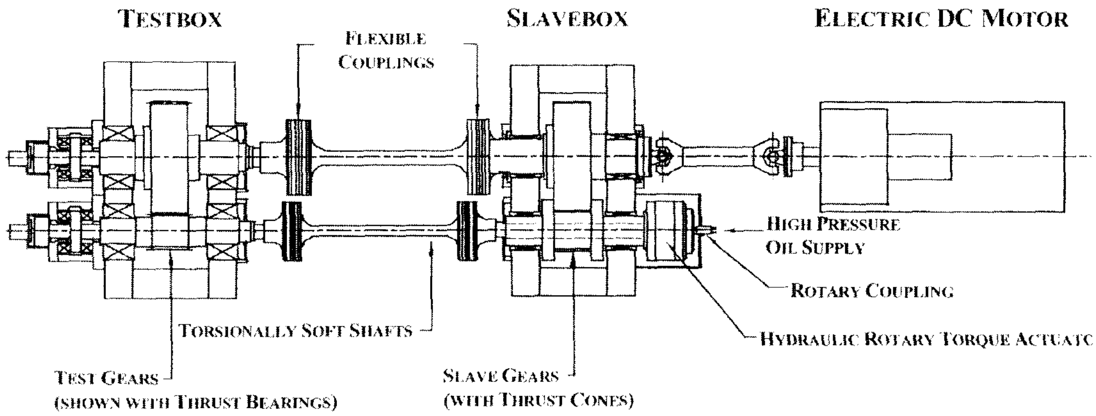

FIG.2 – GENERAL ARRANGEMENT OF MARINE GEARING RESEARCH RIG (PART SECTION)

Transmitted torque is controlled with a hydrostatic rotary actuator developed for this rig, which is described elsewhere.[7] The body of this is mounted on the hollow slave pinion, and the rotor is coupled to the pinion line by a quill shaft passing through the pinion. The rotary actuator has no mechanical seals, so has very low hysteresis making torque control by closed loop servo-hydraulics simple.

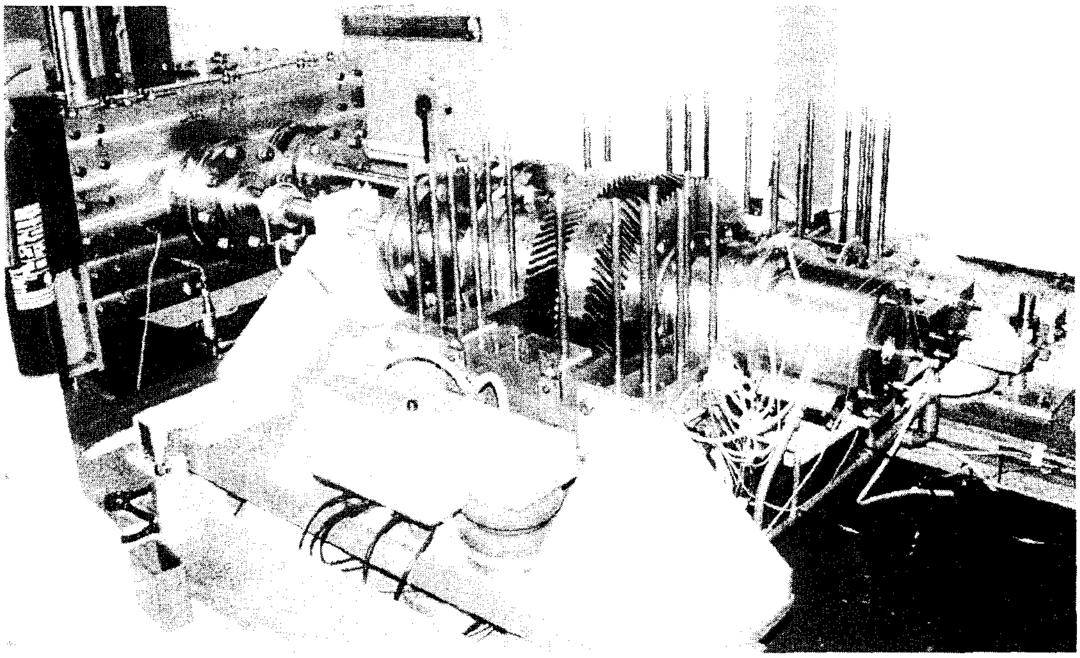

FIG.3 – THE MARINE GEARING RESEARCH RIG

The rig is driven by a 200 kW variable speed DC motor, which overcomes the losses in bearings and gear mesh. The motor is sized to drive the rig at 8 MW, allowing for 1.2% total loss in each gearbox. To avoid the significant losses incurred in high speed thrust bearings, the single helical slave gear pair is fitted with thrust cones. A general view of the test rig (FIG 3). It is known that gear noise and vibration in the engine room and as measured by sonar in the far field is wholly the result of dynamic forces transmitted through the bearings to the gearcase, and then to the surrounding structures such as raft and hull. A measurement of dynamic bearing forces thus gives an excellent, qualitative measure of the dynamic excitation generated by the meshing gears, and thus a direct measure of the 'quality' of the gears in relation to noise and vibration.

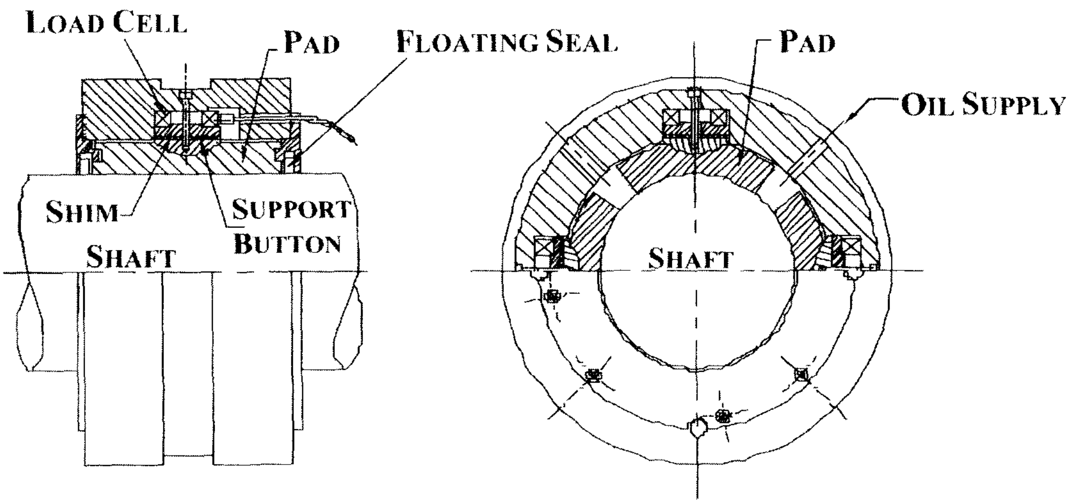

FIG.4 – PART SECTION OF TILTING PAD BEARING SHOWING LOAD DYNAMOMETER

To effect the measurement of dynamic bearing force with high discrimination and good frequency response, a special bearing dynamometer was developed (FIG 4). In place of the more usual cylindrical or lemon bore hydrodynamic bearings, tilting pad journal bearings are fitted. Each of the four pads is directly mounted onto a very stiff piezoelectric, charge mode compression load cell. Net dynamic bearing loads in the base tangent plane are computed by algebraic addition of all four load cell outputs. This arrangement is reliable, has a discrimination of 0.1N in 100,000 N and gives a frequency response of nearly 4 kHz. The use of tilting pad bearings on spherical seatings has an incidental advantage in providing self aligning bearings which will accommodate the large shaft misalignments at which tests were carried out in the research programme.

In addition to the bearing load dynamometer, the test gearbox is also fitted with a high performance optical system for measuring TE developed specifically for this application,[8] and a full set of probes and accelerometers to measure shaft and gearcase vibration. However, the major contribution that the work with this rig has made to the understanding of gear noise and vibration is the ability to measure dynamic bearing forces directly.

In the following sections, the influence of gear design, alignment, and torque on dynamic bearing forces, and hence gear noise, are described.

The effect of Helix Angle on dynamic bearing forces

It is well known that increasing the helix angle, and hence the number of teeth simultaneously in mesh, has a beneficial effect on gear noise. However, this effect had not previously been quantitatively measured in terms of dynamic excitation force. In an early programme of work, the influence of helix angle on gear noise, that is excitation at the bearings, was measured for gears of similar macro-geometry and micro-geometry with 8° and 20° helix angle. The details of the test gears are shown in Table 1.

TABLE 1 – *Gear details*

		High Helix gears		Low Helix gears	
		Pinion	*Wheel*	*Pinion*	*Wheel*
Number of teeth	z	29	87	33	99
Module	M_n mm	6		6	
Pressure angle	α_n°	20°		20°	
Helix angle	ß°	29.54°		8.11°	
Facewidth	b(mm)	200		200	
Hob addendum	h_{ao}(module)	1.4		1.4	
Face contact ratio	ε_β	5.23		1.5	
Transverse contact ratio	ε_α	1.42		1.74	

A very large number of tests were carried out, primarily in the form of run-up tests, where the test gearbox was run from low to high speed at constant torque, and a frequency analysis of dynamic bearing loads carried out. Since gears generate vibration at multiples of shaft rotation, the 'frequency analysis' is plotted for ease of interpretation not on a frequency but an 'order' base, where 1st order is pinion shaft rotation, second order the 2nd harmonic of shaft rotation etc.

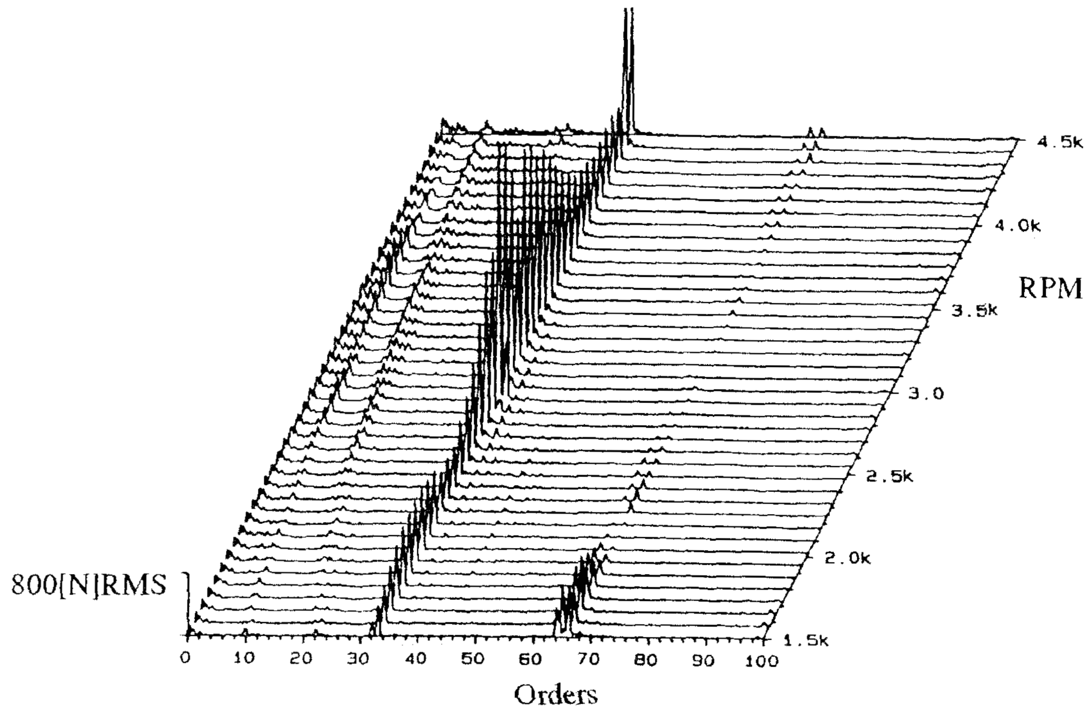

FIG.5A – DYNAMIC BEARING FORCE ORDER ANALYSIS, RUN UP TESTS – LOW HELIX GEAR

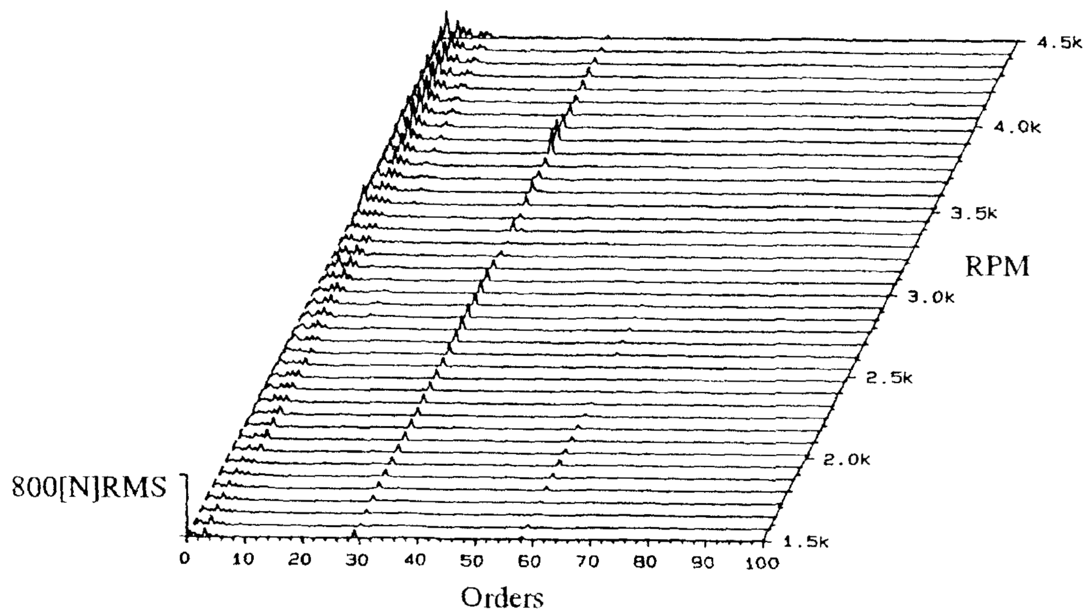

FIG. 5B – DYNAMIC BEARING FORCE ORDER ANALYSIS, RUN UP TESTS – HIGH HELIX GEAR

Typical results from these measurements are shown in (FIGS. 5A and 5b). These show the order analysis of dynamic bearing force, with force (N rms) plotted against order and rotational speed for a run-up test from 1,500 to 4,500 rpm. Figure 5A shows the dynamic bearing force characteristics for the low helix gear, Figure 5B those for the high helix gear, at a constant torque of 8,000 Nm and a nominal (no-load) zero mesh misalignment.

In both cases the measurements from the pinion aft bearing are used, with the gears running Astern (ahead and astern flanks were ground with different micro-geometry). Figure 5A shows significant bearing excitation predominantly at the 33rd order, that is, at Tooth Contact Order (TCO), with a peak dynamic bearing force of over 3,000 N at about 3,00 rpm. In the high helix angle gear, FIG.5B, excitation at TCO, 29th order, is very much less.

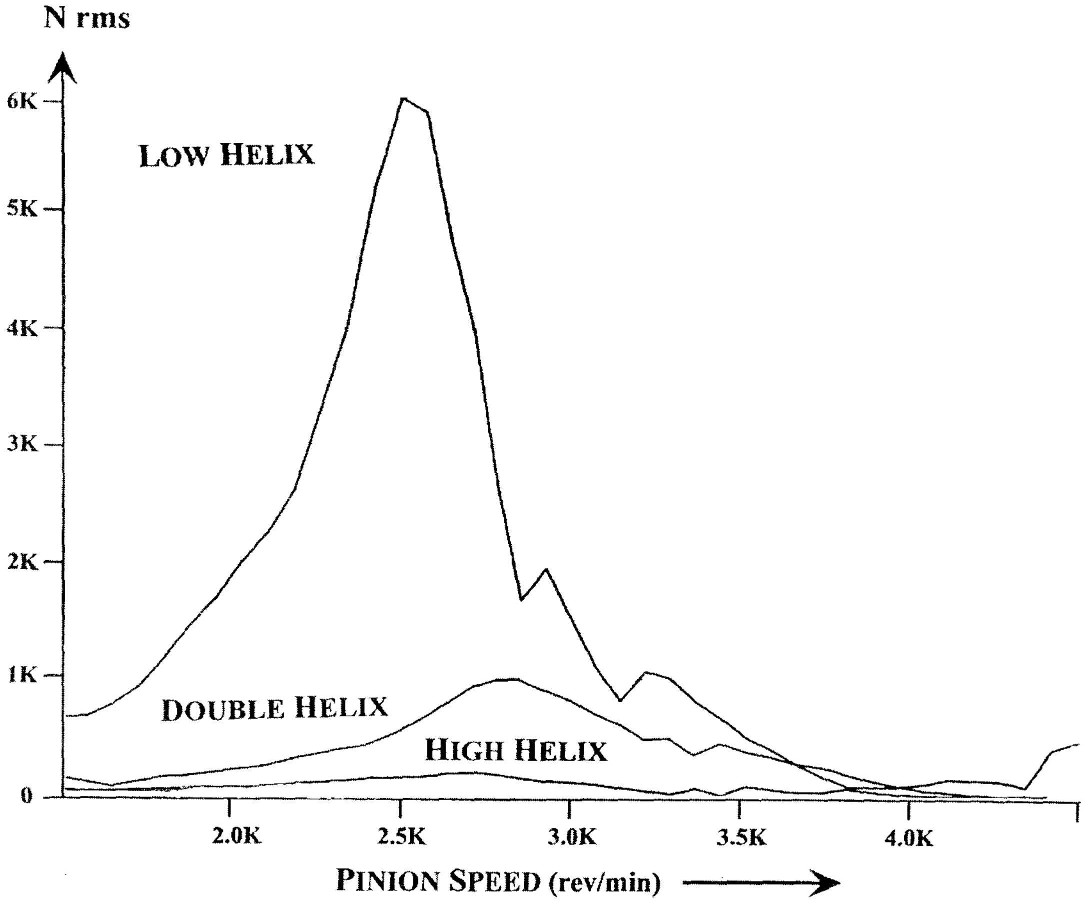

FIG.6 – DYNAMIC BEARING EXCITATION, SINGLE AND DOUBLE HELICAL GEARS

The comparison of the dynamic excitation of different gears can be extended to also consider the difference between single and double helical gears of otherwise identical geometry. (FIG.6) shows the dynamic bearing force (N rms) at TCO as a function of pinion speed. (i.e. Figure 6 can be considered as a 'slice' through the Order Analysis Waterfall plots shown in Figure 5 at TCO (33rd and 29th order respectively). In this case the dynamic bearing forces are plotted at a torque of 4,000 Nm, for the low and high helix angle single helical gears (8° and 30° helix angle) and for a double helical gear of 30° helix angle and identical gear geometry. For all three gears the mesh misalignment was zero. It is noted that the maximum dynamic bearing forces at this operating condition were as follows:

Single helical gear, high helix angle:	200 Nm
Double helical high helix angle:	1,000 N rms
Single helical low helix angle:	6,000 N rms.

The effect of misalignment

The effect of mesh misalignment, that is the axial misalignment between mating gears, has been considered in relation to gear stressing for many years. The effect of mesh misalignment on gear noise has not been as well understood, but can be very significant. (FIG.7) shows the influence of mesh misalignment on dynamic bearing loads at TCO for the low helix gear, ahead flank, measured at the pinion bearing.

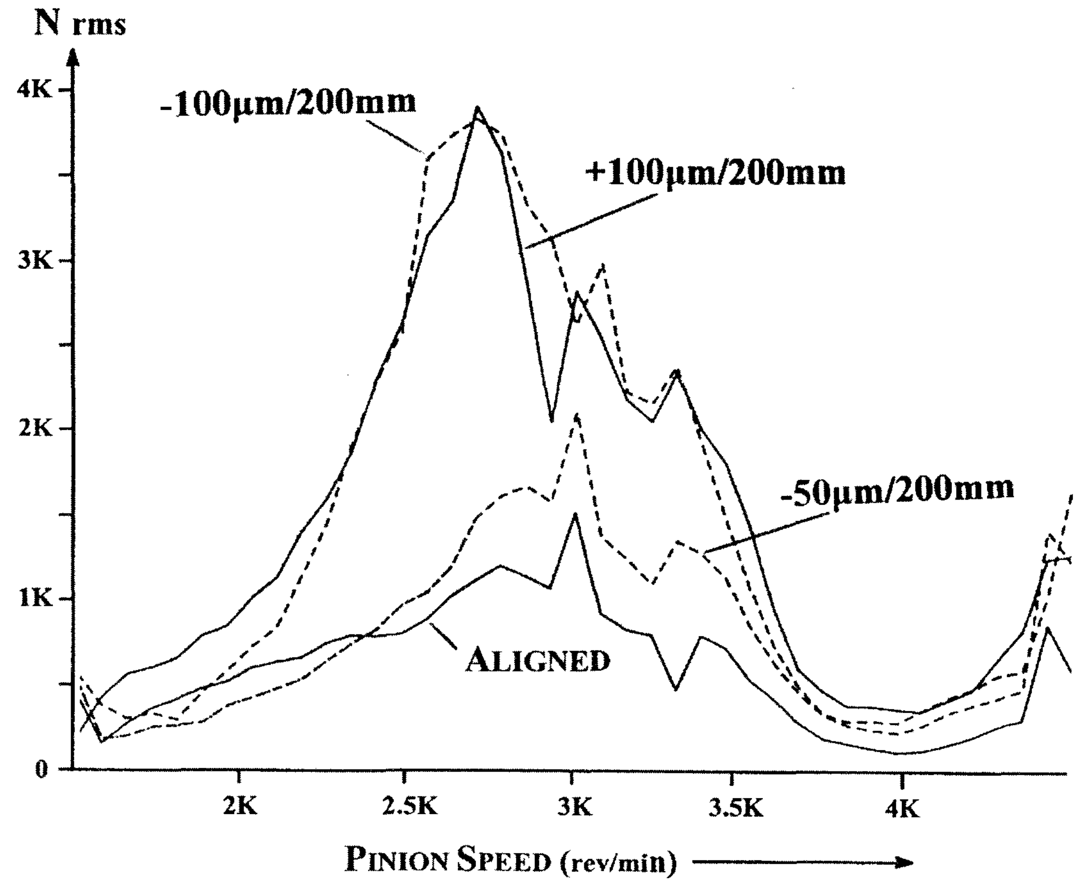

FIG.7 – TE VARIOUS MESH MISALIGNMENTS

From Figure 7 it is clearly seen that misalignment increases maximum dynamic bearing forces at TCO from 1,500 to nearly 4,000 N rms. In other words, significant mesh misalignments can double the noise signature of a particular gearbox.

Dynamic excitation at Pinion and Wheel Bearings

The dominant cause of gear noise and vibration is KE, i.e. the TE between pinion and wheel. The dynamic forces generated between the teeth of pinion and wheel excites movement of these, which in turn results in dynamic forces being excited at the bearings. The bearing forces are thus a function not only of TE but also of the mass of pinion and wheel and of their dynamic response. With a large difference in pinion and wheel mass, it would be expected that dynamic bearing forces would be very much greater at the pinion than the wheel bearings. The dynamic bearing forces at TCO at a pinion and a wheel bearing in the high and the low helix gear pair at identical torques of 4,000 Nm and zero misalignment are shown in (FIGS.8a and 8b).

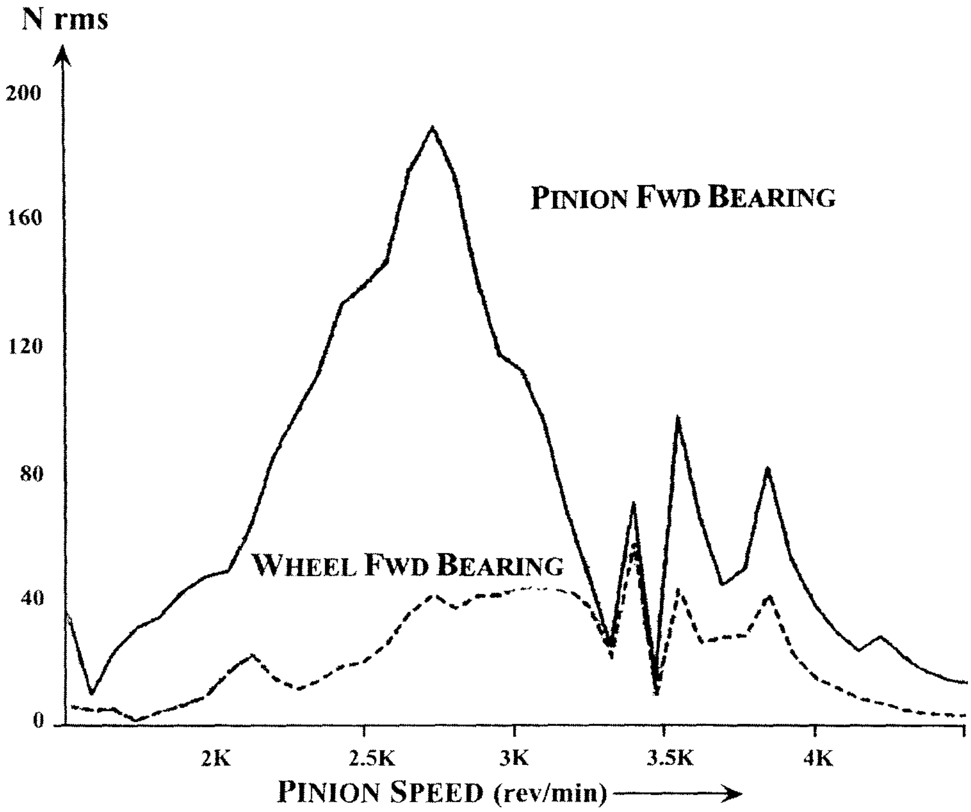

FIG.8A – DYNAMIC BEARING FORCES: HIGH HELIX GEARS (TCO)

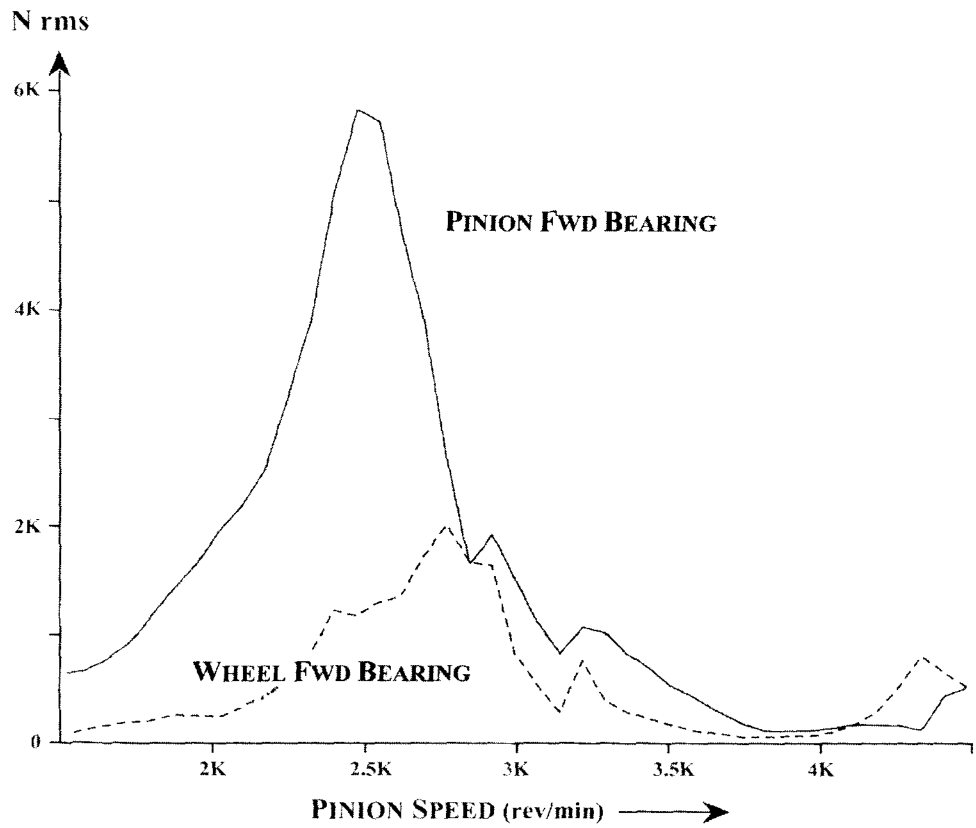

FIG.8B – DYNAMIC BEARING FORCES: LOW HELIX GEARS (TCO)

It is noted, as previously shown, that the peak dynamic bearing force is significantly greater for the low helix gear (6,000 N) than for the high helix gear (190 N). In each case, the peak dynamic force at the pinion bearing is some 3 to 4 times greater than at the wheel bearing. It should also be noted that the dynamic bearing forces are not proportional to speed to the power two, as could be expected, but pass through a number of clearly defined resonance's, particularly a dominant pinion flexural resonance at 2,400 – 2,700 rpm (40....45 Hz).

The effect of torque on dynamic bearing forces

Dynamic bearing forces are the result of

- KE e.g. TE
- Quasi-static bearing load variation.

The dynamic force at the bearings due to kinematic excitation depends primarily on inertia and system modal stiffness, and could thus be expected to be substantially independent of transmitted torque. However, the extent of contact between mating gears, and hence the mesh stiffness, varies with torque, as does the oil film thickness and hence the damping in the tooth contact. The resonant frequencies, and the 'amplification' of KE, can therefore be expected to vary with torque.

The 'quasi-static' bearing load variation due to the axial shift of the locus of the resultant of mesh forces could be expected to be more significantly affected by transmitted torque, since total bearing forces are directly proportional to torque. However, the shift of focus of the resultant is also variable with torque, so that there is no linear relationship between quasi-static bearing force and transmitted torque.

The typical complex relationship between dynamic bearing forces and torque is shown in (FIG.9). This shows dynamic bearing force at TCO for the high helix angle gear pair over a speed range from 1,500 to 4,500 rpm, for four pinion torque levels. (2,000, 4,000, 8,000 and 12,000 Nm). It is noted that:

- Maximum dynamic bearing forces occur at a torque of 2,000 Nm at first flexural resonance of the pinion shaft, at which speed (2,700 rpm) the lowest bearing forces occur at 4,000 Nm torque.
- Outside of resonance, and particularly above 3,000 rpm, not only are dynamic bearing forces very much smaller, but also the difference due to torque is much less.
- The resonance frequency is lowest at 2,000 Nm (2,600 rpm) and greatest at 12,000 Nm (2,850 rpm), demonstrating the effect of increased mesh stiffness at higher torques.

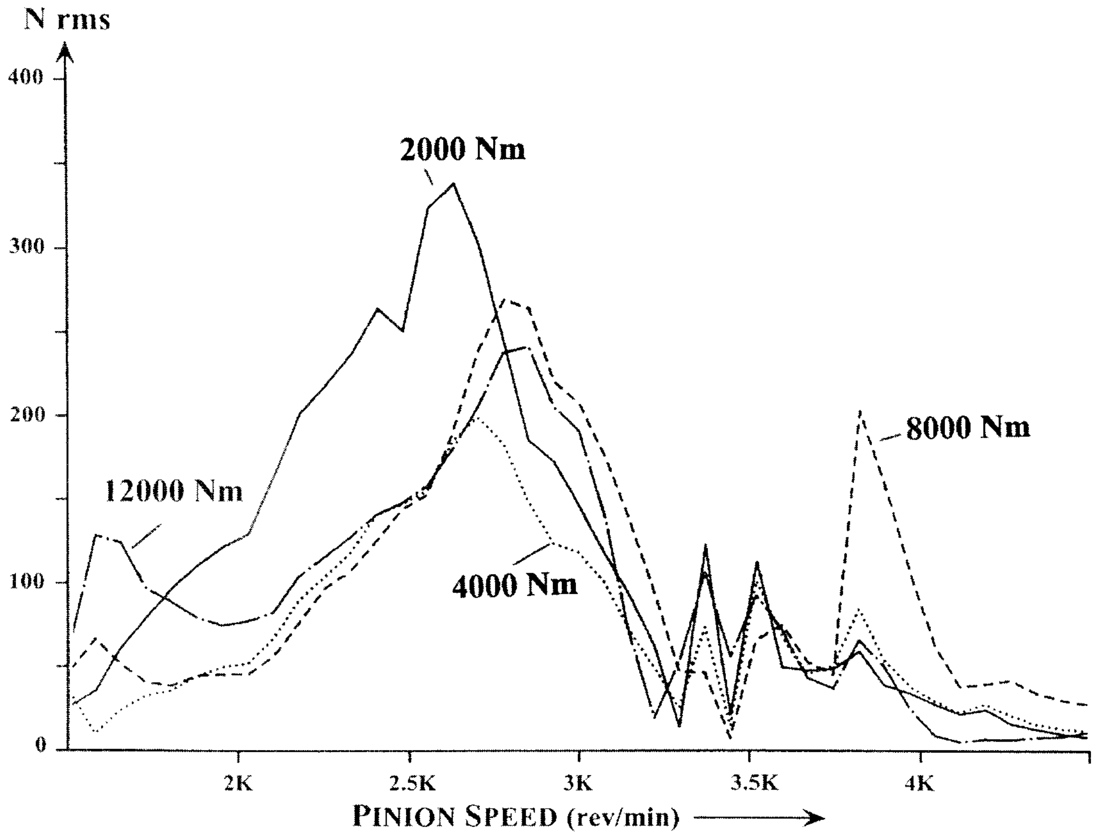

FIG.9 – DYNAMIC BEARING FORCE AT TCO FOR THE HIGH HELIX GEAR, ASTERN FLANK, ZERO MISALIGNMENT

Noise and vibration reduction

When designing the geometry of gears for low noise, it is theoretically possible to correct for variations in mesh stiffness over the course of a single tooth engagement by introducing geometric 'features', which give an equal and opposite TE. This technique can now be implemented as a result of improvements in the accuracy and versatility of gear grinding, and the ability to accurately calculate TE. based on a defined gear geometry.

Note:

> Perfect compensation of elastic TE by equal and opposite geometric TE is possible at only one specific torque. However, it is possible to devise gear geometries on this basis which have very low total TE over a significant range of torque). The use of FEA can be used to reduce noise and vibration in the design stage. For this purpose DU-GATES was designed and validated.

Elastic Mesh Analysis – DU-GATES

The elastic mesh analysis, DU-GATES, is based on modelling accurately the surface topography of pinion and wheel teeth, and defining their position relative to each other, i.e. the gears are 'meshed' at the correct centre distance and with the correct alignment. The full description of the gear geometry allows the calculation of the 'no-load gap' (in the base tangent plane) at a specific phase of mesh at all points that could potentially contact. The local and global deflection of the gears at a specific torque is solved by calculating the load distribution along all contact lines between the gears, setting up equations of compatibility between applied torque, gear displacement and contact forces along the contact lines.[4,5] Repeating this analysis for 32 or 64 increments over one base pitch of tooth engagement gives the change of total elastic mesh deflection, and hence the kinematic error or TE for that gear pair, at a specific torque and a specific alignment between pinion and wheel shafts.

In addition to calculating transmission error DU-GATES also calculates contact and bending stress. It can be used for the analysis of all parallel axis gears, i.e. spur, helical, double helical and single helical gears with thrust cones. The structure of the package is shown schematically in (FIG.10) and can be divided into two parts:

1. FEA and generation of compliance and stress coefficients: for a given macro-geometry this part of the analysis is run only once.
2. Tooth contact analysis. This is run repeatedly to optimize the micro-geometry to achieve minimum TE and gear stress.

The software runs under Windows NT or 98 with interactive data input and job control via a graphical user interface. The macro-geometry for the mating gears needs to be entered (e.g. number of teeth, helix angle, module, tooth depth, addendum modification, protuberance, face-width etc). A dedicated pre-processor automatically generates the Finite Element (FE) model of the gear, and an internal FE solver and a curve fitting procedure generate the compliance and stress coefficients.

The analytical contact analysis is then carried out for a specified micro-geometry (e.g. tip relief, root relief, profile crowning, lead correction, end relief, face crowning or any defined surface topography) at a number of phases of mesh (typically 32 or 64 increments in a base pitch) to determine kinematic transmission error, load distribution and root bending stress at given torque(s) and misalignments(s). The micro-geometry can then be varied to achieve minimum TE and stress for the full range of operating conditions.

Output from the calculation procedure – kinematic TE, bearing load, tooth load, contact and bending stress can be output either as a function of phase of mesh or, in the case of load and stress, as a distribution across the facewidth at a particular phase of mesh. The computation time, depending on the platform used, is typically about 3 hrs per gear pair for the FEA and curve fitting, and about 10 secs for the elastic analysis per phase increment.

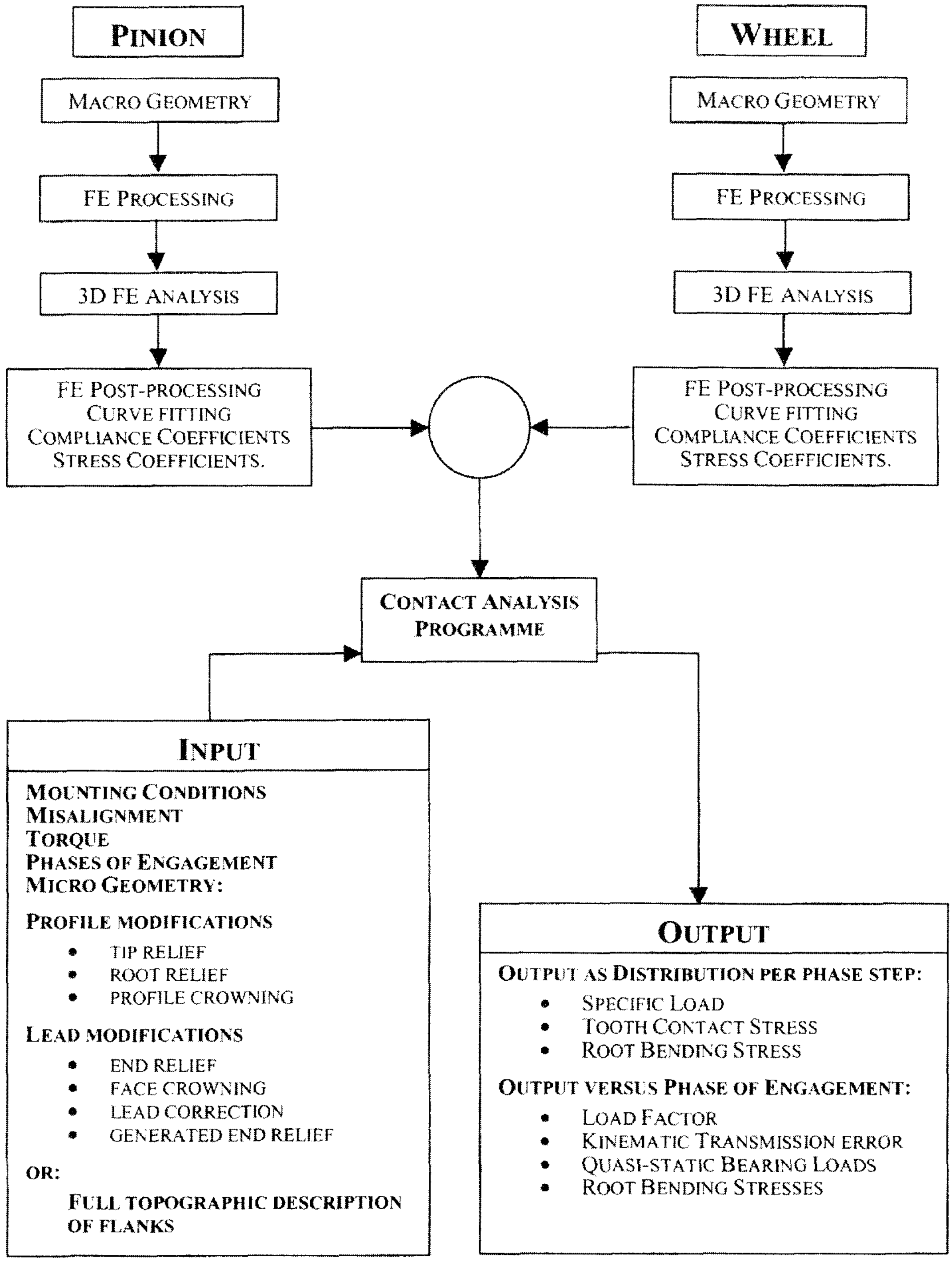

FIG.10 – BLOCK DIAGRAM OF DU-GATES SOFTWARE PACKAGE

Validation of DU-GATES

The test data for a large number of gears, over the torque range 500 to 15,000 Nm and for misalignments between -100 and +100 Nm has been analysed, and compared with the TE calculated for these test gears at those torques and misalignments using the FE based elastic mesh model DU-GATES. Since DU-GATES calculates only the KE, comparison with the measured data is only realistic at one speed. For the purpose of the comparison between calculated TE and measured dynamic bearing force at TCO, a common speed of 1,500 rpm was selected. (FIG.11) shows the relationship between measured dynamic bearing forces at the aft pinion bearing. (N rms at TCO) and calculated TE.

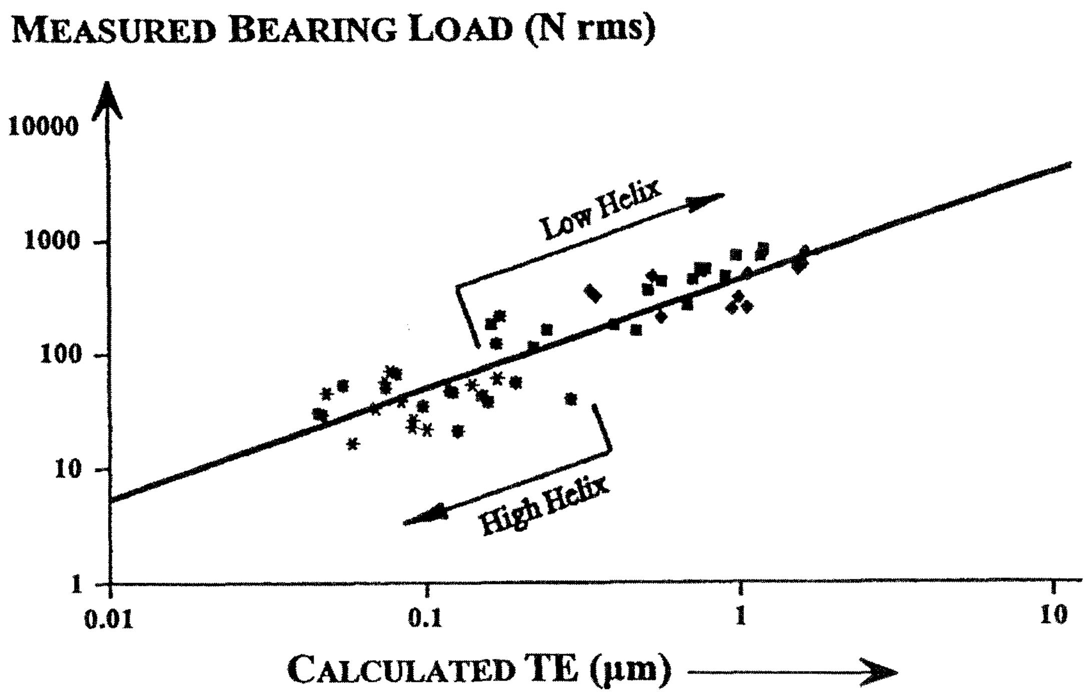

FIG.11 – MEASURED BEARING FORCE AT TCO (PINION AFT BEARING) V COMPUTED TE

The experimental data covers a range of dynamic bearing excitation at TCO from less than 20 N rms to over 1000 N rms, i.e. a range of 50:1, that is a range of 34 dB in equivalent noise and vibration levels. Although there is significant scatter in the relationship between measured bearing load and calculated TE, there is a clear linear relationship corresponding to an excitation of 1000 N rms (at TCO) per µm TE (at TCO). This relationship between dynamic bearing force and TE is only valid, of course, for the particular test gears used, and at this speed outside of resonance. At resonance, the dynamic bearing forces (at TCO) could be 5 to 10 x greater for a given TE.

The design of an 'Ultra-Quiet' Gear Pair

The good correlation between TE and dynamic bearing forces confirms that minimizing the quasi-static TE (using DU-GATES) should result in a low noise gear pair.

Such a gear has been designed for and tested on the MGRR. Both the macro-geometry and the micro-geometry were selected to achieve minimum TE, particularly at low torque where quiet operation is particularly important in a naval vessel. The 'ultra-quiet' gear was designed to have the same overall size as the gears previously tested, with the same torque and speed capability. Since the principle objective was to investigate how quiet a gear pair could be made, not to investigate the effect of mesh misalignment, the gears we designed for a maximum mesh misalignment capability of only ± 20 µm.

To simplify comparison of test data with the previous work, the same number of pinion and wheel teeth were presumed as in the previous investigations (29:87). Otherwise the macro-geometry, that is helix angle (and hence face contact ratio ε_β) and tooth depth (and hence transverse contact ratio ε_α) was optimized for low noise.

The gear geometry for the 'ultra-quiet' gears was developed in two stages. Firstly the macro-geometry was defined, and then the micro-geometry. The major parameters to be defined in the macro-geometry are the helix angle and tooth depth. (FIG.12) shows the relationship between T. and face contact ratio, ε_β, for four torque levels. For this analysis, standard depth teeth with ε_α = 1.4 are used with 10 µm face crowning and zero mesh misalignment. At low torque, where contact is not right across the facewidth, face contact ratio, e.g. helix angle, has little effect. At higher torques, where contact is occurring right across the facewidth, minimum T.E. occurs at or very close to integer face contact ratio, i.e. ε_β = 5 and 6. This effect is most significant at maximum torque, where the average mesh deflection is over 45 µm, and load distribution across the facewidth even with 10 µm crowning is fairly uniform. The TE at this torque is reduced from 0.85 µm to 0.45 µm. It should be noted that the peak to peak transmission error, 45 µm, is less than 1% of the total elastic mesh deflection.

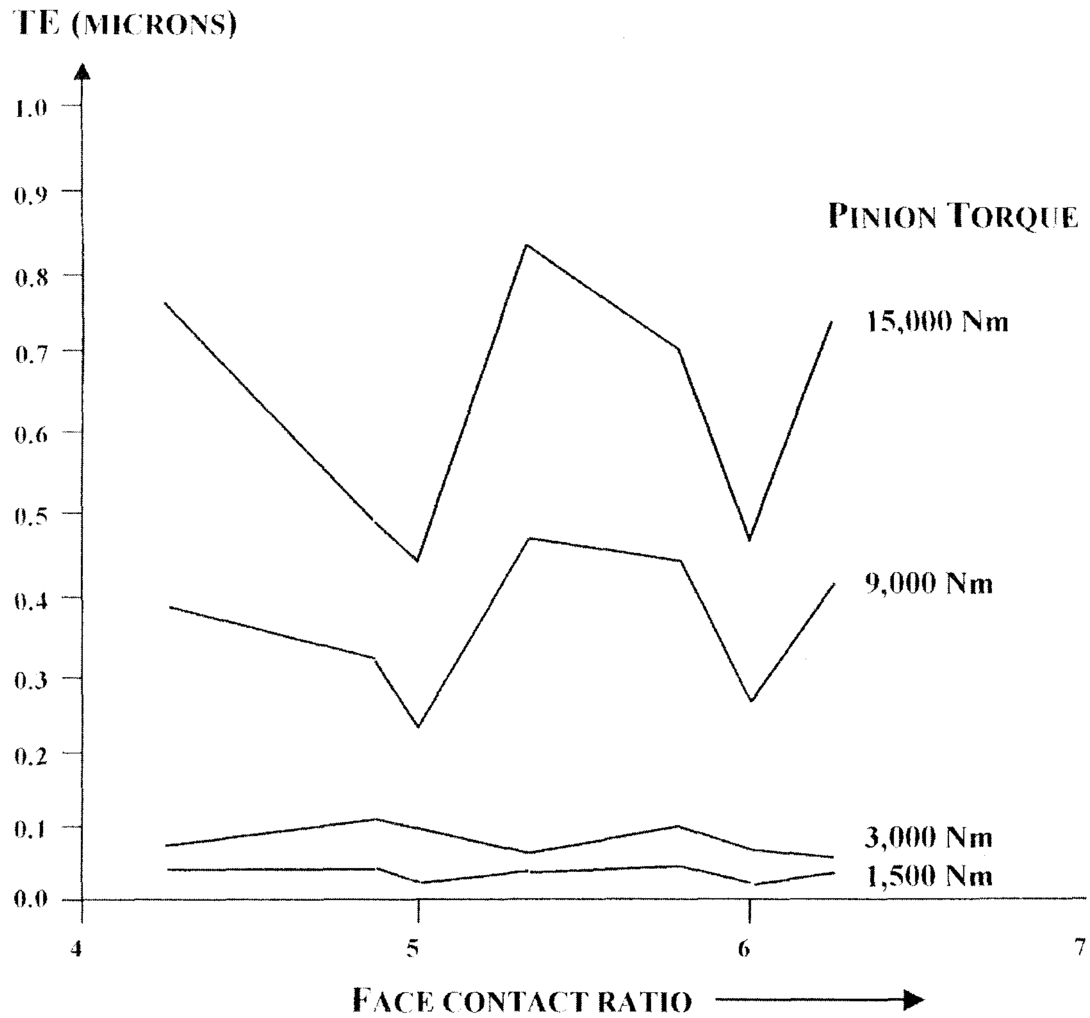

FIG.12 – TE VERSUS FACE CONTACT RATIO. ZERO MISALIGNMENT. FACE CROWNING 10 µM

The above results clearly confirm the well known fact that integer face contact ratio results in the lowest gear noise. Further improvements could be expected from increasing the transverse contact ratio that is, using 'deeper' gear teeth. (FIG.12) shows TE as a function of torque for a gear with 10µm lead crowning, once for standard tooth depth and ε_α = 1.4 and once for 'deep' teeth and ε_α = 2.04.

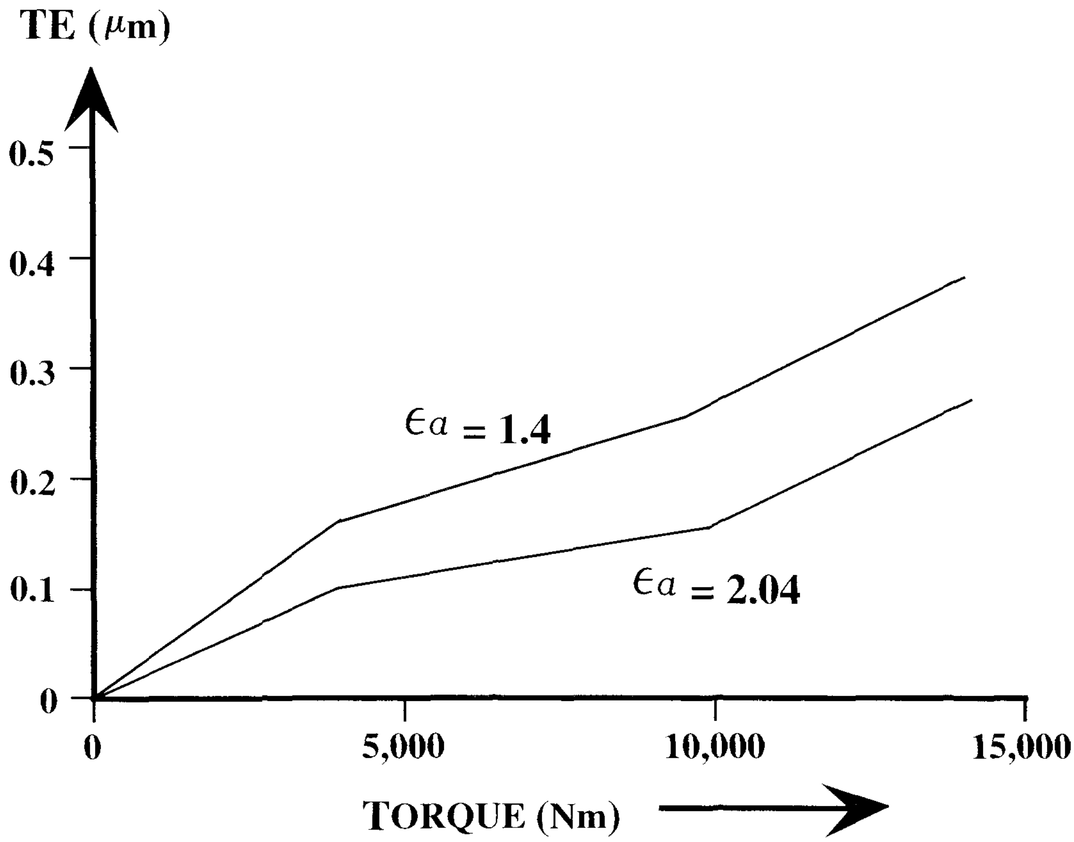

FIG.13 – TE VERSUS TORQUE FOR ε_α = 1.4 AND 2.04

The results of this analysis confirm that, at higher torques, there is a significant advantage in gears with a greater transverse contact ratio. The number of teeth, module, helix angle (ε_β) and tooth depth (ε_α) defines the macro-geometry of the gear that can now be further refined to achieve the best possible TE. The features of the gear geometry, which can be varied to further reduce TE are:

- Lead crowning.
- Topographic lead correction.
- Torsional wind-up correction.
- Tip and root relief.
- Involute crowning.
- Involute pressure angle.

The optimum design will give low TE over a specified torque range and over the full range of misalignment from –20 μm to +20 μm. Parametric optimization as described in reference 5 resulted in the final gear design as summarized in Table 2. It should be noted the pinion is centre driven, and that no additional torsional lead correction was applied.

TABLE 2 – *Gear Geometry – Ultra Quiet Gear*

		Ultra Quiet Gear	
		Pinion	**Wheel**
Number of teeth	z	29	87
Module	M_N (mm)	6.04907	
Pressure Angle	αn (°)	17.5°	
Helix Angle	β (°)	28.676°	
Facewidth	b (mm)	202	
Transverse Contact Ratio	ε_α	2.04	
Face Contact Ratio	ε_β	5.05	
Profile Shift	x	+0.4	-0.4
Tooth Depth	h/m_n	2.79	2.81
Crowning Height	μm	15	Zero
Tip Relief Depth	μm	10	10
Tip Relief Extent (Radial)	mm	2.6	2.6
Lead Correction		Zero	Zero

The TE performance envelope as a function of torque and misalignment, is shown in (FIG.14A) as a three dimensional plot, and in (FIG.14B) as T.E. versus torque with misalignment as parameter. (0, ±10 and ±20 μm misaligned).

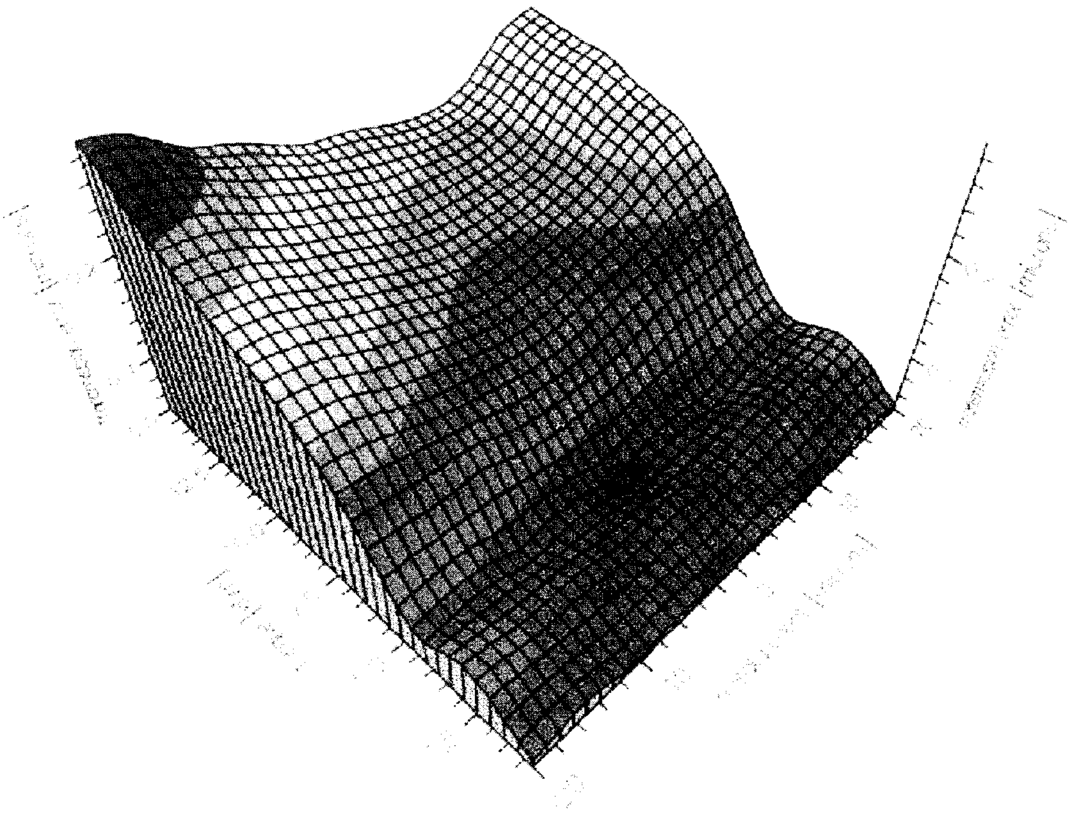

FIG.14A – THREE DIMENSIONAL PLOT

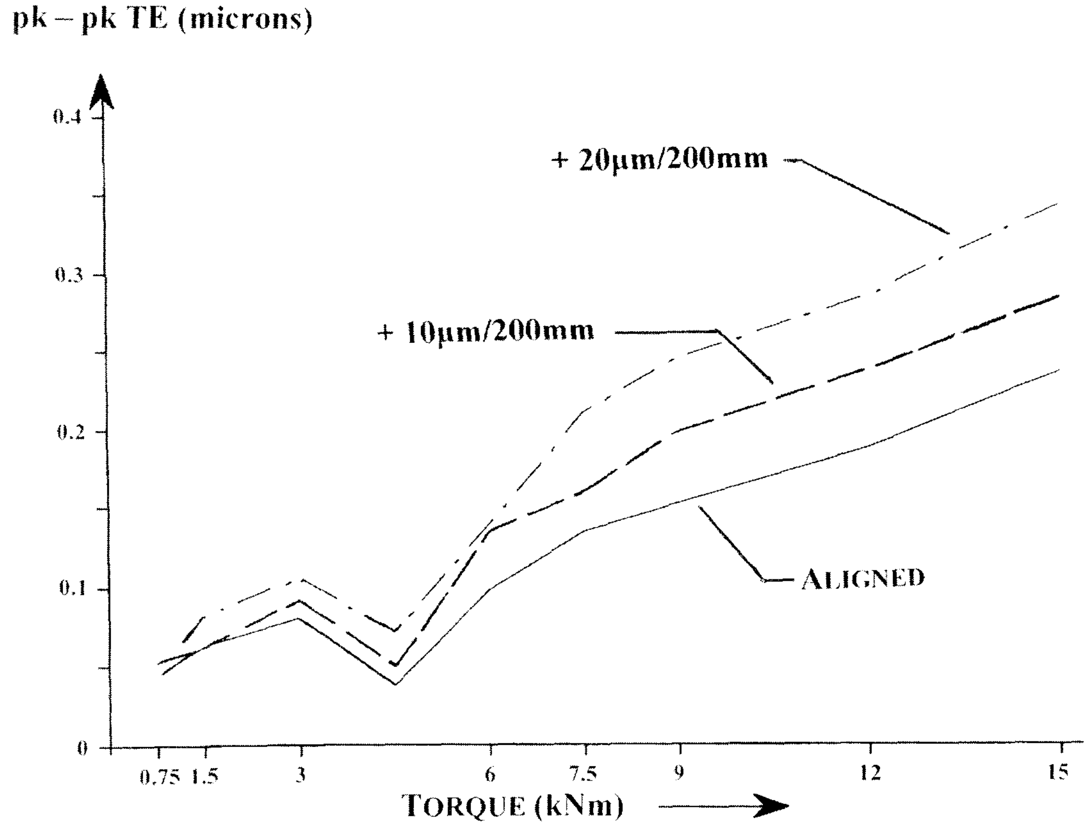

FIG.14B – TE VERSUS TORQUE

FIG.14 – TE AS A FUNCTION OF MISALIGNMENT AND TORQUE (ULTRA LOW NOISE GEARING)

It is noted that, at torques up to 6 kNm (40% FLT) the TE is below 0.1 μm for all alignments. The increase in crowning, and tip and root relief have not significantly improved the TE compared to the uncorrected geometry (see FIG.12). However, at higher torque, the combination of centre drive, and tip and root relief has effected a significant improvement. Relative to the high contact ratio gear without tip and root relief and with only 10 μm crowning, the TE of the optimized gear at zero misalignment has been reduced from 0.36 μm to less than 0.2 μm. In other words, optimum micro-geometry has resulted in reducing TE to 55%, equivalent to a potential noise reduction of –5dB. The ultra quiet gear was manufactured, and extensively tested in the MGRR.

Test Data – Ultra Quiet Gear

Typical results of dynamic bearing load at TCO versus pinion speed (500…5,000 rpm) are shown in (FIGS.15A to 15D). For comparison, the dynamic bearing forces previously measured for the conventional, non optimized high helix angle gear are also shown for the same four torque levels (2,000, 4,000, 8,000 and 12,000 Nm) for tests at zero misalignment.

It is noted that the design optimized, ultra quiet gear generates lower dynamic bearing forces at all torque levels across the speed range. The improvement is particularly noticeable at those speeds where the conventional gear passes through three significant resonances that amplify the dynamic bearing forces.

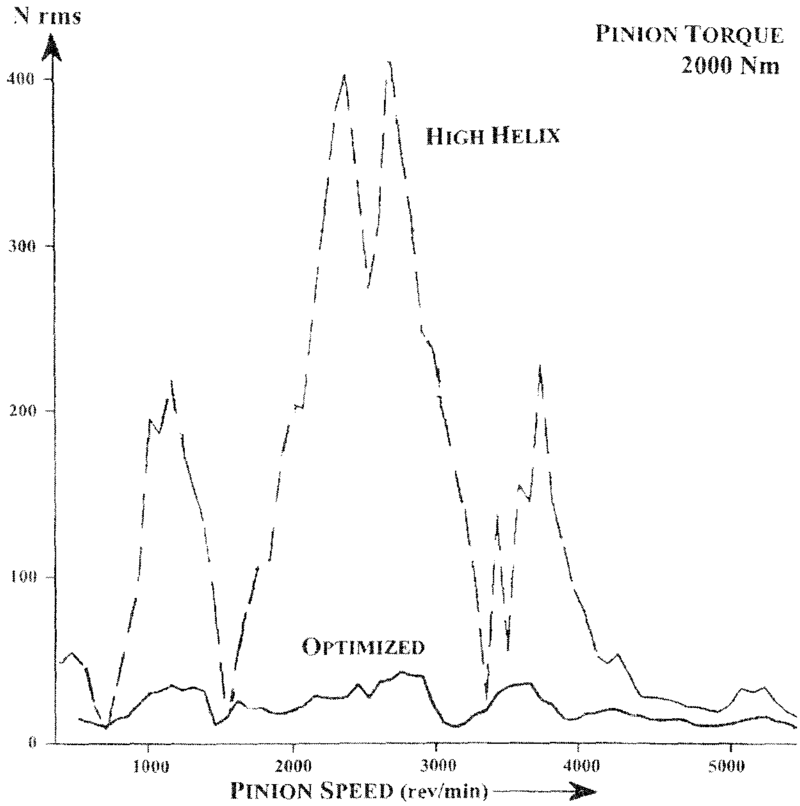

Fig 15a – 2,000 Nm

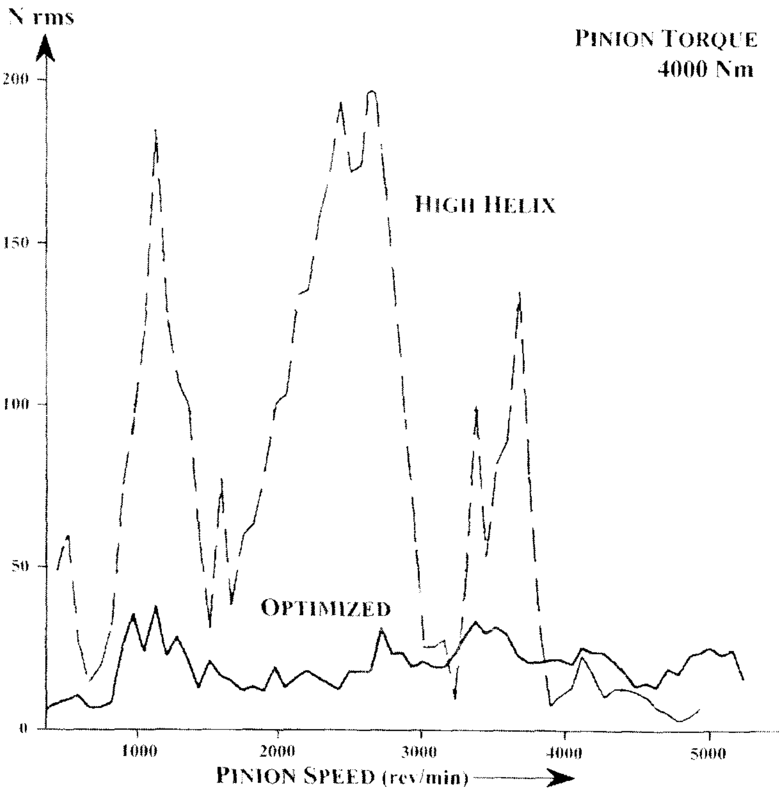

Fig 15b – 4,000 Nm

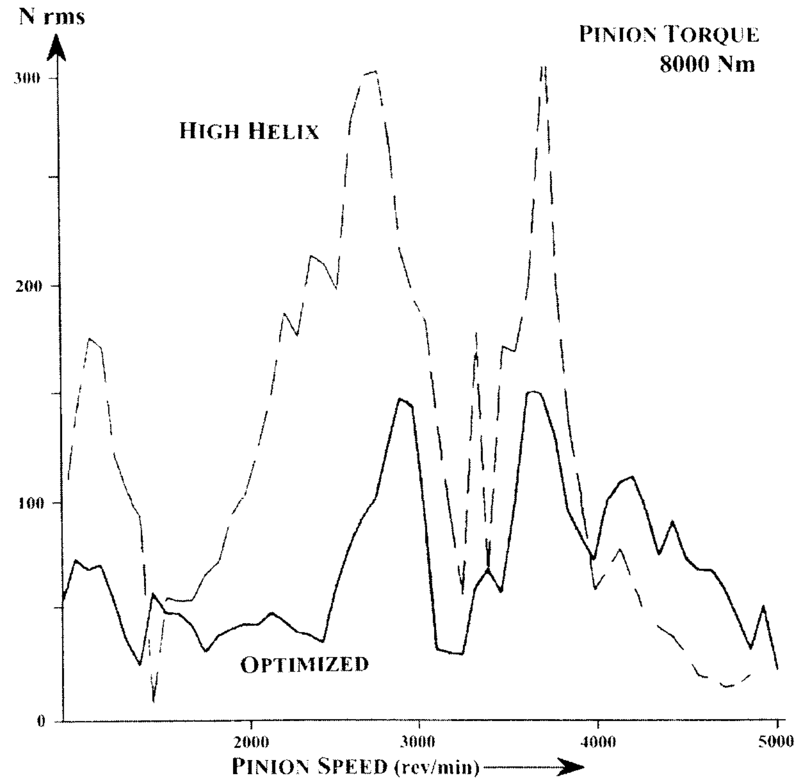

FIG 15C – 8,000 NM

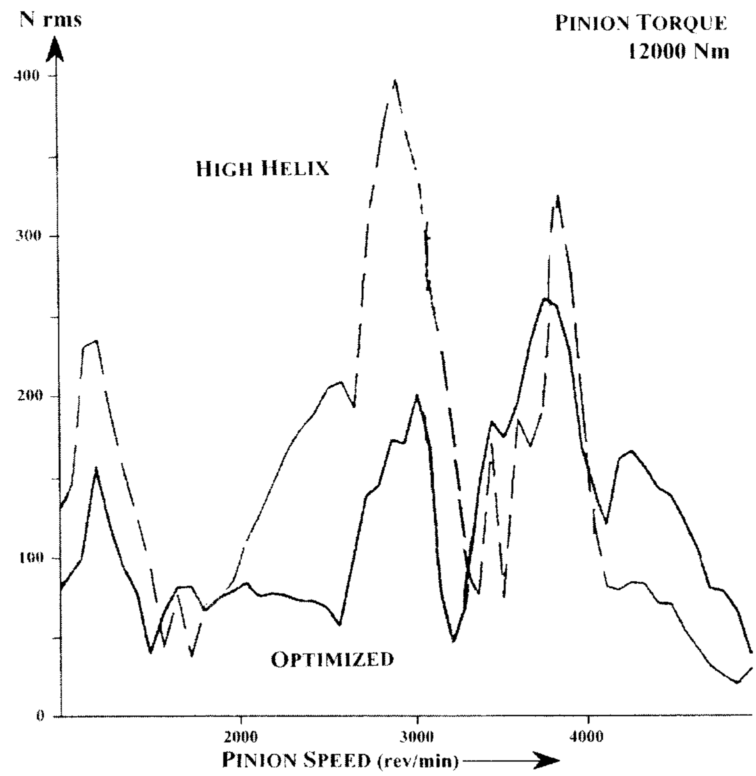

FIG 15D – 12,000 NM

FIG.15 – DYNAMIC BEARING FORCE AT TCO FOR CONVENTIONAL AND DESIGN OPTIMIZED HIGH HELIX ANGLE GEAR ZERO MISALIGNED

The measured data confirms that a significant improvement in dynamic bearing force has been achieved.

Comparison of predicted TE and measured dynamic bearing force

An interesting comparison can now be made between the predicted and actual difference in performance of the conventional high helix angle gear and the design optimized 'ultra quiet' gear.

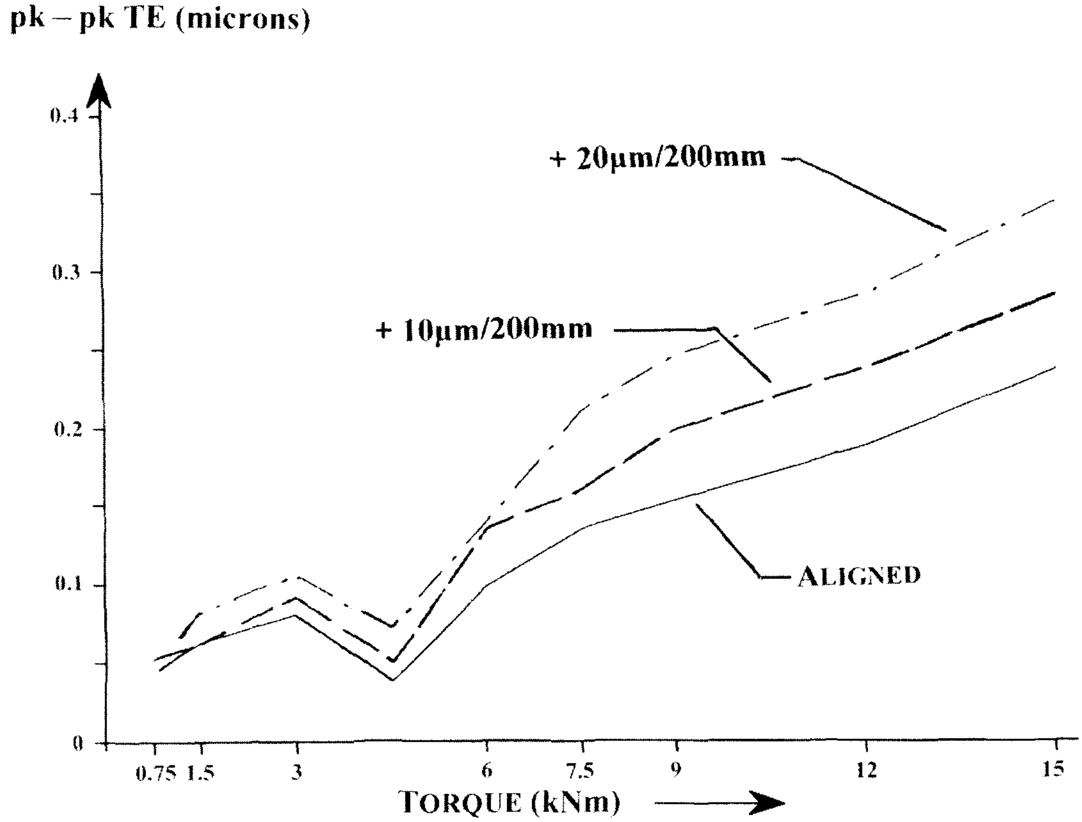

FIG.16 – PREDICTED TE FOR STANDARD AND DESIGN OPTIMIZED GEAR

(FIG.16) shows the calculated TE at TCO (1st Fourier Coefficient) as a function of torque for the conventional high helix and the design optimised gear. (Zero misalignment). This predicts the following reduction in excitation for the design optimized compared to the conventional high helix angle gear (Table 3).

TABLE 3 – *TE reduction at TCO in optimized gear*

Gearbox Torque Nm	Reduction in calculated TE Db
2.000	-16
4,000	-19
8,000	-7
12,000	-2

It is noted that the maximum predicted reduction in TE at TCO occurs at 4,000 Nm (-19 dB) while at 12,000 Nm the predicted reduction in excitation at TCO is very much less (-2 dB). In the case of the 'conventional' high helix gear, the measured dynamic bearing forces at TCO shown in Figures 14A to 14D for 2,000....12,000 Nm torque show extremely large variations of dynamic bearing forces at TCO with speed, due to significant resonances. In contrast, the dynamic bearing forces at TCO excited by the design optimized gear show little amplification due to resonance, and are at extremely low levels. Indeed, it is probable that the dynamic bearing forces due to KE at TCO are below the general 'white noise' level due to other excitations. It thus becomes difficult to compare 'measured' bearing forces with KE. The best comparison and the one that is functionally most relevant to a naval gearbox, would be to compare the measured dynamic bearing forces at TCO at the major resonance, in this case at about 2500...2800 rpm. Table 4 shows the reduction in measured bearing excitation (at TCO) and the reduction in calculated TE for the design optimized gear relative to the conventional high helix angle gear pair.

TABLE 4 – *Reduction in calculated TE and measured dynamic bearing force at TCO in optimized gear*

Gearbox Torque Nm	Reduction in calculated TE dB	Reduction in measured excitation at TCO dB
2,000	-16	-20
4,000	-19	-16
8,000	-7	-6
12,000	-2	-6

From Table 4 it is seen that the magnitude of the reduction in dynamic bearing force at TCO is quite close to the measured calculated reduction in TE (at TCO). At low torque, the optimized gear is 16...20 dB quieter than the standard helical gear.

The smaller reduction in KE at high torque is also reflected in the smaller reduction in measured dynamic bearing forces.

Summary and conclusions

A large number of different gears have been tested on the 8 MW MGRR. These have been typical of current marine practice. The measurements of dynamic bearing forces, that is the excitation forces responsible for gear noise and vibration, show that for otherwise identical gears:

- A gear with 30° helix angle is 15...30 dB quieter than an 8° helix angle gear of otherwise identical geometry.
- A single helical gear is 8...14 dB quieter than a similar double helical gear.

The tests on different gears show that misalignment can significantly affect dynamic bearing forces (i.e. noise).

The major contributor to gear noise and vibration is the pinion, with dynamic excitation at the pinion bearing many times greater than at the wheel bearing.

There is no simple relationship between torque and gear noise. This situation is even more complex in marine drives where torque is related to speed and follows the propeller law.

The noise generated by a gear pair is greatly affected by the dynamic response of the shafts and bearings.

The gears tested in the MGRR at many torques and many different alignments have been analysed with the FE based elastic mesh analysis DU-GATES. A comparison of the calculated TE with the measured dynamic bearing forces shows that there is good correlation between calculated TE and the excitation forces responsible for gear noise and vibration.

The design of an ultra quiet gear has been developed using the calculation procedure DU-GATES. Based on a comparison of calculated TE this is predicted to be some 16...20 dB quieter than the 'conventional' helical gear at low torque, and 2...7 dB quieter at high torque.

Tests on the 'ultra quiet' gear on the MGRR have confirmed the predicted reduction in excitation at TCO.

It is concluded that the optimization of gear geometry, with DU-GATES, to give minimum TE at TCO, is a practicable way of achieving low noise in main propulsion gearing.

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