# The ultra low noise gearbox

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As a result of intensive research over the last 10 years it is now possible to achieve very significant reductions in gear noise. This paper outlines the development of the calculation procedures for low noise gears, and describes the research carried out on the Royal Navy's 8 MW Marine Gearing 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 optimising gear design using these new design techniques can reduce gear noise significantly compared to the best current marine gearing.

## 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 considered only gear strength, which is the fatigue strength against pitting and tooth breakage. Until very recently no techniques were available for optimising 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, there will be decreasing benefits in gear noise.

In the past, the design of gears for low noise has been dominated by the search for ever 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 paper 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.

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**Author's Biographies**
**Ian Atkins** joined the Royal Navy as a Seaman Officer in 1988 following a year in the Merchant Navy. After a period of general training, he undertook Degree training at the Royal Naval Engineering College, Manadon before transferring to the Engineering branch in February 1993. Following applied and engineering sea training, he served as the Deputy Marine Engineer Officer of HMS Campbeltown for 2½ years. Returning ashore, he spent the next 8 months on secondment to the Prince's Trust, teaching character development to young people aged 16-25, prior to completing the Marine Engineer's dagger course at University College, London. Immediately prior to taking up his current appointment as the Gearing Development Officer for the MOD(N) he was based in Sarajevo, Bosnia Herzegovina, where he served as the Personal Assistant to the Chief Press Officer of the NATO led Stabilisation Force.
**Dieter Hofmann** graduated from Kings College, University of Durham in 1962. Following a graduate apprenticeship at A Reyrolle & Co Ltd, Hebburn, he was employed as a design engineer working on high speed switchgear mechanisms. He joined Auto Union (Audi) GmbH in Germany in 1965 as Konstrukteur and worked on a number of new vehicle projects before becoming Assistant to the Chief Engineer. In 1970 he joined the newly formed Design Unit in the University of Newcastle, was appointed Director of the Unit in 1975, and has since developed the gearing and mechanical power transmission R & D activities.
**John Haigh** began an undergraduate apprenticeship with Rolls Royce (aeroengines) Ltd in 1969, graduating from Liverpool University in 1973. On completing his apprenticeship he worked in the Mechanical Research Department until 1978 when he joined the Design Unit in the University of Newcastle as Project and Design Engineer. Over the latter years the major part of his work has been connected with the Marine Gearing Research Rig, for which he has been responsible from the time of conceptual design through to commissioning and subsequent testing.

## 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 that time, the analytical techniques were not available to calculate accurately these changes of velocity ratio or Transmission Error (T.E.).

The development of powerful computers has made possible techniques such as Finite Element Analysis (F.E.A.), which has been developed into a comprehensive tool for calculating the deflection of 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 represent mating gears.

Based on earlier work by A J Pennell and his co-workers at the University of Newcastle upon Tyne (2), (3), the Design Unit has developed a comprehensive gear analysis procedure (DU-GATES – Gear Analysis for Transmission Error and Stress) for the design of high performance, low noise gearing.

Since 1992 the Royal Navy (RN) has supported 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 optimising the detailed geometry of high contact ratio gears as used in marine propulsion gearing (4), (5). Both the concept and the calculation procedure has been exhaustively validated against measured data from the RN's Marine Gear Research Rig. (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 Transmission Error (T.E.). 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 (that is gears with pitch, profile and lead deviation and mounting errors within specified tolerances) are meshed together at zero load with a 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 these geometric errors, and is termed the 'unloaded' or 'geometric transmission error'.

When torque is applied to geometrically perfect gears, there are elastic deflections of the gear teeth at the points of contact due to bending, shear, and contact (Hertzian) forces. The total elastic deflection of pinion and wheel teeth, the mesh deflection, will vary depending on the local stiffness of pinion and wheel teeth, and how this combines along the contact lines. When two geometrically perfect gears are thus meshed together under load 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' transmission error.

The sum of geometric transmission error and elastic transmission error is the total of kinematic T.E. Under dynamic conditions, that is when gear inertia is considered, the acceleration and deceleration of pinion and wheel in response to kinematic excitation (T.E.) 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 and bearings, and also the dynamic characteristics of the gearcase and the stiffness and dynamic behaviour of the mounting between gearcase and hull. This is shown schematically in Fig.1. Until the dynamic response to kinematic excitation 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, thus, reducing excitation, i.e. T.E., and quasi-static bearing force variation, will proportionally reduce gear noise and vibration.

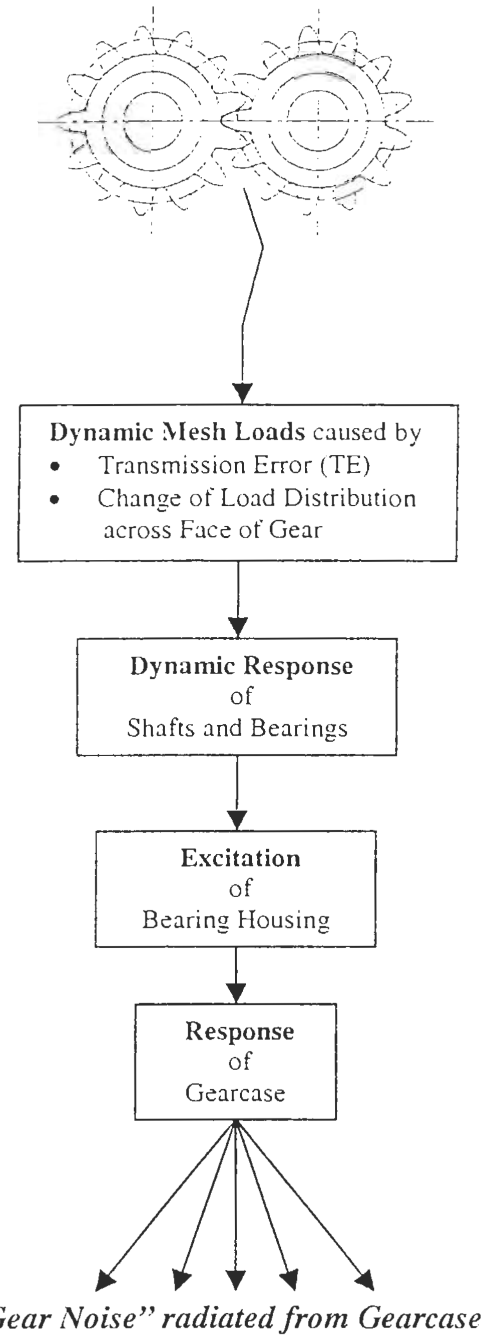



***Fig.1.*** *Transmission of gear noise and vibration*

## THE MARINE GEAR RESEARCH RIG

To study the noise and vibration of typical Naval gearing, the British Royal Navy has funded the setting-up of a Marine Gearing Research Rig (MGRR) at the University of Newcastle upon Tyne (6). 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 facewidth to diameter ratio of about one, at pinion speeds up to 6000 rpm. The size of the test rig was governed by cost, the electric power available in the gear laboratory, and 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 |
| Facewidth | : | 200 mm |
| Maximum speed | : | 6000 rpm |
| Max. torque (pinion) | : | 15000 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 minimise 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.

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.

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 and test gears is shown in Photo 1.

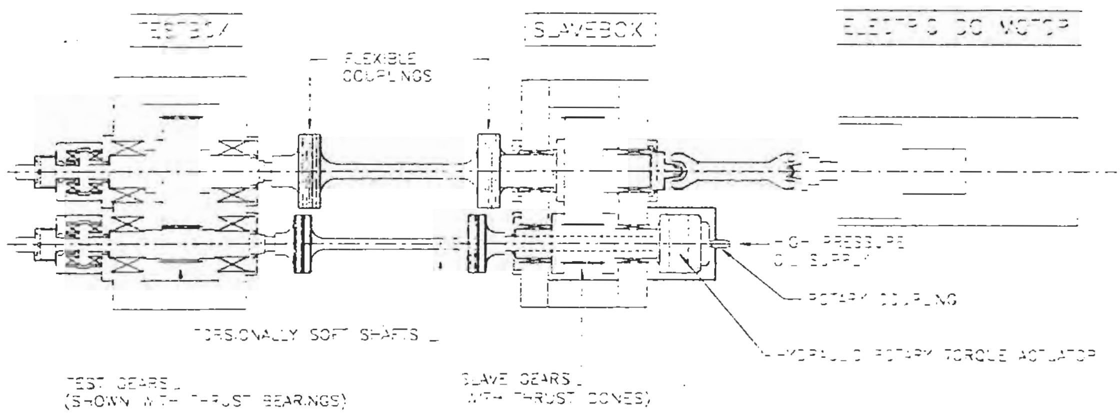
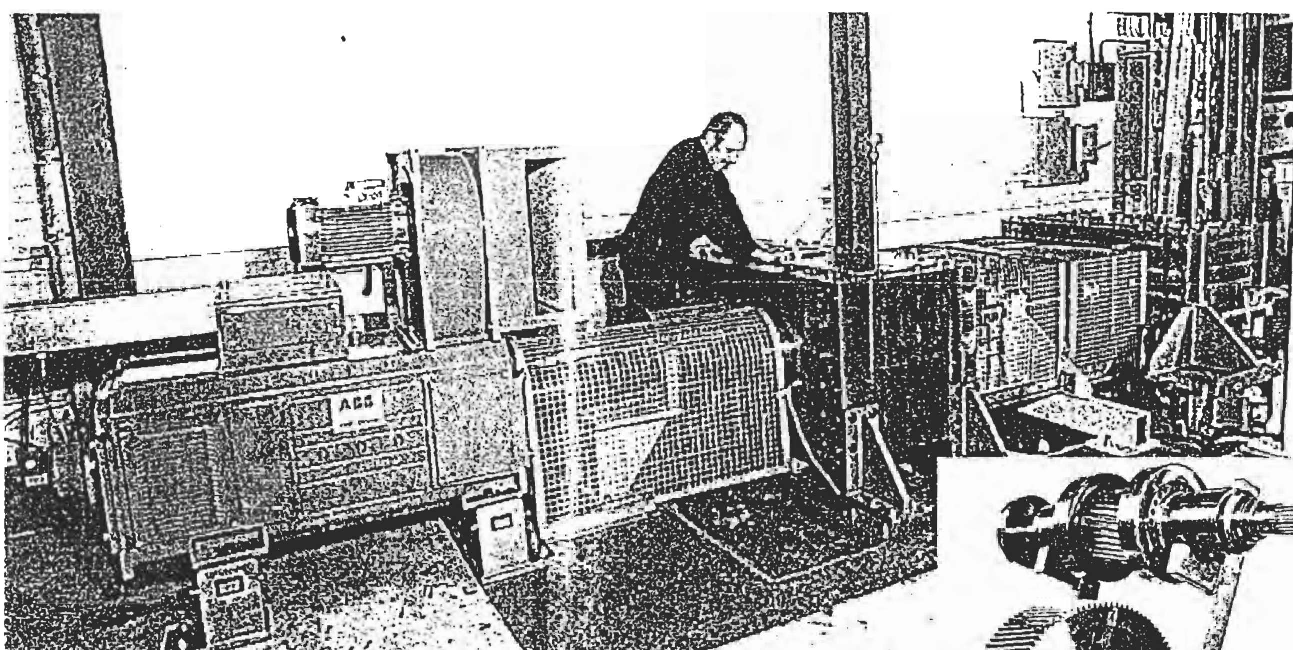



*Fig2. General arrangement of Marine Gearing Research F : (part section)*



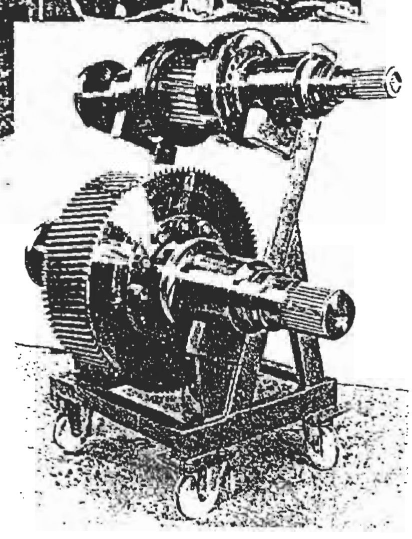


*Photo 1. The Marine Gearing Research Rig (Inset shows Test Gears)*

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.

To effect the measurement of dynamic bearing force with high discrimination and good frequency response, a special bearing dynamometer was developed. This is shown in Fig.3. 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 0.1 N 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.

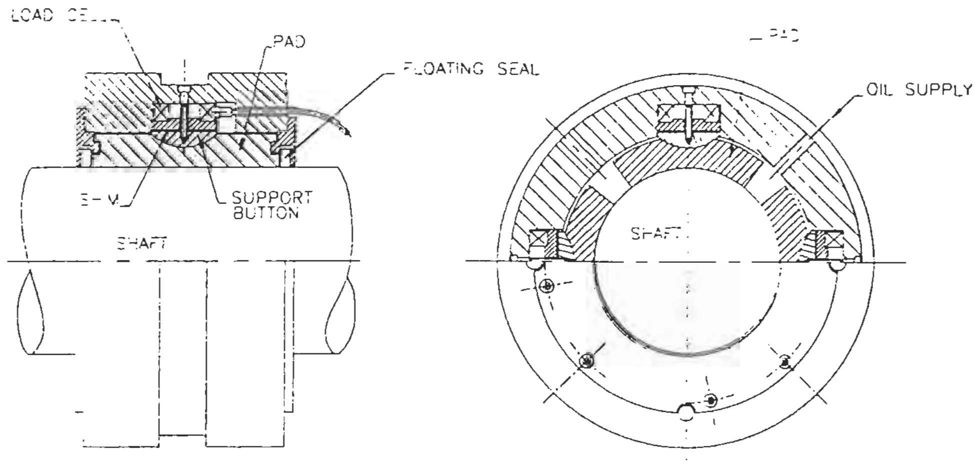



*Fig.3. Part section of tilting pad bearing showing load dynamometer*

In addition to the bearing load dynamometer, the test gearbox is also fitted with a high performance optical system for measuring T.E. 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 but with 8° and 30° helix angle. The details of the test gears are shown in Table 1.

| | | *HIGH HELIX GEARS* | | *LOW HELIX GEARS* | |
|---|---|---|---|---|---|
| | | *Pinion* | *Wheel* | *Pinion* | *Wheel* |
| No. of teeth | $z$ | 29 | 87 | 33 | 99 |
| Module | $M_n$ mm | 6 | | 6 | |
| Pressure Angle | $\alpha_n$° | 20 ° | | 20 ° | |
| Helix Angle | $\beta$° | 29.54 ° | | 8.11 ° | |
| Facewidth | b(mm) | 200 | | 200 | |
| Hob Addendum | $h_{a}$(module) | 1.4 | | 1.4 | |
| Face Contact Ratio | $\varepsilon\beta$ | 5.23 | | 1.5 | |
| Transverse Contact Ratio | $\varepsilon\alpha$ | 1.42 | | 1.74 | |

*Table 1. Gear details*

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.

Typical results from these measurements are shown in Figs. 4a and 4b. 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 1500 to 4500 rpm. Fig.4a shows the dynamic bearing force characteristics for the low helix gear, Fig.4b those for the high helix gear, at a constant torque of 8000 Nm and a nominal (no-load) zero mesh misalignment.

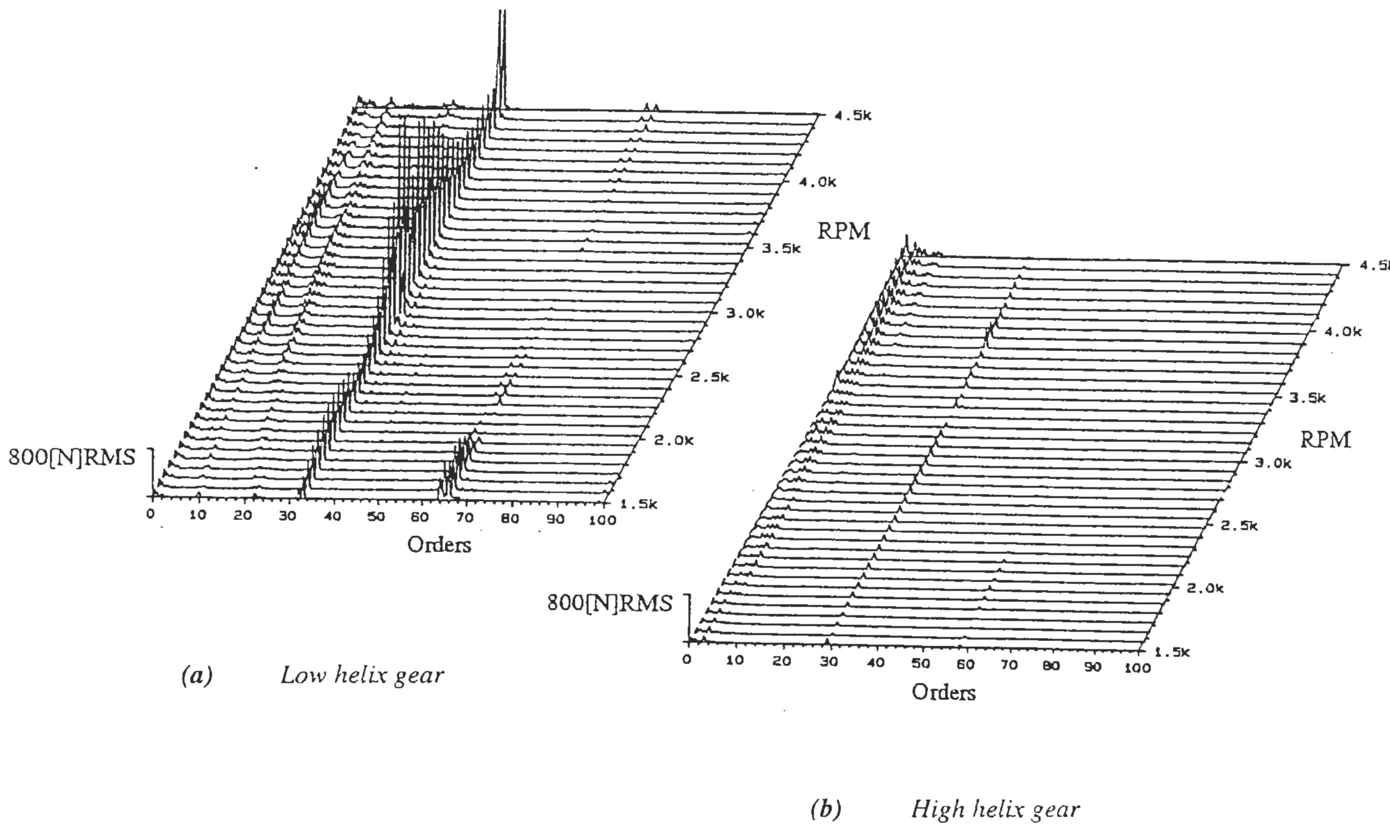



*Fig. 4* *Dynamic Bearing Force Order Analysis, Run-up Tests*

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). Fig.4a shows significant bearing excitation predominantly at the 33rd order, that is, at tooth contact order (TCO), with a peak dynamic bearing force of over 3000 N at about 3000 rpm. In the high helix angle gear, Fig.4b, excitation at TCO, 29th order, is very much less.

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.5 shows the dynamic bearing force (N rms) at TCO as a function of pinion speed. (i.e. Fig.5 can be considered as a 'slice' through the Order Analysis Waterfall plots shown in Fig.4 at TCO (33rd and 29th order respectively). In this case the dynamic bearing forces are plotted at a torque of 4000 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: 1000 N rms
- Single helical low helix angle: 6000 N rms.

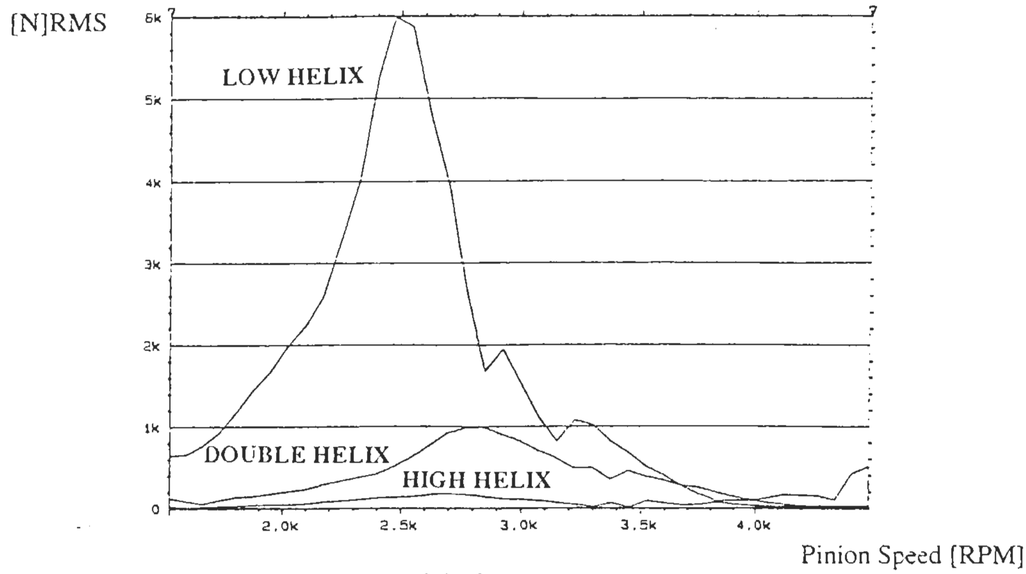



*Fig.5* *Dynamic Bearing Excitation, single and double helical gears*

## 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.6 shows the influence of mesh misalignment on dynamic bearing loads at TCO for the low helix gear, ahead flank, measured at the pinion bearing.

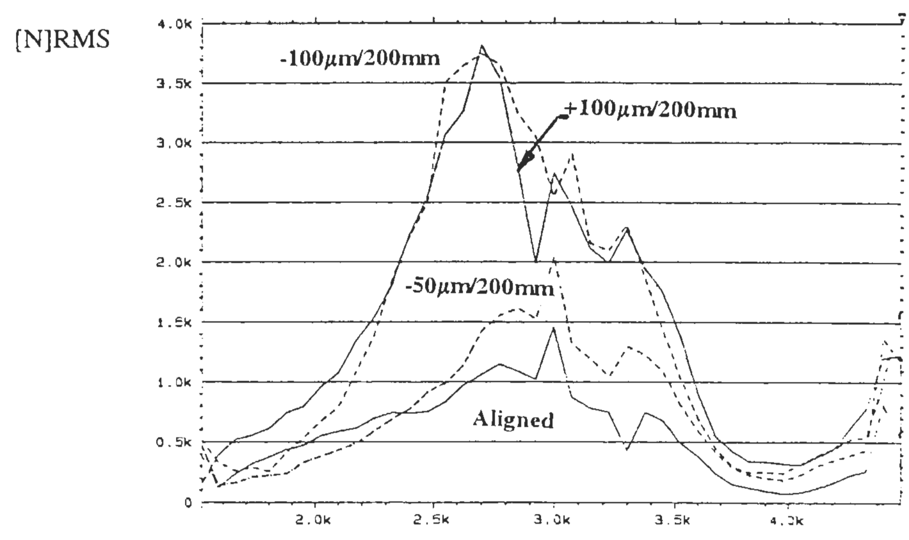



*Fig.6* *T.E. at various mesh misalignments*

From Fig.6 it is clearly seen that misalignment increases maximum dynamic bearing forces at TCO from 1500 to nearly 4000 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 kinematic excitation, i.e. the T.E. 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 T.E. 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 4000 Nm and zero misalignment are shown in Figs.7a and 7b.

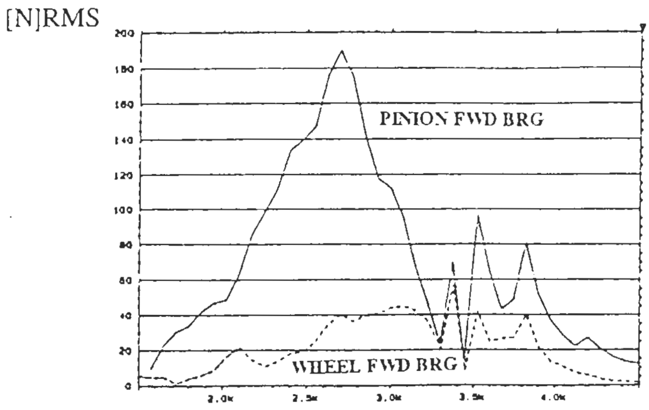


*Fig.7a* *Dynamic Bearing Forces*
*High Helix Gears (TCO)*

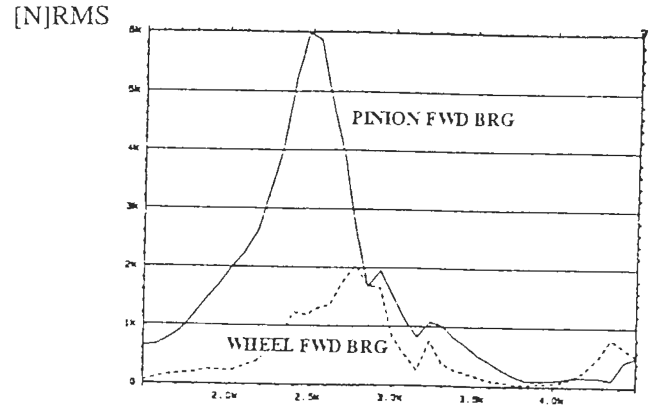


*Fig.7b* *Dynamic Bearing Forces*
*Low Helix Gears (TCO)*

It is noted, as previously shown, that the peak dynamic bearing force is significantly greater for the low helix gear (6000 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 resonances, particularly a dominant pinion flexural resonance at 2400 – 2700 rpm (40….45 Hz).

## THE EFFECT OF TORQUE ON DYNAMIC BEARING FORCES

Dynamic bearing forces are the result of

- Kinematic excitation, e.g. T.E. and
- 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 kinematic excitation, 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 locus 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.8.

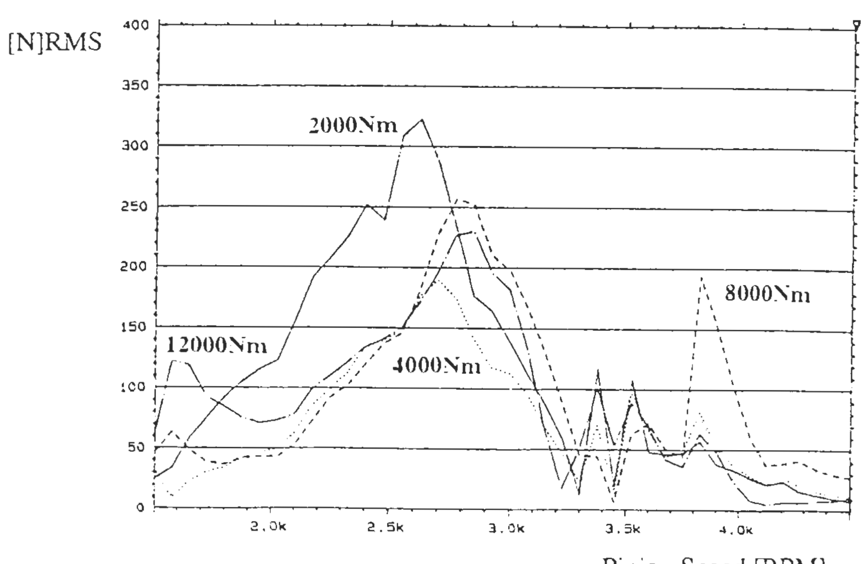



*Fig.8* *Dynamic bearing force at TCO for the high helix gear, astern flank, zero misalignment*

This shows dynamic bearing force at TCO for the high helix angle gear pair over a speed range from 1500 to 4500 rpm, for four pinion torque levels. (2000, 4000, 8000.12000 Nm). It is noted that:

- Maximum dynamic bearing forces occur at a torque of 2000 Nm at first flexural resonance of the pinion shaft, at which speed (2700 rpm) the lowest bearing forces occur at 4000 Nm torque.
- Outside of resonance, and particularly above 3000 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 2000 Nm (2600 rpm) and greatest at 12000 Nm (2850 rpm), demonstrating the effect of increased mesh stiffness at higher torques.

## NOISE AND VIBRATION REDUCTION

Gears will not generate any noise if the T.E. and quasi-static bearing load variation (i.e. the excitation) is reduced to zero. 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 T.E. 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 T.E. based on a defined gear geometry. (Note: Perfect compensation of elastic T.E. by equal and opposite geometric T.E. is possible at only one specific torque. However, it is possible to devise gear geometries on this basis which have very low total T.E. over a significant range of torque).

## ELASTIC MESH ANALYSIS – DU-GATES

The elastic mesh analysis, DU-GATES, is based on accurately modelling 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. (Ref. 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 T.E. for that gear pair, at a specific torque and a specific alignment between pinion and wheel shafts.

In addition to calculating load distribution along the contact lines and elastic mesh deflection (i.e. T.E.) 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.9 and can be divided into two parts:

1. Finite element analysis and generation of compliance and stress coefficients (for a given macro-geometry this part of the analysis is run only once), and
2. tooth contact analysis which is run repeatedly to optimise the micro-geometry to achieve minimum T.E. 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 F.E. model of the gear, and an internal F.E. 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 T.E. and stress for the full range of operating conditions.

Output from the calculation procedure – kinematic T.E., 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 F.E. analysis and curve fitting, and about 10 secs for the elastic analysis per phase increment.

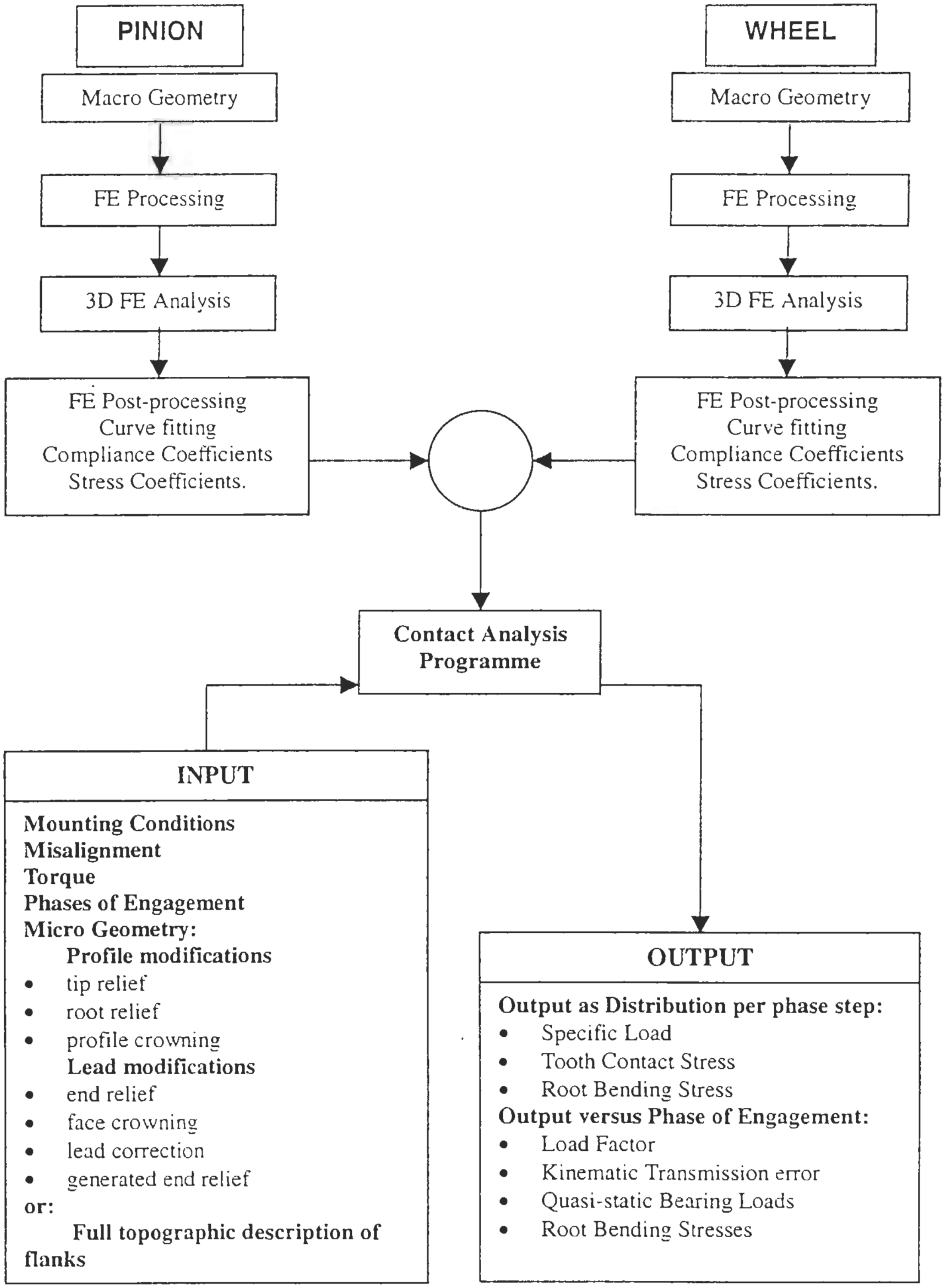



*Fig.9.* *Block diagram of DU-GATES software package*

## VALIDATION OF DU-GATES

Test data for a large number of gears, over the torque range 500 to 15000 Nm and for misalignments between −100 and +100 μm has been analysed, and compared with the T.E. calculated for these test gears at those torques and misalignments using the F.E. based elastic mesh model DU-GATES. Since DU-GATES calculates only the kinematic excitation, comparison with the measured data is only realistic at one speed. For the purpose of the comparison between calculated T.E. and measured dynamic bearing force at TCO, a common speed of 1500 rpm was selected. Fig.10 shows the relationship between measured dynamic bearing forces at the aft pinion bearing (N rms at TCO) and calculated T.E.

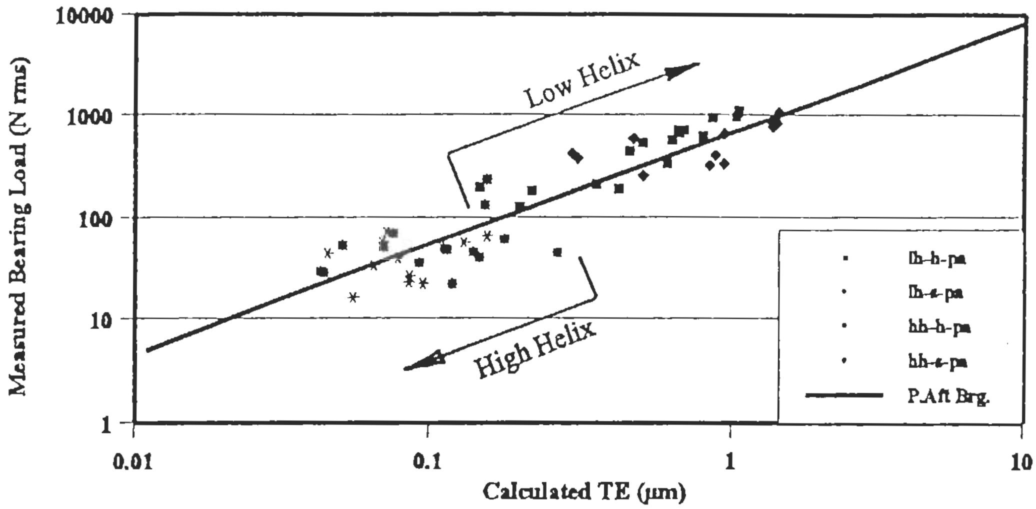



*Fig.10 Measured bearing force at TCO (Pinion Aft Bearing) v computed T.E.*

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 noise and vibration levels. Although there is significant scatter in the relationship between measured bearing load and calculated T.E., there is a clear linear relationship corresponding to an excitation of 1000 N rms (at TCO) per μm T.E. (at TCO). This relationship between dynamic bearing force and T.E. 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 T.E.

## THE DESIGN OF AN 'ULTRA-QUIET' GEAR PAIR

The good correlation between computed T.E. and measured dynamic bearing forces confirms that minimising the quasi-static T.E. (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 T.E., 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 were 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 $\varepsilon_\beta$) and tooth depth (and hence transverse contact ratio $\varepsilon_\alpha$) was optimised 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.11 shows the relationship between T.E. and face contact ratio, $\varepsilon_\beta$, for four torque levels. For this analysis, standard depth teeth with $\varepsilon_\alpha = 1.4$ 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. $\varepsilon_\beta = 5$ 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 T.E. at this torque is reduced from 0.85 μm to 0.45 μm. It should be noted that the peak to peak transmission error of 0·45 μm is less than 1% of the total elastic mesh deflection.

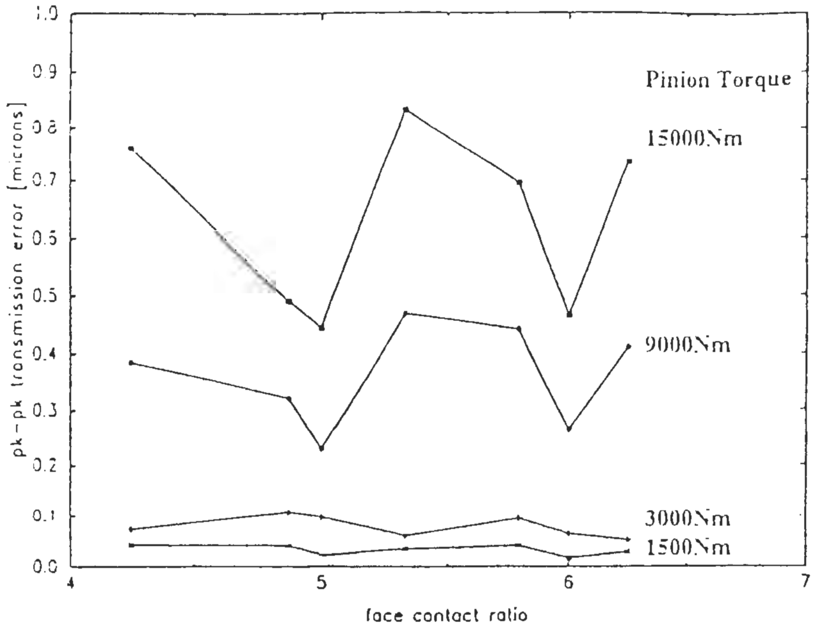



*Fig.11 T.E. versus face contact ratio. Zero misalignment. Face crowning 10 μm*

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 T.E. as a function of torque for a gear with 10μm lead crowning, comparing standard tooth depth ($\varepsilon_\alpha = 1.4$) and 'deep' teeth ($\varepsilon_\alpha = 2.04$).

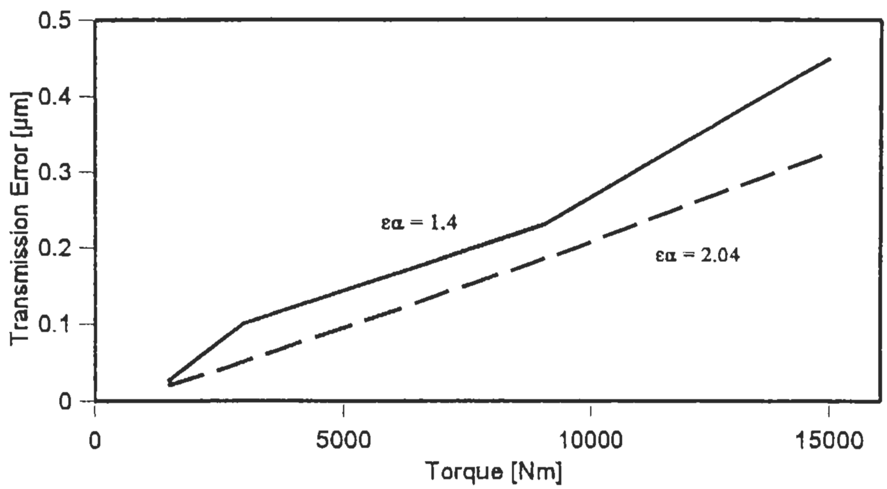



*Fig.12 T.E. versus torque for $\varepsilon_\alpha$ = 1.4 and 2.04*

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 ($\varepsilon_\beta$) and tooth depth ($\varepsilon_\alpha$) defines the macro-geometry of the gear that can now be further refined to achieve the best possible T.E. The features of the gear geometry which can be varied to reduce T.E. 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 T.E. over a specified torque range, and over the full range of misalignment from −20 μm to +20 μm. Parametric optimisation as described in (5) resulted in the final gear design as summarised in Table 2. It should be noted the pinion is centre driven, and that no additional torsional lead correction was applied.

| | | *ULTRA QUIET GEAR* | |
|---|---|---|---|
| | | *PINION* | *WHEEL* |
| Number of teeth | z | 29 | 57 |
| Module | $M_N$ (mm) | 6.04907 | |
| Press Angle | $\alpha n$ (°) | 17.5° | |
| Helix Angle | $\beta$ (°) | 28.676° | |
| Facewidth | b (mm) | 202 | |
| Transverse Contact Ratio | $\varepsilon_{\alpha t}$ | 2.04 | |
| Face Contact Ratio | $\varepsilon_{\beta}$ | 5.05 | |
| Profile Shift | x | +0.4 | -0.4 |
| Tooth Depth | $h/m_n$ | 3.21 | 3.21 |
| Crowning Height | μm | 15 | Zero |
| Tip Relief Depth | μm | 10 | 10 |
| Tip Relief Extent (Radial) | mm | 2.6 | 1.9 |
| Lead Correction | | Zero | Zero |

*Table 2 Gear Geometry – Ultra Quiet Gear*

The T.E. performance envelope as a function of torque and misalignment, is shown in Fig.13a as a three dimensional plot, and in Fig.13b as T.E. versus torque with misalignment as parameter. (0, +10 and −20 μm misaligned).

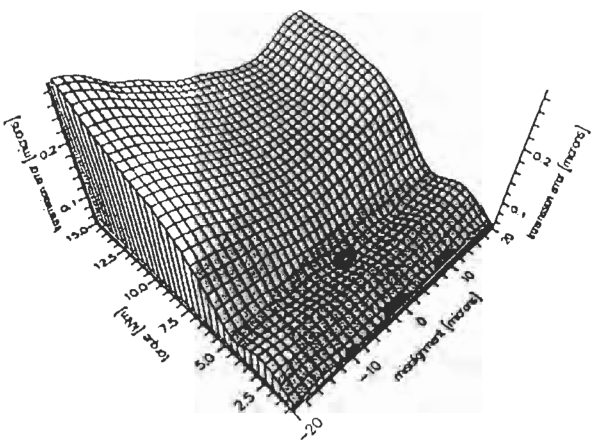



*Fig.13a*

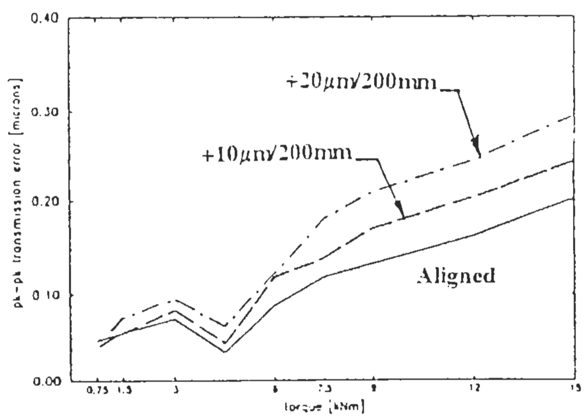



*Fig.13b*

*Fig 13 T.E. as a function of misalignment and torque (Ultra Low Noise Gearing)*

It is noted that, at torques up to 6 kNm (40% FLT) the T.E. is below 0.1 μm for all alignments. The increase in crowning, and tip and root relief have not significantly improved the T.E. 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 T.E. of the optimized gear at zero misalignment has been reduced from 0.33 μm to less than 0.2 μm. In other words, optimum micro-geometry has resulted in reducing T.E. to 60%, equivalent to a potential noise reduction of −4dB. The ultra quiet gear was manufactured, and extensively tested in the MGRR.

## TEST DATA – ULTRA QUIET GEARS

Typical results of dynamic bearing load at TCO versus pinion speed (500...5000 rpm) are shown in Figs.14a to 14d. 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 (2000, 4000, 8000 and 12000 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.

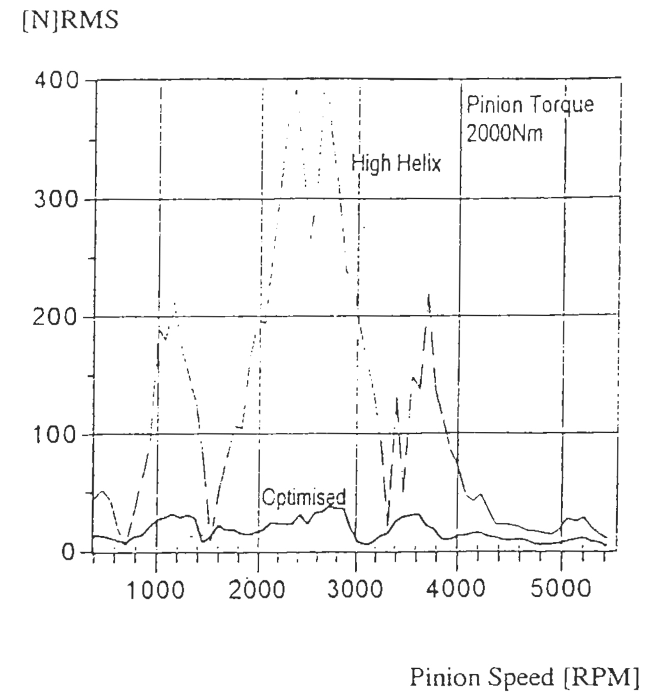



*Fig.14a 2000 Nm*

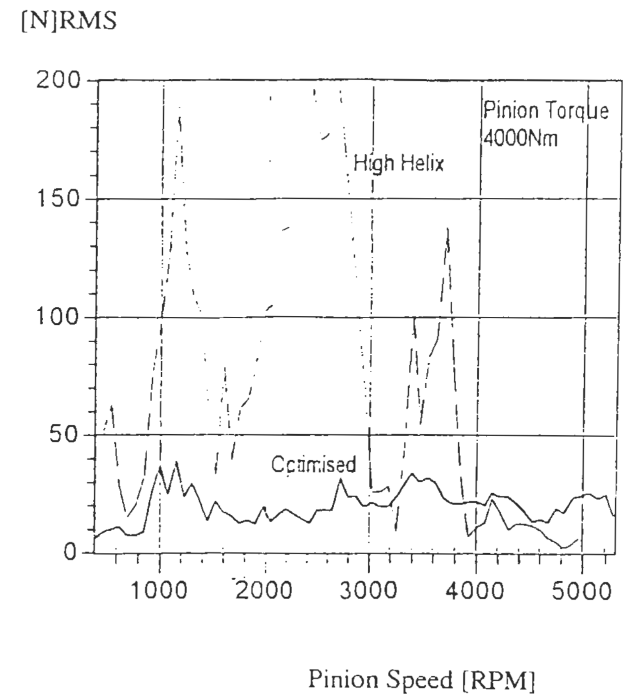



*Fig.14b 4000 Nm*

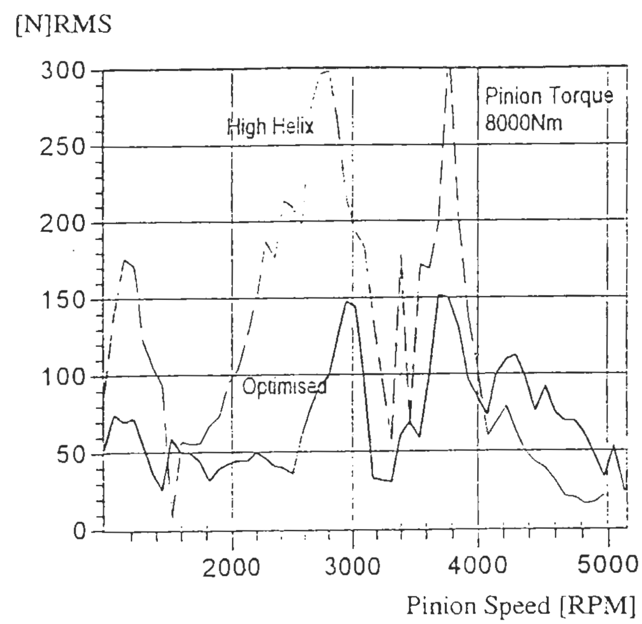



*Fig.14c 8000 Nm*

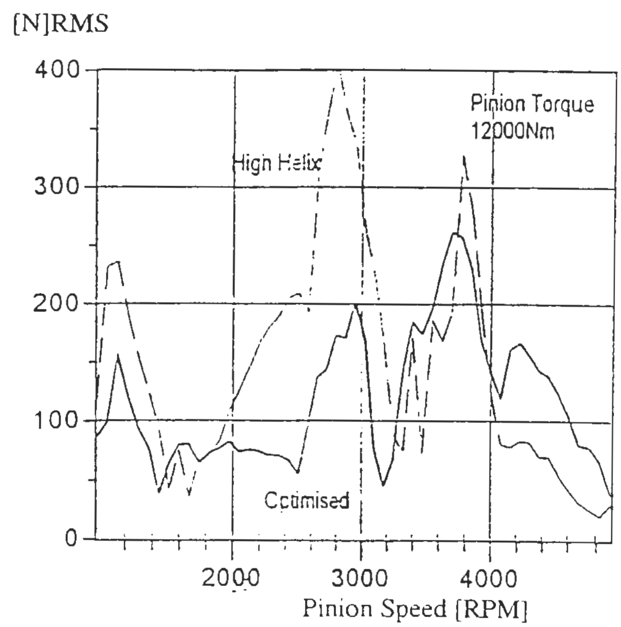



*Fig.14d 12000 Nm*

*Fig.14 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 T.E. 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.

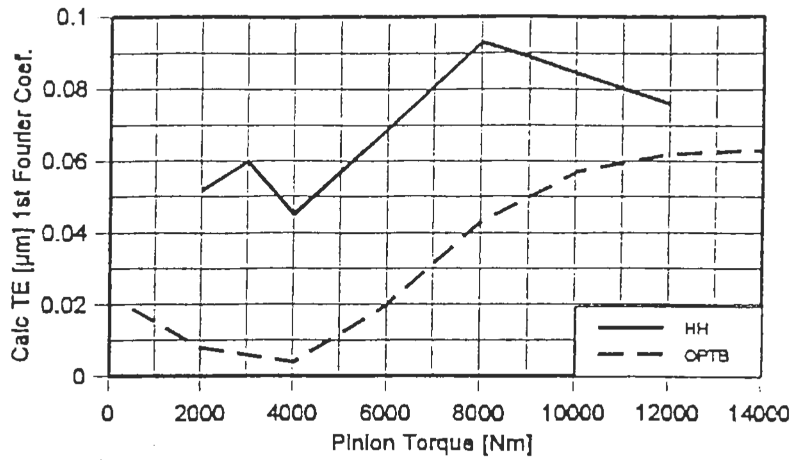



*Fig.15 Predicted T.E. for standard and design optimized gear*

Fig.15 shows the calculated T.E. 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).

| *GEARBOX TORQUE Nm* | *REDUCTION IN CALCULATED T.E. dB* |
|---|---|
| 2000 | -16 |
| 4000 | -19 |
| 8000 | -7 |
| 12000 | -2 |

***Table 3** T.E. reduction at TCO in optimised gear*

It is noted that the maximum predicted reduction in T.E. at TCO occurs at 4000 Nm (-19 dB) while at 12000 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 Figs.14a to 14d for 2000....12000 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 optimised gear show little amplification due to resonance, and are at extremely low levels. Indeed, it is probable that the dynamic bearing forces due to kinematic excitation at TCO are below the general 'white noise' level due to other excitations. It thus becomes difficult to compare 'measured' bearing forces with kinematic excitation. 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 T.E. (at TCO) for the design optimised gear relative to the conventional high helix angle gear pair.

| *GEARBOX TORQUE Nm* | *REDUCTION IN CALCULATED T.E. dB* | *REDUCTION IN MEASURED EXCITATION AT TCO dB* |
|---|---|---|
| 2000 | -16 | -20 |
| 4000 | -19 | -16 |
| 8000 | -7 | -6 |
| 12000 | -2 | -6 |

***Table 4.** Reduction in calculated T.E. and measured dynamic bearing force at TCO in optimised gear*

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

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

## SUMMARY AND CONCLUSIONS

i. A large number of different gears have been tested on the 8 MW Marine Gear Research Rig. These have been typical of current marine practice. The measurements of dynamic bearing force, 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.
- A single helical gear is 8...14 dB quieter than a similar double helical gear.

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

iii. 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.

iv. 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.

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

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

vii. The design of an ultra quiet gear has been developed using the calculation procedure DU-GATES. Based on a comparison of calculated T.E. 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.

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

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

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