THE FAILURE INVESTIGATION AND REPLACEMENT OF A LARGE MARINE GEAR

BY

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This is an edited version of the paper that was presented at the British Gear Association Autumn Technical Meeting (November 2004) and at the American Gear Manufacturers Association Fall Technical meeting (October 2004).

ABSTRACT

Following a routine inspection of HMS *Invincible*'s main propulsion gearboxes, cracking was identified on the starboard main wheel teeth. This article presents a summary of the subsequent work, including: the palliative repair, inspection regime, risk reduction measures, failure investigation, the design and manufacture of replacement gears and the permanent repair.

Introduction

The three Carrier Vertical Strike (CVS) platforms in service with the UK Royal Navy (RN) provide the platform and facilities for the command and control of maritime and joint forces. They are 209m in length, displace 22,000 tonnes and have a complement of 685 ship's company and 386 Air Group personnel. Powered by four 18MW OLYMPUS gas turbines, they are capable of 28kts (FIG.1).

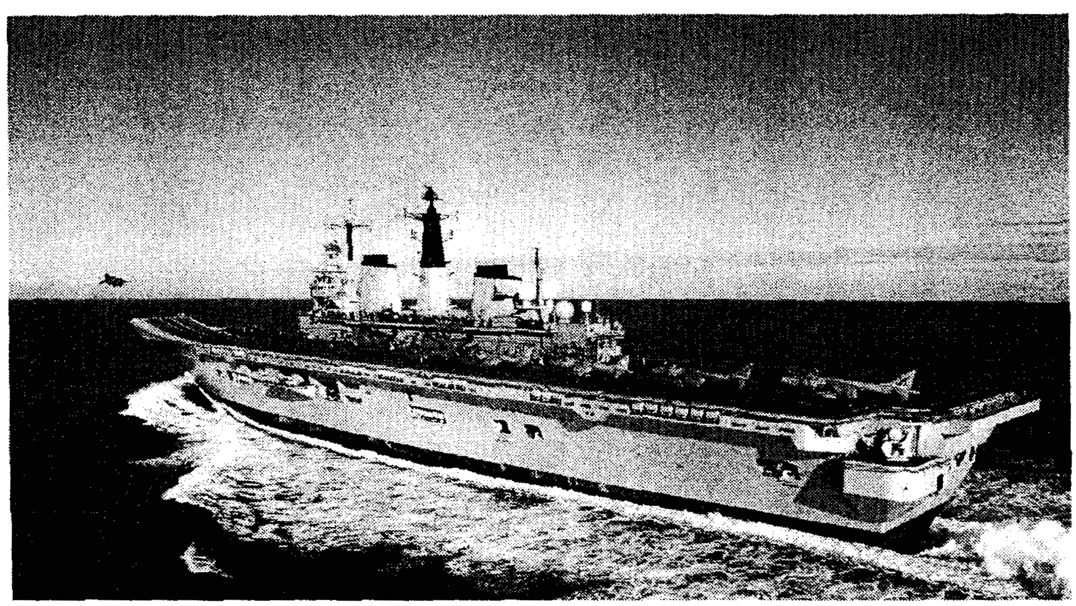


FIG.1 – HMS *INVINCIBLE*

Each CVS is fitted with two main propulsion gearboxes, the largest and most complex in RN service, designed and manufactured by David Brown, now part of the Textron group. The triple reduction, reversing, double helical tandem design transmits power from either one or two OLYMPUS gas turbines to a fixed pitch propeller. Manoeuvring ahead and astern is achieved by means of fluid couplings and for high power ahead operation drive is transmitted through two Synchronous Self Shifting clutches. The gearbox weighs 170 tonnes and contains 19 gear elements, the largest being the main wheel, which is 3m in diameter and weighs 22 tonnes (FIG.2).

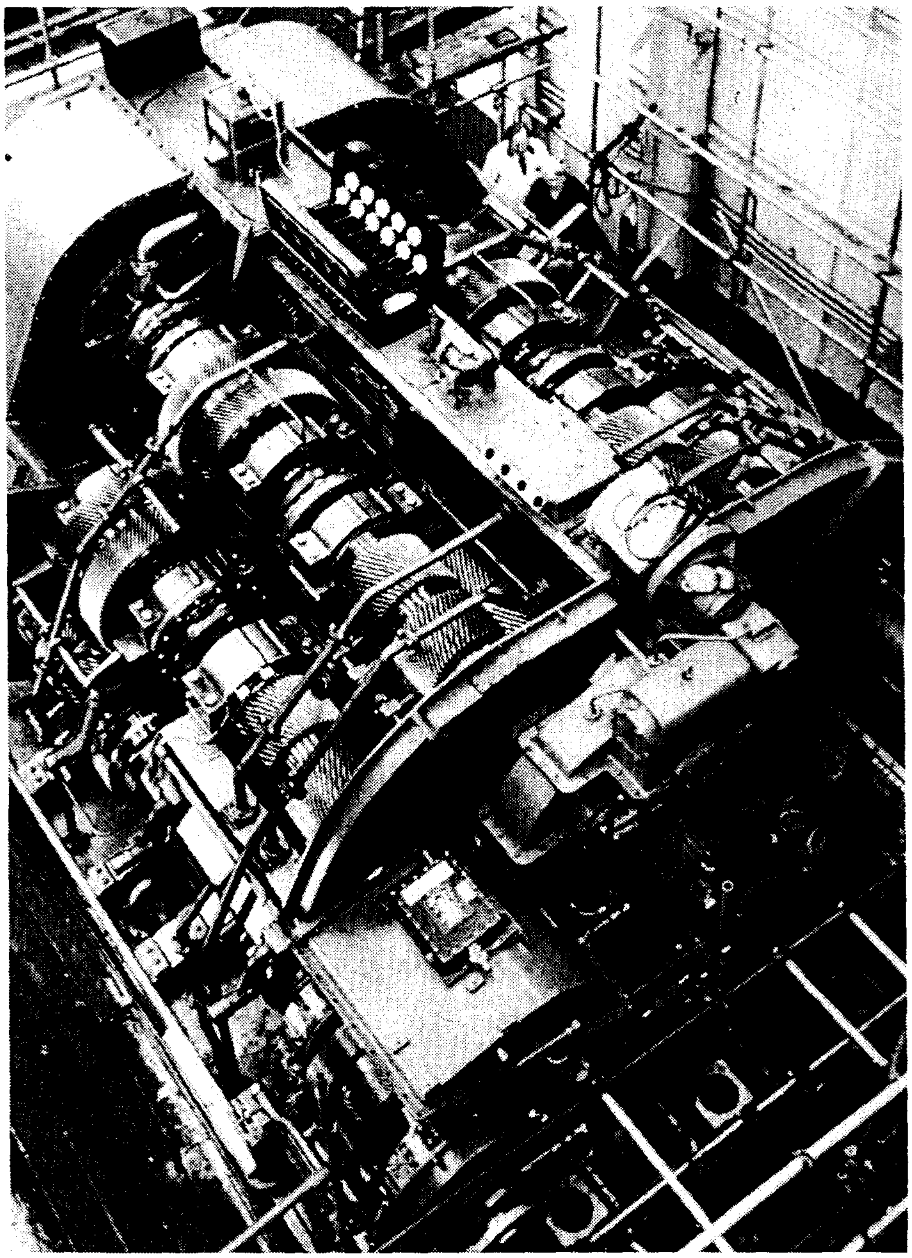


FIG.2 – CVS GEARBOX DURING BUILD

In December 1999, during the Commander (E)'s supercession inspection of HMS *Invincible*'s starboard gearbox, ship's staff identified damage to a number of teeth on the main wheel. A full dye penetrant Non-Destructive Examination (NDE) of the main wheel by David Brown revealed cracks originating from surface pits in 11 teeth. These varied in length up to a maximum of 30cm (FIG.3).

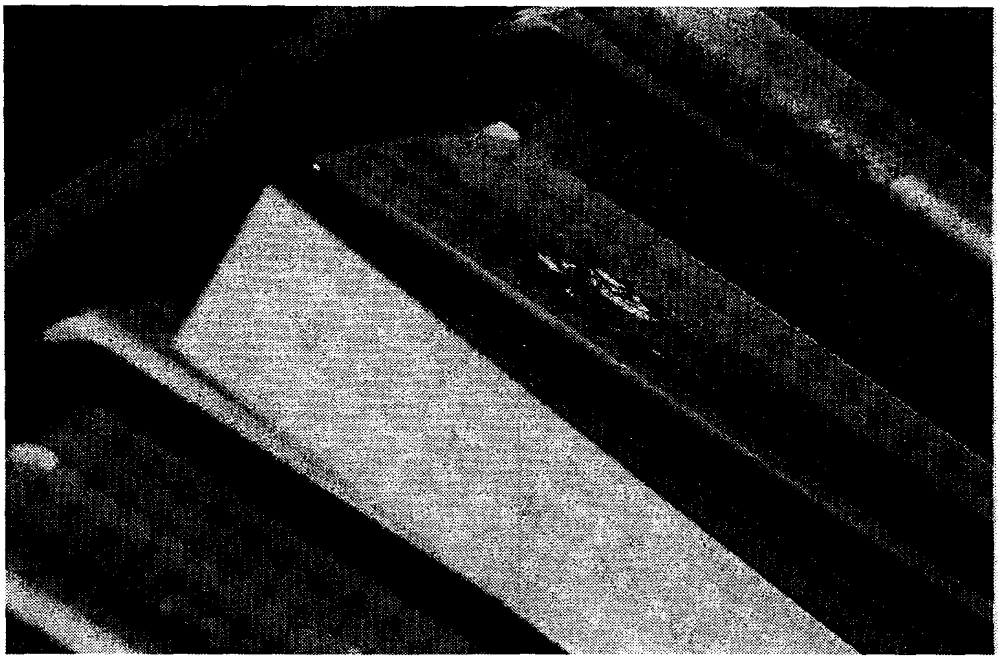


FIG.3 – FLANK PITTING ON A CRACKED TOOTH.

Palliative Repair

Following an initial assessment it was decided to remove the cracks by in-situ dressing, to establish the full extent of the problem and to prevent any further crack growth. An iterative process of grinding and NDE was undertaken, to ensure that no crack indications remained (FIG.4).

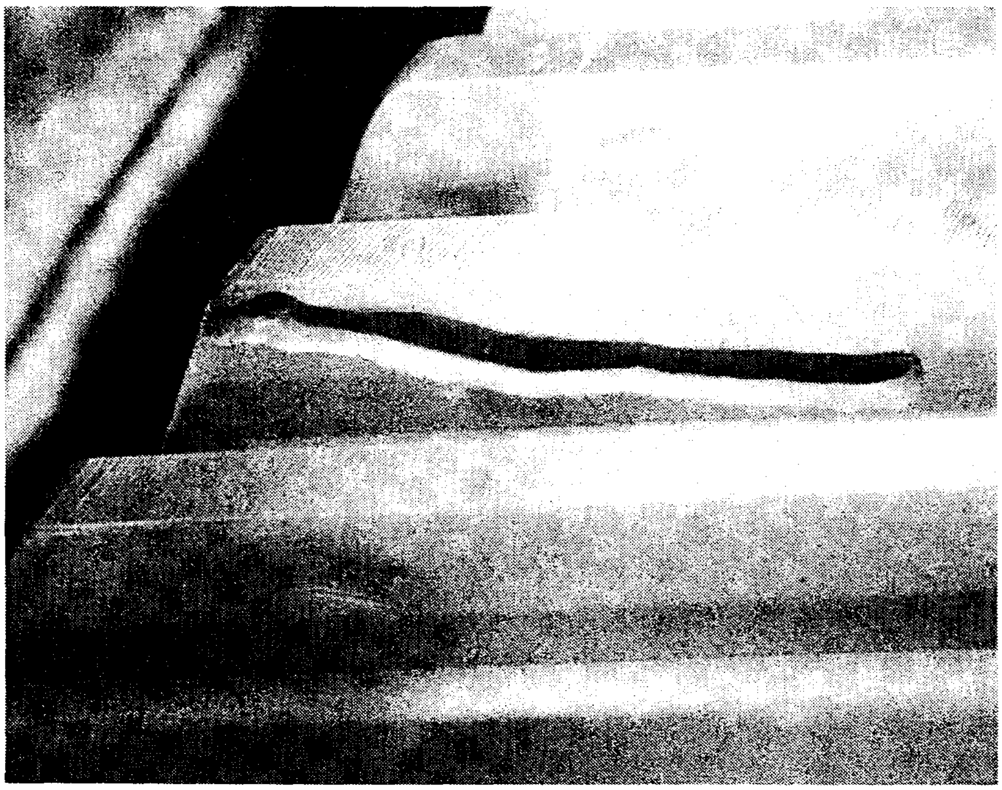
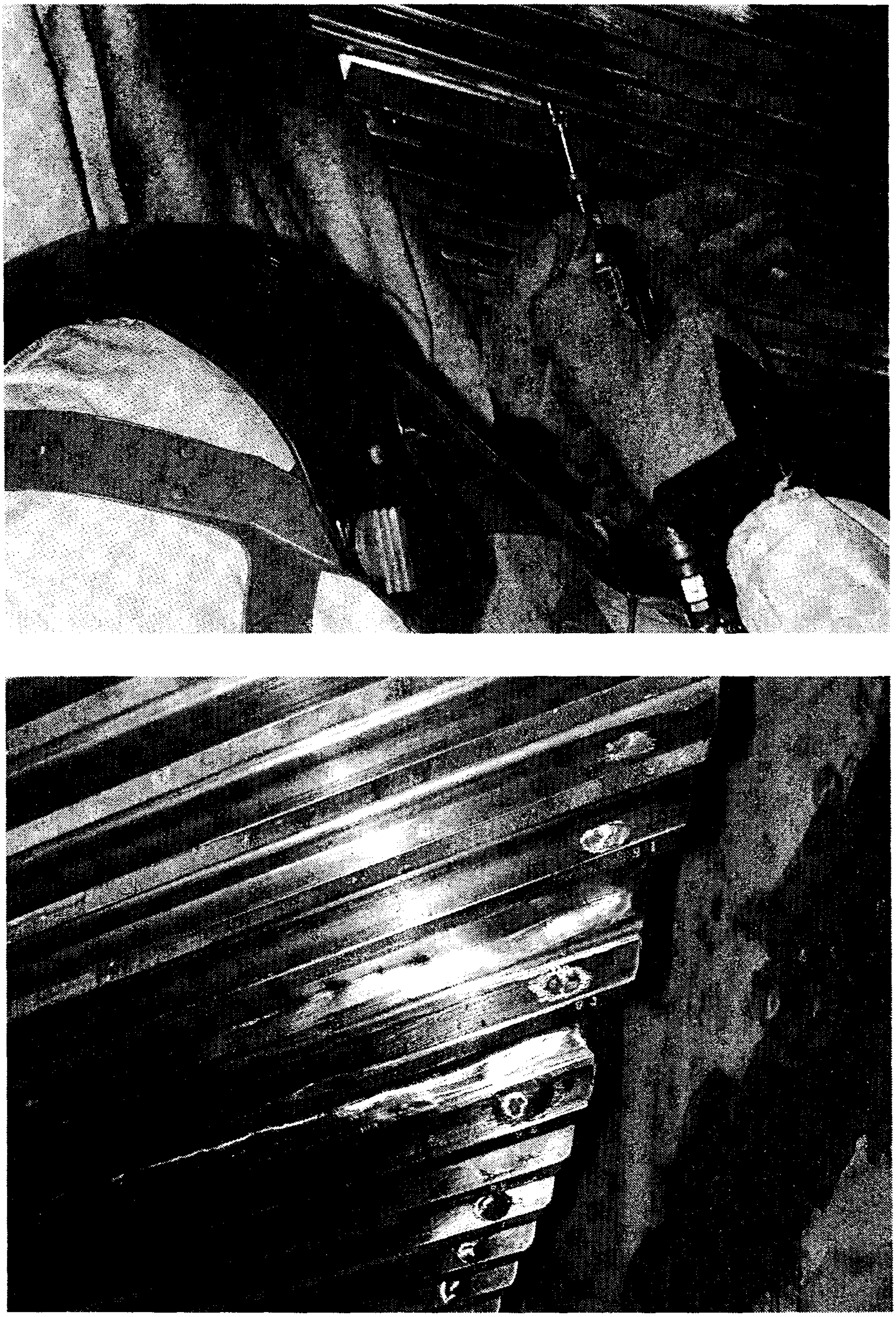


FIG.4 – NDE WHILST REMOVING A CRACK

This resulted in the removal of 9 complete teeth and parts of 2 teeth (FIG.5).



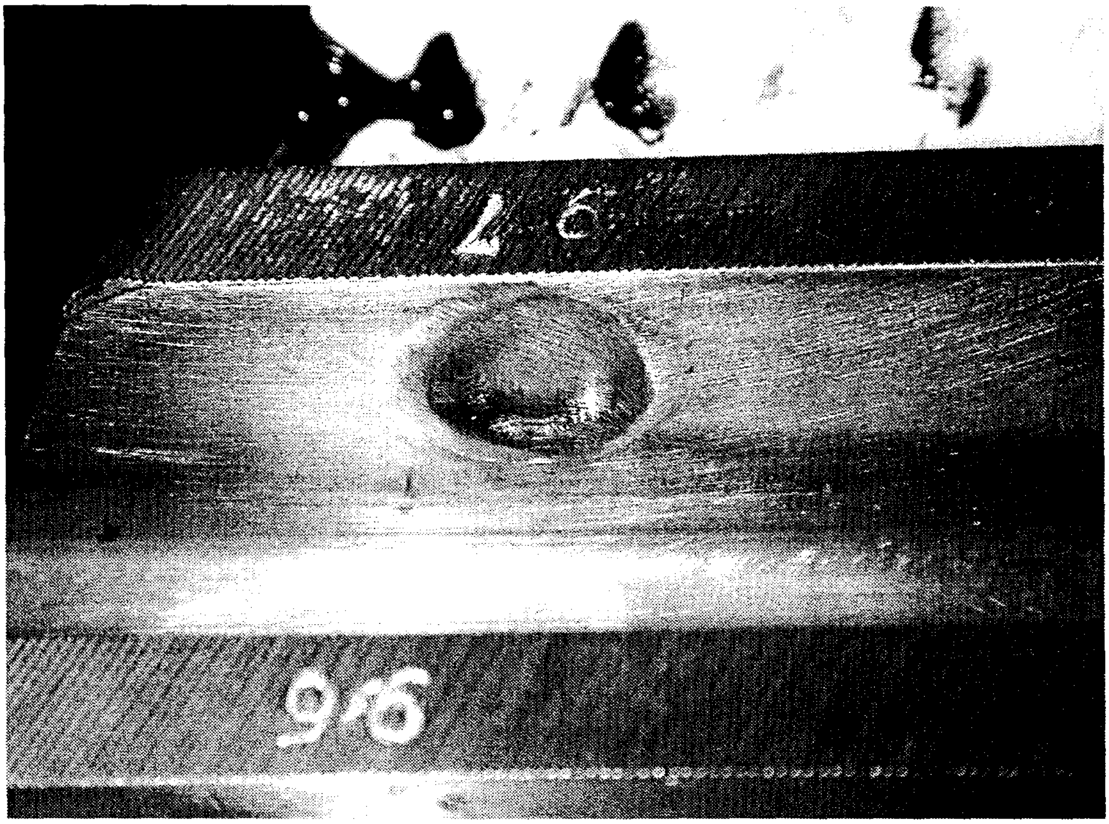


FIG.5 – DRESSING TO REMOVE TOOTH SECTIONS AND AREAS OF PITTING.

A full and detailed inspection of the port main wheel revealed no defects confirming that the problem was isolated to the starboard main wheel.

Reducing the Risk of Further Damage

It was clear that revised propulsion limitations were required if the risk of further gear tooth failures or deterioration was to be minimized. Therefore, to reduce the loads being experienced by the damaged main wheel teeth, revised single and twin engine torque limits were calculated, based on the loss of tooth contact area and an assessment of the allowable stress to theoretically assure infinite life.

For the undamaged gearbox the torque at which additional engines were brought on-line (change-up criteria) was reduced, to avoid maximum tooth stressing at intermediate ship speeds. Whilst this did increase engine running hours and fuel consumption, it provided a margin of error against failure. The original twin engine maximum limitation was retained, but for 'urgent' operational use only and a lower 'routine' operational maximum was introduced to reduce the risk further.

A revised procedure was also introduced to ensure that a good balance of torque was achieved between the OLYMPUS gas turbines, particularly at high torques. With the torsion meter measuring output torque at the main shaft, it had been possible for one engine to be working harder than the other, effectively overloading the main wheel teeth, but for this to go unnoticed because the total shaft torque was within limits.

With these revised limitations and the close control of transients, the risk of further tooth failure was assessed as small, but the ship was still able to meet her operational requirements. Subsequently a monitoring system was installed, which captured torque readings on a continuous basis, to enable the cause of any transients to be investigated and avoided.

Monitoring the Defects

To provide early warning of any further deterioration, an enhanced inspection regime was devised. The defects were initially monitored following post repair trials and then by a rigorous three monthly inspection, undertaken using dye penetrant NDE by a specialist team consisting of MoD, David Brown and Design Unit. Other monitoring techniques, including vibration analysis, were considered, but in tests this had not been able to reliably identify and trend deterioration. It was considered that new cracks would not propagate at a rate that would not be identifiable by the imposed regime of dye penetrant NDE.

The task of full scale dye penetrant inspection on a large wheel is time consuming and unpleasant to carry out, but it was felt that this technique gave the best opportunity of providing full coverage of all teeth with good accuracy for identification and monitoring of the type of cracks experienced previously. Although other methods are available for crack detection (e.g. ultrasonics and eddy current), these were not as easy to apply with confidence in-situ. Being a relatively simple technique to apply, dye penetrant was also made available as a backup during ship's staff visual inspections.

With the revised operating procedures in place, further deterioration did not occur which suggested that the new restrictions had been pitched at about the right level.

Failure Investigation

Having managed the immediate situation through inspection, remedial work and changed operating protocol in order to control and prevent any further degradation of the main wheel, the next stage was to gain a full understanding of the cause.

This was tackled by investigating the basic material properties and failure mechanism, gaining a full understanding of operational characteristics and relating this to the gear design intent with a reassessment of the gear design using up-to-date techniques.

Metallurgical Examination

Using sections from the failed portions of teeth, a full material investigation was carried out by Design Unit to determine if any deficiencies in basic material properties were present that could account for the occurrence of the failure and to identify the mode of failure from fracture surfaces.

The specified material condition is through hardened to a U condition (approximately 290Hv minimum). Hardness measurements taken from polished cross sections through the failed portions of tooth revealed a hardness level in the region of 280Hv. This compared well with manufacturing records, which showed a hardness of approximately 300Hv. Examination of the microstructure did not show anything unexpected for the condition of material and no large inclusions were observed showing the steel to be of the quality level intended.

Examination of the fracture surfaces showed clear indications of fatigue crack growth indicating that the crack had taken sometime to initiate and propagate to give the final failures. The main bending crack that led to failure was found to initiate from below the surface and this was tracked to the presence of flank surface initiated pitting that was seen fairly widespread in the regions of tooth cracking (FIG.6).

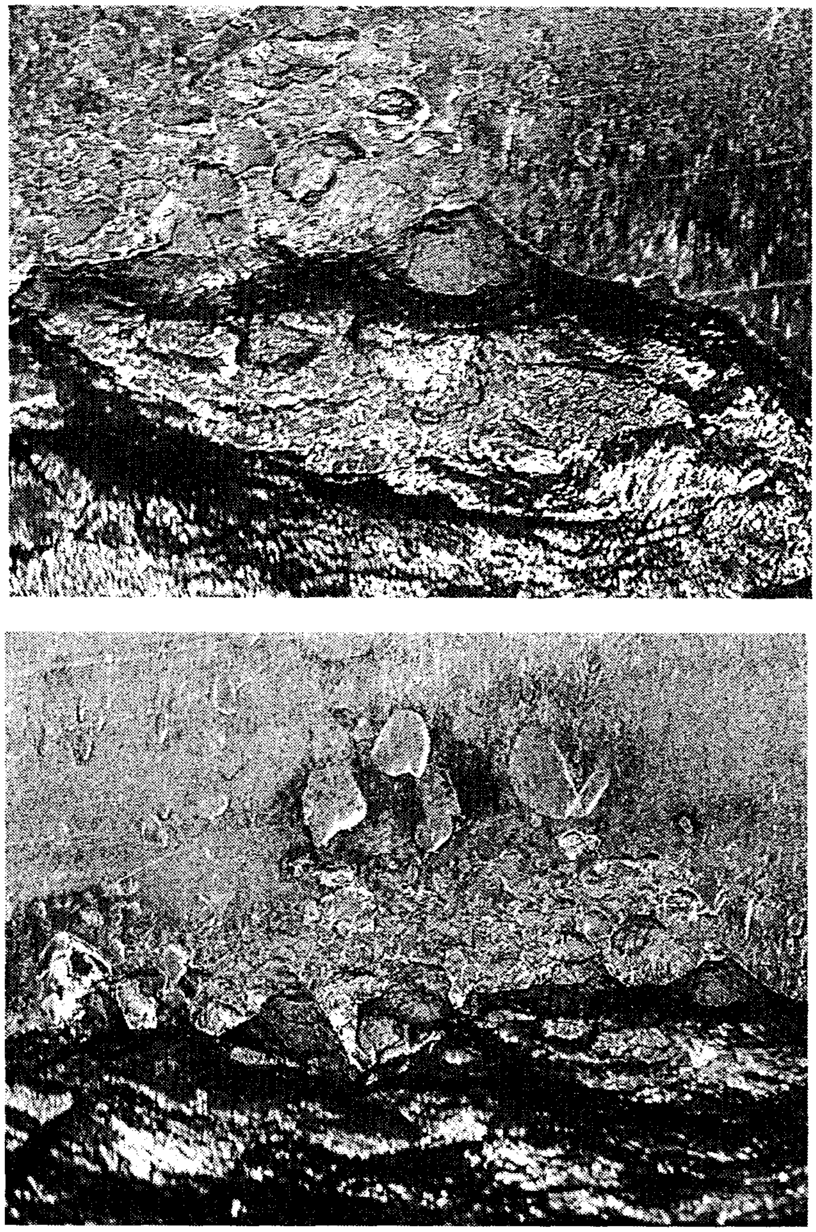


FIG.6 – FLANK INITIATED PITTING LEADING TO BENDING FATIGUE CRACK GROWTH.

Examination of the fracture surfaces was undertaken by using a Scanning Electronic Microscpe (SEM) with Energy Dispersive analysis by X-ray (EDX) for chemical analysis. This confirmed the visual and optical microscopy observations of fatigue initiated from below surface flank pitting. Close examination of the pitting and the main fatigue crack initiation sites did not reveal the presence of any unusual material defects such as large oxide inclusions that could account for the failure (FIGs 7 and 8).

In Figures 7 and 8, the lower image compares backscattered imaging with the normal secondary imaging in the upper image. In this imaging mode oxides would appear black.

It was therefore concluded that the material condition was not the primary cause of failure as it was within the specified condition used in the original design conditions and that no unexpected quality issues had resulted in a lower that expected material strength. The mechanism of failure was traced to a primary contact fatigue failure resulting in flank pitting that then led on to a secondary initiation of a bending (flank initiated) fatigue crack that ultimately resulted in the major tooth bending fatigue failure observed.

From these investigations, it was then necessary to identify what had resulted in gear flank contact stressing sufficient to result in contact fatigue pitting that had been identified as the primary cause of failure.

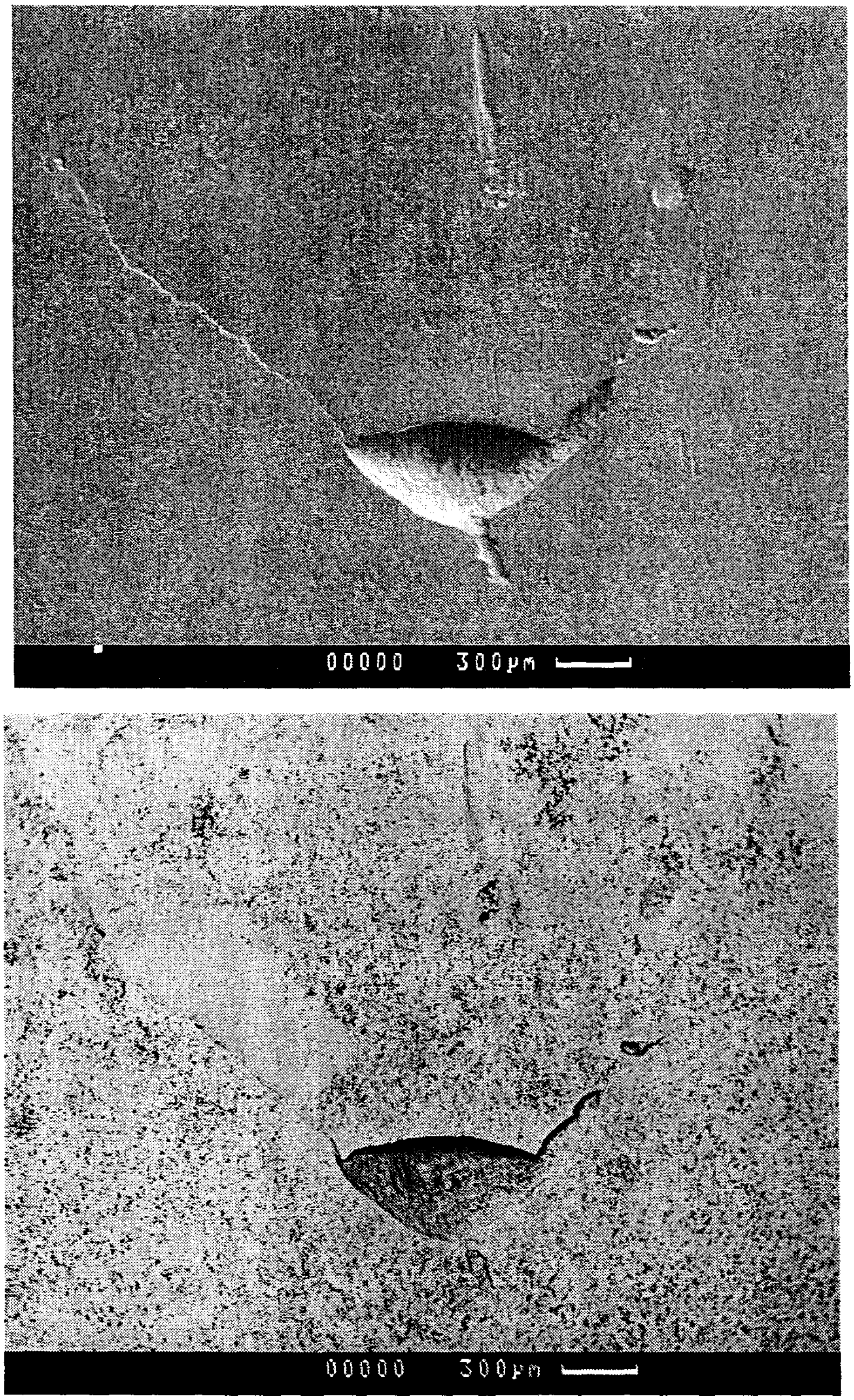


FIG.7 – SEM IMAGES SHOWING FLANK PITTING LEADING WITH ASSOCIATED CRACK GROWTH (TOP IMAGE IS SECONDARY ELECTRON IMAGING AND THE LOWER IS BACKSCATTERED ELECTRON IMAGING)

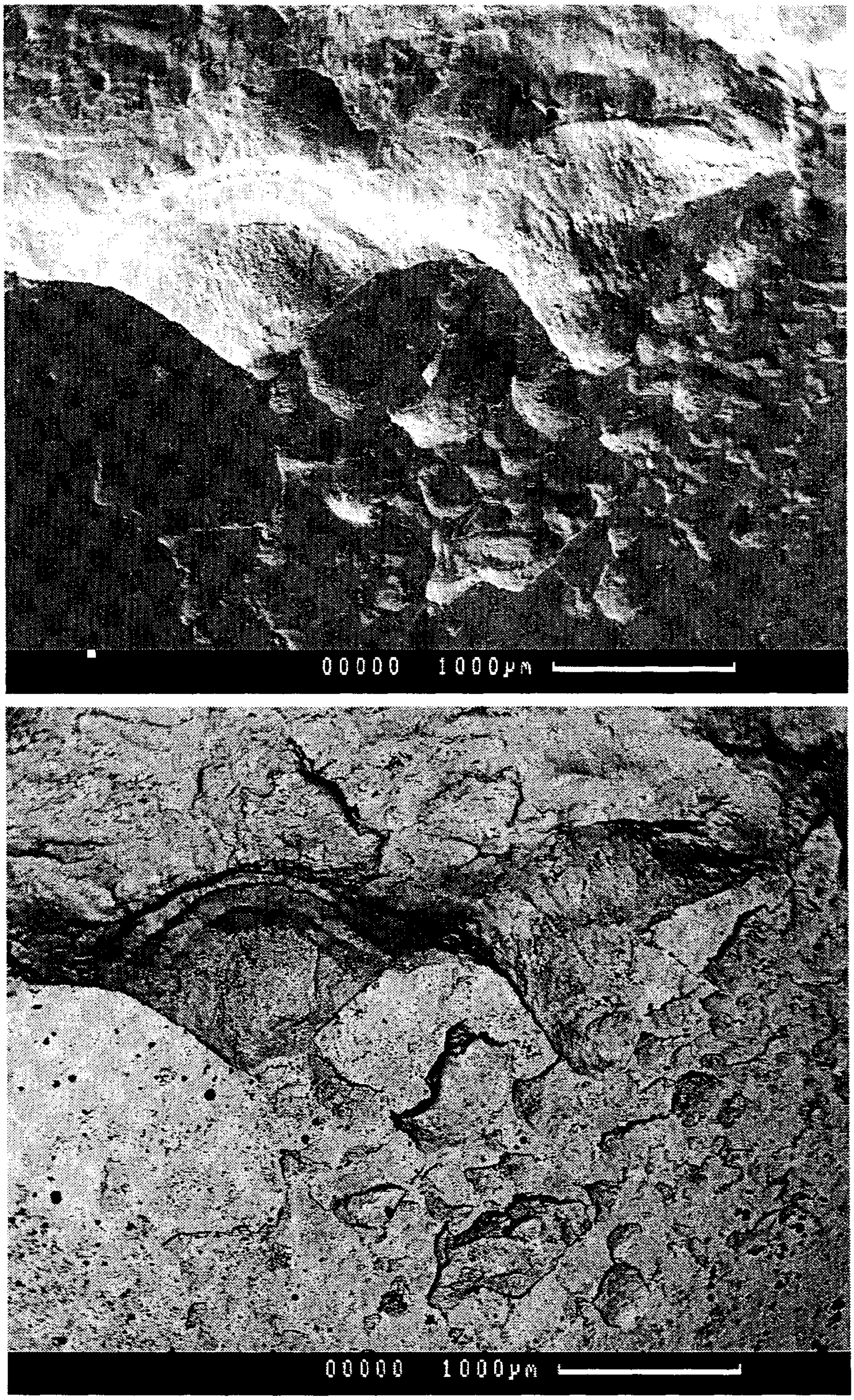


FIG.8 – SEM IMAGES SHOWING FATIGUE CRACKS GROWING FROM FLANK PITTING

CVS Operating Philosophy

To minimize fuel usage and engine operating hours CVS tended to operate on the minimum number of engines at all times. Starting with one engine driving through one gearbox with the other shaft trailing, then two engines, then three, then four. The tendency was to drive up to near maximum torque on each engine before adding the next one. Because of the design of the gearing, with each engine driving through separate drive trains onto the main wheel, each drive train and most crucially the teeth on the main wheel would therefore tend to see something near their maximum design loading for significant periods of time.

The situation was exacerbated by the fact that the maximum power design point was for all four engines in use. In any other mode, the engines that are driving are producing their power at lower speeds and therefore commensurately higher torques. In this situation, particularly with gas turbines, it is easier to introduce transients to the gearing. When accelerating or manoeuvring the vessel the risk of accidentally overloading the power train increases.

The situation is further exacerbated by the fact that torque is only measured at the output shaft. When driving on two engines, it is generally assumed that the inputs are balanced, but this was not necessarily the case. Gas turbine optimization could and did drift and if so, when at maximum two engine torque, one drive will inevitably be overloaded.

HMS *Invincible* Operating History

The operating history of *Invincible* was reviewed and compared with other ships of the class and the following were identified as potentially significant differences:

- Being the first of class, *Invincible* had accumulated considerably more operating hours than the other ships of the class.
- *Invincible* had seen more active service than other CVSs, when the greatest demands are made of the propulsion plants.
- Changing deployment patterns between the 1980's and 1990's, from predominantly anti-submarine warfare operations in the windy North Atlantic, to mainly fixed wing operations in the high temperature and low winds of the Gulf and Adriatic resulted in more high power use to achieve optimum aircraft launch.
- It was also found that undesirable transient over-torques had been experienced during operation.

Material History

The MoD and David Browns undertook a review of the material history of the gearing, which highlighted the following:

- In 1983 pitting was discovered on the main wheel, at the forward end of the forward helix. An investigation identified that this had been caused by the main wheel aft side plate bolts becoming loose, which caused a lack of radial support at the aft end of the rim resulting in a transfer of load to the forward end of the helices. Combined with pinion torsional wind-up and bending, the maximum load intensities were now all at the forward end of the forward helix resulting in pitting.
- Replacement of the side plate bolts arrested the pitting.
- In the early 1990's, slight progression of the pitting was noted which was probably caused by increased usage at high power.

- At some time during the mid/late 1990's cracks initiated from some of the pits, probably due to a small number of overloads. Once initiated, these progressed during normal operation to the extent discovered in late 1999.

Peer Review

A gearing peer review group, comprising the MoD and gearing experts from David Brown, Alstom Gears, VSEL and Design Unit met to review the design, operating history and nature of failure. They agreed that during design it would not have been realized that the configuration and way to achieve fuel economy would lead to high tooth loading for significant periods of time and that this situation would not have been recognized by the design codes of the time.

They concluded that the main wheel was the weak link in a CVS gearbox and that it appeared to have insufficient capacity for its current duty. In this case, the design was now considered to have been operating above its fatigue limits, confirmed by the fact that the failure initiated from what were minor defects.

The Repair

Several options were considered for the future of the ship.

1. Do nothing and live with the power restriction.
2. Replace the main wheel and mating pinions with new or used elements.
3. Replace the complete gearbox.

To restore full capability, replacement gears or a replacement gearbox would be required and following an initial feasibility study and discussions with customers, the decision was taken to fit replacement new gears.

Replacement Gears

The gearbox was designed in the late 1960's and manufacture commenced in 1972. Since then, there had been significant improvements in materials, manufacturing processes and gear technology and it was proposed to take the opportunity to increase the strength of the new wheel. It was also proposed to change the mating pinions, to reduce risk and to provide the opportunity to refine tooth profiles.

The selected main wheel design was a fabricated assembly, with the rim welded to the side plates, in preference to the original bolted interface, which had become loose and allowed the wheel to distort. The new rim was constructed from a stronger grade of 'clean' steel, manufactured to ensure fewer inclusions from which defects can initiate and was thicker than the original, to make it stiffer and to minimize flexure.

Whereas the original wheel was manufactured from a through hardened steel, the replacement teeth were surface hardened using tooth by tooth induction hardening, providing enhanced contact and bending fatigue strength. The result was actually a lower cost option than a replacement original design wheel.

Due to the quill shaft arrangement, the pinions require large holes down the full length of the shaft. This bore then facilitates relatively large shaft bending and torsional wind up as the full load torque is applied. These induced pinion shaft distortions were calculated using proven empirical techniques and added to those taken from Finite Element Analysis (FEA) of the gear wheel structure.

The calculated distortions and the tooth 'as manufactured' micro geometry was entered into DUGATES, a specialist 3D gear FE package developed by Design Unit. This software indicates the load distributions over the tooth surfaces (contact stresses) and tooth roots (bending stresses). Though the original tooth micro geometry had performed well, DUGATES showed that small modifications to the gear micro geometry could give significant stress distribution improvements.

The pinion teeth were also crowned to decrease stressing at the ends of the gear meshes and reduce the sensitivity to misalignment. Improved helix corrections assisted the crowning to give an improved load distribution along the full length of the face width. Revised tip and root reliefs would give smoother running as they passed through the mesh (FIG.9).

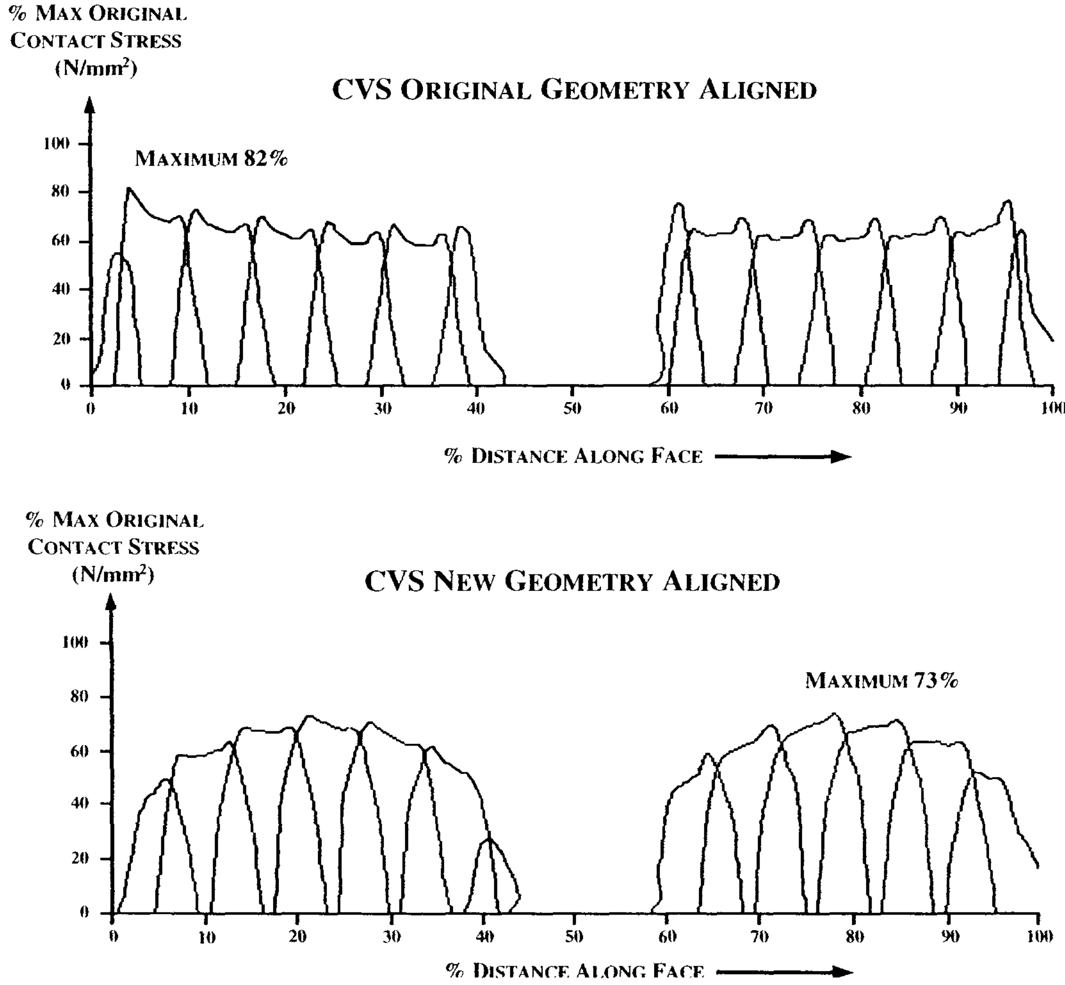


FIG.9 – TOOTH LOAD DISTRIBUTION FOR ORIGINAL AND MODIFIED GEOMETRIES.

The resultant safety factors for the replacement gears are shown in Table 1:

TABLE 1 – *Safety Factors to ISO 6336 — Contact and Bending at Maximum Power Direct Ahead*

	Original Gears Pinion case hardened Wheel Through Hardened	**Replacement gears** Pinion case hardened, crowned and helix corrected Wheel Induction Hardened
Pinion Contact	2.76	2.59
Wheel Contact	**1.38**	**2.00**
Pinion Bending	2.72	2.81
Wheel Bending	**1.79**	**2.12**

Planning the Replacement

At build, the Gearbox had been installed as a complete unit and a procedure for removing a main wheel had never been developed. The ship refitting contractor, Babcocks and David Brown were tasked with preparing a detailed proposal to cover the whole process of dismantling, re-assembling and setting to work the gearbox to achieve a main wheel and final drive pinion exchange. It quickly became apparent that it would be impractical to remove the gearbox as a complete assembly and the gearbox would have to be dismantled in-situ.

To visualise the sequence of the dismantling procedure, an existing wooden model of the gearbox was used. Once the original ship drawings had been updated to electronic format, more detailed work was completed using AutoCAD 3D to model options dynamically.

Before it would be possible to remove the main wheel, it would first be necessary to remove the other rotating elements and the 'A' frame that supports them.

Two options were considered for removing the main wheel from the ship. The first was a vertical lift, through two decks into the hangar, where the main wheel could be craned onto the dockside. The second was to cut a hole in the side of the hull, withdraw the wheel horizontally and then lift it onto the dockside. Due to the volume of electronic cabling in the compartments above the gear room, the decision was made to remove the wheel sideways.

Without cutting though the compartments above, cranes could not be used to lift the 'A' frame or main gear wheel. The solution devised was based on a lifting assembly (FIG.10) consisting of beams lifted by four hydraulic jacks and a stack of cans to provide four columns for the lifting rails to sit on.

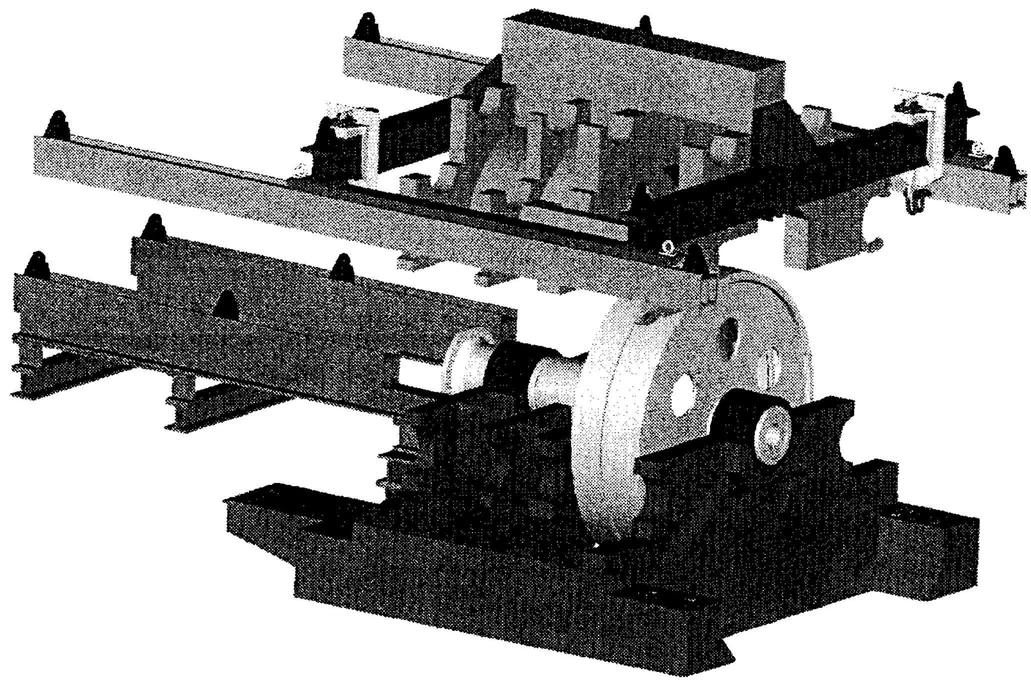


FIG.10 – 3D MODEL OF LIFTING ASSEMBLY

The four jacks would lift the beams and then cans would be placed between the seatings and the lifting rails and the assembly lowered onto the cans. Additional cans would be placed under the jacks and the process repeated until the lifting rails were at the correct height to allow the 'A' frame to be moved.

Due to the size and weight of the 'A frame, removing it from the compartment would have been difficult. Various options were considered including rotating the 'A' frame in a cradle to move it out of the way. The method chosen was to lift the 'A' frame clear of the main wheel, move it aft and then lower it onto a support structure fitted to the thrust block seating.

The same lifting system would be repositioned to lift the main wheel clear of the gearbox. The wheel would then be lowered onto cradles allowing it to be removed from the hull on rails supported on a structure of beams constructed on the dockside alter.

A risk reduction exercise was undertaken to ensure that the work would be completed to programme. This led to detailed procedures being written for the whole task, early inspection of existing removal equipment and ship visits to validate work in way.

The task was undertaken in the following sequence:

- Before the ship docked, work had commenced to remove all items in the way within the compartment, which included:
    - Two auxiliary boilers.
    - Salt water pumps.
    - Oil water separators.
    - Stabilizer power unit.
    - Air coolers.
    - All compressed air systems.

  At this point the hull was cut to create the shipping route for the main wheel and all specialist equipment.
- All ancillary components were stripped from the gearbox.
- A containment area was set up around the gearbox to minimize contamination.
- An optical sweep was undertaken by attaching pillars to specific target points on the 'A' frame and using lasers to measure the relative heights of target points in order to establish the structural alignment of the gearbox. These readings would be repeated during the rebuild to ensure that the affect of any residual hull stresses on the gearbox structure could be assessed. Lasers were also used to establish the alignment of bearing bores relative to one another and these checks would be repeated during the rebuild, to ensure that bearing bores were in line and that the bearing bores of mating gears were parallel.
- The support structures and specialist lifting equipment were installed by Babcocks and John Gibson Projects Ltd.
- The rotating elements were removed, for storage or overhaul.
- The opportunity was taken to remove the mercury filled thermometers fitted as secondary temperature indication to bearings, to reduce a recognized hazard, and to fit modified design transient brake seals, to replace the original unreliable seals.
- The lifting process commenced. The hydraulic rams lifted the whole assembly at the same rate. It was a slow, controlled process that continued round the clock (FIG.11).

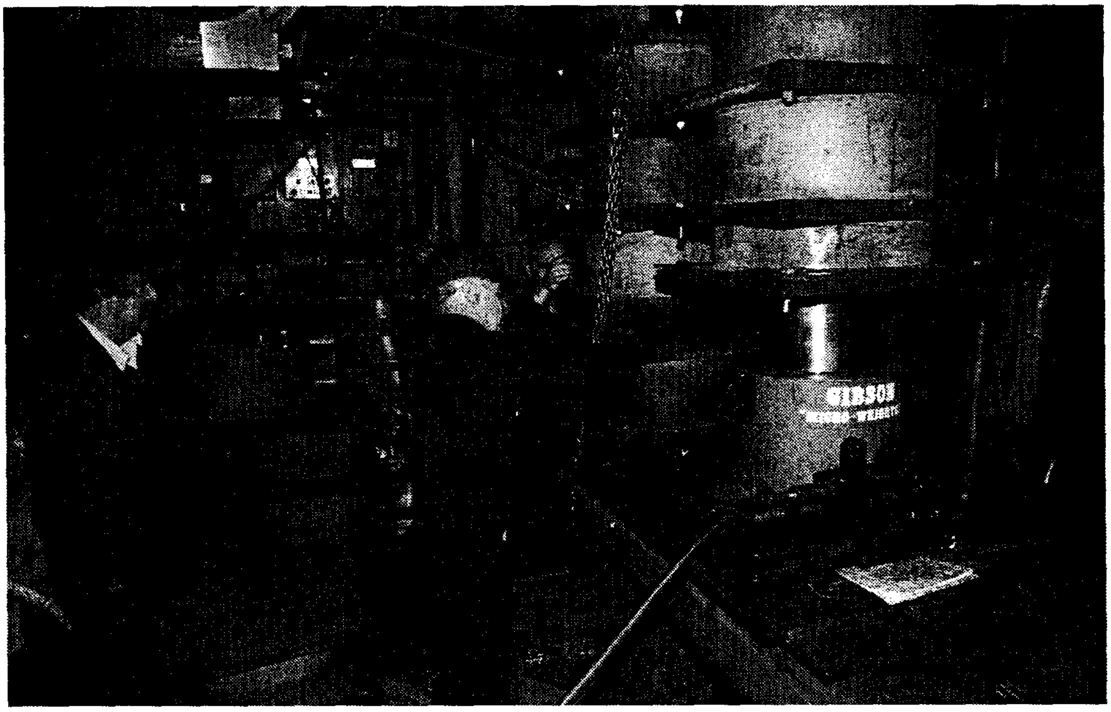


FIG.11 – THE HYDRAULIC JACKS

- It is worth noting that the gearbox is inclined inline with the propeller shaft, which slopes by 2.2 degrees. As a result, the 'A' frame had to be lifted perpendicular to the shaft line to prevent it fouling the main wheel (Fig.12).

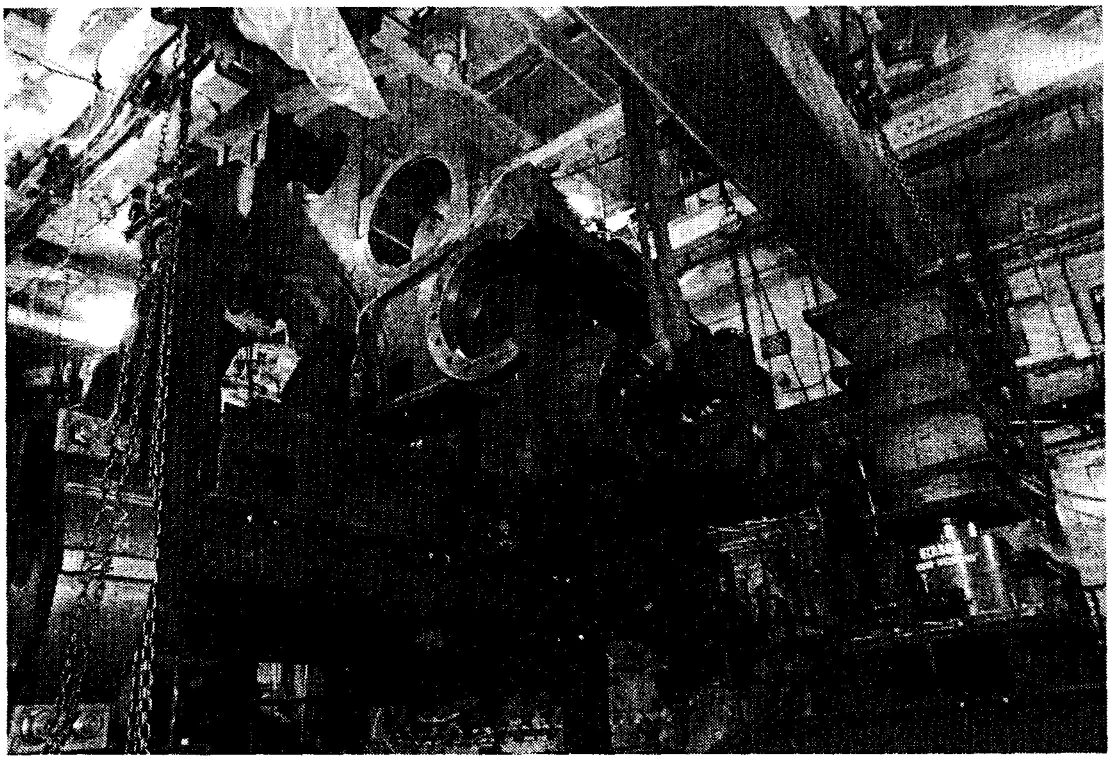


FIG.12 – LIFTING THE 'A' FRAME

- Once the 'A' frame had been lifted clear of the main wheel the lower set of rails were installed, to enable it to be moved aft and lowered onto the support structure created on the thrust block mounting.

- The lifting assembly was then repositioned to lift the main wheel, which was then lowered onto a cradle and withdrawn from the ship sideways, before being craned onto the dockside (FIG.13).

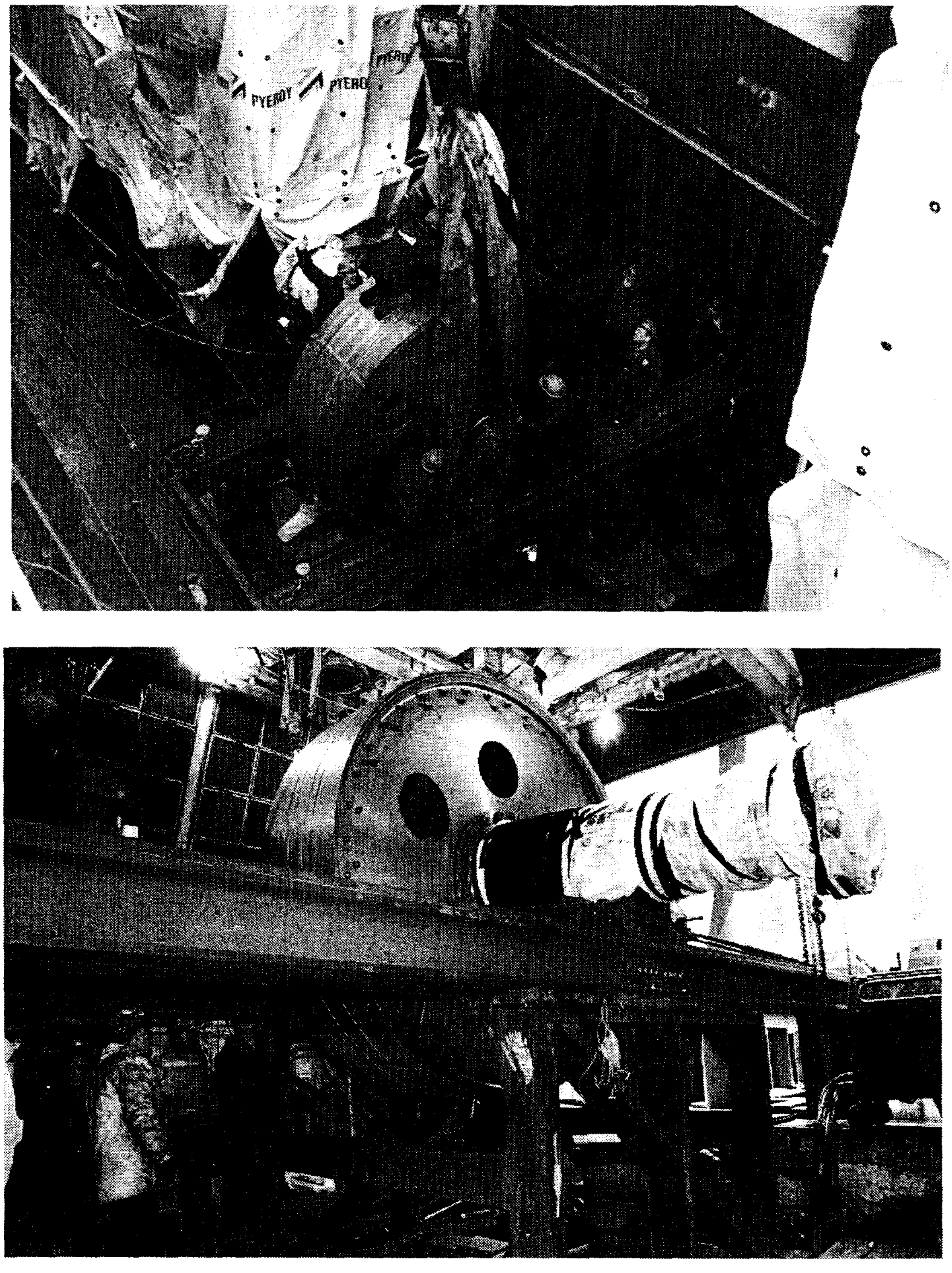


FIG.13 – FITTING THE NEW WHEEL

The delivery of the replacement gears and the gearbox rebuild were on the critical path of the refit, so particular attention was given to monitoring the supply of replacement gears, to ensure that they were completed on time. The gears were provided on time and the rebuild proceeded, with the above steps being undertaken in reverse order. Particular attention was paid to:

- The axial setting of the gears.
- The alignment of bearing bores.
- Bearing clearances.
- Cleanliness.

As a result of the detailed planning and the close working relationships developed between the MoD, Babcocks and David Brown the rebuild was completed to schedule and without any significant technical issues.

Setting to Work and Trials

Once the gearbox had been rebuilt, the lubricating oil system and gearbox were flushed, to remove any remaining contamination. The lubricating oil system was then statically retuned and gear spray patterns confirmed correct.

The trials package centred on a gradual build up in power in the different drive modes, to enable bearing temperatures and lubricating oil pressures to be analysed. After each test point, the gears were visually inspected, to establish whether any surface features were developing. Gear tooth contact markings were checked using blue dye which, was sketched and photographed after each stage. All tooth contact markings were satisfactory with no indication of any angular or axial misalignment. Noise levels and bearing temperatures remained at satisfactory levels throughout the trials.

Following sea trials in late 2002, *Invincible* returned to active service. The new gears continue to give trouble free service, confirmed by routine specialist visual inspections.

Future Work

Fatigue testing of samples taken from the failed gear are planned, to improve our understanding of the failure and to help influence design codes for future geared applications. This work will also confirm if the assumed material fatigue strength used in the design calculations is valid. Samples will also allow a fuller material investigation to be carried out than was possible from the smaller sections of failed tooth and this will allow other properties such as residual stress to be understood in more detail as well as changes that have taken place due to running of the gear at high stress.

Summary

The maximum stress levels that this gearbox was designed to were considered to have been adequate for the duty specified at the time, supported by the fact that the failure occurred late in the ship's life. However, with the current duty cycle, it is possible that the original gearbox design specification may have been modified.

The failure occurred because changes in the role of the platform, coupled with the need to reduce engine operating hours and achieve fuel efficiency, resulted in the main wheel seeing maximum design torque for much more significant periods of time than had been envisaged. Transient overloads that could have been experienced during acceleration or manoeuvring exacerbated the situation.

This article has drawn attention to the fact that reliability and duty require sufficient consideration and margin to be incorporated at the design stage.

Conclusions

The causes of the *Invincible* main wheel failure are now well understood and this knowledge will help with considerations for future MoD geared applications.

As a result enhanced visual inspection techniques have been developed and these are now being deployed within MoD gearing applications.

This has highlighted the importance of understanding the operational duty (and more importantly the potential changing duty) of main propulsion gearboxes.

For new gearboxes, high power density should not be achieved at the expense of properly considering appropriate reliability, duty and adaptability margins.

The timely completion of the wheel change and trouble free trials clearly demonstrated the benefits of the detailed planning undertaken by the MoD, Babcocks and David Brown Textron, which included new computer aided techniques to simulate the dismantling and rebuild procedures.

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