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Naval thrust bearings

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SYNOPSIS

Marine thrust bearings have been developed for naval use from the earliest days of screw propulsion, through multi-collar thrust blocks to the present generations of tilting pad bearings. Following the introduction of the Michell thrust bearing to marine applications in the years around the First World War, there was a long period of relative stability in bearing technology. During this time there were few design changes of major interest. In the past 20 years, however, a much wider range of choices have become available. Bearing casings, although retaining similar internal arrangements, are now designed to suit different shipboard machinery layouts. The increasing requirement for submersible vessels to operate continuously at depth at very slow speeds, has led to methods for enhancing the load carrying capacity of bearings in a situation which is unfavourable for hydrodynamic lubrication. Lubrication systems themselves have been developed with self-contained thrust bearings becoming a realistic choice in some cases. Possible future bearing developments include the use of active magnetic bearings to absorb at least part of normal thrust loadings.

HISTORICAL

The nineteenth century pioneers of screw propulsion quickly discovered that there must be adequate provision within the hull of a vessel to absorb the reaction of the propeller thrust. John Bourne's book on the screw propeller, published in 1852, gives considerable attention to various thrust devices. Figure 1 shows the arrangements in the single screw vessel, HMS *Ajax*. To quote the author: 'The thrust of the screw is received upon a cast iron upright applied to the end of the shaft for that purpose'.[1] Notice that the astern thrust is taken at the end of the shaft aft of the propeller, where best practice was to fit a disc with lignum vitae segments. Such simple bearings on the ends of propeller shafts were soon found to be inadequate and were replaced by multi-collar thrust blocks, in either enclosed or open horseshoe form as shown in Fig 2. These multi-collar systems formed the new standard until the advent of single collar, tilting pad thrust blocks during the First World War.

The horseshoe type of multi-collar thrust with provision for independent adjustment of the shoes was a fine piece of engineering. However as speeds and powers advanced, the multi-collar thrust found it increasingly difficult to accommodate the heavier loads it was called upon to bear. The solution was to provide more and more load bearing surface until, in the case of at least one merchant liner, no less than 22 collars were required in the design. As we now know the conditions for hydrodynamic lubrication were far from ideal and continual adjustment was necessary to ensure an even distribution of load between the collars. Power losses were considerable, with enormous amounts of heat being generated at the bearings. Continuous wear of the bearing elements ensured that frequent replacement was essential.

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The introduction to marine use, at about the turn of the century, of the direct drive steam turbine temporarily eased the situation. In these turbines practically all the thrust was balanced by steam pressure on a dummy piston. However with the later development of geared turbine propulsion systems around 1912, thrust block problems re-appeared in aggravated form. According to J H Gibson,[1] marine engineers of the day were 'at their wits' end' for a solution until it was realised that the remedy was at hand in the form of the single collar, tilting pad thrust bearing which we know today.

A G M Michell had taken out patents covering his tilting thrust pad invention in 1905. In the same year his solution to Reynolds' equation which forms the basis of the invention was published in a German mathematical journal.[2] The first published description in this country of a Michell bearing was given by G B Woodruff in a lecture to the Institute of Marine Engineers in October 1908.[3] The discussion which followed

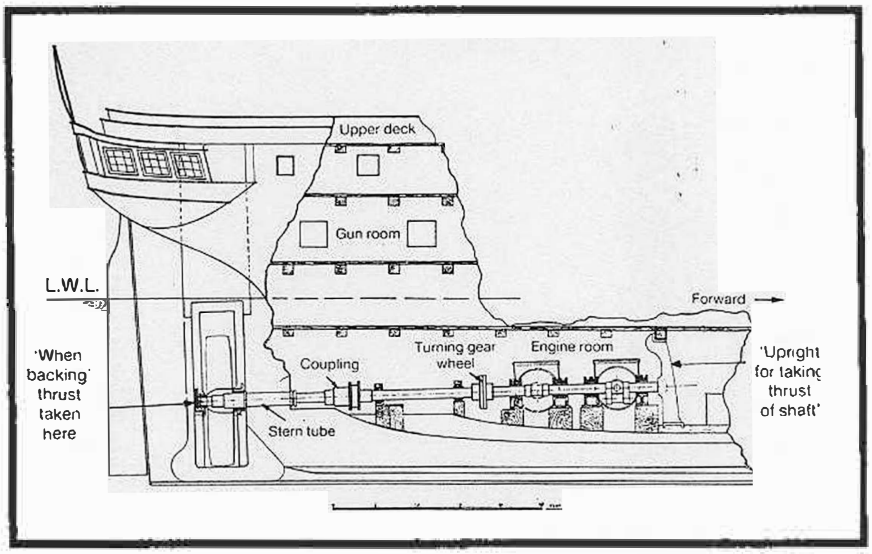

Fig 1: Screw steamship HMS *Ajax* 1848; method of receiving thrust of screw propeller in either direction

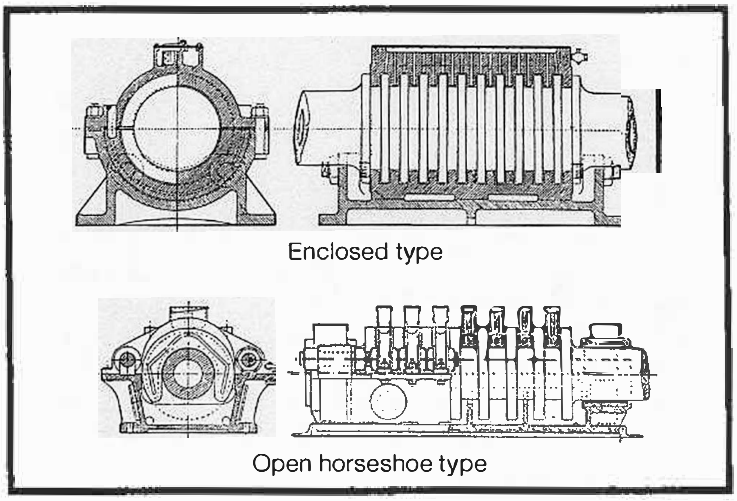

Fig 2: Multi-collar thrust blocks

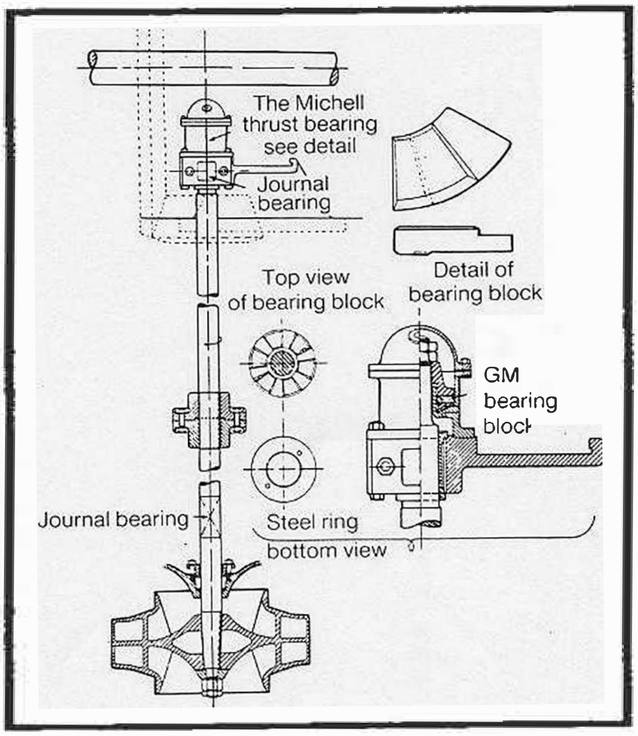

Fig 3: Michell vertical thrust bearing for a centrifugal pump installed at Cohuna on the Murray River, Victoria, 1907

Woodruff's lecture is recorded and gives some idea of the enormous scepticism with which marine engineers approached the revolutionary idea of replacing large, satisfying, multi-collar thrust blocks with what must have seemed a ridiculously small single collar device.

All the early Michell bearings applications were for industrial rather than marine use. Figure 3, reproduced from Woodruff's 1908 Institute of Marine Engineers lecture, is of a centrifugal pump fitted with a Michell thrust bearing in 1907 as part of the Murray River hydro-electric scheme. Some experimental work was started by C A Parsons in the early part of 1912 with a view to testing the suitability of a tilting pad thrust bearing for marine use.[4] Such was the success of these experiments, and of the first vessels incorporating Michell thrust blocks launched in 1913 and 1914, that the new bearings were soon adopted wholeheartedly and universally. The first ship of the Royal Navy fitted with tilting pad bearings ran her trials in August 1914. During the First World War Michell bearings were fitted to propulsion machinery totalling 10M shaft hp, a figure which reflects the massive shipbuilding effort of the time. As an example, no less than 280 destroyers were launched from British Yards during the First World War.[5]

Figure 4 gives a good idea of the state of the art for naval thrust bearings by the end of the First World War. The shaft is carried by two journal bearings incorporated in the casing. The casing is foot mounted and split on the horizontal centre line. The lower half is in one piece; the upper half in three pieces to allow independent access to the journals. The tilting thrust pads have spherical pivots acting on hardened inserts set in massive retaining rings which are themselves spherically seated. The spherical pivots did not provide entirely satisfactory results,[1] and were soon abandoned in favour of line pivots which are still in use for most present day applications.

Apart from minor developments, the configuration shown in Fig 4 remained relatively unaltered for naval vessels for something like 50 years. In the last 20 years, however, there have been a number of important developments in bearing design. The reasons for these developments are various. They include increasing sophistication in calculation and computing techniques giving more reliable predictions of bearing performance; the increasing need for bearings to economise on space and weight, particularly in submarines; improved manufacturing techniques; and changes in bearing operational requirements. The purpose of the rest of this paper is to highlight some of the major recent trends in naval thrust bearing design

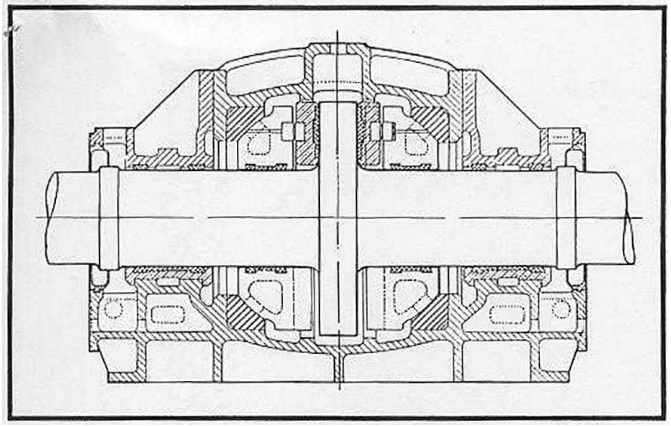

Fig 4: Typical naval thrust bearing about 1918

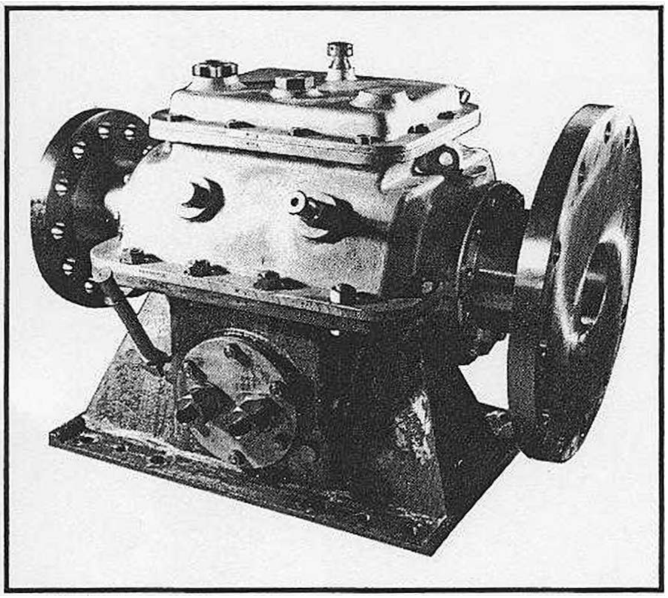

Fig 5: Hunt class minesweeper thrust block, 1970

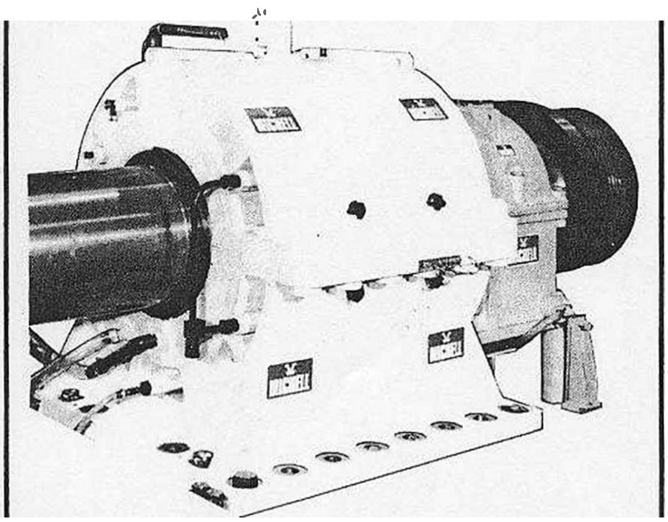

Fig 6: HMS *Invincible*; thrust block on test bed in maker's factory, 1979

for vessels currently afloat and for some of those coming into service in the next decade or so.

BEARING STRUCTURES

The thrust bearing designed in the early 1970s for the Hunt class of minesweepers (Fig 5) can be traced back to the immediate post First World War era. The casing is cast although the materials, having regard for the nature of the vessel, are gunmetal for the lower half and aluminium for the upper half respectively. Internally the double thrust arrangement with journals fore and aft of the collar is very similar to that of the bearing shown in Fig 4 although on a much smaller scale. By contrast, Fig 6 is a photograph of the thrust block supplied for the light aircraft carrier HMS *Invincible*, and her sister ships. Internally the double thrust arrangement is as in earlier bearings although in this case the overall design is made simpler by the shaft being supported by journal bearings which are sited separately from the thrust block. The most serious major development embodied by the bearing shown in Fig 6 is the switch from a cast casing to one which is fabricated. This change, made possible by modern welding and non-destructive testing techniques, is reckoned to provide in excess of a 20% weight advantage compared with a casting of comparable strength.

The casing structure is built around three concentric cylinders. An outer cylinder spans the length of the bearing while two shorter, smaller diameter ones cover the seal areas at each end. A series of ribs radiate from the inner cylinders to the outer skin. This arrangement shown in diagrammatic form in Fig 7 has proved immensely strong and has been widely used for subsequent freestanding naval thrust bearings. The same basic structure is used, for example, for thrust blocks installed in the Dutch Walrus and HMS *Upholder* classes of diesel-electric powered submarines. In a modified and slightly simpler form the casing structure has been employed in the USS *Avenger* class of mine counter measure vessels and in the US Navy TAO-187 class of fleet replenishment ships. Further variants of the same arrangement have been supplied for the current UK and French strategic submarine building programmes.

The bearings shown in Figs 4, 5 and 6 are all designed with a foot mounting which is bolted to a horizontal surface well below the centre line of the shaft. A common alternative, used for example in the US Navy TAO-187 class of ships and in certain submarines, is for the bearing casing to be secured to the vessel by flanges which are extensions horizontally of the mating surface of the lower half casing with the top. Such centre line flange mounting arrangements provide for a very stiff structure and are particularly suitable when overhead space is limited.

A new design variant which may become popular and be appropriate in some instances is for the bearing casing to be located on a vertical bulkhead as shown in Fig 8. This mounting arrangement is to be employed in the forthcoming class of Australian A471 submarines and in the Osprey class MHC 51 minehunters for the US Navy.

The main structural material for modern naval thrust bearings is high quality steel although other materials can be used. The bearings of minesweeping vessels were traditionally made from cast gun-metal with aluminium for the unloaded parts. A present day alternative is to use fabricated aluminium plate throughout for loaded and unloaded parts alike. There is some evidence that designs incorporating low magnetic materials will be sought in future for vessels other than minesweepers to assist the degaussing process.

OPERATING REQUIREMENTS

Hydrodynamic thrust bearings have proved enormously reliable in naval use. For surface ships the application is an ideal one in that there is no load at zero speed and the load increases with speed in line with the ability of the bearing to form an efficient and relatively thick hydrodynamic film. In the early years of tilting pad bearings typical maximum specific thrust loadings were about 2 MPa. Through experience and improved computer based analytical tools this figure has been gradually increased over the years to be in the region of 4.2 MPa.

In the case of submersible vessels the situation is more complicated. At or near the surface, the bearing experiences conditions similar to those of surface ships, ideal for the development of a thick hydrodynamic oil film separating the thrust collar and thrust pads. At depth, however, hydrodynamic thrust forces are augmented by hydrostatic forces acting on the propeller, external to the pressure hull, leading to increased loading of the bearing. If subsurface speeds are low, it is possible for the hydrostatic forces to be considerably larger than any propulsive loading. These conditions of high loads and very low speeds are not ideal for hydrodynamic lubrication and when this situation immediately follows a period of high speed operation then there may be a particular risk of failure. The reason for this is that the period of high speed running will have led to high temperatures in the bearing giving rise in turn to reduced oil viscosity and thermal distortion of the thrust pads.

Two techniques, which can be used separately or together, have been developed to overcome this problem. One approach is for oil under very high pressure to be introduced between the thrust pads and the collar, from a secondary lubrication system, to form a temporary hydrostatic bearing. The way this can be effected, using a dumb-bell connection into each thrust pad, is shown in Fig 9. The principal drawback of this arrangement, apart from increased cost, is potential operating noise associated with the high pressure pumps and motors of the secondary system. It can also be argued that the hydrostatic or 'jacking' system introduces an extra element of potential mechanical unreliability into the bearing. However, it can be pointed out that the very existence of the secondary lubrication system is precisely to enhance the overall reliability of the total operation. The second approach is to employ a multi-part thrust pad, also shown in Fig 9. The three piece construction allows component distortion due to thermal deflection to be balanced by bending due to load. As a result the thrust pad surface area remains approximately flat, thus reducing the possibility of failure.

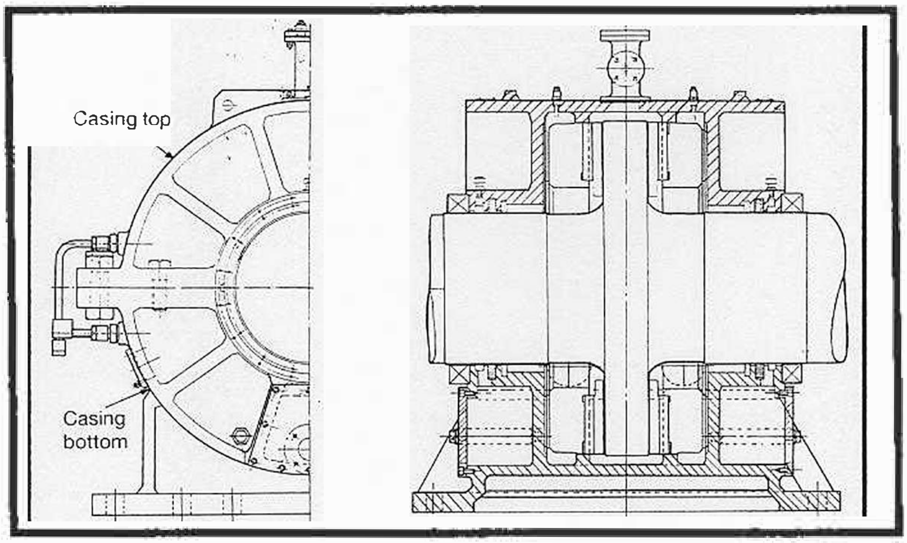

Fig 7: Concentric cylinder casing structure of a modern naval thrust bearing

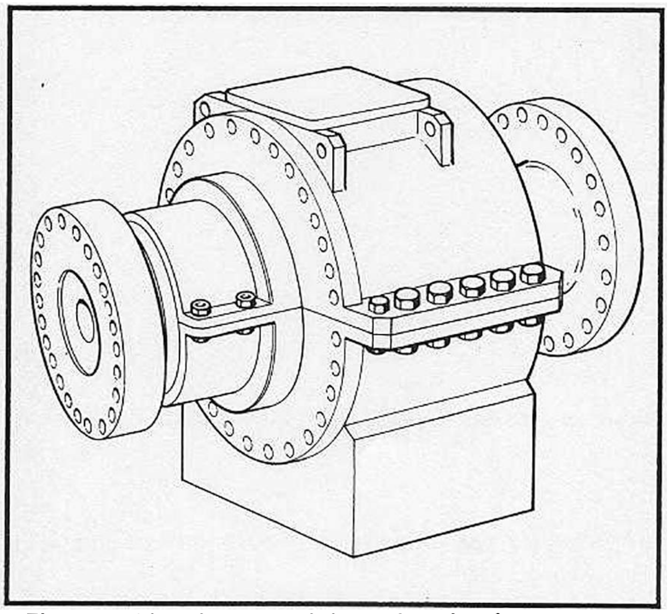

Fig 8: Bulkhead mounted thrust bearing for naval use

LUBRICATION SYSTEMS

Most naval thrust bearings are designed to be operated with the working parts immersed in oil which is supplied continually to the bearing casing from an external source which is also feeding other items of ship's machinery. Conventionally oil is supplied to the bearing under moderate positive pressure via an inlet in the lower part of the casing. Lubricant returns to the vessel's general reservoir and cooling system via an outlet in the upper half casing.

In some cases, for example in diesel-electric submarines and smaller surface ships, it is now possible to design for self-contained lubrication in which the lubricant used is retained solely in the bearing and cooled by water tube coolers built into the casing. In these cases oil is circulated by means of the thrust collar which rotates through an oil sump created by the casing bottom. An oil scraper located in the top half of the casing removes the oil from the periphery of the thrust collar and directs it to the thrust pads in the upper half of the bearing (Fig 10). The configuration of the scraper is such that the distribution of the oil to forward and aft thrust surfaces (loaded and unloaded faces) is matched to their requirements. The benefits of this arrangement are that the system is automatic, robust and needs no external system to support it other than a supply of cooling water.

For a number of years it has been a common design practice

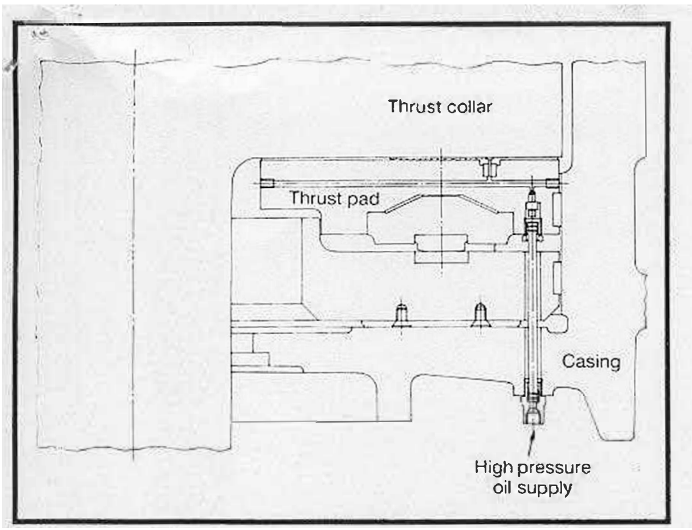

Fig 9: High pressure jacking and three piece thrust pad construction

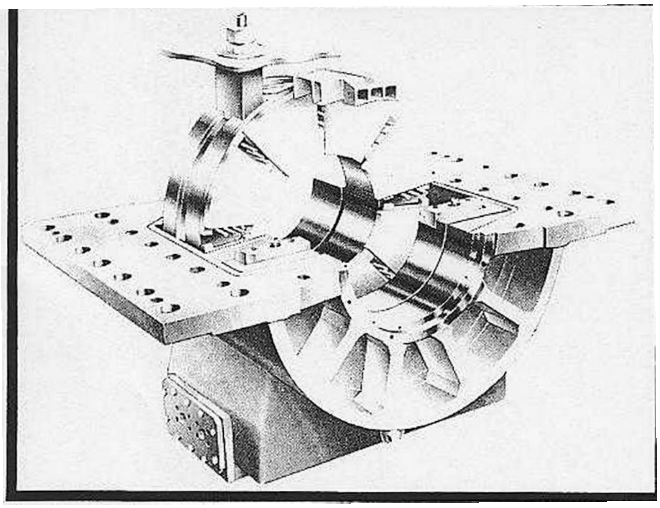

Fig 10: Self-contained thrust bearing showing oil scraper location in upper half casing

for ships with CODOG or CODAG machinery layouts to incorporate the main propulsion thrust bearing in the marine gearbox. This has been the case, for example, for Type 21 and Type 22 frigates and Type 42 destroyers. In each of these vessels a conventional flooded lubrication arrangement is used similar to that described earlier. The latest generation of Anti-submarine warfare frigate, the Type 23 Duke class, has a CODLAG machinery configuration.[6] In this case the self-contained operation described above is combined in an innovative arrangement with a low pressure circulation system to provide the vessel with two distinct modes of thrust bearing operation to suit the role of the vessel.

When the ship is in a sprint condition the main thrust bearing and radial bearings, located at the aft end of the casing, are lubricated from an external low pressure oil source taken from the main supply to the gears. If propulsion is by means of the main electric motor, however, the gear train is decoupled, the main lubrication system turned off and a secondary self-contained lubrication system takes over. The construction of the gearbox is such as to provide an overflow weir in the vicinity of the thrust collar and create the necessary oil sump through which the thrust collar rotates. Oil on its periphery is transferred to the upper part of the casing where a scraper similar to that described earlier distributes the oil to the working surfaces. In this mode cooling of the oil is simply by radiation via the gearbox surface. It follows that when the vessel is in its ASW role, the bearing, needing no external lubrication resources, contributes to reducing the overall noise signature of the vessel.

FUTURE DEVELOPMENTS

The history of naval propulsion since the end of the Second World War has been one of increasing attention to the reliability and acoustic characteristics of machinery. One consequence of the changing nature of naval operations has been to give increased importance to low speed conditions. Hydrodynamic bearings are not themselves significant sources of noise and vibration but their role as transmitters of disturbance is likely to receive more attention. In the past and in some present cases 'resonance changing' systems in which the whole bearing is supported by a tuned hydraulic damping arrangement[7] have made a significant contribution to vibration reduction. It has been suggested that there may be a future similar role for active magnetic bearings in marine applications.[8] While there are now a significant number of electromagnetic bearings in industrial applications, the specific pressures required for marine use and naval reliability requirements mean that magnetic bearing and damping systems are likely to find a place in parallel, and in support of, existing hydrodynamic systems rather than as a substitute for them.

The continuing development of super conducting generators and motors suggests future interest in an all electric frigate with a final drive which allows the prime mover to be divorced from the main propulsion shaft. The removal of the need for a gearbox could mean the reintroduction of free standing thrust blocks for the majority of vessels.

The reliability of hydrodynamic thrust bearings under a wide variety of operating conditions and shock loadings is well attested by experience and these bearings are likely to remain the main way in which propulsive thrusts are absorbed for the foreseeable future. There will, however, be room for improvement and change and a number of current trends giving an increase in the design options available have been described in this paper. These include moves towards a wider use of low magnetic materials, more variety in bearing mounting arrangements to suit new ship designs and an increasing role for self-contained lubrication arrangements.

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