

## 推進軸系の 舉動에 關한 研究 現況

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The Present Status of Study on the Behaviour of the Line Shafting

### Abstract

Past and present developments in the alignment procedures of the line shafting are discussed. It is shown that the effects of the propeller hydrodynamics and of the oil-films, built up in the journal bearings, can essentially change the static alignment. However, the inclusion of these phenomena in the alignment is not yet standard.

Also the dynamic effects of natural frequencies and vibratory excitations are not yet accounted for in the general case, although also these aspects can influence the safe operation of the line shafting. Therefore a plea is made to develop a more complete and thus a more rational alignment procedure.

### 1. Sketch how the past transferred into the present

The behaviour of the line shafting is importantly influenced by its alignment. With the alignment for a given propulsion shafting the well-or malfunctioning of the bearing bushes is determined, the rate of the wear down of the bushes, the effective support of the shaft, the natural frequencies of the lateral vibrations, etc. Therefore the alignment is the procedure to guarantee a trouble free operation of the line shafting during a certain period and many research

### 概 要

軸系配置方法에 關한 옛날과 오늘날의 發展에 關하여 論하였다. 프로펠러의 流體力學의 影響과 자아널·베어링에서 形成되는 油膜의 影響이 靜的配置를 根本적으로 變化시킬 수 있음을 提示하였다. 그러나 軸系配置에서 이러한 現象들을 包含시키는 것이 아직 標準化 되지 못하고 있다.

또한 固有振動數와 起振力의 動的效果는 비록 이러한 事項들이 軸系の 安全運轉에 影響을 미칠 수 있음에도 不拘하고 아직은 通常의 配慮되고있지 않다.

그러므로 좀더 完全하고, 따라서 좀더 合理的인 軸系配置方法을 開發할것을 提議하는 바이다.

### 1. 옛날의 方法으로부터 오늘날의 方法에로의 變遷

軸系の 舉動은 配置狀態에 의하여 重大한 影響을 받는다. 주어진 推進軸系에 대한 配置에 따라 베어링붓슈의 良好한, 또는 不良한 動作이 決定되고 붓슈의 磨滅率, 軸의 效果的인 支持, 橫振動의 固有振動數 등이 決定된다. 그러므로 配置는 一定期間中에 있어서 軸系の 故障없는 作動을 保障하는 길이며 많은 研究가 最適配置를 實現하기 위하여 行하여져 왔다.

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is devoted to realize an optimum alignment.

The present status of the studies in this respect is the product of the past and therefore a small look back into the past may help to describe better the present.

The shaft alignment in the past was done very simply. The most straightforward method was taken to align the shafting system: the straight line. Easily be defined, easily to be justified, easily to be performed.

But it could be done along that line because the transmitted power was low, the rate of shaft revolutions rather high, so that rather slender and flexible intermediate shafts resulted. Also the propellers were rather light due to the low power they had to deliver.

This may sound a good reason but maybe it is not the main reason. Maybe the lack of technical possibilities is a much better reason. At that time one did not by far have the instrumentation we have nowadays available. Although the economic requirements of a safe and optimum operation must have existed also in those early days, these requirements could not be met to the degree they can be met now.

With the increase in installed power on the shaft the need of a more sophisticated alignment became necessary. From the straight line one has to turn over to the curved line. Measuring and calculation techniques were improved to realize this. Because of these improvements a more optimum alignment became possible but as another result also the requirements became more severe. And so on.

And this spiral of improving capabilities and sharper specification is still existing. Continuously forcing the designers in more and better in shorter time and at lower cost.

More specifically we see that terms like "rational alignment" and "judicious distribution of bearing loads" were born in the sixtieths of this century. [1] In order to be able to predict and to guarantee any safe and reliable operation, the precise operational condition of the line shafting has to be known beforehand. This precise condition is given by the conditions between the

이 方面에 있어서의 研究現況은 過去의 研究結果에 의한 所産이며 따라서 옛날을 잠깐 回顧하는 것은 現況을 좀더 잘 理解하는데 도움이 될 것이다.

過去의 軸系配置는 아주 單純하였다. 가장 單純明瞭한 方法, 即 直線으로 軸系를 配置하는 方法이 採擇되었었다. 定하기 쉽고 고치기 쉬우며 일하기도 쉽다.

그러나 이것은 傳達하는 出力이 작고 軸의 回轉率이 높으며 따라서 어느정도 가늘고 柔軟한 軸으로 되었기 때문에 그러한 方針에 따라 施行할 수 있었던 것이다. 또한 프로펠러도 傳達해야될 出力이 낮았기 때문에 어느정도는 가벼웠었다.

이것은 그럴듯한 理由이지만 根本的 理由는 아닐 것이다. 아마도 技術的인 能力이 없었던 것이 훨씬 그럴만한 理由일 것이다. 當時에는 우리가 오늘날 利用할 수 있는 計器를 전혀 갖지 못하였다. 安全하고 最適의 作動에 대한 經濟的 要求가 이 當時에도 있기는 하였으나 이러한 要求事項들을 오늘날 滿足시킬 수 있는 程度로 들어줄 수 없었다.

軸에 걸리는 出力이 增加함에 따라 좀더 高尚한 配置가 必要하게 되었고 直線配置로부터 曲線配置로 바꾸지 않을 수 없었다. 計測技法과 計算技術이 이것을 實現할 수 있을 程度로 改善되었다. 이러한 改善德澤으로 좀더 適切한 配置가 可能하게 되었으나 또다른 結果로서 이에 대한 要求가 더욱 까다롭게 되고 이러한 狀況이 交互로 反復되었다.

그리하여 이러한 改善의 能力和 좀더 嚴한 仕様에 대한 螺施狀 進行過程이 아직도 繼續되고 있다. 設計者로 하여금 짧은 時間에 더 많은 것을, 더 좋게, 보다 낮은 값으로 逐行 하도록 끊임없이 督勵하고 있다.

"合理的 配置", "베어링 荷重의 慎重한 配分"과 같은 낱말이 今世紀의, 60年代에 나타났음을 매우 뚜렷하게 알고 있다. [1] 安全하고 信賴性있는 動作을 豫測하고 保障할 수 있기 위해서는 軸系의 正確한 作動狀態를 미리 알고 있지 않으면 안된다. 正確한 狀態는 軸系의

journals of the shafting and the bushing material of its bearings. Only in case the bearing loads can be kept within certain bounds for all normal operating conditions of shaft and ship, one can guarantee the safe operations of these bearings and thus of the shaft.

The rational alignment is generally subdivided into 3 parts:

- the static alignment
- the stationary running alignment
- the dynamic alignment.

In normal practice the static alignment is applied, the stationary running alignment as well as the dynamic alignment are not commonly applied. Partly because they are not yet familiar, partly because of many uncertainties in these techniques which requires still research.

In this paper some thoughts are spent to this field of research in the alignment of the shaft. In Appendix A a proposal is given as produced some years ago by one of the technical committees of ICMES, the International Cooperation on Marine Engineering Systems. This proposal considers the standardization of terms and symbols as used in modern shaft alignment.

For the safe and reliable operation of the line shafting it is also important to keep the vibration level low. Generally this means that resonances have to be avoided. For the lateral shaft vibrations this can be done by a rational shaft alignment, accounting also these aspects. For the torsional and longitudinal vibrations the alignment has no effect and other means have to be used.

From the very beginning of engine propelled ships torsional vibrations have plagued the designer. By trial and error first but later on by increased understanding and by better calculating and measuring techniques, the marine engineers have succeeded to arrive at accurate and safe design procedures. It may be stated that torsional vibrations in the line shafting does no longer need fundamental research.

This does not hold for the lateral and longitudinal vibrations. In the fiftieths a

자아닐과 그의 베어링 붓싱材料間的 條件에 의하여 주어진다. 베어링荷重을 軸과 船舶의 모든 正常的인 作動狀態에 대하여 特定範圍以內로 維持할 수 있을 경우에만 이러한 베어링들, 나아가서는 軸의 安全한 作動을 保障할수 있다

合理的인 配置는 通常 3가지 境遇로 나누어진다.

- 靜的 配置
- 定置的 運轉에 대한 配置
- 動的인 配置

普通의 경우에는 靜的인 配置가 適用된다. 動的인 配置뿐만 아니라 定置的 運轉에 대한 配置도 普遍的으로 適用되고 있지 않다. 한편으로는 그들이 아직 生疎하기 때문이고 다른 한편으로는 이러한 技法들에는 아직도 究明할 必要가 있는 많은 不確實한 事項들이 存在하기 때문이다.

이 글에서는 軸配置에 있어서의 이 分野의 研究에 대하여 若干 考察하여 보기로 한다. 附錄A에 ICMES(The International Cooperation Marine Engineering Systems)의 한 技術委員會에 의하여 數年前에 發表한 提案을 그 대로 引用하였다. 이 提案은 最近의 軸系配置에서 使用되고 있는 用語와 記號의 標準化를 考慮하였다.

軸系の 安全하고 信賴할만한 配置를 위해서는 振動程度도 낮게 維持하는 것이 重要하다. 一般的으로 이것은 共振狀態를 避하여야 하는 것을 뜻한다. 軸의 橫振動의 경우 이러한 狀況들을 配慮한 合理的인 軸配置를 함으로써 共振狀態回避를 達成할 수 있다. 비틀림振動과 縱振動에 대하여 軸系配置는 影響을 미치지 않으며 다른 手段을 講究해야한다.

船舶을 機關으로 推進하기 始作한 바로 그때부터 비틀림振動은 設計者를 괴롭혀왔다. 처음에는 試行錯誤法에 의하여, 나중에는 理解가 깊어지고 좀더 나은 計算과 計測技術에 의하여 船舶技師들은 正確하고 安全한 設計技法을 터득하는데 成功하였다. 軸系에서 비틀림振動은 이제 基本的 研究가 必要하지 않다

first practical design method was developed on the lateral vibrations, then indicated as the critical whirling speeds [2], although the theory of whirling was already known decennary earlier.[3] But not only the gyroscopic effects of the propeller, also the hydrodynamic effects of the entrained water in the propeller disc play an important role [4] as well as the oilfilm in the stern bearing and the support stiffness in the stern-bearing. [2,4]

Only because of the availability of the high speed digital computers such problems can be analysed and investigated. The finite element method, to represent the structure, the finite difference method, to estimate the oil film behaviour, and the lifting surface theories, to calculate the propeller hydrodynamic, make it possible to study complex systems and to investigate the sensitiveness to certain parameters.

The analysis of the longitudinal vibrations is basically not more difficult to perform than for the lateral vibrations. However, the studies of these vibrations are hampered by the fact that the effective thrust block participation in these longitudinal vibrations is expensive, tedious, time consuming and, therefore, in general rather difficult to be determined. [5,6] Future research may be needed in which much full scale information has to be considered. No further attention to this kind of vibrations will be given.

Nowadays the digital electronic computer is available in many forms, running from the pocket calculator via the mini's to the mainframes. Its usefulness in design work is now generally acknowledged. However, only 30 years ago it was stated [2]: "It is obvious that, useful as the digital computer may be, it is not suitable for direct everyday application by the designer".

Obviously research and development in the intermediate time have changed the possibilities and thus the philosophies thoroughly. Also in marine engineering this applies and it takes a lot of effort to be kept upgraded.

고 말할 수 있을 것이다.

橫振動과 縱振動은 그렇지가 않다. 50年代에 橫振動에 대한 最初의 實用的 設計方法이 開發되었는데 [3] 휘돌림의 理論은 이미 이보다 10年前에 알려진 것이지만 이 當時에는 危險 휘돌림速度로 表現되었었다. [2] 그러나 프로펠러圓板의 자이로効果뿐만 아니라 附加水の 流體力學的인 效果가 船艙베어링에 있어서의 油膜이나 船艙베어링에 있어서의 支持剛성과 마찬가지로 [2,4] 重要な 役割을 한다. [4]

高速디지털計算機의 利用에 의하여서만이 이러한 問題를 解釋할 수 있고 研究할 수 있다. 構造物을 解析하기 위한 有限要素法, 油膜의 舉動을 推定하기 위한 有限差分法, 프로펠러의 流體力學的 舉動을 計算하기 위한 揚力面理論등이 複雑한 系를 研究하고 特定 媒介變數에 대한 應答程度를 研究하는 것을 可能하게 하고 있다.

縱振動的 解析은 基本的으로 橫振動보다 더 어려운 것은 아니다. 그러나 縱振動的의 研究는 이들 振動에서의 有効한 推力베어링 關與程度를 아는 것이 값비싸고 귀찮으며 時間이 많이 所要된다는 事實때문에 妨害를 받고 있으며 따라서 大體로 結論짓기가 若干 어려운 形便이다. [5,6] 앞으로의 研究는 實物크기에 관한 많은 情報를 導入할 必要가 있을 것이다. 이러한 種類의 振動에 대하여서는 더 이상 言及을 하지 않겠다.

오늘날 디지털電子計算機는 포켓計算器로부터 미니計算機를 거쳐 大形計算機에 이르기까지 여러가지 形態로 利用할 수 있다. 設計에 있어서 그의 有用性은 이제 一般的으로 認定되고 있다. 그러나 겨우 30년전만 하여도 "비록 디지털計算機가 便利하기는 하여도 設計者가 直接 日常的으로 利用하기에는 適合하지 않다는 것은 明白하다"고 말하였던 것이다. [2]

그동안의 研究와 開發은 分明히 可能性과 따라서 哲學을 송두리채 바꾸어 놓았다. 또한 船用工學에 있어서도 이것은 利用되고 있으며 性能을 向上시키는데 있어 많은 努力이 이루어 졌다.

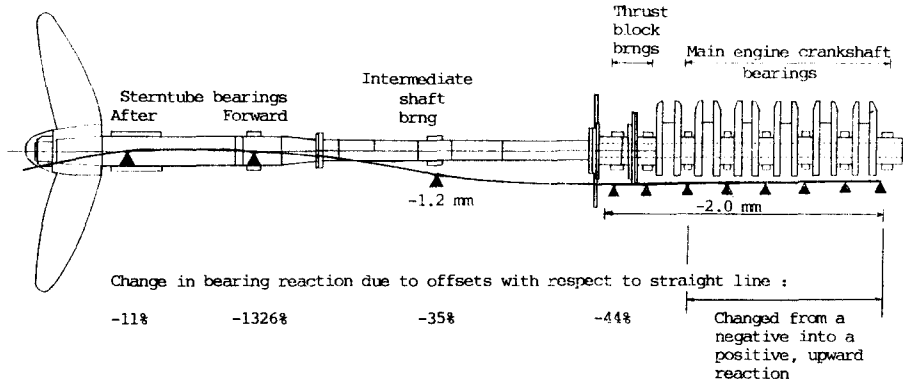


Fig 1. Effect of the alignment on the bearing reactions.

## 2. General considerations on the alignment.

### 2.1 Familiar procedures.

Nowadays it seems at the yards to be general practice that the propeller-shafting system is aligned along a curved line with the purpose to have all bearings loaded in a similar way. For conventional, single screw ships with the engine room aft this means that the sterntube, the intermediate bearings and the engine bearings are positioned with respect to a reference line. This reference line can be drawn in various ways. Mostly it is drawn horizontally with respect of the boring of the aft stern bearing. Also the forward stern bearing is generally centered along this reference line, so that only the intermediate bearings and the main engine are oriented with respect to the reference line. Figure 1 gives an example of such an alignment with respect of the curvature as well as the relative change of the bearing reactions with respect to the straight line alignment.

Considering Figure 1 the following observations can be made.

In the straight line condition the bearing reaction of the forward stern bearing was very small. Lowering the intermediate bearing brought the forward stern bearing involved in carrying the shaft. Of course

## 2. 軸系配置에 있어서의 一般的 配處

### 2.1 通常的인 方法

오늘날 프로펠러 軸系는 모든 베어링에 같은 狀態로 荷重이 加하여지도록할 目的으로 曲線에 따라 配置하는 것이 一般的인 方法이 되고 있음을 造船所에서 볼 수 있다. 船艙機 關室을 갖는 傳統의인 單一 프로펠러船에 대하여 이것은 船艙管, 中間軸베어링과 機關베어링이 參照線에 따라 配置됨을 뜻한다. 이 參照線은 여러가지로 그릴 수 있다. 大概是 後部 船艙베어링의 口徑에 대하여 水平으로 긋고 있다. 또한 前部 船艙베어링은 통상 이 參照線에 中心이 오도록 함으로써 中間베어링과 主機關만이 參照線에 따라 位置를 잡게 된다. Fig.1은 直線配置에 대한 베어링反力の 相對的變化뿐만 아니라 曲線에 관한 이러한 配置의 1例를 보여주고 있다.

Fig.1을 보고 생각하여 볼 때 다음과 같은 말을 할 수 있다. 直線狀態에서 前部船艙管베어링의 베어링反力은 매우 적다. 中間베어링을 낮추면 前部船艙管베어링은 軸을 支持하게 된다. 또한 勿論 機關의 變位가 影響을 미치나 密接해 있기 때문에 中間베어링의 影響이 더 強하다.

크랭크軸에 대하여서는 直線配置狀態에서

also the displacement of the engine will have an effect, but because of its close location the effect of the [intermediate bearing will be stronger.

For the crankshaft the bearing reactions were negative in the straight line condition. By changing the location of the main engine and the intermediate bearing this resulted into positive reactions at the crankshaft bearings.

### 2.2 Non-familiar aspects.

Clearly it follows from the results of Figure 1 that by introducing some freedom in the vertical position of the bearing a much more rational distribution of the bearings is possible. But also it follows that having the position of the forward stern bearing variable will increase the possibilities in alignment enormously. This is shown in Figure 2. From this Figure also follows that for long sleeves, as the aft bearing, there may be large advantages in using also the slope of a long bearing as a parameter to influence the distribution of the bearing reactions.

This slope boring of the after stern bearing, even multi-slope boring, has been done in different cases. However, it is still not very popular and therefore may need more attention and research. In this report no further attention is spend to this, because it seems an easy improvement of the existing calculation techniques.

More important and more fundamental are the two following aspects. The first one is the hydrodynamic load on the propeller due to its operation. The second is the oil film that is built up between the journals and the bushing. These two hydrodynamic effects may have an important influence on the future realization of the alignment and they are discussed in detail in the following two sections.

베어링反力이 負(-)로 되었다. 主機와 中間 베어링의 位置를 바꿈으로써 크랭크베어링에서의 反力이 陽(+)으로 된다.

### 2.2 通常의이 아닌 狀況

Fig.1의 結果로부터 베어링의 垂直位置 選定을 若干 더 自由롭게 함으로써 훨씬 더 合理的인 베어링荷重配分이 可能하게 됨이 明白하게 되었다. 그러나 또한 前部船尾管베어링의 位置를 可變으로 하면 軸系配置에서의 可能性이 顯著히 增加하게 된다. 이것을 Fig.2에 보였다. 이 그림으로부터 또한 後部베어링과 같은 긴 슬리브에 대하여서는 베어링反力分布에 影響을 미치는 파라미터로서 긴 베어링의 傾斜를 또한 利用하는 것이 크게 有利하게 될 것이다.

後部 船尾管베어링의 이러한 傾斜보오링, 심지어는 多傾斜보오링이 여러가지 경우에 施行되고 있다. 그러나 이것은 아직도 그다지 人氣가 있는 것이 아니며 따라서 더 많은 關心과 研究가 必要할 것이다. 이 글에서는 이 問題에 關하여 이 以上 더 論及을 하지 않을 것이다. 왜냐하면 現存하는 計算技法을 고치면 쉽게 이루어질 것으로 보이기 때문이다.

보다 더 重要하고 基本的인 것은 다음의 두가지 事項이다. 첫번째 것은 프로펠러의 作動에 의한 그에게 걸리는 水力學的 荷重이며 두

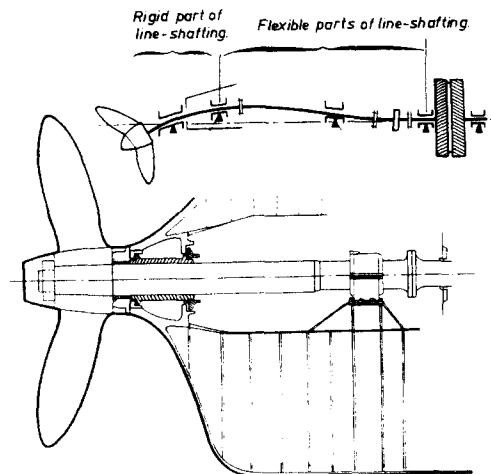


Fig. 2. Example of rational alignment(1)

### 3. The propeller induced hydrodynamic effects on the alignment.

#### 3.1 General remarks on the propeller hydrodynamics.

The task of the propeller is to transform the torque delivered by the engine into the thrust that brings the ship in motion.

In first consideration one expects that thrust and torque are symmetric with respect to the shafting. However, the water flow over the propeller disc is not homogeneous and therefore the load distribution is also unevenly over the propeller disc, leading to an unsymmetric application of thrust and torque. This unsymmetry in thrust and torque influences the alignment of the shafting and therefore the natural frequencies of the lateral shaft vibrations. Some detailed attention is devoted to this phenomenon in the following.

In Figure 3 a two dimensional representation is given of a profile operating in a steady inflow. This profile may be regarded as a section of a propeller blade at a certain radius.

The inflow velocity  $U$  is composed of the local circumferential velocity  $\omega r$  and the

번져 것은 자아널과 붓슈間에 形成되는 油膜이다. 이들 두가지 流體力學的 効果는 앞으로의 軸系配置를 實現함에 있어 重要한 影響을 미치게 될 것이며 다음의 두章에서 詳細히 다루게 된다.

### 3. 軸系配置에 미치는 프로펠러 誘發 流體力學的 影響

#### 3.1 프로펠러의 流體力學에 관한 一般的 考察

프로펠러의 役割은 機關이 供給하는 토오크를 推力로 變하는 것이며 이 推力는 배를 움직이게 한다.

우선 生覺하건대 推力과 토오크가 軸에 대하여 對稱일 것이라고 豫想하게 된다. 그러나 프로펠러圓板에로의 물의 흐름은 均一하지 않으며 따라서 負荷의 分布도 프로펠러圓板에 대하여 不均一하여서 推力과 토오크가 非對稱으로 作用하게 된다. 이 推力과 토오크에 있어서의 非對稱性은 軸系의 配置에 影響을 미치며 따라서 軸橫振動의 固有振動數에도 影響을 미친다. 다음에 이러한 現象에 대하여 약간 仔細하게 살펴보기로 한다.

Fig. 3에 定常流에서 作動中인 프로펠러 날개斷面形狀을 2次元으로 表示하였다. 이 形狀은 特定 半徑에서의 프로펠러 날개斷面으로 볼

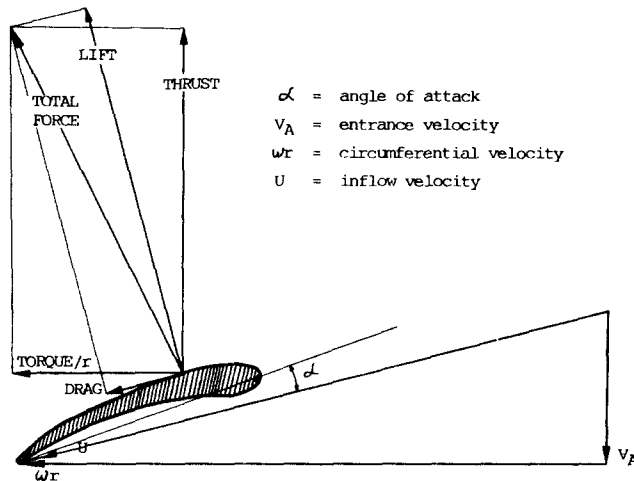


Fig. 3. The hydrodynamic forces on the sectional profile of a propeller blade at a certain radius  $r$ .

axial inflow of the water in the propeller disc  $V_A$ . To this inflow velocity  $U$  belongs a lift and a drag which form together the section force. This section force is decomposed into a component that is parallel to the thrust and a component that contributes to the torque. [7, 8]

The propeller operates in a non-uniform inflow of water. Therefore its circumferential load distribution is not uniform. For single screw ship the water velocity  $V_A$  in the top region of the propeller disc is appreciably lower than elsewhere over the disc. That means that the propeller blades in the top position are heaviest loaded, because here the angle of attack is the largest. Then the averaged load on the propeller, that means the thrust, does not apply nicely at the center of the propeller, but at a point above the center. Thus a moment is exerted to the propeller, tending to lift the propeller and consequently also the shaft.

The contribution to the torque is also largest for the top position and therefore, in addition to the mean torque applying at the center of the propeller, also a transverse force applies to the propeller. For a right-handed rotating propeller (looking in forward direction) this transverse force is directed to port side and horizontally oriented.

In the above only the axial inflow into the propeller disc was considered. However, in general the flow into the propeller disc has an oblique direction. Therefore the flow velocity is decomposed into 3 components:

- the axial component
- the tangential component
- the radial component

The radial component is perpendicular to the plane in which the sectional profile is moving, in fact parallel to the wing or the propeller blade. It therefore does not directly contribute to the generation of forces as sketched in Figure 3 and generally is not considered in propeller theories.

The tangential component acts parallel to the circumferential speed and therefore has effects. For a single screw ship this effect can be described as follows.

For a single screw ship there is a vertical

수 있을 것이다.

流込速度  $U$ 는 局部的 圓周速度  $\omega r$ 와 프로펠러 圓板으로 흘러들어오는 물의 軸流速度  $V_A$ 로 構成된다. 이 流込速度  $U$ 에는 斷面力을 構成하는 揚力과 抗力이 屬하게 된다. 이 斷面力은 推力과 平行한 分力과 토오크에 寄與하는 分力으로 分解된다. [7, 8]

프로펠러는 물의 不均一한 흐름속에서 作動하게 된다. 單一프로펠러船에 대하여 프로펠러 圓板의 上方領域에서의 물속도  $V_A$ 는 圓板의 모든 다른 곳에서보다 相當히 느리다. 이는 上方位置에 있는 프로펠러날개는 가장 크게 荷重이 걸린다는 것을 意味한다. 왜냐하면 여기서 入射角이 가장 크기 때문이다. 그렇다면 프로펠러의 平均荷重, 即 推力은 프로펠러中心에 正確히 作用하는 것이 아니라 中心上方의 한 點에 作用하게 된다. 이리하여 프로펠러에 모멘트가 作用하게 되고 프로펠러와, 따라서 軸도 또한 들어올려지는 傾向이 생기게 된다.

토오크에 대한 寄與도 또한 上端位置에서 가장 크며 따라서 프로펠러 中心에 作用하는 平均토오크에 더하여 橫方向의 힘도 프로펠러에 作用하게 된다. 右비轉프로펠러에 대하여 (前進方向으로 보아) 이 橫方向힘은 水平方向에서 左舷으로 向하게 된다.

위에서는 프로펠러 圓板으로 流込하는 軸流만을 生覺하였다. 그러나 一般的으로 프로펠러 圓板으로 流込하는 흐름은 비스듬한 方向을 갖는다. 그러므로 흐름속도는 3가지 成分으로 分解된다.

- 軸方向成分
- 接線方向成分
- 半徑方向成分

半徑方向成分은 斷面形狀이 움직이고 있는 平面, 實際로는 날개 即 프로펠러블레이드에 平行한 平面에 대하여 直角이다. 그러므로 이것은 Fig. 3에 그린것 처럼 힘을 發生하는데 直接寄與를 하지 않으며 一般的으로 프로펠러 理論에서 考慮되지 않는다.

接線成分은 圓周速度에 平行하게 作用하며



upward stream of the waterflow. That means that for the blade, that is moving in the upward direction, the relative velocity with respect to the waterflow, which is originally  $\omega r$ , is reduced. Thus the angle of attack in Figure 3 is reduced and therefore the lift and the drag, thus also the contributions to the mean thrust and torque, are less than elsewhere.

However, the blade that has passed the top moves against the waterflow so that its relative velocity with respect to the flow is increased and the angle of attack is increased as well in Figure 3. For a right handed propeller this results in a movement of the thrust to starboard side and there is applying a vertical upward lateral force due to the increase in the contribution to the torque.

For a single screw ship, provided with a right handed propeller, the following general description can be given. The thrust applies at the propeller in the 1 to 2 o'clock position, having an eccentricity between 10% to 25% of the propeller radius. See also Figure 4. Keeping the torque centrally applying at the propeller, a lateral force has to be considered also centrally applying at the propeller. The magnitude of this lateral force is given as a part of the ratio of the torque and the propeller radius:  $Q/R$ , this ratio lies between 20% and 40%. So the lateral force runs from 20% to 40% of  $Q/R$  and it is directed into the 10 to 11 o'clock direction, almost perpendicular to the thrust eccentricity.

It must thus be expected that due to the operation in a nonuniform wake flow the propeller may experience large steady loads that affects its alignment. In how far these effects are important will be shown in the next sections.

### 3.2 Example of the magnitudes of the mean propeller loads.

Figure 5 shows the contours of the axial velocities in the plane of a propeller of a single screw ship. These contour lines are connecting the points with the same axial

따라서 影響을 미친다. 單一프로펠러船에 대하여 이 影響은 다음과 같이 說明할 수 있다.

單一프로펠러船에 대하여서는 水流의 垂直上方向흐름이 있다. 이것은 上方向으로 움직이고 있는 날개에 있어 元來  $\omega r$ 인 水流에 대한 相對速度는 減少한다. 이리하여 Fig.3에서의 入射角은 減少하며 따라서 揚力과 抗力, 그러므로 平均推力과 토크에 대한 寄與도 또한 다른 어떤곳 보다 작게된다.

그러나 上端을 通過한 날개는 水流에 거슬면서 움직이게 되므로 그의 흐름에 대한 相對速度는 增加하며 入射角은 Fig.3에서와 같이 亦是 增加하게 된다. 右回轉프로펠러에 대하여 이것은 推力이 右舷側으로 움직이게 하며 토크에로의 寄與가 增加하기 때문에 垂直上方向의 橫方向힘이 作用하게 된다.

右回轉프로펠러를 裝備하는 單一프로펠러船에 대하여 다음과 같은 一般的 說明을 할 수 있다.

推力은 프로펠러의 1~2時方向에서 프로펠러 半徑의 10~25%間에 偏心을 갖는다. 또한 Fig.4를 보자. 프로펠러에서 토크를 中心에 加한채 橫方向힘을 프로펠러中心에 加하였다고 또한 生覺하지 않으면 안된다. 이 橫方向힘의 크기는 토크와 프로펠러半徑比  $Q/R$ 의 分數로 表示되며 이 比는 20~40%間에 있다. 따라서 橫的인 힘은  $Q/R$ 의 20%에

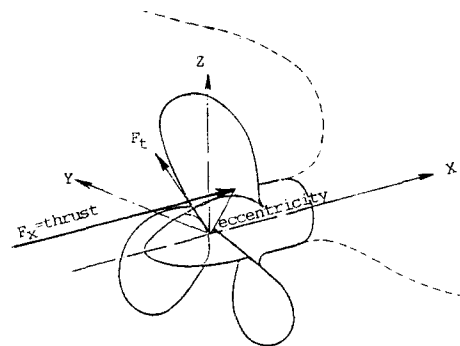


Fig. 4. Illustration of the eccentric application of the thrust and of the application of the lateral force, both resulting from the steady propeller operation in a non-uniform wakefield.

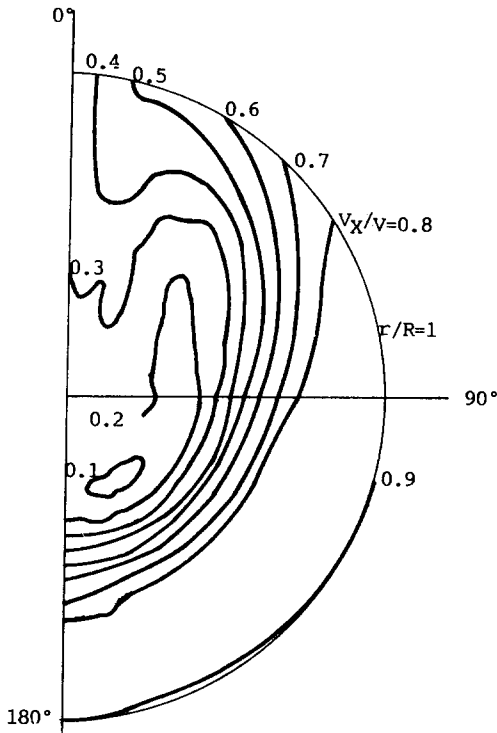


Fig 5. Iso-axial velocities in the propeller plane.

velocity. Starting from the top position this is done over 180° only. For a single screw ships the vertical plane is considered as a plane of symmetry.

From this Figure follows clearly the low axial inflow speed of the water in the top region of the propeller disc.

Figure 6 gives an other way of presentation. Here, per radius, the circumferential distribution of the water velocity is given. In this Figure 6 also the distribution of the tangential component of the velocity is given, the positive direction is given by the angular rotation, which is righthanded.

In this velocity field of the water works the propeller, which general characteristics are shown in Figure 7. This a righthanded propeller, looking from aft. It is designed to propel a 30,000 DWT bulk carrier and has to deliver maximum 12,500 KW at about 95 revolutions per minute.

In Figure 8 the coordinate system used in this paper is shown, together with the description of the forces and moments that

서 40%間에 있으며 10~11時方向으로 向하고 있어 거의 推力偏心에 垂直으로 된다.

따라서 不均一한 伴流内에서의 作動때문에 프로펠러는 軸系配置에 影響을 미치는 큰 定常荷重을 經驗하게 될 것으로 豫想하지 않으면 안된다. 어느 程度 이 影響이 重要한가를 다음 章에서 보여주도록 하겠다.

### 3.2 平均프로펠러荷重의 크기의 例

Fig. 5는 單一 프로펠러船의 프로펠러面에서의 軸方向速度輪廓을 보여주고 있다. 이들 輪廓은 같은 軸方向速度를 갖는 點을 連結한 것이다. 頂點位置로 부터 始作하여 180°에 대하여서만 이것을 보였다. 單一프로펠러船에 대하여서 垂直面은 對稱面이라 生覺한다.

이 그림으로부터 볼 때 프로펠러圓板의 上部領域에서 물의 軸方向流込速度가 分明히 느리다.

Fig. 6은 또다른 表示方法을 보여주고 있다. 여기서는 半徑마다 水流速度의 半徑方向分布를 보여주고 있다. 이 Fig. 6에서 速力의 接線成分分布도 또한 表示되어 있는데 陽(+의) 方向은 右回轉으로 주어진다.

이러한 물의 速度場에서 프로펠러가 作動하며 Fig. 7에 그의 一般의 特性이 表示되어 있다. 이것은 後方에서 보아 右回轉프로펠러이며 30,000 DWT 撒物船을 推進하기 위하여 設計된 것으로 약 95 RPM에서 最大 12,500 KW를 供給해야 한다.

Fig. 8에 이 글에서 使用하는 座標系를 프로펠러에 作用하는 힘과 모멘트의 說明과 함께 表示하였다. 이들의 流體力學的 힘과 모멘트를 Fig. 7에 보인 프로펠러와 Fig. 5에 보인 軸方向伴流에 대하여 計算하였다. 이들의 計算에는 文獻[9]에 說明하고 있는 것과 같은 揚力面理論에 基礎를 둔 計算프로그램을 利用하였다. 結果를 Table 1에 表示하였다. 그 외에 接線方向伴流의 效果도 또한 이 表에 表示하였다.

軸方向흐름内에서 作動하면 주로 水平力과 水平모멘트가 存在하게 됨을 Table 1로부터

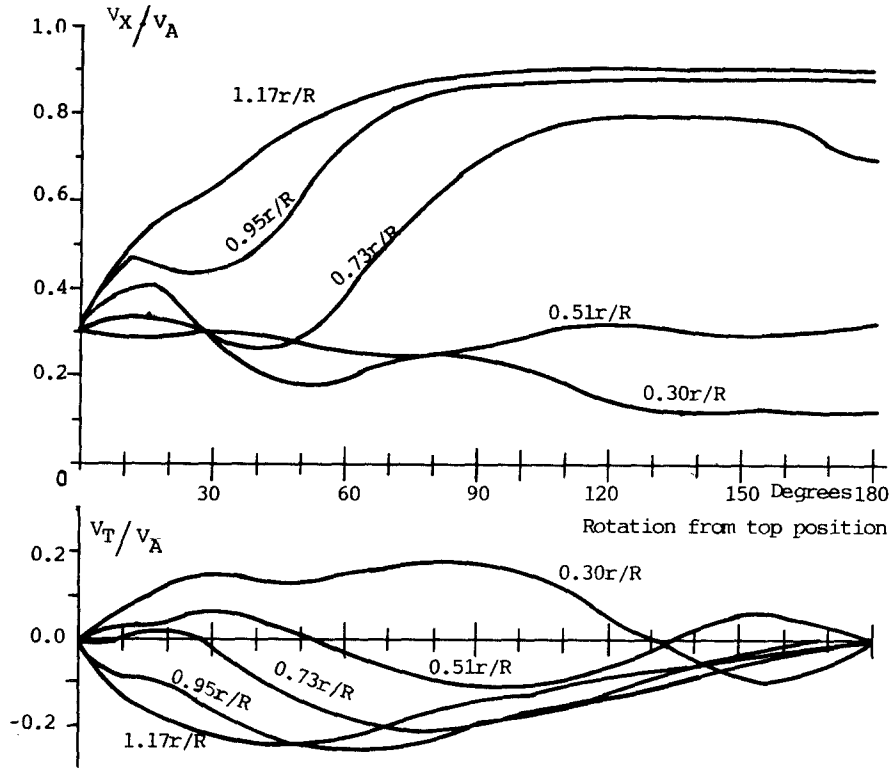


Fig. 6. Circumferential variation of velocity component.

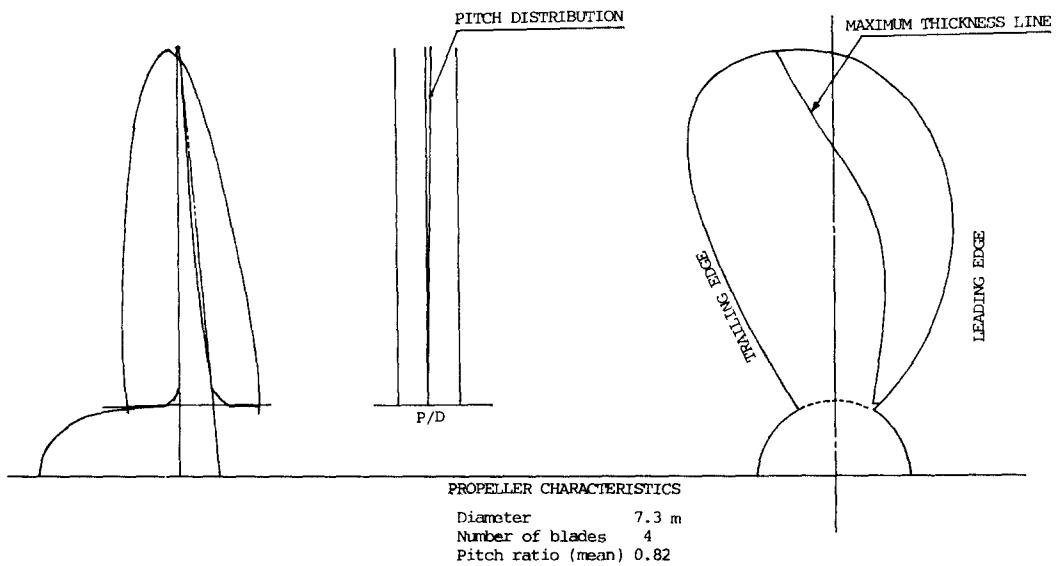
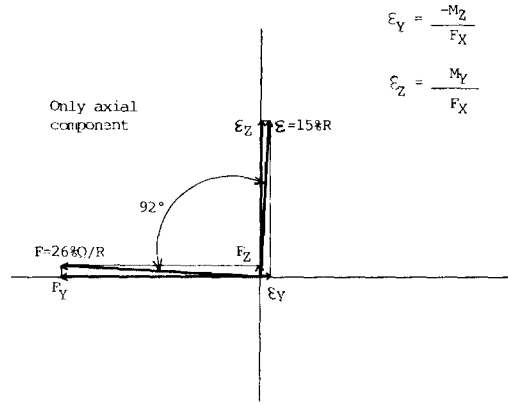
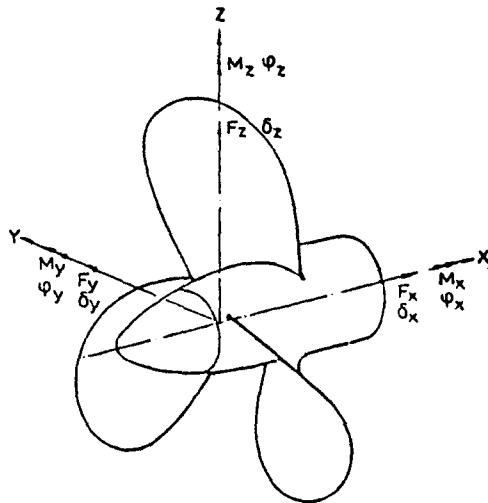


Fig. 7. Propeller outlines and main characteristics.

**Table 1. Mean forces and moments caused by the steady operation of a propeller (Fig. 7) in a non-uniform wakefield (Fig. 5 and 6.).**

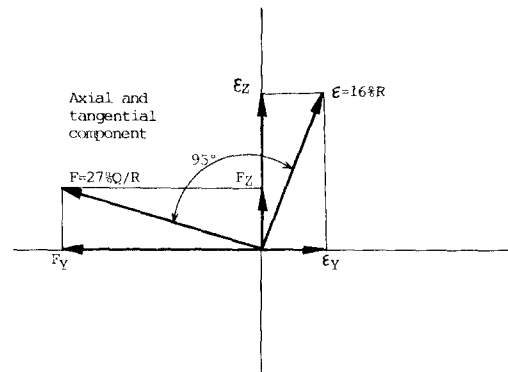
		Only axial flow	Effect of tangential flow
$F_x$ (thrust)	kN	1,340	+1.2%
$-M_x$ (torque)	kNm	1,355	+1.2%
$F_y$ (Hor. force)	kN	96.8	+0.8%
$F_z$ (Vert. force)	kN	3.15	+874%
$M_y$ (Hor. mom)	kNm	761	-0.2%
$M_z$ (Vert. mom)	kNm	47.5	+602%
$F$ (Transv. force, % Q/R)		27	27
$e$ (eccentricity % R)		15	16



**Fig. 8.** Choice of the coordinate system and the definition of forces and moments that apply at the propeller.

act on the propeller. These hydrodynamic forces and moments have been calculated for the propeller indicated in Figure 7 and the axial wake flow shown in Figure 5. For these calculations use has been made of a computer program based on a lifting surface theory like the one as described in reference. [9] The result is shown in Table 1. In addition also the effect of the tangential wake is shown in this Table.

Operating in the axial flow only we see from Table 1 that there is mainly a horizontal force and horizontal moment. The horizontal force pushes the propeller to portside, the horizontal bending moment lifts the propeller upwards. Also there are



**Fig. 9.** Vectorial representation of the thrust eccentricity  $\epsilon$  and the lateral force  $F$  due to the non-uniformity of the wakefield.

a small vertical force and a vertical bending moment. These small effects are caused by several complicating matters that are not easily described, but have to be taken into account in a sophisticated computer program.

However, a much larger effect is found from the inclusion of the tangential component of the wakefield. A strong increase in the vertical force and vertical bending moment is shown, although the resultant vector is hardly effected. The directions of the vectorial presentation of the transverse force and of the eccentricity are more affected as shown in Figure 9.

**3.3 The effect of the mean propeller forces on the alignment.**

It is still a subject of research how detailed knowledge of the hydrodynamic forces and moments is needed to perform the alignment with sufficient accuracy. As an example the effects of the above calculated hydrodynamic loads are shown in Table 2.

In this table the original alignment is shown. The offsets were chosen to fulfill the requirements of the bearing mean pressures. These calculations only referred to the vertical aspects. Also the alignment was planned to be done only for the vertical plane. Therefore the attention is restricted to this plane.

볼 수 있다. 水平力은 프로펠러를 左舷으로 밀고 있으며 水平굽힘모멘트는 프로펠러를 上方向으로 들어올린다. 또한 작은 垂直方向 힘과 垂直方向 굽힘모멘트가 있다. 이들 작은 影響들은 쉽게 說明할 수 없는 여러가지 複雜한 要因들에 의하여 發生하는데 仔詳한 計算機프로그램에서는 考慮에 넣어야만 한다.

그러나 훨씬 더 큰 影響은 伴流場의 接線成分이 包含됨으로써 일어난다. 비록 合成벡터는 거의 影響을 받지 않으나 垂直方向 힘과 垂直方向굽힘모멘트가 크게 增加함을 볼 수 있다. Fig. 9에서 보는바와 같이 橫方向 힘의 벡터表示方向과 偏心の 方向이 더 많이 影響을 받고 있다.

**3.3 軸系配置에 미치는 平均프로펠러 힘의 影響**

充分한 精密度를 갖고 軸系配置를 遂行하기 위해서는 얼마나 仔細한 流體力學的 힘과 모멘트에 관한 知識이 必要한지 아직도 研究課題로 남아있다. 한가지 例로서 위에서 計算한 流體力學的 荷重의 響影을 Table 2에 表示하였다.

이 表에서 元來의 配置를 보여주고 있다. 軸套트는 베어링 平均壓力의 要求를 滿足하도록 選擇되었다. 이들 計算은 단지 上下方向에

**Table 2. The effect of the hydrodynamic propeller loads on the alignment in the vertical plane.**

		Bearing				
		After sterntube	Forward sterntube	Interm. shaft	Thrust shaft	Aft crank shaft
Original alignment	Offsets	0.0	0.0	-1.2	-2.0	-2.0
	Reactions	379	89	66	106	86
Hydrodynamic propeller loads	Vert. force	-37	+13	-2	+2	-1
	Hor. mom	-234	+264	-41	+31	-26
Combined original alignment		108 (-72%)	366 (+412%)	24 (-64%)	139 (+31%)	59 (-32%)
Realignment	Offsets	0.0	0.0	+0.6	+0.5	+0.5
	Reactions	186	209	129	140	35

Bearing reactions are given in kN.

Offsets are given in mm, Positive in upward direction.

Values were derived from shafting in Figure 1, provided with the propeller shown in Figure 7 and operating in the wakefield of Figure 6.

Considering the hydrodynamic loads, that apply to the propeller, the corresponding bearing reactions were calculated. In these calculations the effects of the gravity were omitted. For the vertical force and the horizontal bending moment the general effects on the bearing reactions are understandable. Important is the large effect of the bending moment. The changes in the bearing reactions due to this hydrodynamic load are of the order of magnitude of the original bearing reactions.

These strong effects are more explicitly demonstrated in calculating the combined effect of the gravity and of the hydrodynamic loads, keeping the original offsets in the alignment. From this result (5th line in Table 2) follows that the bearing reactions are essentially changed when considering the hydrodynamic loads. A new alignment is needed as shown in the last 2 lines of Table 2. The optimal offsets show to be completely different from the original offsets in which the hydrodynamic propeller loads were not accounted for.

Research on alignment is done already for many years to face these problems[10] and it is still continuing.[11] The above described considerations should become standard in shaft alignment procedures. Then the margins become known within which the shaft has to operate. Only in that case one can make a "rational alignment." [1] However, we are still far from that situation as shown in reference[12] which lacks these considerations.

#### 4. The effect of the oilfilm on the alignment

##### 4.1 The building-up of an oilfilm

Due to the shaft rotation an oilfilm is built up between bush and journal and thus offsetting the shaft from its original position. This offsetting occurs in a horizontal and a vertical way as shown in Figure 10. It is a function of the rate of revolutions, the bearing load, the radial clearances and so on.

關한 것으로 限定하였다. 또한 配置도 단지 上下方向으로만 行하도록 計劃되었다. 그러므로 이 方向에만 注意를 限定시킨다.

프로펠러에 作用하는 流體力學的 負荷를 考慮하여 相應하는 베어링反力을 計算하였다. 이러한 計算에서 重力의 影響은 省略하였다. 上下方向과 水平方向굽힘모멘트에 대하여 베어링反力에 미치는 一般的인 影響을 理解할 수 있다. 굽힘모멘트의 큰 影響이 重要하다. 이 流體力學的인 負荷에 의한 베어링反力の 變化는 元來의 베어링反力 크기에 비금가는 것이다.

이들 강한 影響들은 軸系配置에서 本來의 欸셋트를 그대로 維持한채 重力과 流體力學的 荷重의 結合된 影響을 計算함으로써 보다 分明하게 表示될 수 있다. 이런 結果로부터 (Table 2에서 5번째 줄) 流體力學的인 荷重을 생각할 때 베어링反力은 根本的으로 變化하게 된다. Table 2의 마지막 두줄에서 보는 바와 같이 새로운 配置가 必要하다. 最適의 欸셋트는 流體力學的인 荷重을 考慮에 넣지않은 元來의 欸셋트와는 完全히 差異가 있음을 보게 된다.

配置에 관한 研究는 이러한 問題들을 對處하기 위하여 여러해 동안 이미 遂行되어 왔으며[10] 아직도 繼續되고 있다.[11] 위에서 說明한 考察들은 軸配置方法에 있어 標準으로 되어야 한다. 그러면 軸이 作動하게 되는 限界值들을 알 수 있게된다. 그러한 경우만이 "合理的인 配置"가 可能하다. 그러나 이러한 配慮가 欠한 것들을 文献[12]에서 볼 수 있는 바와같이 合理的인 配置 狀態로부터 아직 멀리 떨어져 있다.

#### 4. 軸系配置에서의 油膜의 影響

##### 4.1 油膜의 形成

軸의 回轉으로 因하여 油膜이 붓수와 자아 널間에 形成되며 따라서 軸을 그의 本來位置로부터 偏心시킨다. 이러한 偏心은 Fig. 10에서 보는바와 같이 水平과 垂直方向에서 發生하게된다. 그것은 回轉率과 베어링荷重과 半

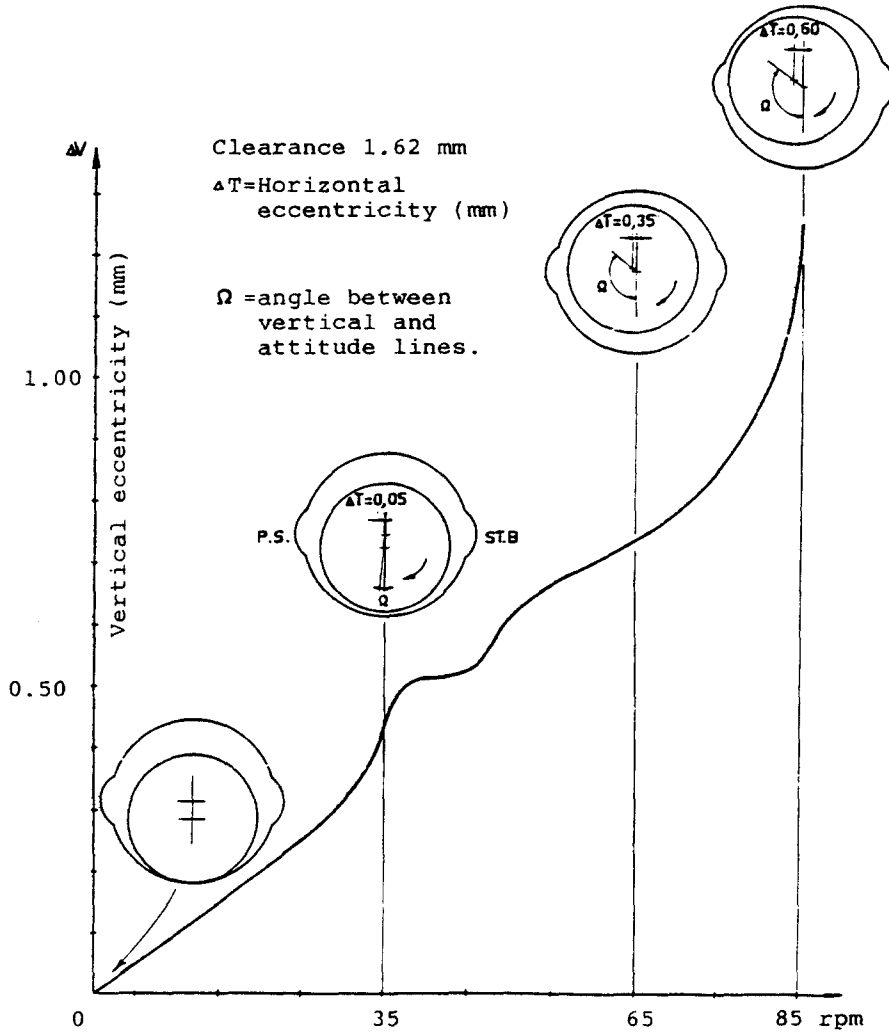


Fig. 10. Measured position of the shaft journal in the bush as function of the rpm and the external loads [10].

By means of the finite difference method or the finite element method[13] it has now become possible to calculate the formation of an oilfilm between journal and bush, see Figure 11. These calculated values confirm our expectation of a high pressure at the aft side of the after sterntube bearing. Plotting the maximum pressures over the length of this bearing (see Figure 12) is a further confirmation of this expectation. However, scrutinizing the results of Figure 11 learns us that the real situation is more complex than suggested by Figure 12.

徑方向等의 函數이다.

有限差分法이나 有限要素法[13]으로 이제는 Fig. 11에서 보는 바와 같이 자아널과 붓슈間的의 油膜形成을 計算하는 것이 可能하게 되었다. 이러한 計算值들은 後部船尾管베어링의 後部側에서 高壓으로 될 것이라는 豫想을 確認하여 주고 있다. 이 베어링의 길이에 따라 最大壓力을 그려보면(Fig. 12를 보라) 이러한 豫想을 더욱 더 確認하여 준다. 그러나 Fig. 11의 結果를 仔細히 살펴보면 實際狀況은 Fig. 12에 의하여 提示되는 것보다 훨씬 더 複雜하다는

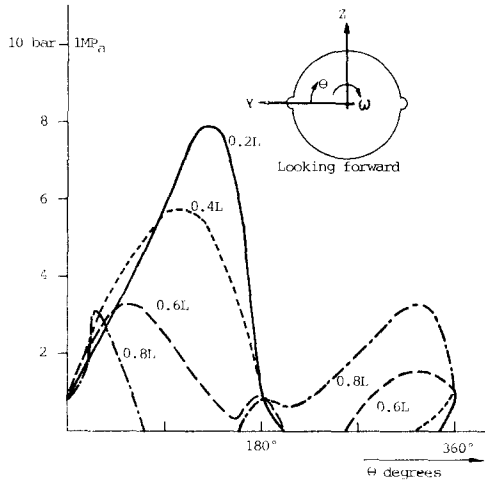


Fig. 11. Calculated pressure distribution in the after sterntube bearing of a 60,000 DWT tanker.

From Figure 11 it also follows that over the bearing length the maximum pressure occurs at different angles. Therefore in Figure 13 a vectorial presentation of the maximum pressures is shown as well as the vectorial presentation of the distribution of the eccentricity over the length of the bearing.

In this respect it has to be remarked that only by a trial and error method the correct orientation of the journal in the bearing can be obtained. This is because of the fact that only for a given eccentricity distribution the equations for the oilfilm pressure can be derived and solved. Based on the resulting pressure distribution or on the bearing reaction, derived from integrating the pressures, the considered eccentricity is evaluated and a new one is estimated. This trial and error or iterative method is caused by the non-linearities in the Reynold's equation. [13]

For the selected distribution of the eccentricity in Figure 13 it follows that the maximum oilfilm pressures vary widely over the bearing length. Therefore, in the alignment procedure account has to be taken of the oilfilm built up in the bearings, especially for the rather long after stern-

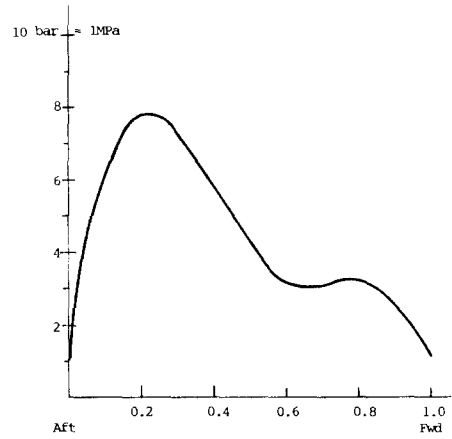


Fig. 12. Distribution of the maximum values of the oilfilm pressure over the bearing length (derived from Figure 11).

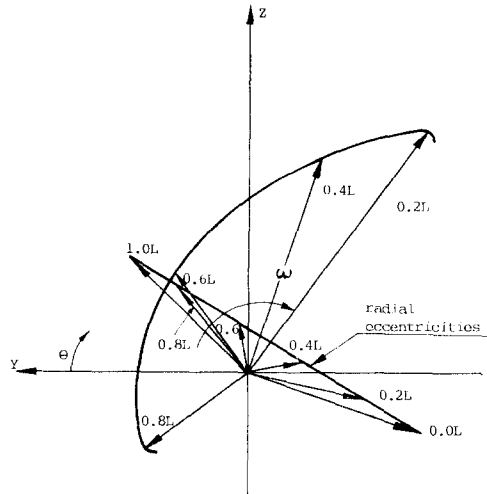


Fig. 13. Vectorial presentation of eccentricities of the journal and maximum pressure of the oilfilm in the after sterntube bush of a 60,000 DWT tanker (Figure 11).

것을 알 수 있다.

Fig. 11로부터 베어링길이에 따라 最大壓力은 相異なる 角度에서 發生하고 있음을 또한 알 수 있다. 따라서 Fig. 13에 베어링길이에 따라 偏心の 分布를 벡터로 表示하였을 뿐만 아니라 最大壓力의 벡터表示로 또한 나타내고 있다. 이러한 觀點에서 오직 試行錯誤法에 의하여



tube bearing.

Normally in this bearing only one support point is considered, situated in the aft part. The choice of its location will depend on the experience of the people involved. This choice is a good start for the first calculation runs. However, in the final runs the stationary-running condition has to be considered, in which the oilfilm together with the mean propeller loads, have to be taken into account.

The distribution of eccentricity in Figure 13 may seem unrealistic, but that has not to be the case. Considering the bending moment and the transverse force due to the operation of the propeller in the non-uniform flow, one can imagine that also horizontal forces have to be counteracted by the bearing and not only vertical forces as caused mainly by the gravity. In the next chapter some informations will be given on the effects of these hydrodynamic forces and moments on the alignment.

#### 4.2 Considerations of the offsets and the hull deformations

The bearing clearances over the shaft line differ considerably. At the stern tube bearing the radial clearance is of the order of 0.5 mm or larger. For the intermediate bearings the radial clearance is of the order 0.5mm or a little less and in the main engine the bearings have a clearance of the order of 0.15mm.

Nevertheless, in the alignment calculations these differences in the radial clearances are not always accounted for. [12] That means that the bearing reactions and the values for gap and sag can not be the same during installation or during operation as the calculated values.

The same applies to the deformation of the hull girder and the double bottom. During construction and launching the hull will set. During operations the hull will set further due to local plastic deformations in highly stressed regions. Further elastic deformations of the hull girder and the double bottom occur due to the variations

서만이 베어링內에서의 자아널의 올바른 方位를 決定할 수 있다. 이것은 特定 偏心分布에 대하여서만이 油膜壓力의 方程式을 誘導할 수 있고 풀수있다는 事實때문이다. 壓力을 積分함으로써 誘導되는 壓力分布나 베어링反力을 根據로 하여 생각중인 偏心을 評價할 수 있고 새로운 偏心을 推定하게 된다. 이러한 試行錯誤法, 即 反復法은 Reynold의 方程式에 있어서의 非線型性에 起因한다. [13]

Fig. 13에서 選定된 偏心分布에 대하여 最大 油膜壓力은 베어링길이에 대하여 多樣하게 變하고 있다. 그러므로 軸系配置過程에서는 베어링에, 特히 기다란 後部 船尾管베어링에 形成되는 油膜을 考慮에 넣지않으면 안된다.

通常 이 베어링에서는 後部에 位置하는 단지 한 支點만을 生覺한다. 그 位置의 選擇은 關聯되는 사람들의 經驗에 따르게 된다. 이것의 選定은 最初의 計算始作에 대한 重要 始點이 된다. 그러나 最終計算에서는 定置運轉狀態가 考慮되어야하며 이 때에 平均프로펠러 荷重과 함께 油膜도 考慮에 넣지않으면 안된다.

Fig. 13에서 偏心分布는 非現實의 일지 모르나 事實은 그렇지가 않다. 不均一한 흐름內에서의 프로펠러動作으로 因한 굽힘모멘트와 橫方向의 힘을 생각하여 볼 때 주로 重力에 의하여 發生하는 垂直力뿐만 아니라 水平方向의 힘도 또한 베어링으로 받혀주어야 한다는 것을 알 수 있다. 다음 章에서 이들 流體力學的인 힘과 모멘트가 軸系配置에 미치는 影響에 관한 몇가지 情報를 提示하겠다.

#### 4.2 軸心옴셋트와 船體變形에 대한 考察

軸系の 길이에 따른 베어링間隙은 相當히 差異가 있다. 船尾管베어링에서의 半徑方向間隙은 0.5mm臺이거나 이보다 크다. 中間軸베어링에 대하여 半徑方向間隙은 0.5mm臺이거나 이보다 작고 主機베어링에서는 0.15mm臺의 間隙을 갖는다.

그럼에도 不拘하고 軸系配置計算에서 半徑方向間隙에서의 이러한 差異들은 반드시 考慮된다고 볼 수 없다. [12] 이것은 베어링反力과 겹(gap) 및 색(sag)의 값이 設置中에나 運轉

in the draft.

Whereas most ships operate reasonably free from severe troubles, due to a wrong alignment, it may be concluded that there is a reasonable margin available in the alignment before the occurrence of damages sets in. Especially for fast ships and navy ships these margins may be not so wide as for a bulker, so that a more accurate alignment is needed in those ship types. With the present techniques on calculation and measurement it must be possible to quantify these margins per ship type or per propulsion type and research should also be devoted to such aspects.

## 5. The lateral shafting vibrations.

### 5.1 General comments.

The lateral vibrations of the line shafting concentrate generally at the propeller and propeller shaft vibrations. This is caused by the fact that the propeller is mounted on the free end of the shaft and therefore the fundamental mode and natural frequency are dominated by the propeller vibrations. Therefore these vibrations were also called whirling vibrations. However, not only the whirl motion of the propeller, but also its hydrodynamic effects and the oilfilm in the aftermost bearing have a coupling effect on the two normal components of the lateral vibrations. Therefore it is better to speak of lateral vibration, which horizontal and vertical components are coupled by several phenomena.

In this chapter the attention will be restricted to the most important items, which are:

- the alignment
- the hydrodynamic effects of the propeller.

### 5.2 The effect of the alignment.

The alignment determines the bearing reactions. In a first approximation the oilfilm stiffness in a bearing is proportional

중에 計算値와 같은 값으로 되지 않는다는 것을 意味한다.

같은 事實이 船體거어더와 二重底의 變形에도 適用된다. 建造와 進水中에 船體는 내려앉는다. 運航中에도 높은 應力을 받는 領域에서의 局部的 塑性變形으로 더욱더 내려앉는다. 船體거어더와 二重底의 그 以上の 塑性變形이 吃水의 變化로 인하여 發生하게 된다.

大部分의 船舶이 그릇된 軸系配置로 因하여 심한 障害를 받지않고 적당히 運航되고 있는 것은 損傷이 일어나기에 앞서 軸系配置에 相當한 餘裕가 있었던 것으로 結論을 내릴수 있을 것이다. 특히 高速船과 海軍艦艇에 대하여서는 이러한 餘裕가 貨物船만큼 크지 못하므로 이러한 種類의 배에서는 더욱 正確한 配置가 必要하다. 現在의 計算技法과 測定技術에 의하면 船型, 또는 推進型式마다 이러한 餘裕를 定量化하는 것이 可能하여야 하며 이러한 方面에 대한 研究도 또한 이루어져야만 한다.

## 5. 軸系の 橫振動

### 5.1 概要

軸系の 橫振動은 一般的으로 프로펠러와 프로펠러軸의 振動에 集中되고 있다. 이것은 프로펠러가 軸의 自由端에 設置되고 따라서 基本振動型和 固有振動數는 프로펠러振動에 의하여 支配된다는 事實에 起因한다. 그러므로 이러한 振動들을 휘돌림振動이라 부르기도 한다. 그러나 프로펠러의 휘돌림運動뿐만 아니라 그의 流體力學의 影響과 最後部베어링에서의 油膜이 橫振動의 두가지 正常的 成分에 聯成效果를 갖는다. 그러므로 橫振動이라 부르는 것이 더 좋으며 이 橫振動의 水平成分과 垂直成分은 몇가지 現象에 의하여 聯成하게 된다.

이 章에서는 가장 重要的 項目들에 限定하여 살펴보기로 한다. 이들은

- 軸系配置
- 프로펠러의 水力學의 影響

### 5.2 軸系配置의 影響

軸系配置는 베어링反力을 決定한다. 最初의

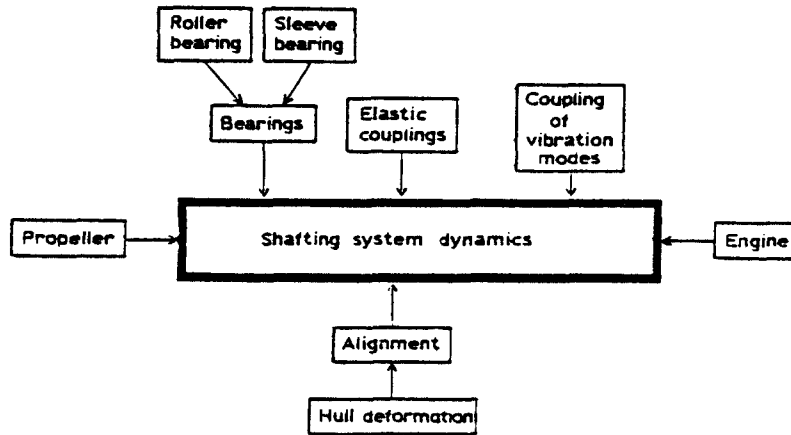


Fig. 14. Integrated propulsion shafting system. [14]

with the mean bearing reactions. Therefore it is important that the load distribution over the bearings is done judiciously. At least each bearing has to be loaded by a clearly defined force. This aspect is depicted in Figure 14.

In case a bearing is not really loaded by the shaft it follows that in the vibration analysis this bearing can not be considered as a support point, hence a drastic reduction in the natural frequency may follow. Therefore it is obvious that the alignment has a paramount influence on the lateral vibrations. As a consequence the alignment can be used to avoid resonances in certain speed ranges.

This task of the alignment is as important as the other task which is to avoid excessive wear or destruction of the bushing materials. However, also lateral vibrations can be the origin of extreme wear of the bushing, so that the alignment should always to have been performed in considering the vibrational consequences. It can be stated that the alignment is directly responsible for a low vibration level of the line shafting as suggested in Figure 14.

### 5.3 The role of the propeller.

In alignment and in vibration the propeller plays an important role. This is because of its large mass, its location at the free end

近似에 있어서 베어링의 油膜剛性は 平均베어링荷重에 比例한다고 한다. 그러므로 베어링에서의 負荷分布가 妥當하게 이루어져야함이 重要하다. 最少한 各베어링은 明白히 定義되는 힘에 의하여 荷重이 加하여져야만 한다. 이러한 狀況은 Fig. 14에 說明하였다.

어떤 베어링이 軸에 의하여 實際로 荷重이 加하여지지 않을 경우에는 振動解析에 있어 이 베어링을 支點으로 생각할 수 없으며 따라서 固有振動數가 急激히 減少하게 된다. 그러므로 軸系配置가 橫振動에 무엇보다 顯著한 影響을 미치고 있음이 明白하다. 結局 軸系配置는 特定速度範圍에서 共振을 回避하는데 利用할 수 있다.

이러한 軸系配置作業은 붓싱材料의 過度한 磨滅이나 破損을 回避하기 위한 餘他の 作業만큼 重要하다. 그러나 橫振動도 또한 붓싱의 過度한 磨滅의 原因이 될 수 있으므로 軸系配置는 恒常 振動的인 結果를 考慮하여 遂行하여야한다. 軸系配置는 Fig. 14에서 示唆한 것처럼 軸系의 振動程度를 낮게하는데 直接 關係이 있다.

### 5.3 프로펠러의 役割

軸系配置와 振動에서 프로펠러는 重要한 役割을 하고 있다. 이것은 그의 質量이 크고 그것이 軸의 自由端에 位置하며 周邊의 물과 相

of the shaft and its interactions with the surrounding water.

In the alignment its weight and overhang have to be accounted for as well as the buoyance effects. For the running propeller the mean hydrodynamic loads can seriously affect the static alignment, as was already described in chapter 3, and therefore can have a large effect on the natural frequencies.

In the vibration analyses care has also to be taken of the interaction with the entrained water, like added mass, coupling terms and damping terms, [15, 16] as already recognized in the fiftieths by Jasper[2] and his contemporaries. However, these quantities are a function of the propeller geometry and not of the mass or inertia moments of the propeller as previously assumed. Figure 15 clearly shows how the blade area ratio affects these inertia terms. Also the pitch has an effect. [15, 16]

The hydrodynamic coupling terms are less known than the added mass term, but they are sometimes of comparable importance. In the lateral vibrations the coupling terms force the shaft in an elliptical vibratory motion, just like the gyroscope does. Their effect is at least as important as gyroscope. [17] Also the hydrodynamic damping plays a role in the response analysis, as well as with regard to the response as with regard to the coupling of the 2 normal components of the lateral vibration.

互作用하고 있기 때문이다.

軸系配置에서는 그의 무게와 외팔보로 되어 있는 점을 浮力の 影響과 마찬가지로 考慮하여야 한다. 作動中の 프로펠러에 대하여 平均 流體力學的 負荷는 이미 3章에서 說明한 바와 같이 靜的인 配置에 深刻한 影響을 미칠수 있으며 따라서 固有振動數에 큰 影響을 미칠수 있다.

振動解析에 있어서는 이미 50年代에 Jasper [2]와 그 時代 사람들에 의하여 알려졌던 것처럼 附加質量과 같은 附加水, 聯成項과 減衰項[15, 16] 等과의 相互作用도 考慮하도록 注意하여야 한다. 그러나 이러한 값들은 以前에 推定하였던 것처럼 프로펠러의 質量이나 慣性 모멘트의 函數가 아니고 프로펠러의 幾何學的 形狀의 函數이다. Fig. 15는 날개面積비가 이들 慣性項에 어떻게 影響을 미치고 있는지 明白하게 보여주고 있다. 릿치도 또한 影響을 미친다. [15, 16]

流體力學的 聯成項이 附加質量項보다 덜 알려져 있으나 이들이 때로는 比較的 重要하다. 橫振動에서 聯成項이 軸으로하여금 마치 자이로 효과가 그러한 것처럼 橢圓型의 振動을 일으키도록 한다. 그들의 影響은 最少한 자이로스쿠우프效果 만큼 重要하다. [17] 또한 流體力學的 減衰는 橫振動의 두가지 規準成分의 聯成에 대한 것처럼 應答에 대하여서 뿐만아니라 應答解析에서도 重要한 役割을 한다.

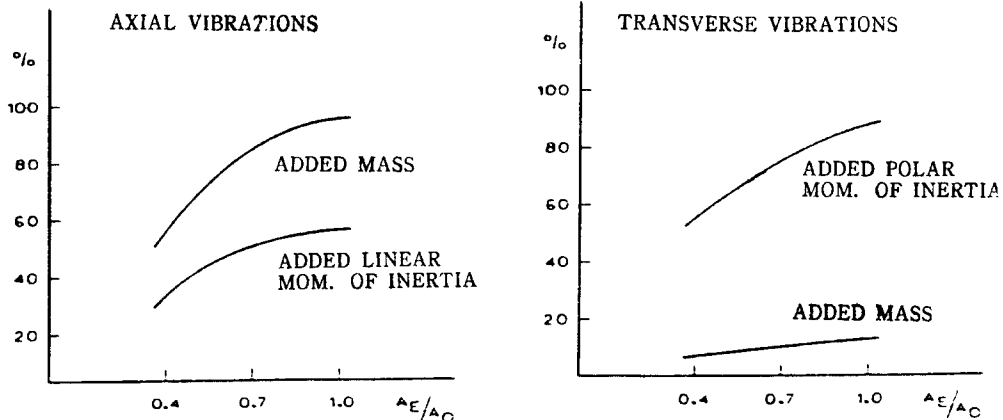


Fig. 15. Effect of blade area ratio on the inertia effects of the entrained water for a vibrating propeller. [15]

An other hydrodynamic effect is the fact that the propeller operates in a non-uniform wakeflow, so that there are also unsteady forces and moments on the propeller. The frequency of this unsteadiness is the shaft rate of revolutions times the number of blades. This is called the blade frequency. Of course also excitations with multiples of the blade frequency occur, but their effect is only small and therefore most times only the blade frequency phenomena have to be considered.

The origin of the blade frequency is easily understood in looking to the propeller. Then one realizes that after the propeller has rotated over one blade angle the system is as at the beginning, thus everything repeats after one blade angle has been traversed.

These unsteady or fluctuating forces and moments consist of 6 components as shown in Figure 8. To have some idea about their value they are often related to the mean thrust and torque as is done in Table 3, which gives some information on their magnitudes. This does not mean that there is a strict relation between the fluctuating components and the mean thrust and torque. This is because of the fact that the thrust and torque are generated by the mean values of the wake velocities and the fluctuating forces are generated by very specific harmonic components of the wakefield.

However, it is of practical importance to express the thrust fluctuations as a percentage of the mean thrust, and *vice versa* the torque fluctuations as a percentage of the mean torque. One reason is that a similar method is used to express the mean values for the bending moment and the transverse force in values related to the mean thrust and torque (see chapter 3). In Table 3 the general range of the percentages for these unsteady loads is given. No direct relation between the height of this percentage and the ship type has yet been found.

In this Table also the range for the lateral forces and moments is given. Whereas the lateral forces are basically due to the blade resistance and thus due to torque like phenomena, they are compared with respect to

또다른 流體力學的인 影響은 프로펠러가 均一하지 않은 伴流內에서 作動하며 따라서 非正常的인 힘과 모멘트가 프로펠러에 作用한다는 事實이다. 이 非正常性의 周波數는 軸의 回轉數에 날개數를 곱한 것이다. 이것은 날개數振動數라 한다. 勿論 날개數振動 倍數의 것도 發生하나 그들의 影響은 적고 따라서 大概의 경우 날개數振動만을 考慮할 必要가 있다.

날개數振動的의 根源은 프로펠러를 觀察하면 쉽게 理解할 수 있다. 이때에 프로펠러가 날개角度하나 만큼 回轉한 다음 系는 처음과 같게되며 따라서 날개하나의 角度가 經過한 다음 모든 것이 다시 反復됨을 알 수 있다.

이들 非正常的인, 또는 變動하는 힘과 모멘트는 Fig. 8에 보이는 것처럼 6個의 成分으로 構成된다. 그들의 값을 나타내는데 종종 Table 3에 보인 것처럼 平均推力과 토크에 대하여 나타낸다. Table 3은 이들의 크기에 관한 몇 가지 情報를 提供하고 있다. 이것은 變動成分과 平均推力 및 토크間에 嚴格한 關係가 있다는 것을 뜻하지 않는다. 이것은 推力과 토크가 伴流速度의 平均値에 의하여 發生한다는 것과 變動力이 伴流場의 特定調和成分에 의하여 發生한다는 事實 때문이다.

그러나 推力變動을 平均推力의 퍼센트로, 마찬가지로 토크變動을 平均토크의 퍼센트로 表示하는 것은 實用的인 면에서 重要하다. 한가지 理由는 비슷한 方法이 굽힘모멘트와 橫方向힘에 대한 값을 平均推力과 토크에 關聯하는 값으로 表示하는데 利用되고 있다는 點이다(3章을 보라). Table 3에 이러한 非正常的인 負荷의 一般의인 퍼센트範圍가 提示되어 있다. 이 퍼센트의 크기와 船型間에 아무런 直接的인 關係도 아직 發見되지 않고 있다.

이 Table에 또한 橫的인 힘과 모멘트에 대한 範圍가 提示되어 있다. 橫的인 힘이 基本的으로 날개의 抵抗에, 따라서 토크와 같은 現象에 起因하는 까닭에 그들은 프로펠러半徑에 의하여 나눈 平均 토크에 관하여 比較된다. 같은 理由로 굽힘모멘트는 平均推力을 프

**Table 3. Range of magnitudes of the blade frequency components of the propeller-shaft excitations.**

Thrust fluctuations	1-10% mean thrust ( $T$ )
Torque fluctuations	1-10% mean torque ( $Q$ )
Lateral force fluctuations	4-40% mean torque/Prop. radius ( $Q/R$ )
Bending moment fluctuations	2-20% mean thrust times Prop. radius ( $T \cdot R$ )

Note: These values apply to single screw ships.

the mean torque divided by the propeller radius. For the same reason the bending moments are related to the mean thrust times the propeller radius. This is a similar procedure as the one that is done for the mean lateral force and bending moment.

In chapter 3 it has been shown that the steady lateral force and bending moment can lead to appreciable changes in the bearing reactions. From Table 3 it follows that the unsteady lateral force and bending moment can have the same magnitude as the steady loads. In vibration there is additionally the dynamic magnification so that the effect of the unsteady loads can be more important than from a simple, static consideration follows. Therefore the need to avoid resonance is beyond any discussion.

But what about the off-resonance operation? What can be tolerated then? In this field no quantified information is available yet. Although general experience has not yet explicitly necessitated extensive research in this field, it is felt evident that further research is needed. This need is argued as follows.

In Figure 16 is shown [1] how strongly the lateral vibrations were reduced by applying a larger skew back for the propeller blades. In this way the unsteady loads on the propeller were reduced, from which the lower level of vibration follows. Whereas the cost for this modification must have been enormous, the need to reduce the vibrations is clear.

Therefore it is stated that in the alignment procedures also the forced vibration level has to be analysed, together with considerations on the steady lateral force and bending moment, the running condition and other aspects. Only such a complete alignment procedure can really be called "ra-

로펠러半徑으로 곱한 값에 關係지어진다. 이것은 平均的인 橫方向힘과 굽힘모멘트에 대하여 行한 것과 비슷한 方法이다.

3章에서 正常的 橫方向 힘과 굽힘모멘트는 베어링反力을 相當히 變化시킬 수 있음을 보인 바 있다. Table 3으로부터 非正常的인 橫方向힘과 굽힘모멘트는 正常荷重과 같은 크기를 갖을 수 있음을 알 수 있다. 振動에서는 附加的으로 動的 增大가 發生하기 때문에 非正常的인 荷重의 影響은 單純한 靜的인 것으로 生覺할 때보다 훨씬 더 重要하게 될 수 있다. 그러므로 共振을 避해야 된다는 것은 論議할 必要가 없다.

그렇다면 共振을 벗어난 곳의 運轉은 어떠한가? 그 때에는 얼마나 許容될 수 있는가? 이 分野에서 아직도 定量的인 情報은 아무것도 利用할 수 없다. 비록 一般的인 經驗이 이 分野에 있어서의 廣範圍한 研究를 아직은 明白하게 必要로 하고 있지는 않으나 더욱더 많은 研究가 必要함을 分明히 느끼게 된다. 이러한 必要性을 다음과 같이 論할 수 있다.

Fig. 16에서 프로펠러 날개에 대한 큰 스쿠백을 주게 되면 橫振動이 매우 줄어들 수 있다[1]. 이러한 方法으로 프로펠러에 대한 非正常的인 負荷가 減少하고 그 結果로 振動이 줄어들게 된다. 이러한 修正에 대한 經費는 莫大하겠지만 振動을 줄여야 될 必要性은 明白하다.

그러므로 軸系配置에 있어서는 強制振動의 程度도 正常的 橫方向힘과 굽힘모멘트 運轉條件과 餘他問題에 關한 配慮와 함께 分析되어야 한다고 말할 수 있다. 이러한 完全한 配慮過程만이 참으로 合理的이라 말할 수 있으며 이러한 完全한 措置가 欠한 모든 過程은 이러한 말에 適合하지가 않다.

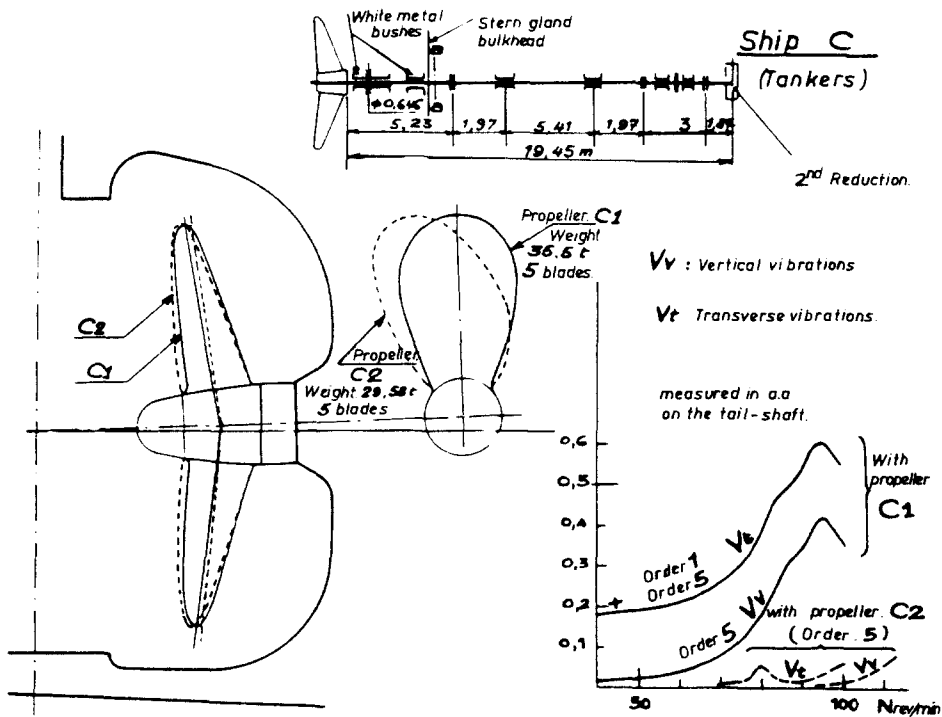


Fig. 16. Influence of the skew of the propeller blade on the flexural vibrations of the tail-shaft. [1]

tional”, all procedures that lack this complete treatment do not deserve this qualification.

With present computer facilities and with present knowledge of the various physical phenomena it is no longer difficult, neither expensive nor time consuming to perform a complete alignment procedure. It only requires the effort to formalize and to standardize these procedures. Therefore some further study has to be performed so that these procedures can be defined.

### 6. Closing Remarks

Considering the remarks in the previous chapters one wonders why generally [12] only the static alignment, that is the alignment for the non-running shaft, is considered. This surprise is further increased by the fact that obvious possibilities are not yet standard. Although one is talking of the

現在の 計算機能力과 여러가지 物理的 現象에 대한 오늘날의 知識으로써 完全한 軸系配置過程을 遂行하는 것이 이미 어려운 것도 아니며 비싼 것도 아니고 時間을 많이 뺏기는 일도 아니다. 그것은 단지 이러한 過程들을 公式化하고 標準化하는 努力이 必要할 뿐이다. 그러므로 이러한 過程들을 規定할 수 있도록 좀더 깊은 研究를 遂行하여야만 한다.

### 6. 結 言

앞章에서의 說明을 읽고 사람들은 어쩌서 作動하고 있지 않는 軸에 대한 配置인 靜的인 配置만을 一般적으로 考慮하고 있는지 異常하게 생각할 것이다[12]. 이러한 놀라움은 分明히 可能한 것도 아직 標準化되고 있지않은 事實에 의하여 더욱더 커지게 된다. 비록 사람들은 “合理的”인 配置를 遂行해야 될 必要性을 이야기하고 있기는 하나 日常의 適用에 있

need to perform a "rational" alignment, the rationality in the daily application is only little.

Maybe the reason is that at heart people is lazy and conservative and therefore will easily stay at old and well-known procedures. Only a catastrophe may awake man from his languishment. But also his intelligence.

As stated in the beginning, the increased possibilities lead to higher demands, which once again provokes the utilization of these better possibilities, etc. This mutual influence of demand and possibility spirals around the optimum solution and maybe has born the present slogan of quality control.

In the line shafting problems this quality control can be enhanced by a further computerization of the procedures, accounting for all the known phenomena. This will importantly extend the experience, because the designer becomes more specifically aware of the possible consequences of his decisions.

This computerization of the alignment procedure is a further and important step in rationalizing the alignment. No much further fundamental research is needed. It is only a utilization of existing knowledge and computer programs.

Ulsan, June 1986.

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## Appendix A.

In 1982 the Technical Committee No.6: "Shafting Syste Dynamics" from the International Cooperation on Marine Engineering system(ICMES) made a proposal on standardization of nomenclature and symbols that are used in the alignment of the shaft line. The present paper uses this proposal, which is repeated for completeness:

### Nomenclature and symbols in ship propulsion-shaft alignment Rational alignment of shafting

Position of individual bearings relative to the reference line in vertical and transverse planes\*), in order to realize a judicious distribution of bearing reactions.

#### Primary elastic alignment

Alignment conditions which take into account the elasticity of the line shafting only.

#### Secondary elastic alignment

Alignment conditions which take also into account the elasticity of bearing material and bearing support.

#### Static alignment

Alignment conditions when the shaft is non-running, which take into account, besides the elasticities, only the mass of shafts, propeller, gear wheel, crankshaft, etc..

#### Stationary-running alignment

Alignment conditions which take into account, besides the elasticities of the line shafting, bearing material and supporting structures, also the effects of oil film, the mean values of forces and couples externally applied to the system and the masses.

#### Dynamic alignment

Alignment conditions which take also into account the vibratory response of the line shafting to the dynamic excitations.

#### Definition of terms needed in alignment

##### Alignment reference line

Reference line drawn either between the centre of the first and last bearings of the line shafting or between the centre of the last and the last-but-one bearings of the line shafting.

offsets and slopes of bearings are referred to this line in defining the shaft alignment.

##### Bearing offsets

Distance between bearing centre line and reference line in a plane normal to the axis, to be stated as vertical and transverse components.

\*) transverse(plane) is used for the horizontal transverse (plane).

### **Bearing slopes**

Inclination of bearing center line with respect to the reference line in both transverse and vertical plane.

### **Reaction influence number (RIN)**

Change in the bearing loads for unit displacement of one bearing in both vertical and transverse planes (stiffness matrix of line shafting). Used to calculate the effects of bearing offsets in order to derive the judicious distribution of bearing loads.

### **Jack-up factor**

Ratio of the actual load on the bearing to the load measured by jackup test.

## **Misalignment**

Misalignment is the non-parallelism between shaft and its bush (or bearing shell). (It is an inevitable phenomenon, even in rational alignment, and it depends on the alignment technique used or the operational condition considered).

### **Definition of terms to express misalignment**

#### **Angular misalignment : $g = \tan\alpha$**

Angle between the centre lines of shaft and bearing respectively.

#### **Inplane misalignment**

Misalignment in vertical or transverse plane only.

#### **Skewed misalignment**

Misalignment in vertical and transverse plane. (the shaft is lying tilted and across the bearing).

#### **Static misalignment**

Misalignment for non-running conditions, taking into account only the mass of shafts, propeller, wheel, crankshaft, etc..

#### **Stationary-running misalignment**

Misalignment for given running conditions also taking into account effect of oil film and the mean values of forces and couples externally applied to the system.

#### **Dynamic misalignment**

Misalignment which also takes into account the

vibratory response of the line shafting to the dynamic excitations.

## **Bearing and supports**

### **Distinction bearing and support**

Under Support(s) is understood bush(es) of stern tube incorporated into the steelwork of the aft part of the ship. For alignment purposes there can also be temporary supports.

### **Slope boring of the stern tube**

Non-parallel position of boring axis of housing of bush (es) in respect to the reference line (in view to realize a rational alignment).

### **Slope boring of the bush**

Non-parallel position of boring axis of working surface of the bush(es) in respect to the reference line in view to realize the most homogeneous contact conditions alongside the bush(es) between the journal(s) of the tail shaft and the working surface of the bush. It can be single, double or even multi-sloped.

## **Alignment procedures and checkings**

### **Light line method**

Checking of offsets of bearing centre by means of optical devices.

### **Sag and gap method**

Method of setting or checking the alignment using the relative position between two consecutive coupling flanges (eccentricity and opening).

### **Jack-up tests**

Checking the static bearing reactions by actual measurement of the static reaction or R.I.N..

### **Bending moment method**

Calculation of reaction values(static and dynamic) in way of equivalent support/bearings from results of measurements of bending moments, in way of several sections of line shafting rationally chosen.

### Symbols in ship propulsion-shafting alignment

(For clarification, the dimensions of each symbol are given between square brackets, using length (L), mass (M), and time (T) as the basic quantities. The M. K. S. units are readily derived, —stands for non—dimensional, like the Reynold's and Sommerfeld's

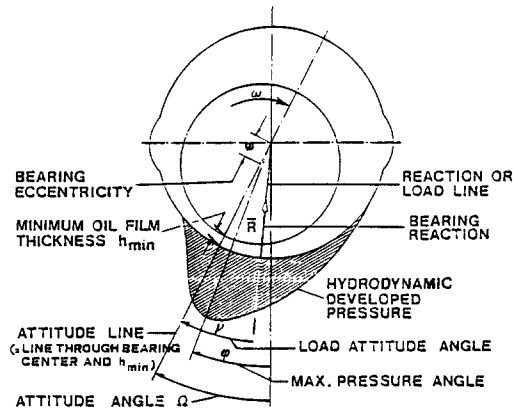


Figure A1. Illustration of some symbols in ship propulsion-shafting alignment,

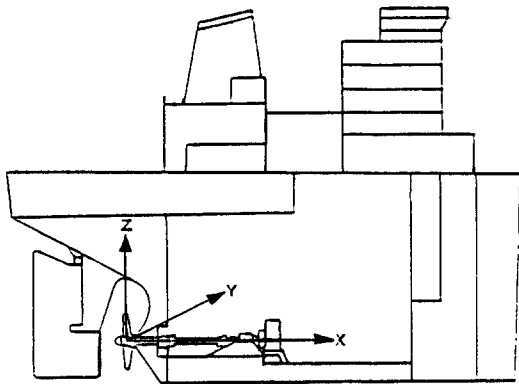


Figure A2. Choice of the co-ordinate system. (XYZ axes are right-handed : X pointing ahead Y pointing port side Z pointing upwards)

numbers, etc.. See Figure A1 for further definition of the symbols and Figure A2 for the co-ordinate system).

symbol	Definition
$c$	radial clearance of bearing ( $= R - r$ ) [L]
$d$	shaft diameter [L]
$D$	bore of bearing [L]
$E$	modulus of elasticity (Young's modulus) [ $L^{-1}MT^{-2}$ ]
$e$	eccentricity between shaft and bearing centerlines [L]
$F$	external transverse force [ $LMT^{-2}$ ]
$f_{iy}, f_{iz}$	offset from reference line of bearing $i$ in transverse (Y) and vertical (Z) direction [L]
$g$	gravity acceleration [ $LT^{-2}$ ]
$h$	oilfilm thickness, a function of location (and time) [L]
$h_{min}$	minimum oilfilm thickness [L]
$I$	area moment of inertia [ $L^4$ ]
$k$	jack-up factor [-]
$K_{iyy}, K_{izz}$ $K_{iyz}, K_{izy}$	stiffness terms of bearing $i$ in transverse (Y) and vertical (Z) directions [ $MT^{-2}$ ]
$l$	bearing length [L]
$L$	bearing span [L]
$M$	external bending moment or couple [ $L^2MT^{-2}$ ]
$N$	number of shaft revolutions per minute [ $T^{-1}$ ]
$n$	number of shaft revolutions per second [ $T^{-1}$ ]
$p$	load per unit length ( $= \frac{W}{l}$ ) [ $MT^{-2}$ ]
$p_{mean}$	mean pressure on projected area ( $= \frac{W}{ld} = \frac{p}{d}$ ) [ $L^{-1}MT^{-2}$ ]
$p_{max}$	maximum pressure in oilfilm [ $L^{-1}MT^{-2}$ ]
$q$	tangent of angular misalignment $\alpha$ [-]
$r$	shaft radius ( $= \frac{d}{2}$ ) [L]
$R$	bearing reaction [ $LMT^{-2}$ ]
$RIN$	reaction influence number [ $MT^{-2}$ ]

$R$	bearing radius ( $=\frac{D}{2}$ ) [L]	$\eta$	dynamic viscosity [ $L^{-1}MT^{-1}$ ]
$R_e$	Reynold's number ( $=\frac{Uc}{\nu}=\frac{Uc\omega}{\nu}$ ) [-]	$\nu$	kinematic viscosity $=\left(\frac{\eta}{\rho}\right)$ [ $L^2T^{-1}$ ]
$S_o$	Sommerfeld's number $\left(=\frac{P_{mean}}{\eta n}\left(\frac{c}{r}\right)^2\right)$ [-]	$\rho$	density [ $L^{-3}M$ ]
$T$	propeller thrust [ $LMT^{-2}$ ]	$\phi$	angle of maximum pressure with respect to vertical upward axis, positive in the direction of shaft rotation [-]
$U$	shaft peripheral velocity [ $LT^{-1}$ ]	$\Omega$	attitude angle, angle between vertical upward axis and attitude line, positive in the direction of shaft rotation [-]
$W$	bearing load [ $LMT^{-2}$ ]	$\omega$	angular speed of shaft [ $T^{-1}$ ]
$X, Y, Z$	the longitudinal, transverse *) and vertical co-ordinates see figure 2		Comment: bearing load $W$ and bearing reaction $R$ are opposite each other.
$\alpha$	angle between the axes of shaft and bearing [-]		
$\gamma$	angle between load and attitude line, positive in the direction of shaft rotation [-]		
$\varepsilon$	relative eccentricity ( $=\frac{e}{c}$ ) [-]		

\*) transverse is used for horizontal transverse.