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Modeling of Shaft Couplings for Alignment and Vibration Calculations

No. 20

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ABSTRACT

Shaft alignment and vibration calculations must account for the inertial, flexibility and damping properties of the couplings used to connect components in the propulsion train. After a brief description of the most common mechanical, fluid and electric couplings, the paper outlines the basic requirements of the numerical methods used to analyze linear and nonlinear shaft systems. Lumped parameter models are then proposed to account for the role of rigid, dental, disk, diaphragm and fluid couplings. Numerical examples and user guidance are provided with reference to three specific ship designs. Recommendations are included for further research along analytical and numerical lines.

INTRODUCTION

Shaft components in marine propulsion systems are joined by means of couplings that provide permanent connections while the ship is in service. When frequent disconnects are desired, the power transmission path is equipped with clutches, brakes or special disconnect couplings.

Despite their ubiquity, the teshnical literature on marine couplings in surprisingly meager. The basic source of information consists of manufacturer catalogs but their content does not always concentrate on technical characteristics. A recent, very informative book by Mancuso, Ref. (1), goes a long way to mimimize this information void, but since it addresses many industries it is not particularly specific for the marine designer.

Brief treatments of the design, installation and maintenance of couplings can be found in standard texts of marine engineering, References (2,3), in books dealing with prime movers, References (4,5,6), and in publications dealing with geared transmissions, such as References (7,8,9,10).

The goal in this paper is to briefly describe the most common types of marine couplings and to present simplified methods for incorporating their characteristics in shaft system mathematical models for use in alignment and vibration calculations. An additional objective is to highlight those areas of analytical and experimental research that are needed so as to better understand the behavior of such devices.

TYPES OF COUPLINGS

Marine couplings can be divided into three main classes as outlined in Fig. 1. The first class includes all types of mechanical couplings, which are the most common. The second class includes all forms of fluid couplings. The last class includes electromagnetic couplings that are not very popular at the present time.

Mechanical Couplings

Mechanical couplings can be divided into two main sub-classes, rigid couplings and flexible couplings.

Flange couplings, whether forged integral with the shaft components or of the removable type, are almost universally employed in main shafting systems, and sometimes within gears and engines. The integrally forged couplings are sized in accordance with standard industry guidelines, e.g., Ref. (11); on the basis of classification society rules, e.g., Ref. (12); or in accordance with Navy specifications, e.g., Ref. (13).

Non-removable flange couplings are assembled either with body-bound bolts with heads or with tapered bolts without heads, as shown in Fig. 2. Removable flange couplings come in different configurations and may employ

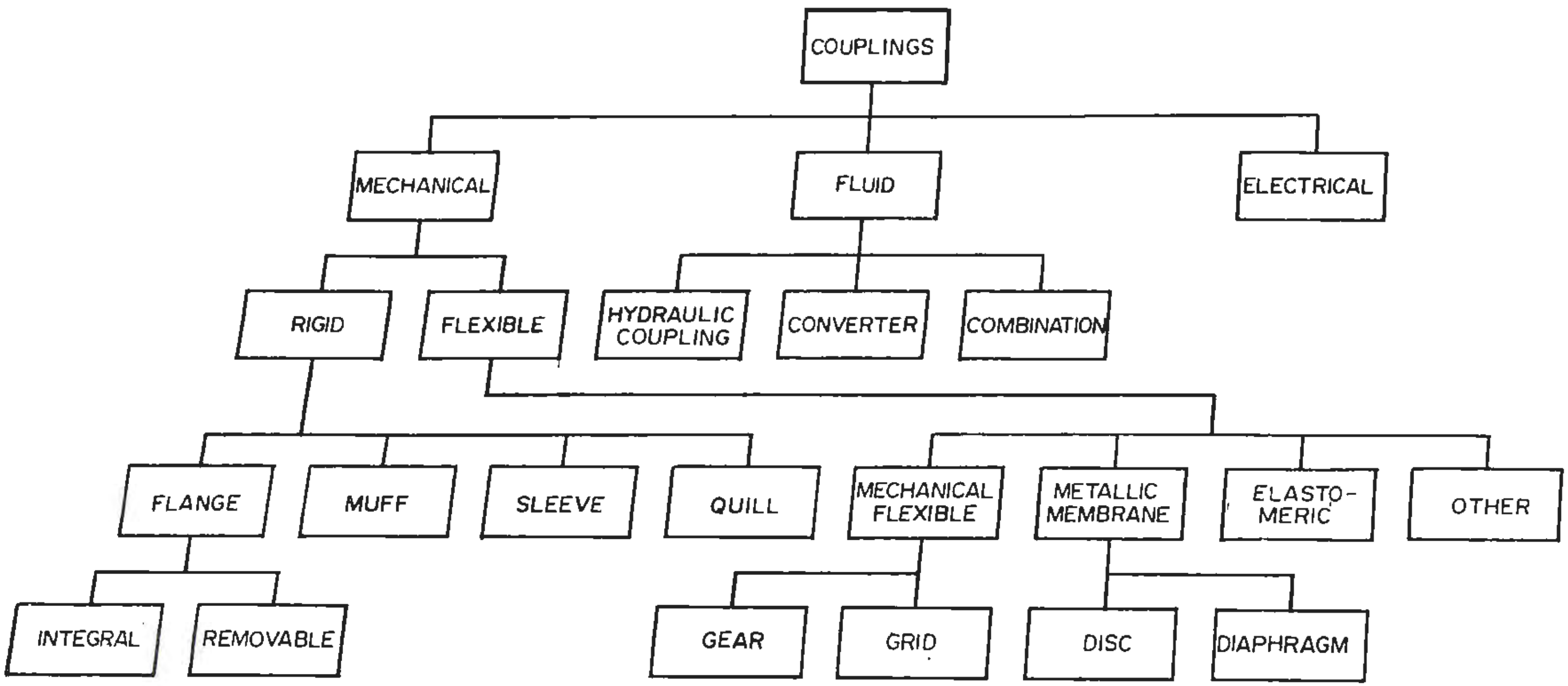


Fig. 1 Classification of Marine Couplings

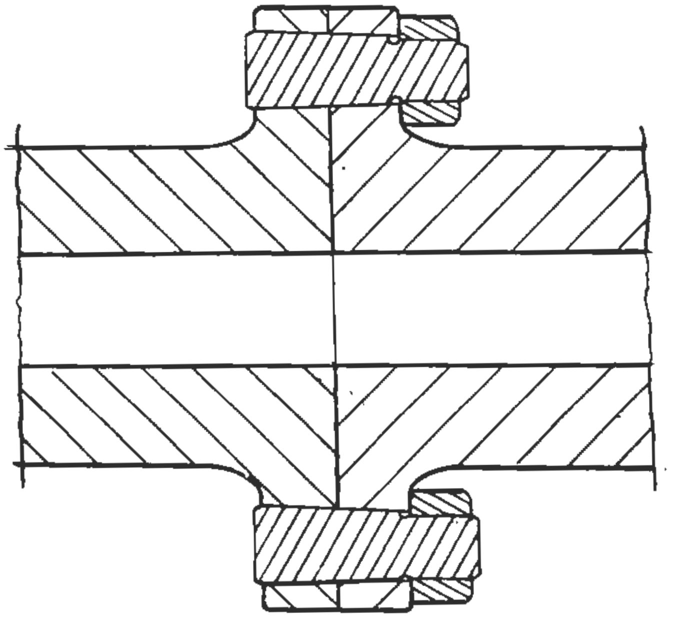


Fig. 2 Integral Flange Coupling

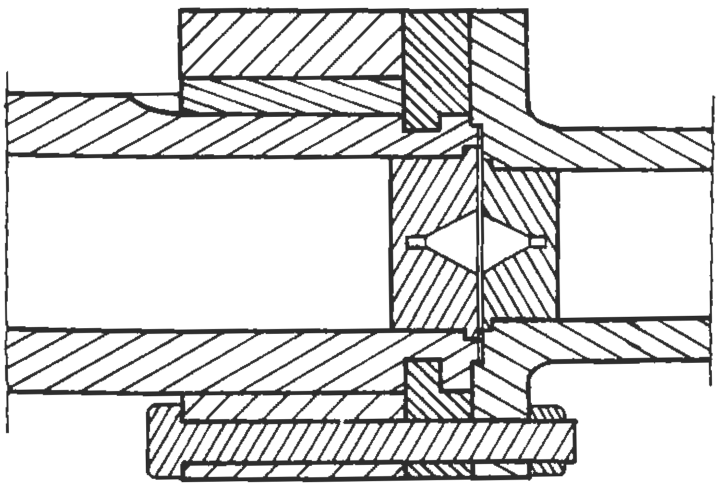


Fig. 3 Removable Flange Coupling

hubs on either one or both of the connecting shafts. The hubs are screwed onto the shafts which, as shown in Fig. 3, are usually keyed or ribbed.

Other types of rigid couplings that are sometimes found in shaft lines include muff couplings, as shown in Fig. 4, for mainly outboard sections, or sleeve, press-fitted couplings for tailshaft connections between inboard lineshafts, Fig. 5. Short length, quill shaft type (or make-shaft) couplings, used to connect prime movers to lineshafts and engines to gears, are but slight variants of the integral flange type. Mention should also be made of the ribbed flange coupling which is not often found in marine drives, Ref. (1).

Flexible Couplings

With few exceptions, flexible couplings are rarely found in main shaft systems. The use of such couplings is primarily confined within gearboxes, and their connections to prime movers such as diesel engines, gas turbines and steam turbines.

There are several types of flexible couplings and these can be

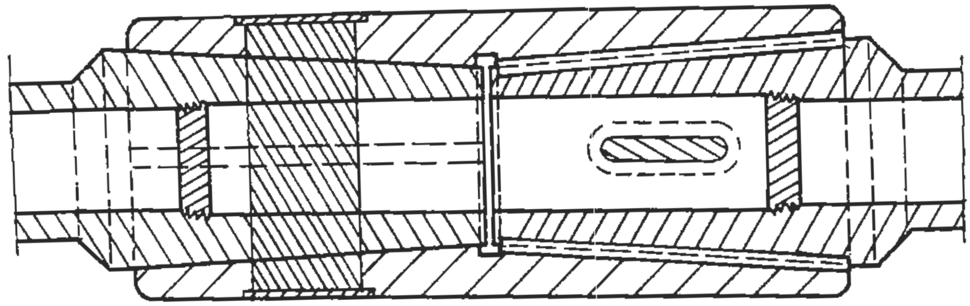


Fig. 4 Muff Coupling

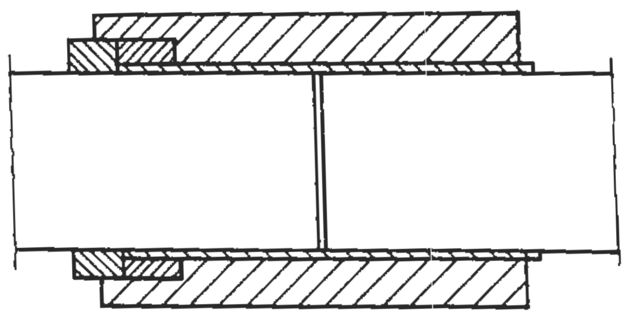


Fig. 5 Sleeve Coupling

conveniently divided into mechanically flexible, metallic membrane, elastomeric and other lesser known types. A brief review of such couplings appeared several years ago in Ref. (14). For industrial purposes the interested reader may also find a large number of papers in Ref. (15).

The behavior of dental (or gear type) couplings was investigated by Boylan, Ref. (16), by Mancuso, Ref. (17), and others. Gear couplings take many forms and an example of a two-mesh system is illustrated in Fig. 6.

Most of the flexible couplings employed in the marine industry are situated betweed diesel engines and gearboxes for the purpose of either detuning or dampening the torsional characteristics of such systems. The most popular amongst these couplings are grid type couplings, such as the one shown in Fig. 7. These have been treated extensively by several investigators, as reported, for example, in References (18) and (19).

Considerable use is also being made in modern marine applications of metallic membrane couplings, of which the disc type, Fig. 8, and the diaphragm type, Fig. 9, are the most representative, Ref. (20).

Fluid Couplings

Fluid drives, such as that shown in Fig. 10, have been used for many years between diesel engines and gearboxes. Interest in hydraulic couplings has recently been accentuated with the development of the reversible converter coupling, References (21,22). This device affords a convenient method of manufacturing a reversing reduction gear for unidirectional prime movers, such as gas turbines. The first application to a new U. S. Navy ship class is currently underway, Ref. (23), and, if successful, will provide a

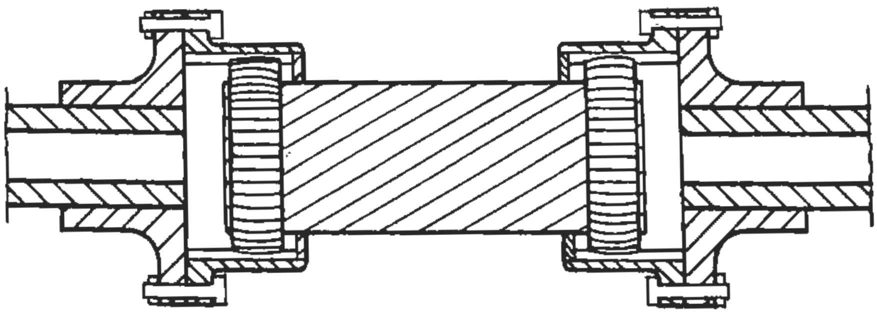


Fig. 6 Gear Coupling

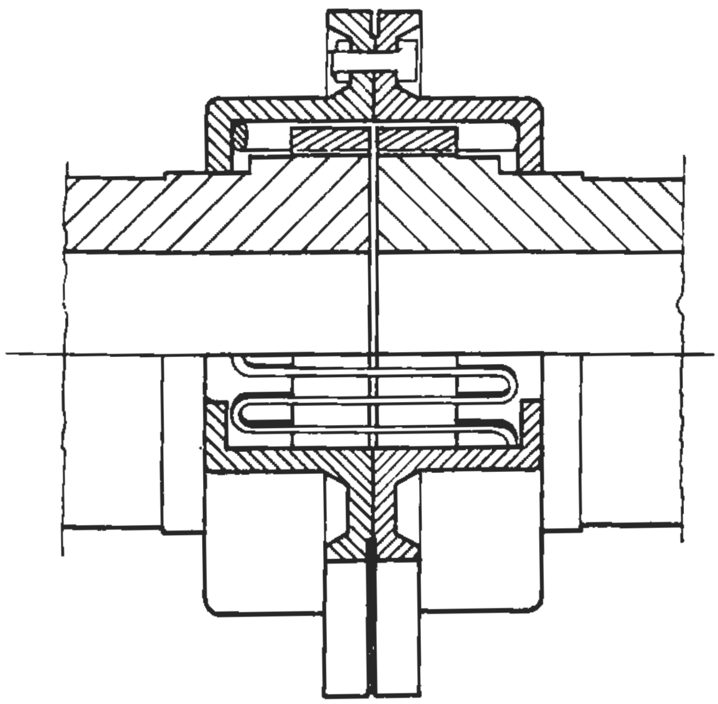


Fig. 7 Grid Coupling

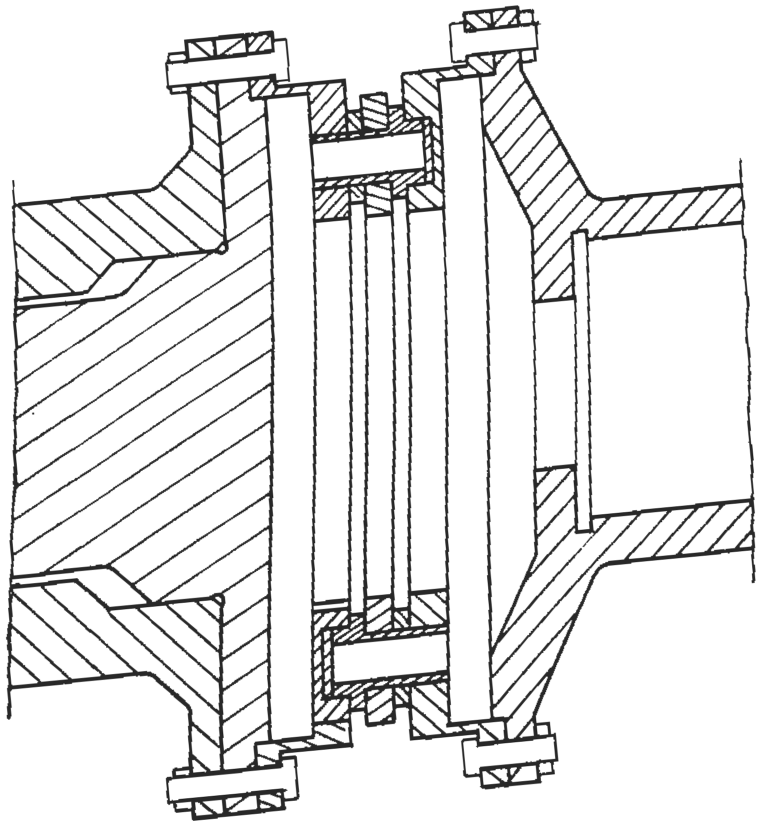


Fig. 8 Disk Coupling

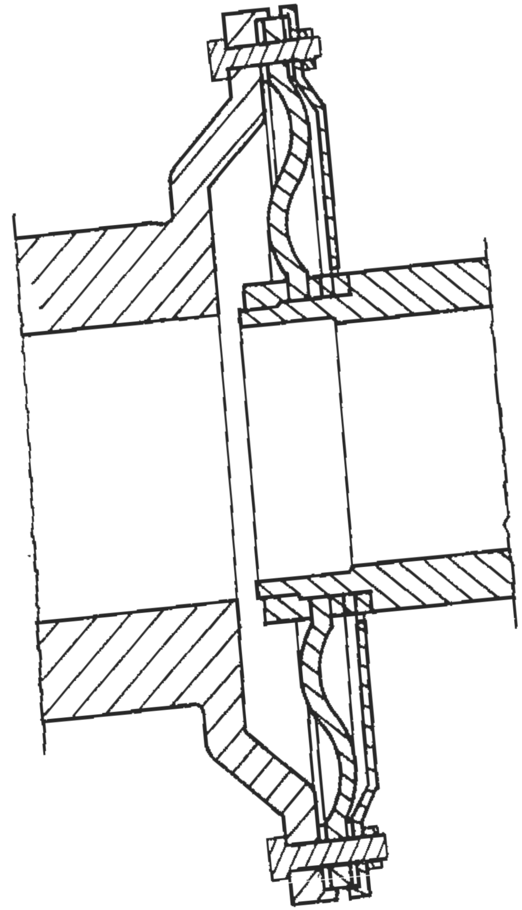


Fig. 9 Diaphragm Coupling

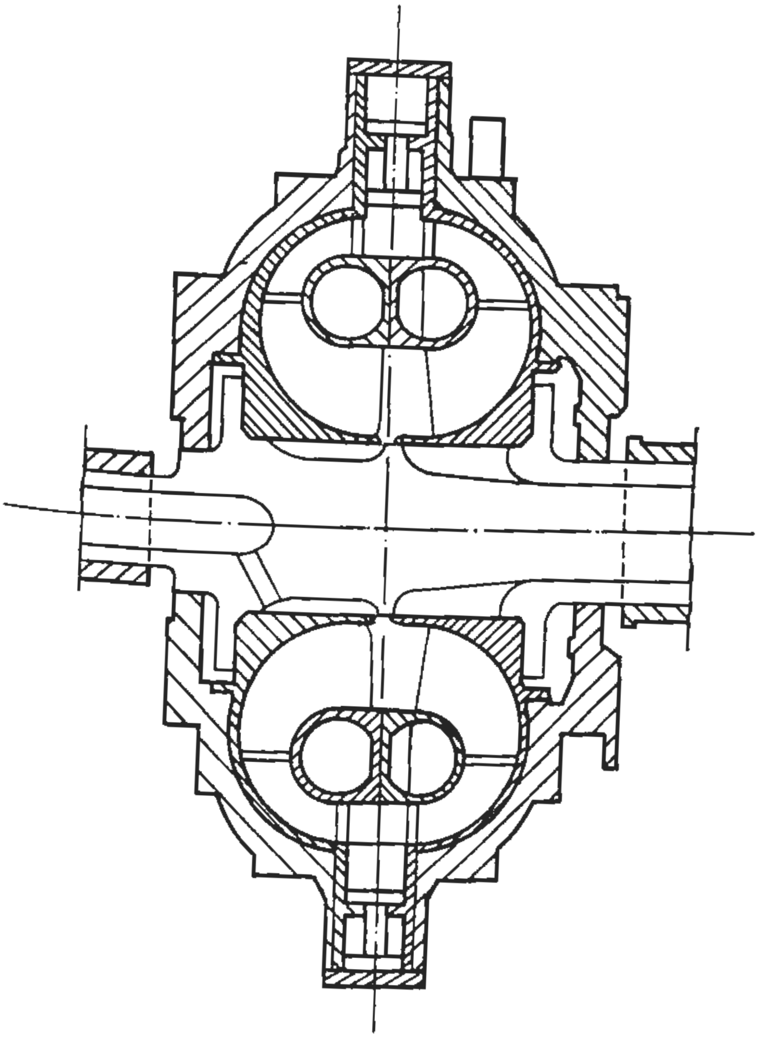


Fig. 10 Fluid Coupling

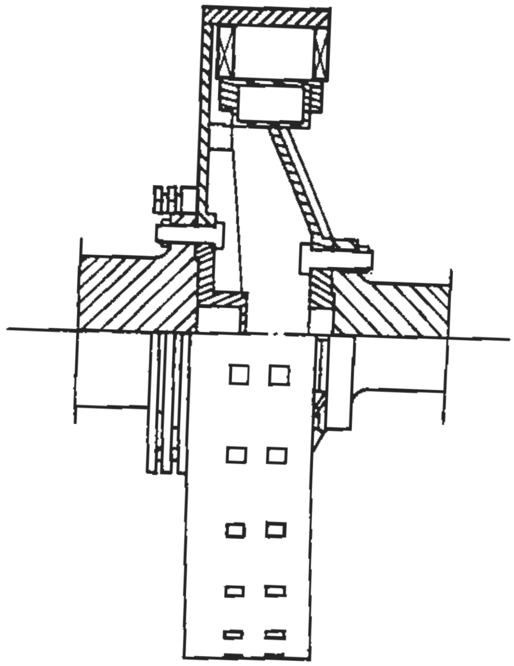


Fig. 11 Electric Coupling

challenging propulsor system to controllable pitch propellers that are universally adopted nowadays on gas turbine ships.

Electric Couplings

The transmission of power through electro-magnetic couplings of the type shown in Fig. 11 is briefly discussed in References (2,3). These couplings are outside the scope of this paper because they are not extensively utilized at the present time. A relatively recent treatment of electric couplings in the context of this paper may be found in Ref. (24).

SHAFTING SYSTEM CALCULATIONS

Scope

During the design and construction stages, it becomes necessary to predict the static and dynamic behavior of all shaft components in a main propulsion shafting system.

From a static standpoint, attention is primarily focused on the state of alignment, Ref. (25). Calculations are also performed to determine the total longitudinal deformation under the action of the propeller thrust, Ref. (26). Static twist calculations are rarely performed, except when checking full-scale calibrations for measuring shaft shear modulus.

From a dynamic standpoint, it becomes advisable to undertake calculations for axial, torsional and transverse vibrations. For linear systems, such computations are currently carried out for both free and forced vibrations using computer programs that are based on transfer matrix theory, Ref. (27), finite element techniques, Ref. (28), or finite difference methods, Ref. (29).

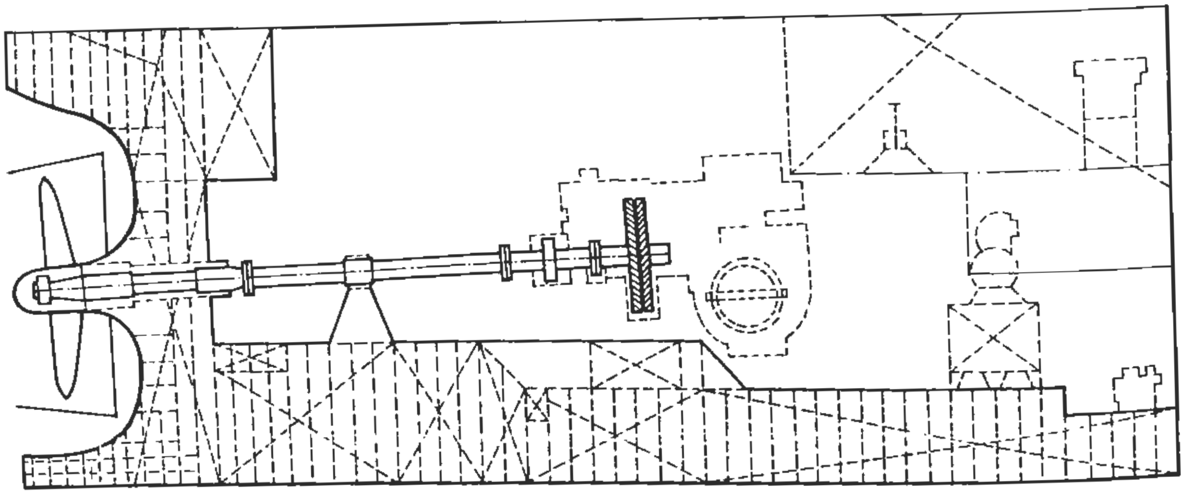


Fig. 12 Main Propulsion Shaft of ULCC Tanker

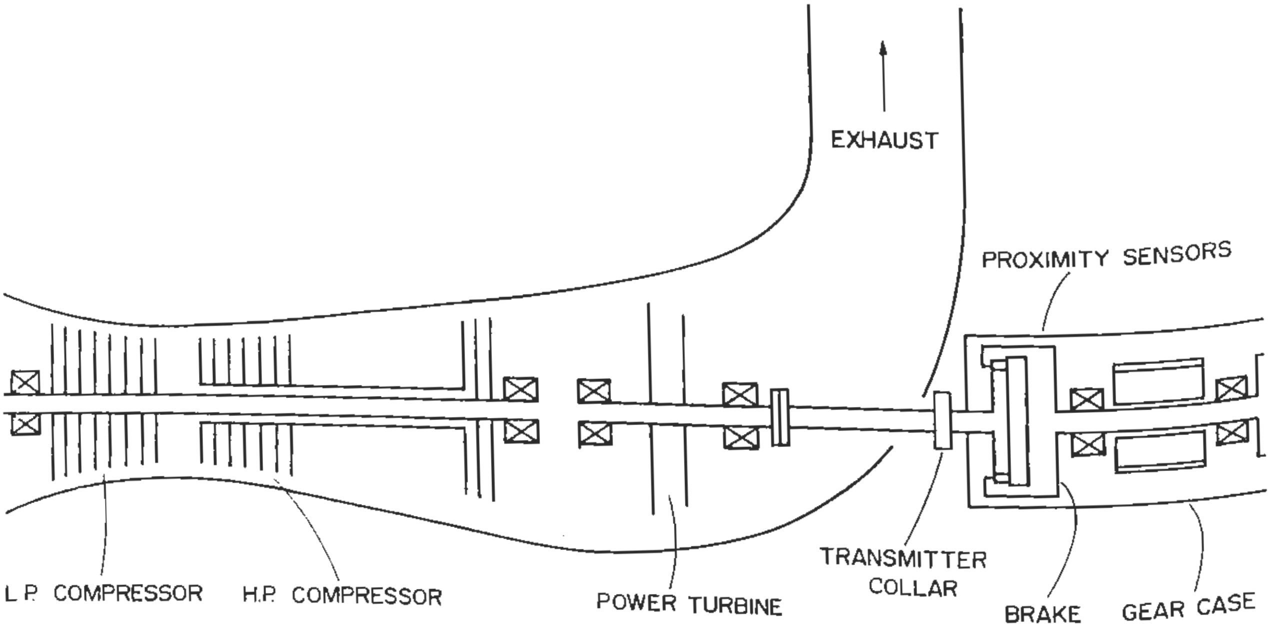


Fig. 13 Gas Turbine-Gearbox Shaft of USCGC HAMILTON

When significant nonlinearities are present, response calculations are performed in the time domain using suitable numerical methods, Ref. (30).

Ship Examples

To highlight the role played by couplings in such systems, reference will be made to the following three representative ship designs:

(1) A typical Ultra Large Crude Carrier (ULCC), the main shaft system of which is outlined in Fig. 12;

(2) A high-endurance, U.S. Coast Guard cutter, for which Fig. 13 depicts the arrangement between each gas turbine and gearbox;

(3) The Fast Combat Support Ship, which will be equipped with a

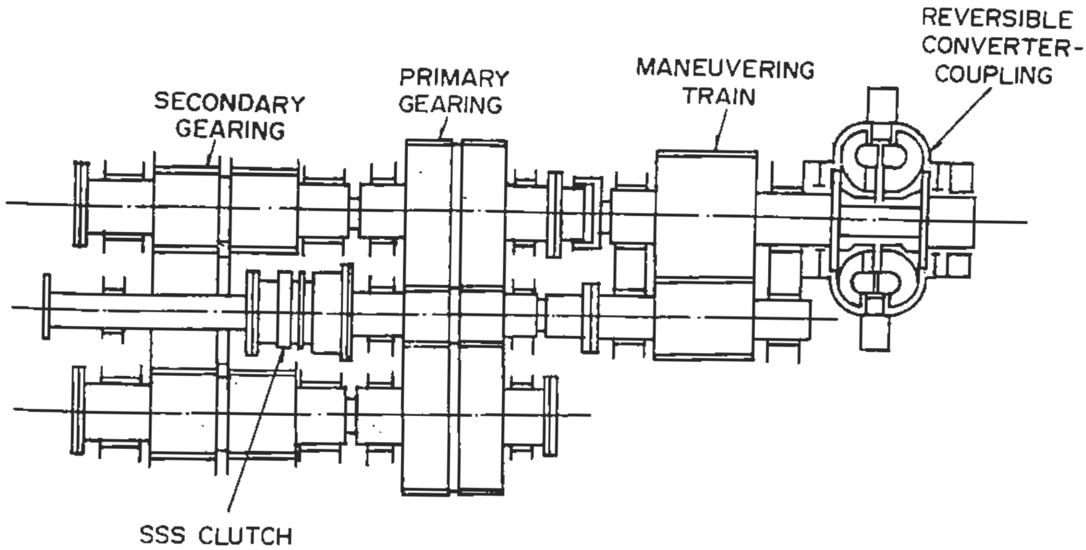


Fig. 14 Arrangement of Modern Reversing Reduction Gear

gearbox similar to that shown in Fig. 14, taken from Ref. (21).

The first example shows the main shaft of the S.S. SEAWISE GIANT, which at the time of its construction was the largest ship afloat. When gear failure occurred during trials very detailed analyses were undertaken to identify the shaft system dynamic characteristics including the state of alignment in the operating condition.

The second example deals with a ship class now undergoing renovation. When first built, this class became famous for its bold innovations in power plant technology, even though the leadship experienced some unusual problems later in life, Ref. (31).

The third example relates to the internal arrangement of a modern gearbox which is characterized by the simultaneous presence of diaphragm, disk and fluid couplings and is currently in the design stage.

All of the foregoing examples pertain to line structures which consist of mainly uniform shaft segments that are connected by couplings of various types. Within the scope of this paper interest arises in the manner in which the mass, stiffness and damping properties of such couplings can be taken into account.

Transfer Matrices

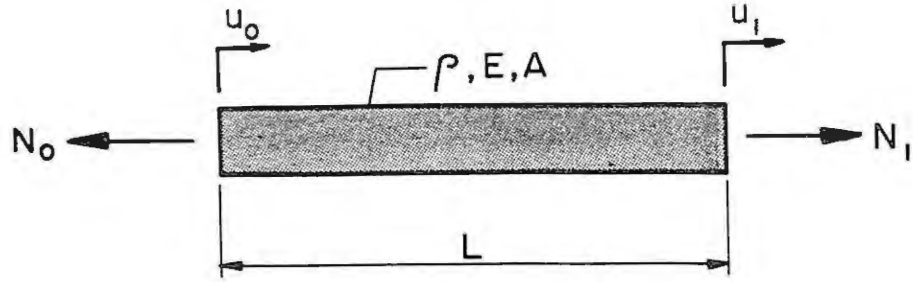
Despite its limitations, the transfer matrix approach offers considerable insight into the behavior of a system and will be used here to model couplings for linear analyses, Ref. (32).

A transfer matrix is an array of elements that relates the state (or boundary) vectors at any two points in a structure. The state vector at any point of an elastic system is a column vector the components of which are the displacements at that point and the corresponding internal loads. For static problems the transfer matrix elements depend only on system geometry and structure material properties. For dynamic problems some elements also become frequency dependent.

As long as the structure is uniform from one point to another, the transmission of information proceeds uninterrupted and without difficulty. When the structure topology changes, however, due to the presence of flexible couplings, supports or branches, special steps must be taken to account for the sudden changes in some of the elements of the state vector.

In contrast to shaft components that are structures with continuously distributed elasticity, mass and damping properties, couplings behave as lumped elements whose action occurs at a point. Such occurrences can be handled with discontinuity matrices which relate only some of the elements of the boundary vectors.

In the case of axial vibrations, the state vector at a point in the shaft system is given by



$$F = \begin{bmatrix} \cos\beta L & \dfrac{\sin\beta L}{AE\beta} \\ -\mu L\omega^2 \dfrac{\sin\beta L}{\beta} & \cos\beta L \end{bmatrix}$$

$$\beta = \omega\sqrt{\frac{\rho}{EA}}$$

Fig. 15 Field Matrix for Uniform Rod in Axial Vibration

$$\underline{s} = \{u, N\} \qquad (1)$$

where u is the axial displacement and N is the internal axial force. The corresponding vector for torsional vibrations is

$$\underline{s} = \{\phi, Q\} \qquad (2)$$

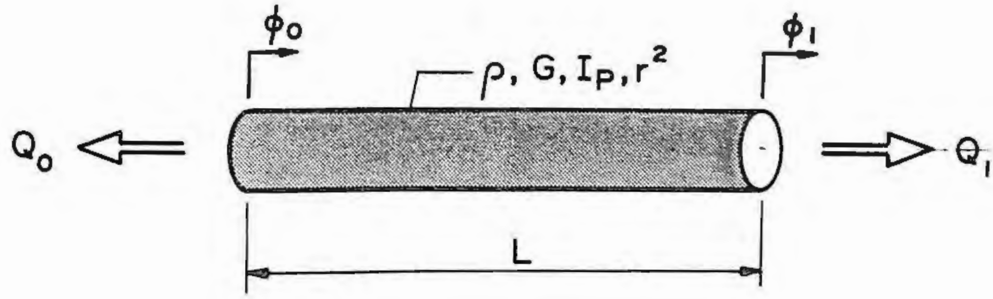
where ϕ is the torsional angle and Q is the internal twisting moment. For bending vibrations, the order of the state vector doubles since there are two displacements and two internal loads. For bending in one plane the state vector becomes

$$\underline{s} = \{w, \theta, M, V\} \qquad (3)$$

where w is the deflection, θ is the slope, M is the internal bending moment and V is the internal shear force.

The field matrix, $\underline{F}$, for a uniform rod in axial vibration is given in Fig. 15. The corresponding results for a uniform shaft undergoing torsional vibrations and a simple Euler-Bernoulli beam experiencing flexural vibrations are given in Figures 16 and 17, respectively.

Couplings will be modeled, in part, with lumped elements such as concentrated masses, springs and dampers. At the point at which they occur, the boundary vectors just before and just after the occurrence will be simulated by a point matrix, $\underline{P}$. It can be shown that the point matrix is the sum of the identity matrix $\underline{I}$ and a discontinuity matrix, $\underline{D}$. The discontinuity matrices for lumped mass, longitudinal spring and damper elements are provided in Fig. 18. The corresponding results



$$F = \begin{bmatrix} \cos\beta L & \dfrac{\sin\beta L}{GI_p \beta} \\ -\rho r^2 \omega^2 \dfrac{\sin\beta L}{\beta} & \cos\beta L \end{bmatrix}$$

$$\beta = \omega r \sqrt{\frac{\rho}{GI_p}}$$

Fig. 16 Field Matrix for Uniform Shaft in Torsional Vibration

for inertia, torsional spring and damper elements are given in Fig. 19. Fig. 20 illustrates the discontinuity matrix for a rotary hinge consisting of a coil spring and a moment release. The discontinuity matrix of a pure moment release is more complex and cannot be presented in a simple form. For its derivation it is necessary to use information from adjacent in-span occurrences and the interested reader is referred to Ref. (32) for details.

The attractiveness of a transfer matrix approach is exemplified by the fact that the state vector at point n can be obtained by multiplication of all transfer matrices in such a way that

$$\underline{s}_n = \underline{T}_n \, \underline{s}_o \qquad (4)$$

where

$$\underline{T}_n = F_n P_{n-1} F_{n-2} P_{n-2} \cdots F_2 P_1 F_1 \qquad (5)$$

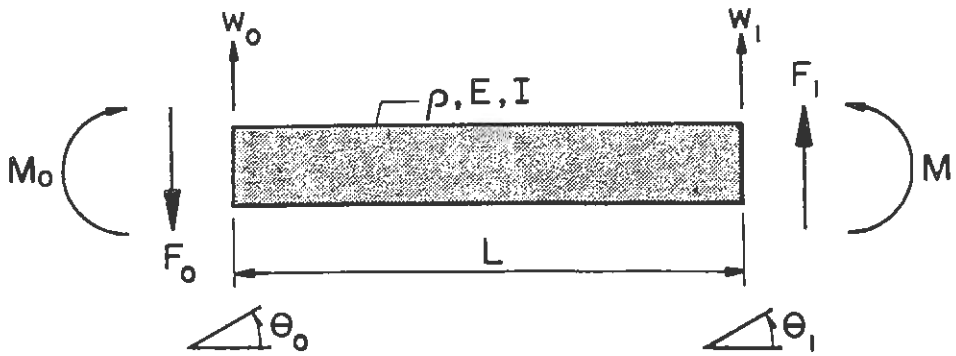
and

$$\underline{P}_i = \underline{I}_i + \underline{D}_i \qquad (6)$$

for i = 1,...,n.

Nonlinear Systems

There are instances in which the load-deformation relations for specific couplings are nonlinear. Typical examples are the axial force versus axial travel in a disk coupling or the torque-twist relation for a gear coupling when backlash between teeth is important. Under these circumstances, prediction of dynamic



$$F = \begin{bmatrix} A_3 & A_1 & -\left(\frac{1}{\beta\alpha^3}\right)A_2 & \left(\frac{1}{\beta\alpha^2}\right)A_4 \\ \alpha A_2 & A_3 & -\left(\frac{1}{\beta\alpha^2}\right)A_4 & \left(\frac{1}{\beta\alpha}\right)A_1 \\ -\beta\alpha^3 A_1 & -\beta\alpha^2 A_4 & A_3 & -\alpha A_2 \\ \beta\alpha^2 A_4 & \beta\alpha A_2 & -\left(\frac{1}{\alpha}\right)A_1 & A_3 \end{bmatrix}$$

$$A_1 = \frac{1}{2}(\sinh\alpha L + \sin\alpha L)$$

$$A_2 = \frac{1}{2}(\sinh\alpha L - \sin\alpha L)$$

$$A_3 = \frac{1}{2}(\cosh\alpha L + \cos\alpha L)$$

$$A_4 = \frac{1}{2}(\cosh\alpha L - \cos\alpha L)$$

$$\beta = EI$$

$$\alpha^4 = \frac{\omega^2\mu}{EI}$$

Fig. 17 Field Matrix for Euler-Bernoulli Beam Undergoing Transverse Vibrations

behavior can be pursued by iterative methods using linear models or by resorting to more complex nonlinear computational algorithms.

The iteration approach can be carried out as follows. A stiffness is assumed corresponding to a specific element displacement. The calculation is then performed and if the resulting displacement differs from that assumed, a new stiffness is selected corresponding to the computed response and the calculation is then repeated. Convergence can be accelerated by utilizing the equivalent linearization method, which has been used successfully to study nonlinear ship motions, e.g., Ref. (35).

The study of such problems with nonlinear methods requires that the

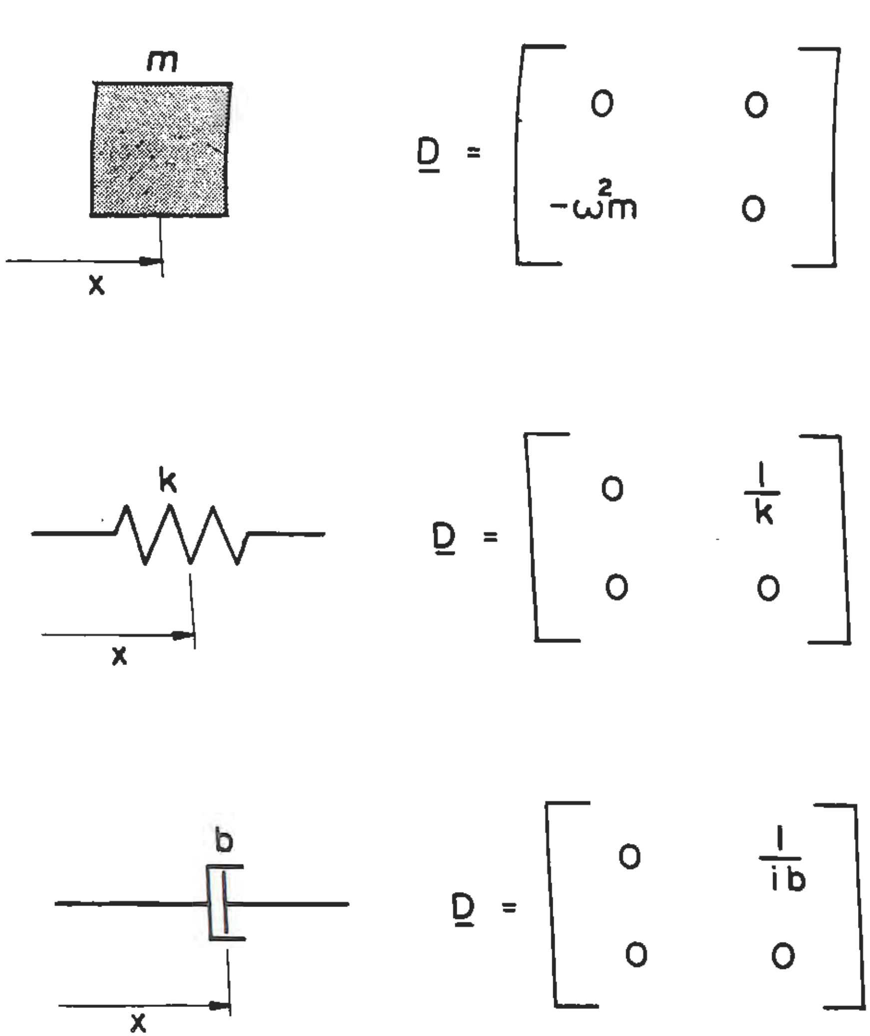


Fig. 18 Discontinuity Matrices for Lumped Mass, Spring and Damper Elements

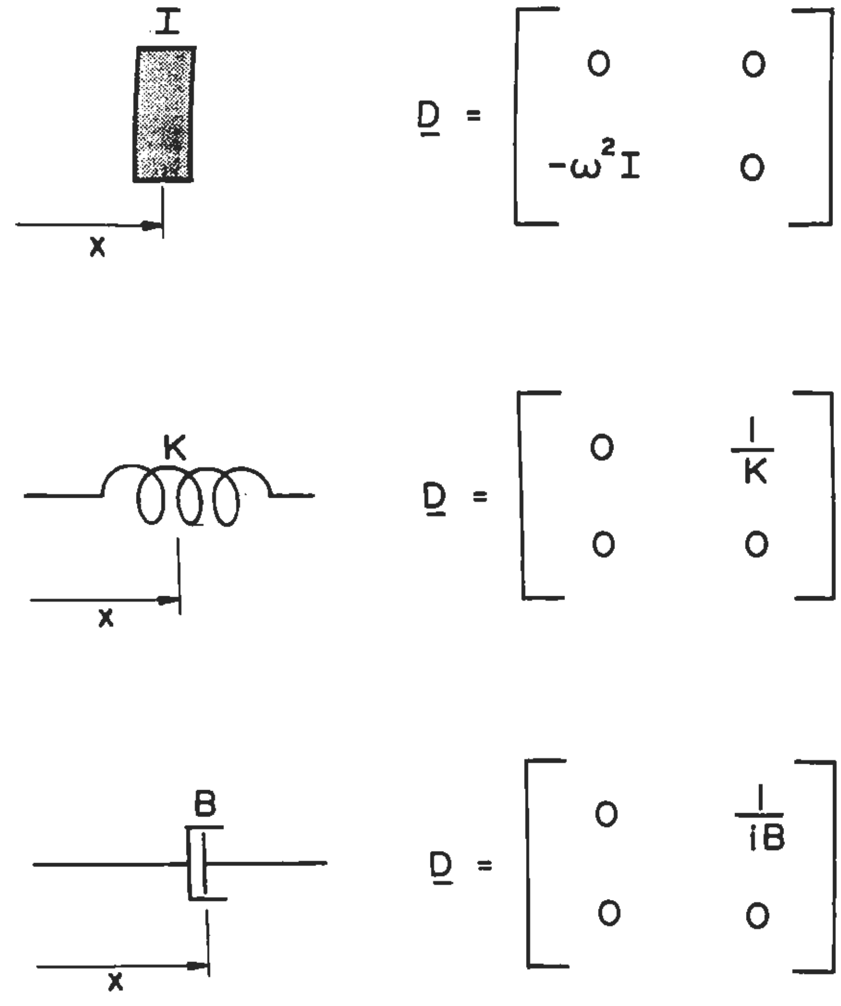
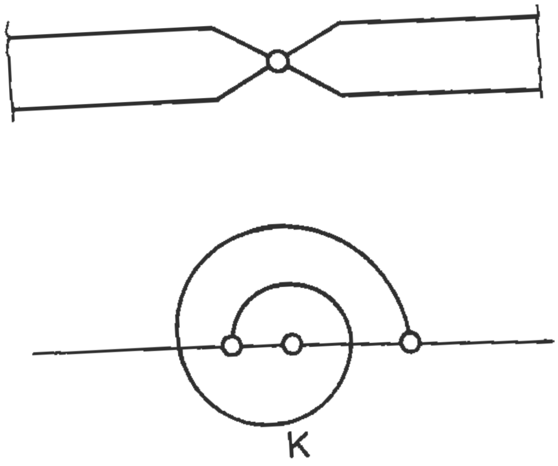


Fig. 19 Discontinuity Matrices for Lumped Inertia Torsional Spring Torsional Damper Elements



$$D = \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 0 & \frac{1}{K} & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix}$$

Fig. 20 Discontinuity Matrix for Moment Release With Lumped Spring in Bending

system be modeled by means of finite element or lumped parameter models and the analysis carried out in the time domain by numerical integration. Finite element models can be analyzed by a variety of integration schemes, Ref. (34). Lumped parameter models usually require that the equations of motion be recast in first order form which can then be integrated by well-established methods for solving numerically differential equations, Ref. (35). Computations of nonlinear responses are time consuming and expensive, particularly if the system models are comprehensive.

MODELING OF FLANGE COUPLINGS

As may be seen from Fig. 12 a flange coupling represents a sudden increase in diameter over a short distance along the shaft length. For this reason, it cannot be treated as a different beam segment with the same diameter, when alignment calculations are considered.

The flexibility of shafts involving abrupt changes in cross section has received attention and was investigated both theoretically and experimentally, Ref. (36). An exact

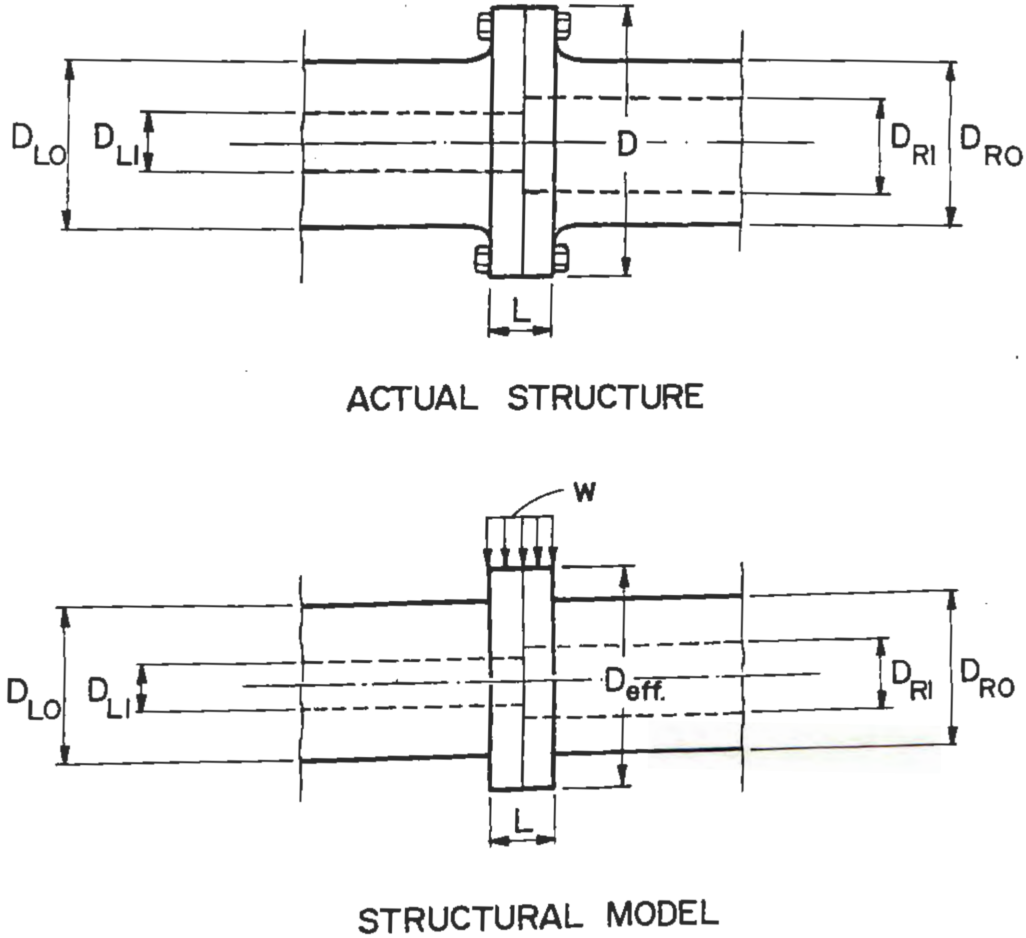


Fig. 21 Modeling of Integral Flange Couplings

solution to a simplified problem statement has also been obtained and a computer program FLANGE has been developed to determine the effective diameter of an equivalent shaft segment.

Fig. 21 shows how a shaft coupling is best simulated as an equivalent shaft segment of reduced diameter so as to better model the effective stiffness of that section. Since the weight properties do not alter, it is necessary in numerical computations to account for the weight lost from this type of calculations, as shown in Fig. 21.

Partial design guidance on the extent to which the shaft segment outer diameter must be reduced is given in Fig. 22 which has been prepared with the FLANGE program for shaft couplings designed in accordance with Ref. (11). The data apply to solid adjoining shafts of identical diameter. The values predicted by this scheme are certainly more accurate than those obtained by either ignoring completely the sudden increase in diameter or taking it fully into account, both of which are practices followed by the profession.

Fig. 23 shows the shaftline shape in the vertical plane for the hot operating alignment plan of the ULCC propulsion system mentioned earlier. In this case, proper modeling of the three shaft discontinuities near the reduction gear had a noticeable influence on local system flexibility and hence the gear bearing reactions.

In the case of torsional vibrations, flange couplings can be modeled with the use of formulae that appear in standard texts on the subject, e.g., Ref. (18). In most studies of longitudinal vibrations the assumption is made that flange couplings do not contribute any extensional stiffness and merely act as lumped masses. Transverse shaft vibration studies treat flange

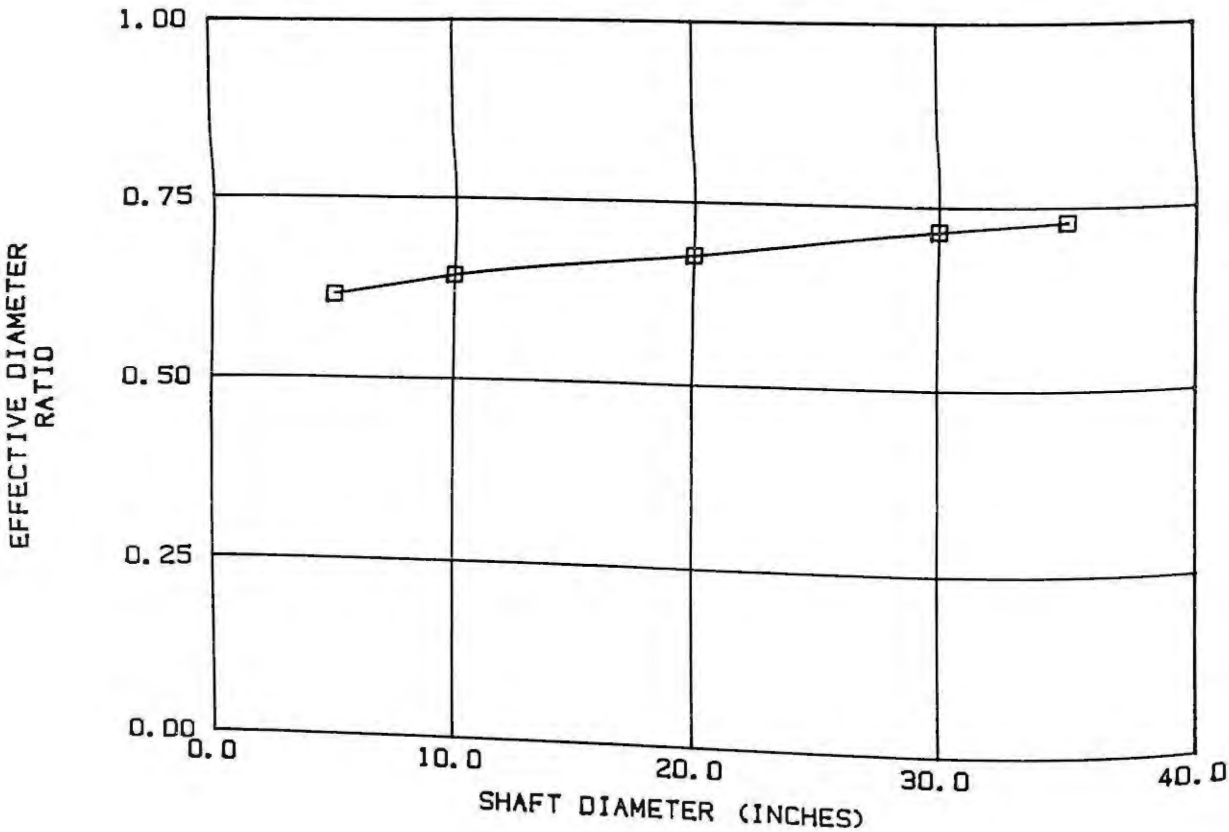


Fig. 22 Effective Diameter Ratio for Solid Shaft Flange Couplings

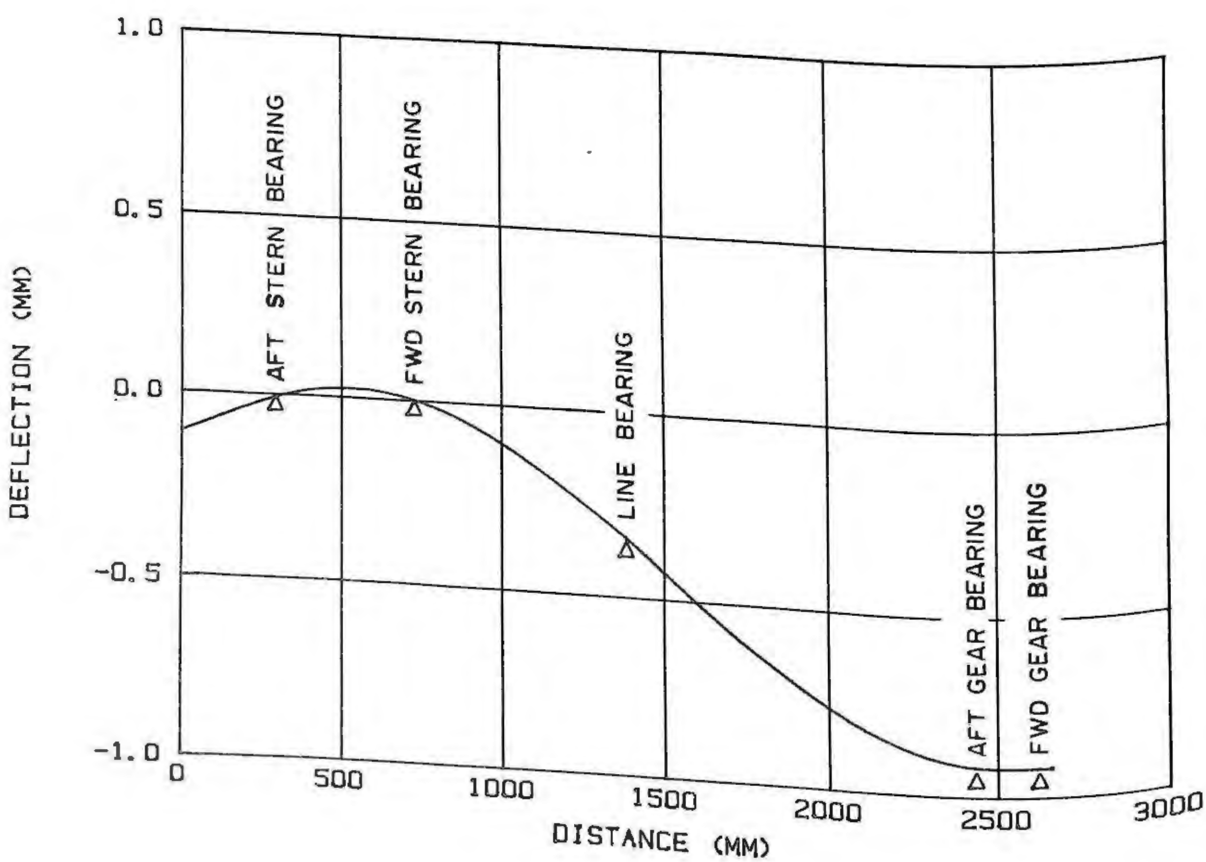
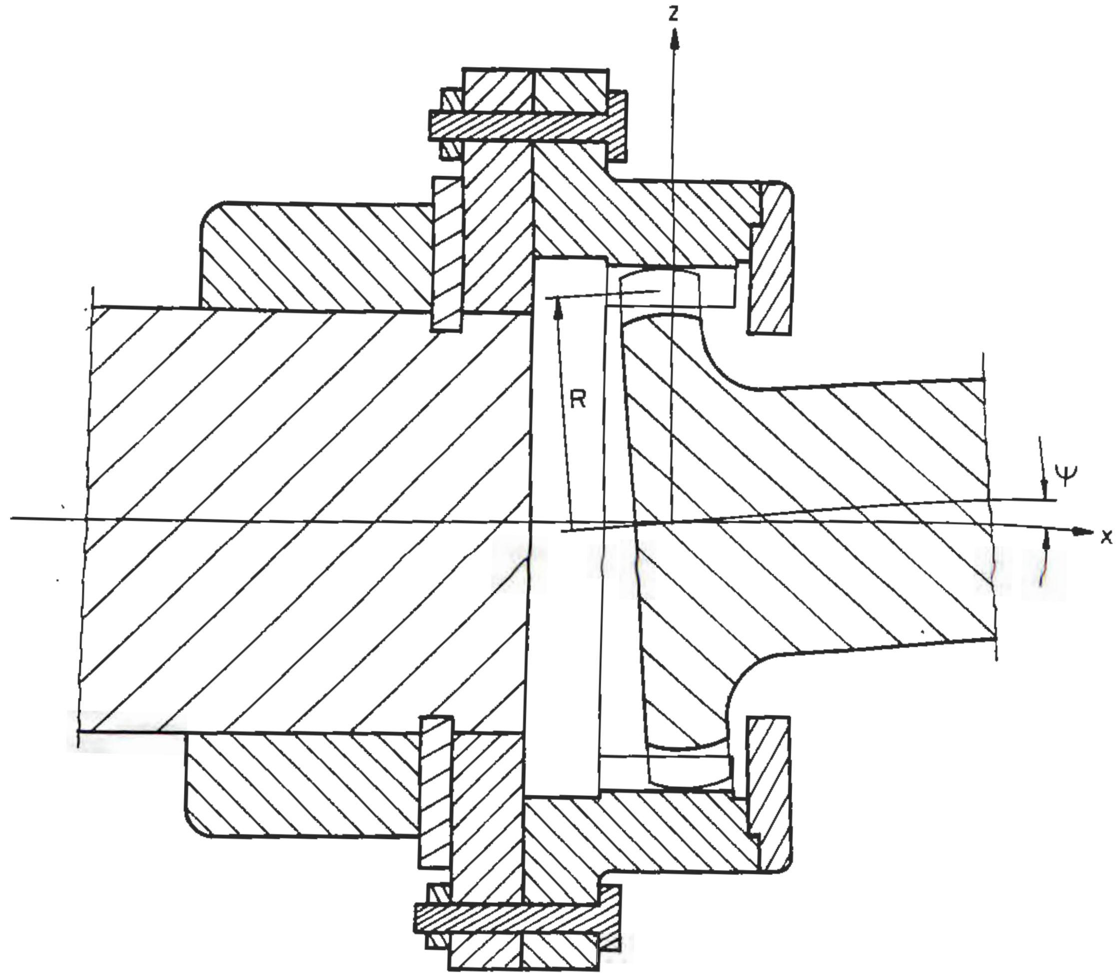


Fig. 23 Vertical Plane Shaftline Shape for ULCC System



$$M_y = \frac{M_x}{R}\left(\frac{\mu}{3} R \sin\frac{\epsilon}{2}\cos\psi + R_c \sin\psi\right)$$

$$M_z = \frac{M_x(\sin\psi + \mu\cos\psi)\sqrt{R^2 - [(\frac{R}{3})\sin(\frac{\epsilon}{2})]^2}}{R(\cos\psi - \mu\sin\psi)}$$

Fig. 24 Mancuso Formulae for Dental Coupling Moments

couplings in the same manner as that discussed above for alignment investigations.

MODELING OF GEAR COUPLINGS

The behavior of gear (or dental) couplings is perhaps the most complex to model for either static or dynamic conditions because of the role that friction plays between the hub and sleeve teeth. The difficulties associated with this type of coupling were discussed by Boylan in Ref. (16), and despite the passage of time, the state of the art has not advanced to any significant degree.

A simplified solution for the moments generated by a misaligned gear coupling was proposed by Mancuso in Ref. (17). The formulae derived for M_y and M_z, the moments in the vertical and horizontal planes respectively, are given in Fig. 24. In these equations, M_x stands for the steady

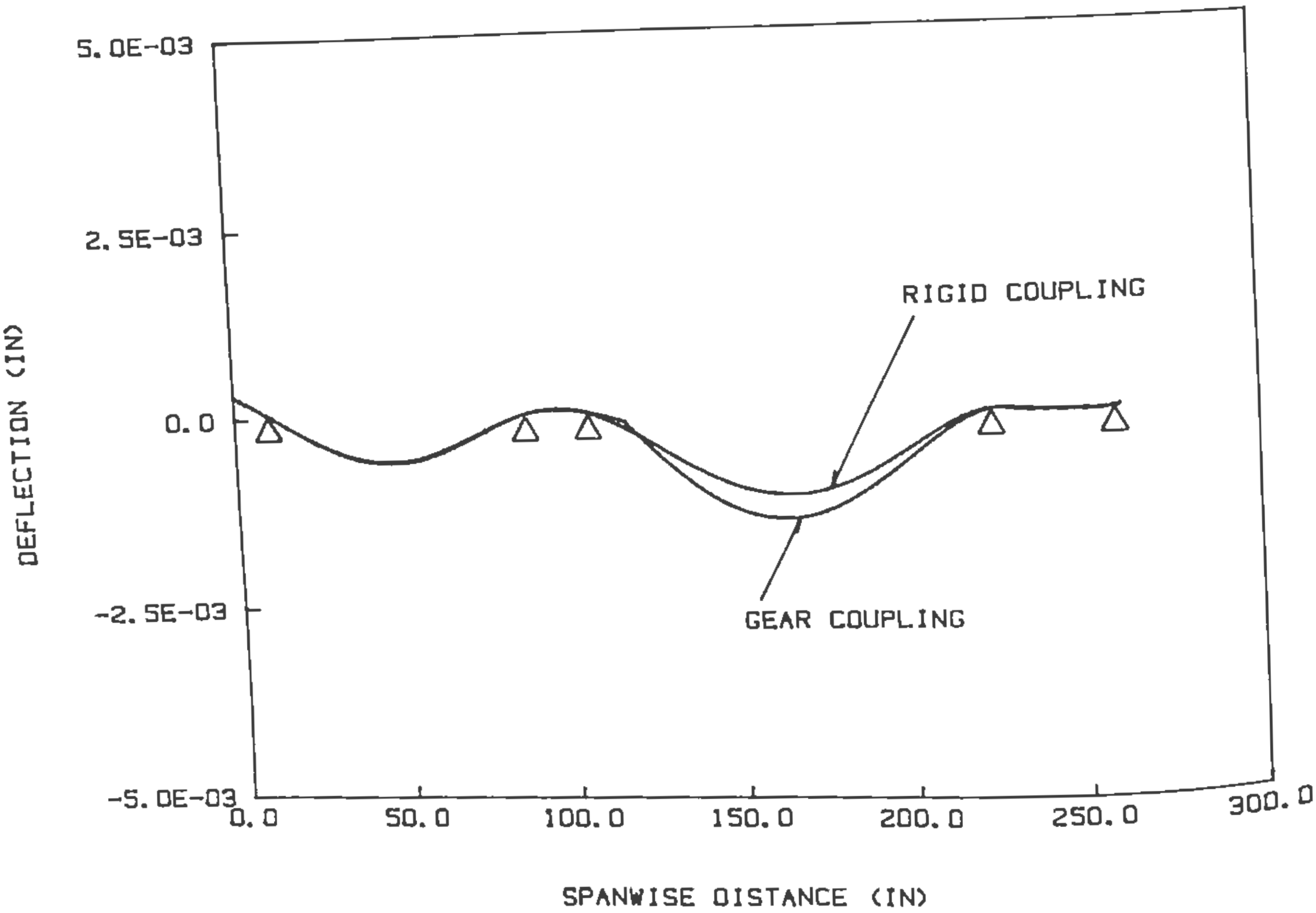


Fig. 25 Straight Line Alignment Plans for USCGC HAMILTON Gas Turbine-Reduction Gear Shafts

shaft torque, ψ is the angle of misalignment, μ is the coefficient of friction, R is the pitch radius of the gear, χ is the angle subtended by the tooth contact range and R_c is the hub tooth curvature radius.

For shaft static alignment calculations, gear couplings can be modeled as moment releases. At the location of that release, the slope will experience a sudden change but the deflection will be the same just before and after the coupling. To illustrate the nature of the results obtained with this modeling, attention is directed to Fig. 25 which shows plots of deflection for the renovated gas turbine-to-gearbox shaft systems on the USCGC HAMILTON. The results were computed with the ALNMEA program on the assumption that the system is straight aligned in all five bearings supporting the system. Also shown is a plot made under the assumption that the gear coupling was replaced by a solid flange coupling. The cusp in the deflection curve is more evident in Fig. 26 which is an enlarged plot of Fig. 25 in the vicinity of the coupling.

When the shaft rotates, the moment produced by the gear coupling is a function of the misalignment angle which may alter because of centrifugal and other effects. The coupling stiffness properties are best modeled by a moment release together with a lumped bending spring whose nodes gap the coupling length. Since the mis-alignment angle is not known a priori, the solution must be obtained iteratively starting with an assumed misalignment, computing the transmitted moment and comparing the assumed and computed misalignments until the results converge to within a small acceptable error.

In the case of axial vibrations, "lock-up" of the gear teeth in the mating parts will occur if friction is large enough to prevent free relative motion. Sliding between the coupling parts will occur so long as the axial force is larger than the frictional force.

In the case of torsional vibrations, a gear coupling offers new phenomena for treatment because of the backlash between the inner and outer

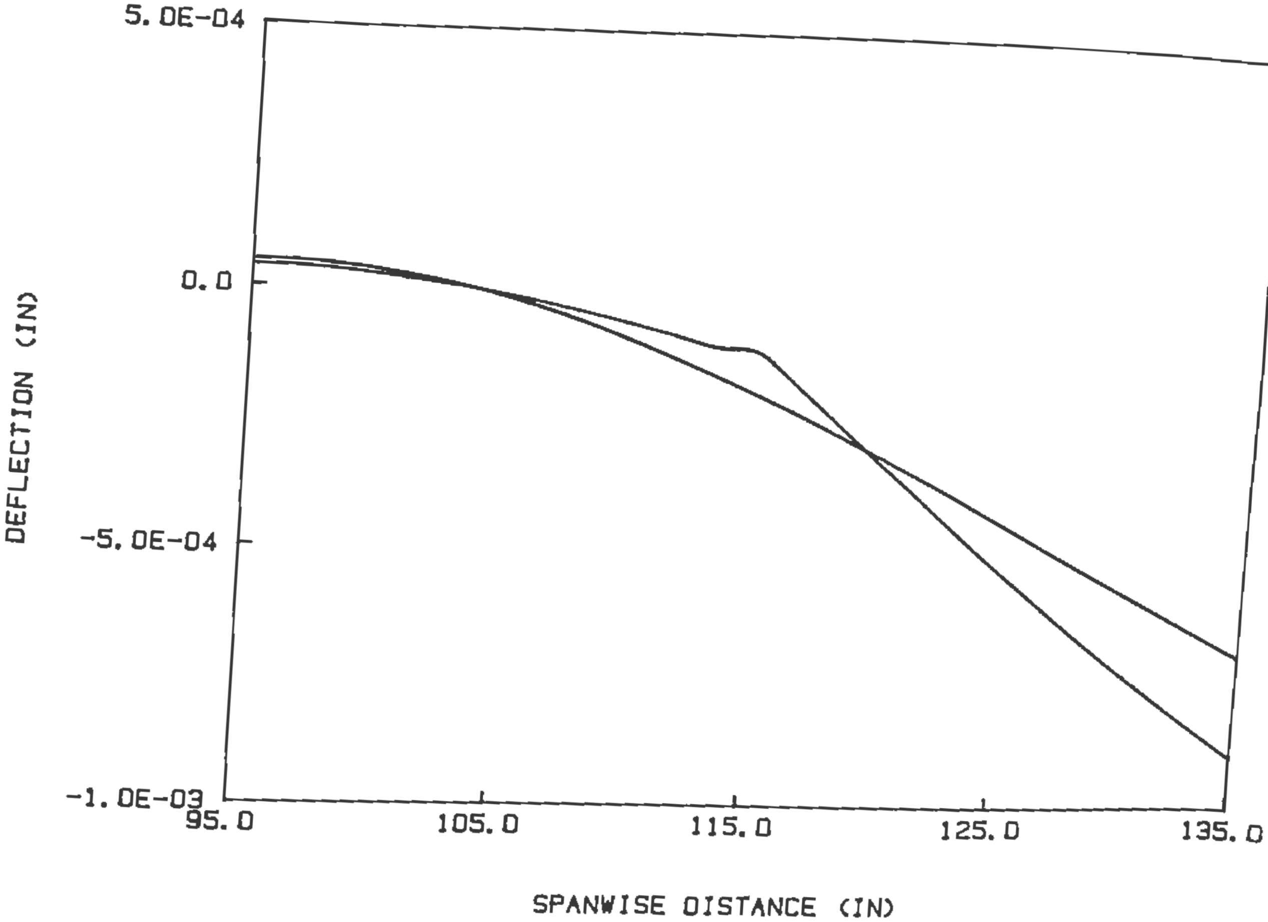


Fig. 26 Enlarged View of Fig. 25 Near Coupling

teeth and the flexibility of the teeth. Backlash becomes significant when the unsteady torque in the system exceeds the mean torque, a phenomenon that will occur when the system is idle or free-wheeling or in the course of transient maneuvers, such as when backing or stopping. Backlash induces non-linearities in the system and thereby necessitates analyses by more complex procedures than those associated with linear systems, Ref. (30). Gear tooth flexibility can be estimated using the procedures described in Ref. (37).

MODELING OF MEMBRANE COUPLINGS

Disk and diaphragm flexible couplings have been used for many years to make connections between reduction gear rotors and there exists a substantial database on coupling characteristics for use by designers. To illustrate the variations exhibited by various types of couplings in this class, Fig. 27 has been reproduced from Ref. (1) to show typical longitudinal vibration stiffness characteristics.

In the absence of any specific data, estimates of bending moment-slope relationship for diaphragm type couplings can be made using results from circular plate theory, Ref. (38). The stiffness to be used in a lumped spring-moment release representation would in this instance be made with the relationships given in Figures 28 and 29.

It is also noteworthy that for relatively high speeds, the axial stiffness of diaphragm type couplings is influenced by centrifugal effects. The trend experienced in such a case may be appreciated from Fig. 30 which has been prepared from manufacturer's data.

Disc couplings exhibit nonlinear characteristics in all modes of vibration. To illustrate the degree to which forces are not proportional to displacements, the reader is referred to Fig. 31. This plot shows the force versus axial displacement characteristic for a disk coupling proposed for the novel gearbox to be adopted in the AOE-6

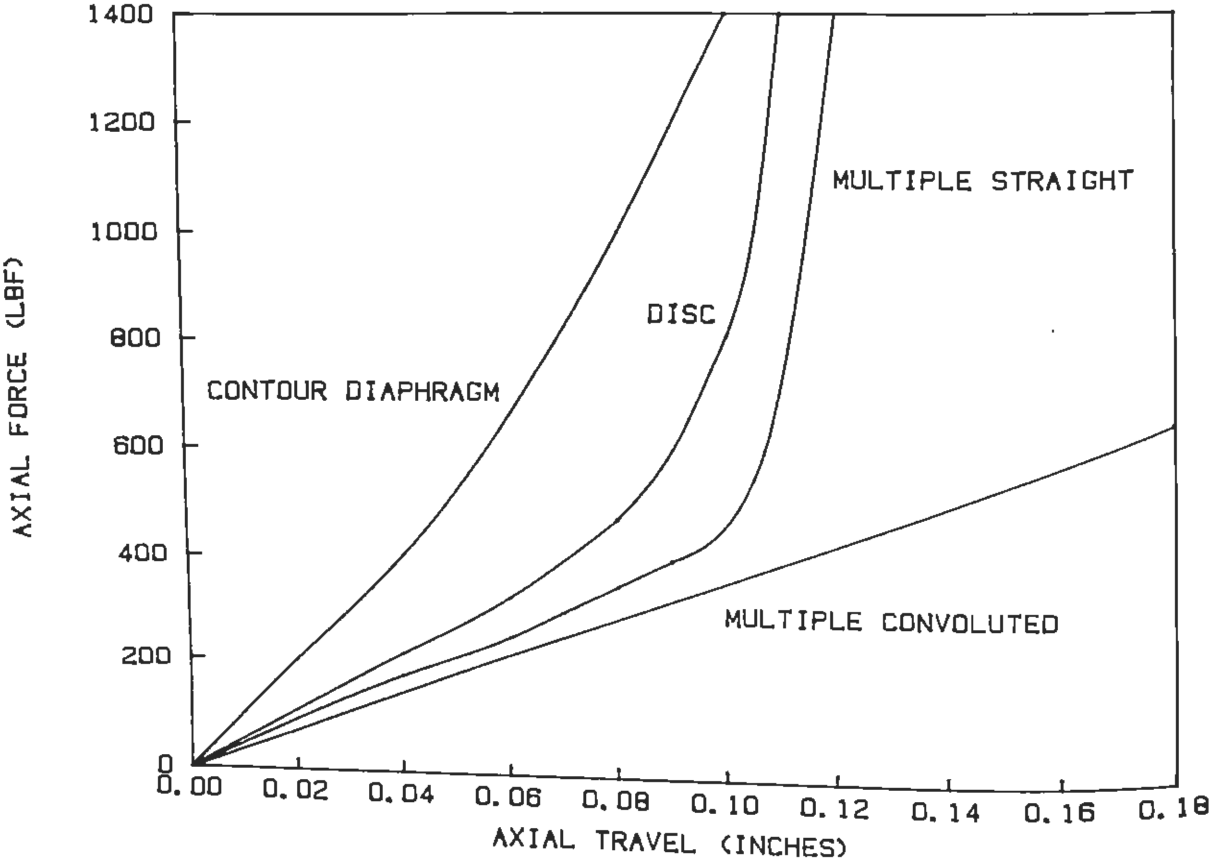
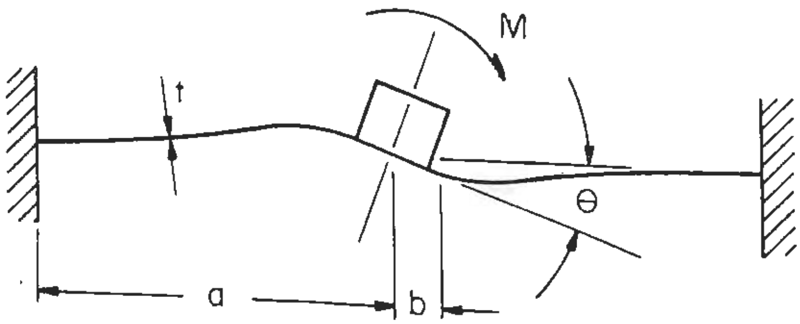


Fig. 27 Longitudinal Force-Deformation Relations for Membrane Couplings



$$M = \frac{E t^3}{\alpha} \Theta$$

Fig. 28 Moment-Angle Relation for Circular Plate

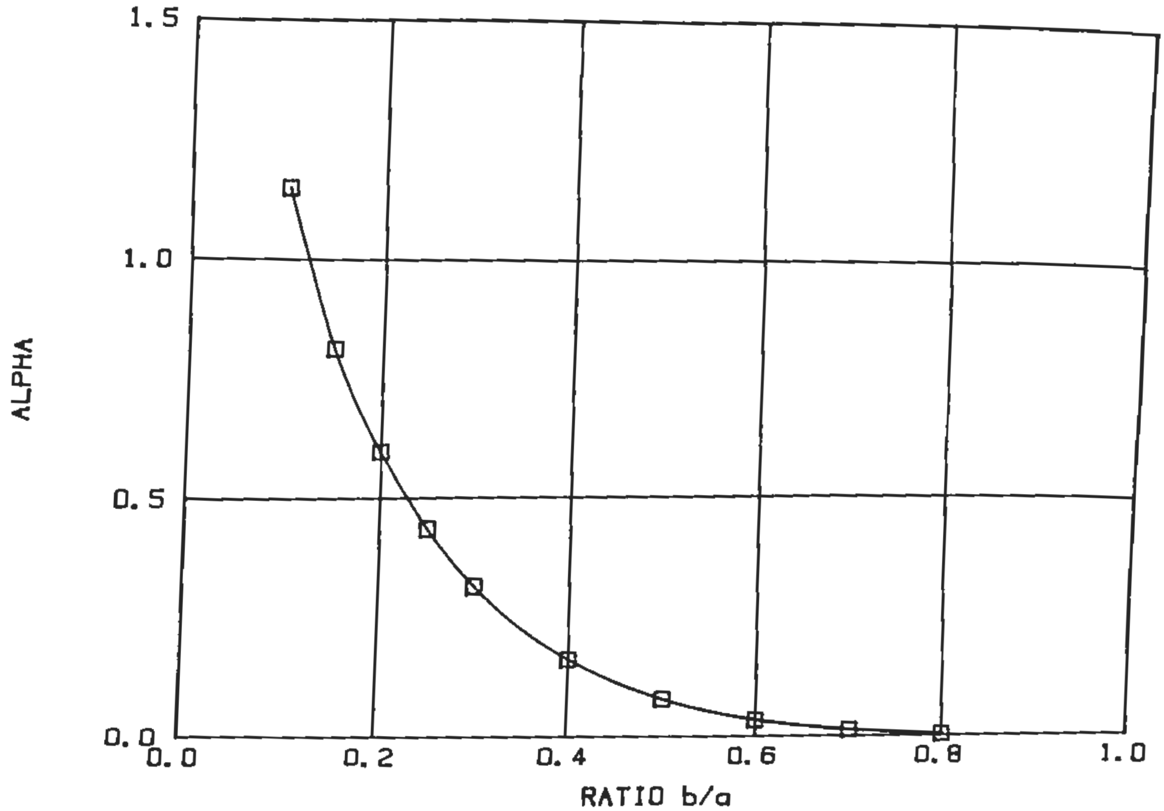


Fig. 29 Coefficient for Formula of Fig. 28

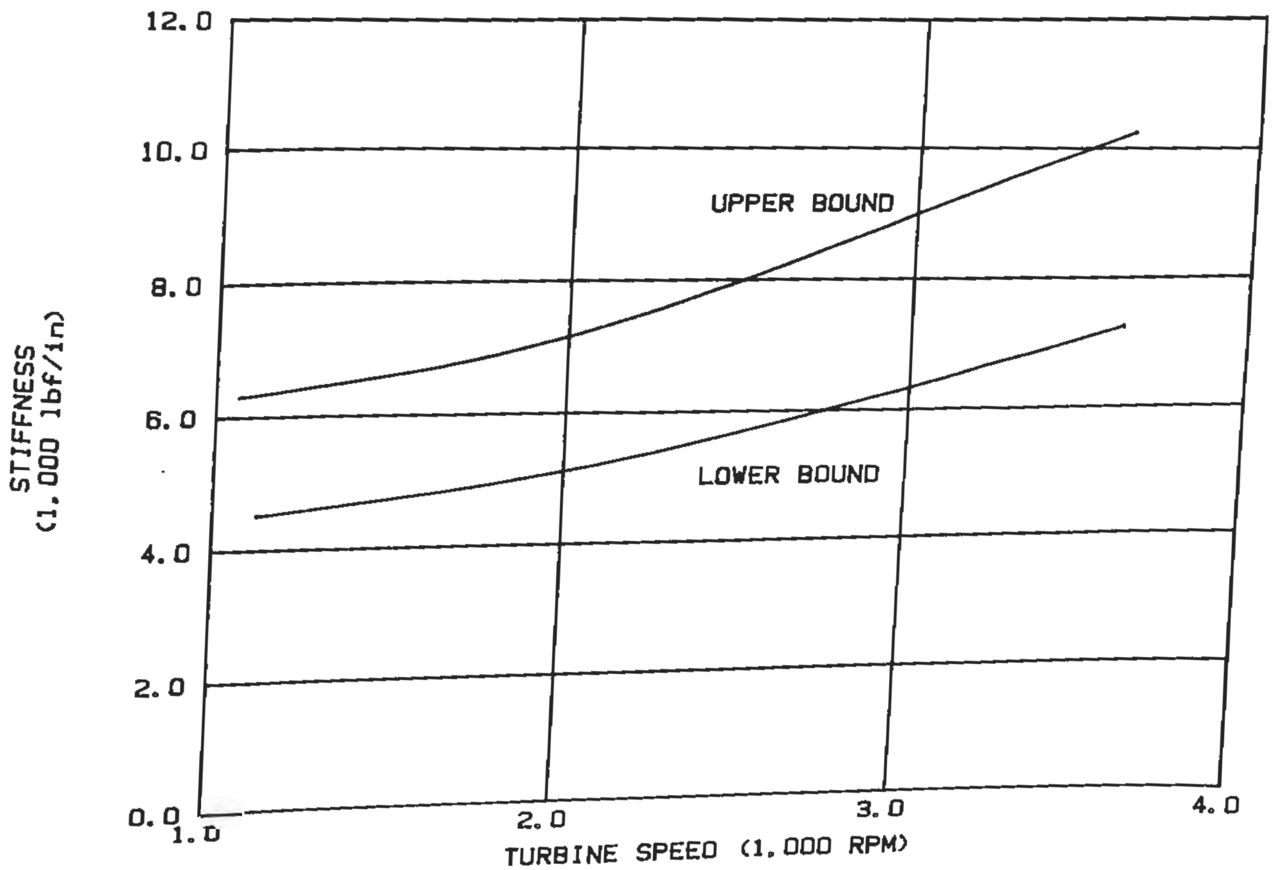


Fig. 30 Influence of Speed on Diaphragm Coupling Axial Stiffness

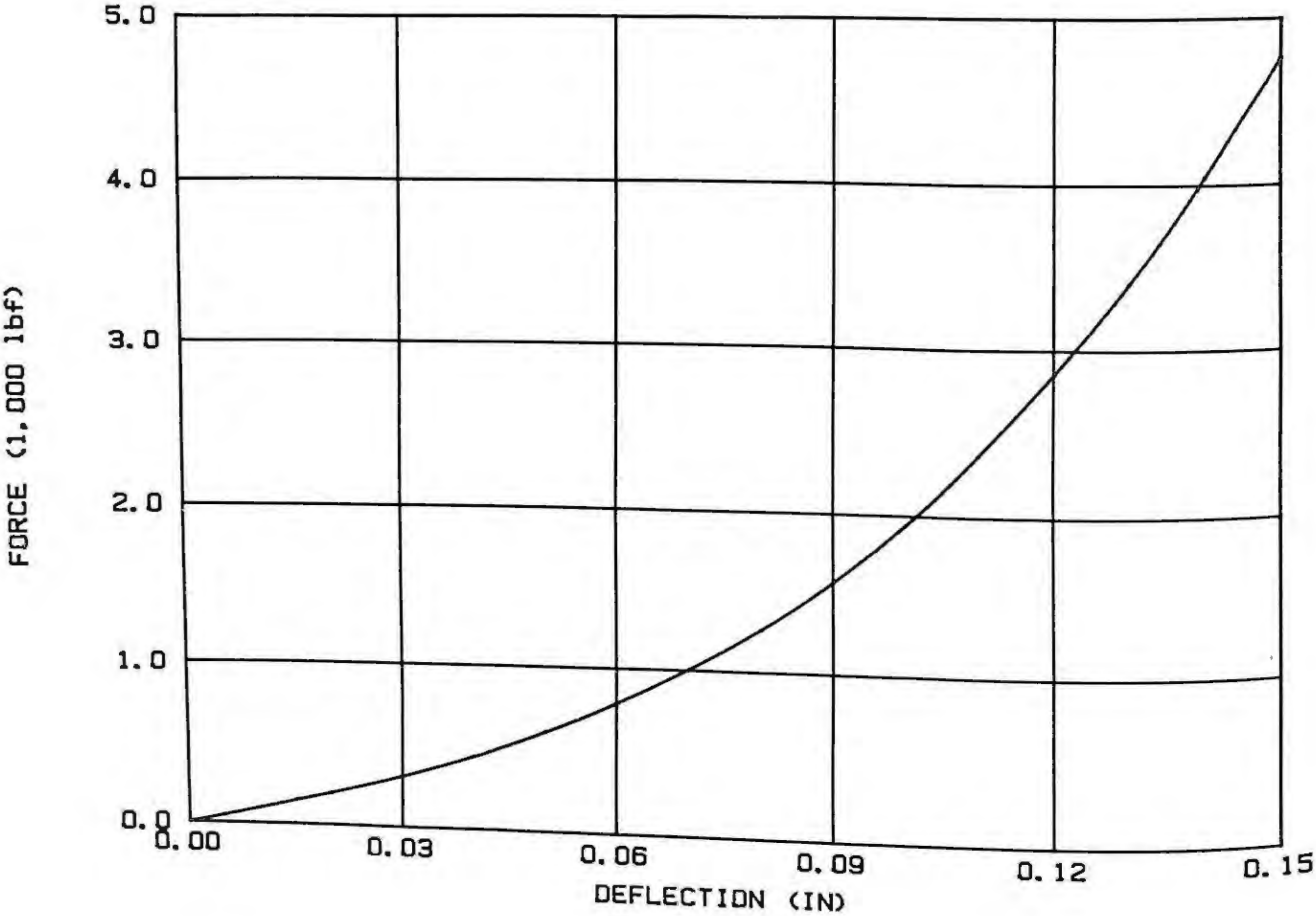


Fig. 31 Nonlinear Force-Deformation Relation for Disc Coupling

class of ship. When incorporated in an overall system model, such as that shown in Fig. 32, for half of the AOE-6 gearbox model, it becomes necessary to utilize nonlinear methods for predicting the system responses to unsteady forces.

MODELING OF HYDRAULIC COUPLINGS

Even though hydraulic couplings have been used for many years, there are only few descriptions in the literature regarding their dynamic parameters.

From the viewpoint of static alignment and transverse vibrations, such couplings offer no bending or shear flexibility because the driving and driven ends, are always separated. As a result, their only role is to add concentrated masses at the ends of the driving and driven shafts in question.

In the case of torsional vibrations, the coupling parts can be modeled as lumped inertias that are connected by a lumped spring and a lumped damper in parallel arrangement. The damping and stiffness parameters are, in general, functions of the relative rotational speeds of the two parts, and can be determined on the basis of the formulae proposed by Draminsky, Ref. (39). Figures 33 and 34 provide plots of these characteristics as a function of speed for the system of Fig. 14.

The variations of these coefficients with shaft speed are not unexpected, since the same phenomenon arises with the ship's propellers, Ref. (40). The significance of this is that forced vibration calculations must be carried with different system parameters for each frequency, a task not particularly well suited to large scale computer programs.

The authors are not aware of any procedure or data related to the axial stiffness and damping of reverse couplings, but both are suspected to be quite small, so that the driven and

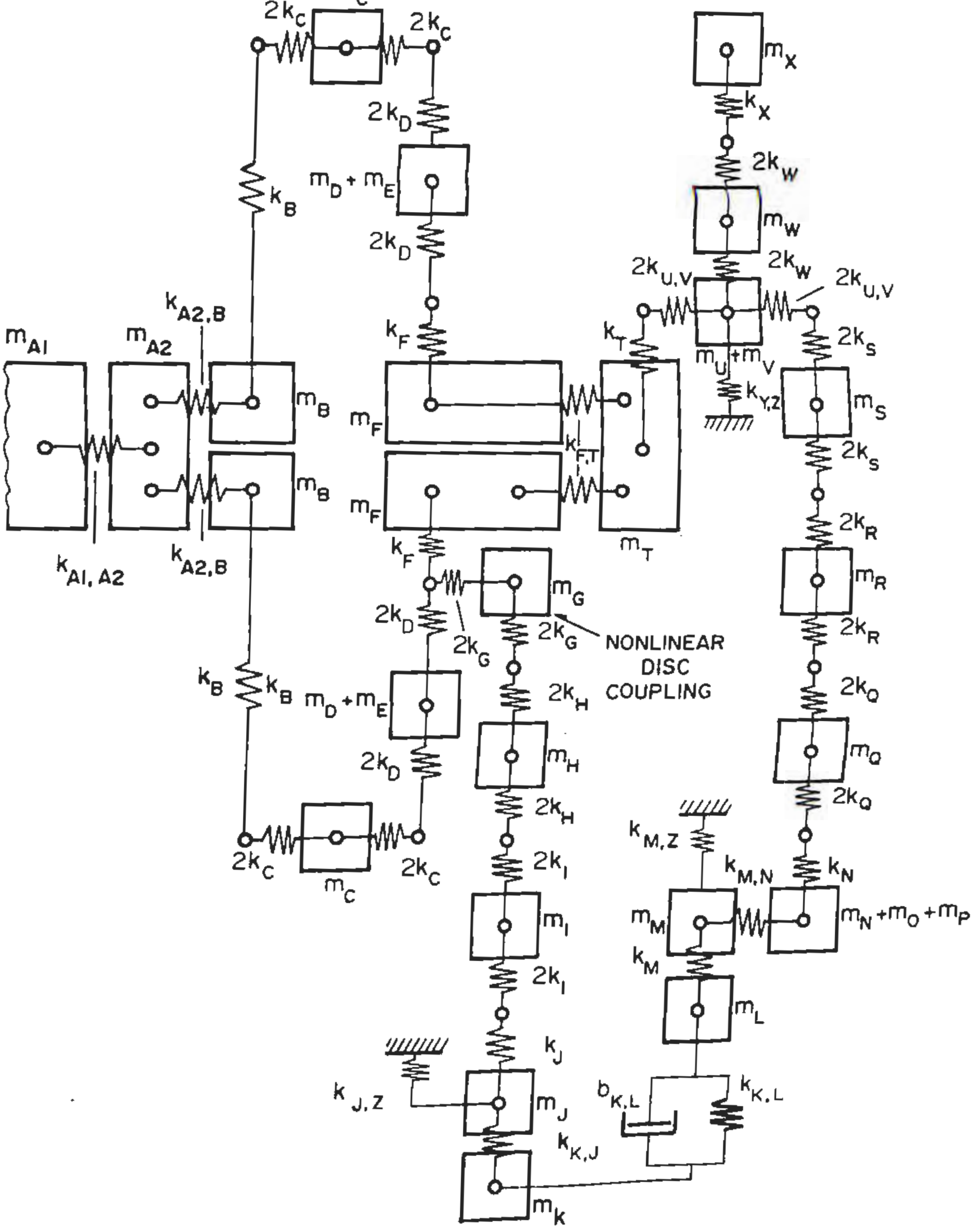


Fig. 32 Lumped Parameter Model of Reduction Gearbox (Starboard Side) for Axial Vibrations

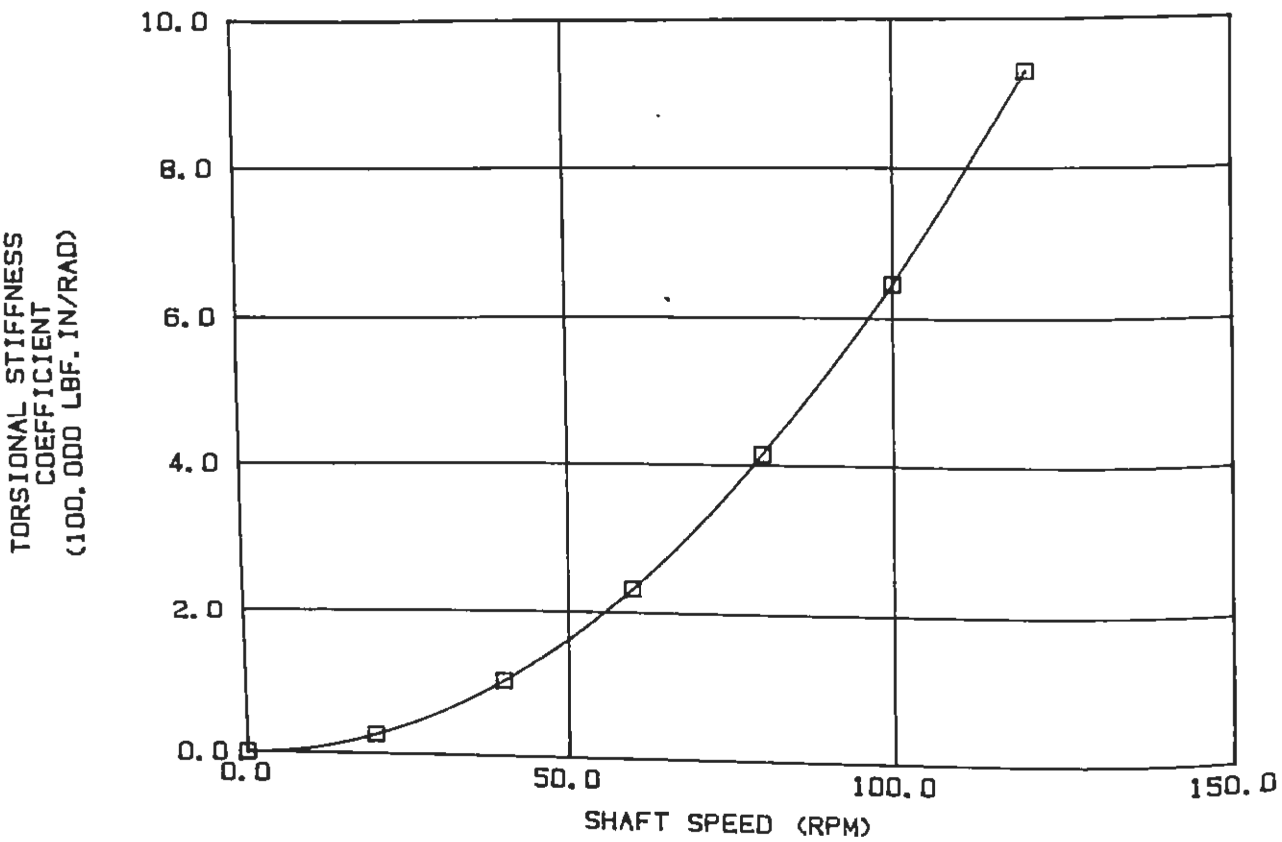


Fig. 33 Variation of Torsional Stiffness Coefficient for Hydraulic Coupling

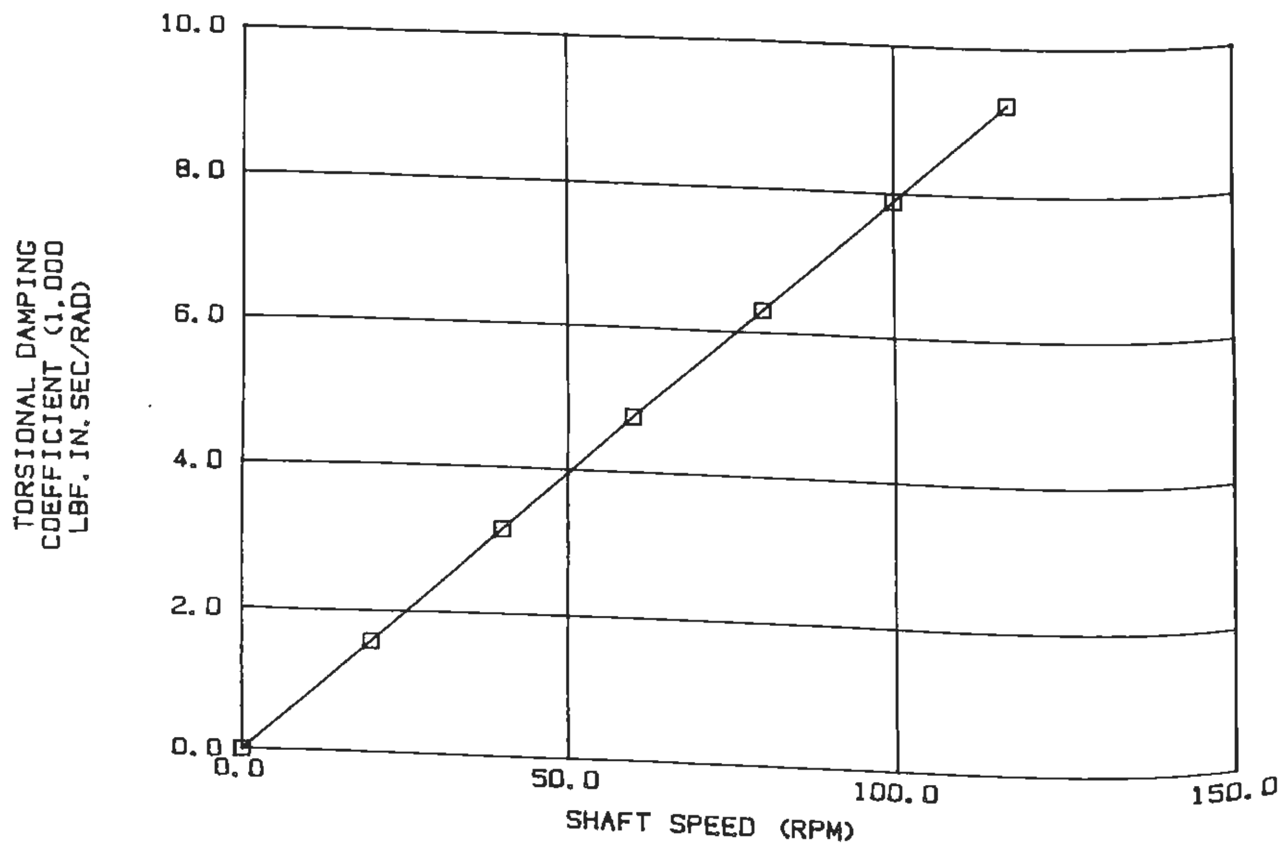


Fig. 34 Variation of Torsional Damping Coefficient for Hydraulic Coupling

driving parts are essentially isolated for the purposes of longitudinal vibration calculations.

FUTURE RESEARCH

It is evident from the contents of this paper that, although it is currently possible to incorporate coupling characteristics in most design studies, the need still exists for further work to help understand the behavior of shaft couplings.

On the numerical and analytical side, the need exists to develop more precise models of dental gear behavior and to help identify the crucial role played by friction. Simplified procedures for estimating mass, stiffness and damping properties are needed in the early design stages for all types of couplings. These can then be refined after the manufacturer of a specific unit has been selected. In that respect, the need exists for a compilation and correlation of coupling data that appear in many catalogs and brochures and their standardization for use by the profession.

It is unlikely that purely theoretical methods can suffice for predicting the behavior of shaft couplings. Besides, if even that were the case, the need would still exist to establish by measurement an experimental data base to be used for correlation with theory. To this end, the need exists to carry out tests not only under laboratory conditions but also with ships at sea, in order to better understand how couplings behave under realistic operating conditions. The initial experiments mentioned by Boylan in Ref. (16), can be used with telemetry methods such as those discussed in Ref. (41), to measure axial forces, torsional moments and bending moments with strain gages, and motions with proximity probes and "flying" accelerometers.

Last but not least, mention should be made of the fact that in other industries the role of couplings has been augmented to include the measurement of power and alignment, as discussed for example, in Ref. (42). Such devices may also prove to be of use to marine and offshore platform designers.

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