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ANALYSIS OF PROPULSION TRANSMISSION COMPONENTS

FOR GREAT LAKES ORE CARRIERS

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This report extends the earlier analysis to deflections of individual components, with deflections either imposed by the hull deflections, or by loads that may not have been analyzed adequately in traditional design processes. In particular, analysis here is applied to shafting, reduction gear pinions, clutch drums and spacers, and reduction gear casing structure.

All of the analyses apply the finite element method, using the University of Michigan MSAP system.

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ABSTRACT

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CONTENTS

INTRODUCTION

Great Lakes bulk carriers built since 1970 are propelled by geared diesel machinery. The machinery technology is standard, since the engines, reduction gears, clutches, bearings, and shafts are typical of those used extensively in other services. (River towboats constitute a typical and numerous example.) Nonetheless, failures of transmission components have occurred in numbers sufficient to attract notice. A possible source of the problems experienced may be the interaction between a more flexible hull (i.e. more flexible than earlier Lakes hulls) and the machinery. In pursuit of this possibility we have in an earlier project made detailed finite element analyses of the deflections of two typical contemporary hulls, and were able to get confirmation of the calculated results by measurements that were being made on the corresponding ships. The effect of hull deflections on the shafting was then explored. (See reference 1.)

Although displacement of shaft bearings affects that most important link in a geared transmission -- the mesh between teeth of pinion and bull gear -- (reference 2) deflections within components such as pinions can also have significant consequences. These deflections can occur because of loads imposed by the deflections of the hull, or as a result of loads that the components are designed to carry, e. g. torques and thrusts.

Our previous work offers data on deflections experienced at transmission foundation points, so that we have immediately available a starting point for further analysis. Torques and thrusts come easily from known speeds, shaft speeds, and powers of the typical ships. Guidance on which transmission elements demand analysis is given by reports of difficulties being experienced by the Lakes ships; information of this nature was generously furnished by industrial sources during the progress of our project.

Our efforts are concentrated on clutch parts, pinions, shafting, and gear case structure. For the clutch, we investigate the possibility that deflection of the hull imposes a bending moment on the transmission parts between engine and reduction gear, a moment that the clutch is not designed to carry. For the pinion, we investigate the possible inadequacy of the traditional analysis of torsion and bending loads. For the shafting, we investigate the causes of changes in bearing reactions, particularly those changes that may be caused by propeller thrust. For the gear case structure, we investigate deflections caused by thrust and by hull deflections, with the objective finding the consequences to pinion-gear tooth mesh.

The work and its results here reported are not based on a particular ship. Data on hull deflections developed earlier and reported in reference 1 is taken to be generally typical of contemporary Lakes ships, and likewise the data on machinery component dimensions and loadings is typical rather than particular. Exceptions to this generality are identified where appropriate.

CONVENTIONAL ANALYSIS -- THE FIRST FOUR TASKS

The first three tasks are combined into a single group; they are the following:

1. Examining conventional methods of analysis

2. Obtaining exact analysis results

3. Comparing conventional and exact analysis.

The rationale for this group is the belief that design (e. g. sizing for acceptable deflection and stress) was based on elementary "strength of materials" formulas learned by practicing engineers at a time when tools such as the now-popular finite-element methods were unknown, and that such a process may not be adequate for the more careful design needed in contemporary propulsion plants. A likely example is the the traditional analysis of bending and torsional deflection of reduction gear pinions (see page 319 of reference 3).

Our approach was to begin with the methods in use, proceed to more exact conventional methods (e. g. methods such as solutions from elasticity theory), then proceed to the finite element approximation. Comparisons would be made, then the method for use would be chosen. Such things we have indeed done, but in a somewhat narrower scope than planned at the proposal stage. One narrowing is caused by the difficulty in establishing just what design procedures are used; many components are adaptations or extrapolations of previous designs, which are in turn extrapolations of previous designs, etc. A second narrowing comes from the impracticality of applying the hoped-for "exact" analysis to geometries that no textbook ever dared treat. Nonetheless we have examined the components of interest, namely reduction gear pinions, clutch spacers and drums, shafting, and gear case structure. These are treated in individual sections following.

A fourth task is to evaluate the effects of hull deflection on components, and this we have done where appropriate. The discussions and results are also found in the following sections.

During the course of the project an unplanned element entered in the form of shaft bearing load measurements on two Great Lakes ships. These, furnished us in private communications from the ship owners and builders, indicate a strong dependence of bearing loading on propeller thrust. Our earlier project (reference 1) includes calculated hull deflections in consequence of thrust, leading to an attempt to relate the measurements to the calculations. Since thrust loading of the reduction gear structure might well be involved, and since we intended analysis of this structure, the effect of thrust loading was emphasized in this part of the work.

REDUCTION GEAR PINION DEFLECTION

Analysis of reduction gear pinions for bending and torsional deflections is a recommended part of design, since these deflections can cause an opening of the mesh with the bull gear teeth. Standard formulas are given on page 319 of the Society of Naval Architects and Marine Engineers book MARINE ENGINEERING (reference 3). The formula for torsional deflection is

$$y_1 = 7.95 * C * J (F_e/d) F_e * 10^{-9}$$

$$y_2 = 2.65 * C * J (F_e/d) F_e * 10^{-9} \text{ inches}$$

where y_1 = tooth separation at driving end

y_2 = tooth separation at driven end

$C = d / (d^4 - do^4)$

d = pinion diameter

do = inner diameter

J = tooth loading, lbf/inch-inch

F_e = effective face width, inches

The formula for bending deflection is

$$f = 0.885 * C * J (F/d)^3 * F * 10^{-9} \text{ inches}$$

where f = tooth separation due to bending

F = distance between ends of bearings

(The three components of tooth separation are combined as explained in the reference.)

These formulas have their origin in elementary beam theory; with minor rearrangements and changes in symbols they may be found in any textbook on beam theory. Obvious possible deficiencies in these simple formulas are the absence of shear deflections and of any end fixity for the bending component.

The deficiencies may be investigated by amplifying the simple beam theory to include shear effects and end moments that represent fixity, and we have done just this. However, our calculations are done by the finite element method with the pinion modeled as a series of axisymmetric beams. Results are shown in Figure 1.

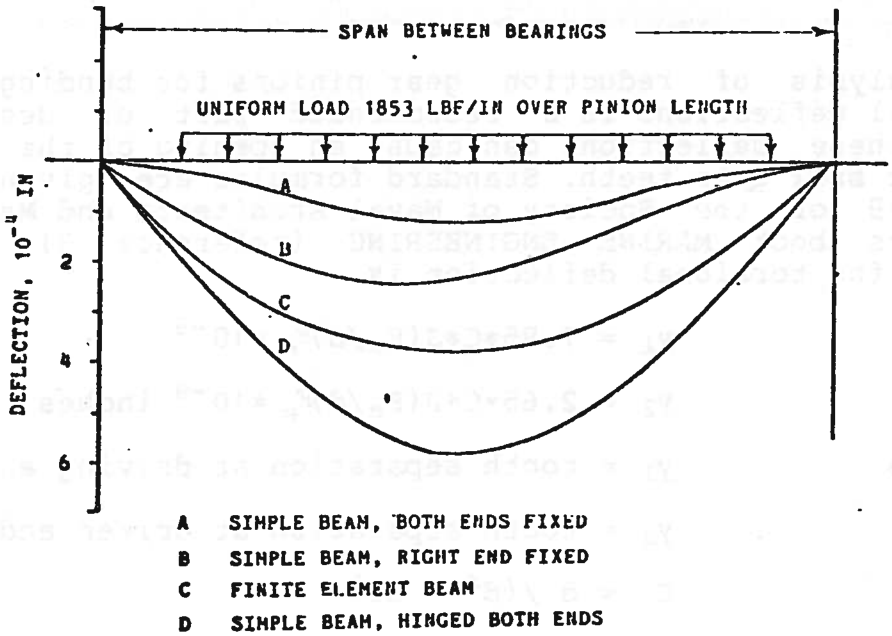


Figure 1. Deflection of Reduction Gear Pinion by Several Methods

The pinion is typical of recent Great Lakes gear sets. It nominally transmits 3500 hp at 900 rpm; the loading shown in the figure corresponds to this service. Four curves are included, with labelling to indentify each. Curve D is the result of application of the simple MARINE ENGINEERING formulas; curve C is the result of a multi-element finite-element analysis, with shear deflection but without end fixity; the other two curves are produced by the MARINE ENGINEERING formulas, but with the indicated end fixity added. Observe that curve D predicts the greatest deflection.

The finite element model is based on a smooth cylinder having pinion pitch radius, just as is the simple model. The teeth can be added to the finite element model, but we have not found that the toothed model produces a curve significantly different from curve C, and so believe that the added complexity of model is not justified.

Since the simplest model (i.e. curve D) produces a conservative estimate of pinion deflection, we conclude that for general purposes it should continue in use. A possible exception would be a design procedure in which teeth are being contoured to compensate for pinion deflection under load; in that case the most accurate calculation is wanted, and we suggest the use of a multi-element finite-element calculation. ("Suggest" rather than "recommend" because we have no measurements to confirm accuracy of calculation.)

ANALYSIS OF STRESSES IN CLUTCH SPACERS AND DRUMS

Figure 2 illustrates the pneumatic clutch that typically connects the reduction gear pinion to a short drive shaft from the engine flywheel. Figure 3 is a sketch that illustrates the possibility of a bending moment being applied to the clutch by movement of the two pinion bearings. Of course, the pneumatic element between drum and spacer presumably allows a considerable flexibility that can ameliorate the consequences. On the other hand, several instances of clutch failure occurred on new Lakes ships during the late 1970s (private communication from Bay Shipbuilding), suggesting that analysis be undertaken.

Both the clutch spacer and the clutch drum can be modeled as circular plates, and so are seemingly amenable to "classical" (i.e. not finite element) analysis via elasticity theory. However, the textbook (see references 4 and 5 for instance) treatments of flat circular plates includes only a concentrated normal load at the plate center and a normal load uniformly distributed over the plate. Neither, of course, introduces bending about a horizontal diameter, the case of interest to us. Now, with reasonable diligence and care one may proceed to the derivation of a circular flat plate case with loading by a

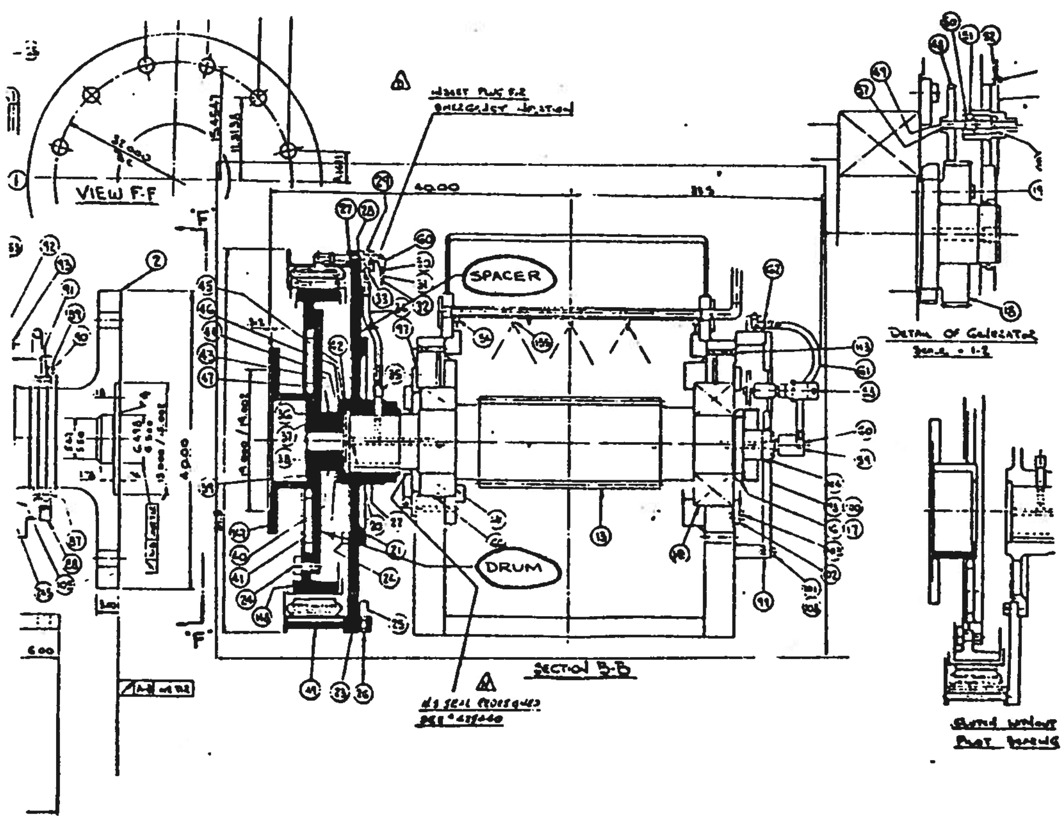


Figure 2. Typical Pneumatic Clutch

normal couple. The resulting formula is impractically complicated, especially when it must stand in competition with an easily constructed and easily solved finite element model. Our efforts were therefore seduced away into the analysis by the latter method.

Our finite element analysis takes two courses. The first of these is indicated by the sketch of Figure 4. It represents a circular plate connected at its center to a circular shaft. The shaft is supported on two bearings, with the more distant one being moveable in the vertical direction. The plate is either clamped at its edge or is simply supported there. The model is therefore of a clutch spacer distressed by a bending of the connected shaft in a way not intended by the designer. On the other hand, it is not intended to be an accurate representation of an actual clutch-plus-shaft. Although overall dimensions approximate those found in Great Lakes clutches, the geometry is greatly simplified, and the clamping of the plate edges is surely a more severe constraint than the actual pneumatic element of a clutch imposes. The intention is to examine the generality of the geometry and type of load as a guide to possibility of high stresses. With a vertical movement of 0.1 inch at the far end of the shaft, the highest value of membrane stress in the plate is approximately 7,900 psi. Although this suggests that no distress is to be expected from attached shaft bending (that 0.1 inch is an extreme movement), we advance to the second course, mostly from the fear that the model just discussed might -- through its very simplicity -- allow a potential high stress point to go undetected.

The second course is to model the clutch drum and clutch spacer in thorough detail, to apply loads of likely type and magnitude at the points where these components are mated by an inflated pneumatic element, and then to calculate the maximum stress. Figure 5 illustrates the finite element models used in this process. The elements are axisymmetric; the views are therefore of cross sections of the elements. The elements that are numbered (one in each sketch) and highlighted by darkening are the ones where the highest value of stress occurs.

Figure 6 shows the five load cases that were followed. In each case the total force between drum and spacer is that exerted by the pneumatic element inflated to 100 psi, but distributed in five different ways, i. e.

1. Uniform distribution over mating surfaces
2. A uniformly distributed shear load
3. Normal force distributed in triangular fashion
4. Concentrated force at one end of the drum
5. Concentrated force at the other end of the drum.

All stresses found are very low. The highest value in the drum is 633 psi and in the spacer is 560 psi (see locations in Figure 5.)

Our conclusion is that hull deflection does not threaten integrity of reduction gear clutches. The actual clutch failures are believed (private communication from Bay Shipbuilding) to be caused by thermal stress as a result of overheating. Our analysis does not confirm this opinion, but eliminates one alternative.

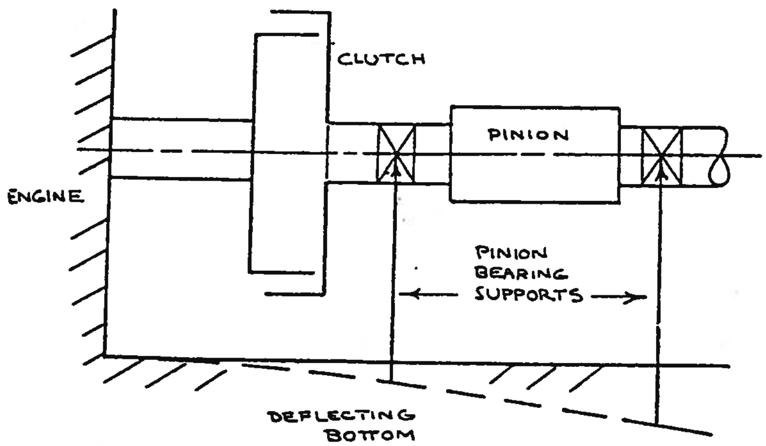


Figure 3. Possible Bending Load on Clutch

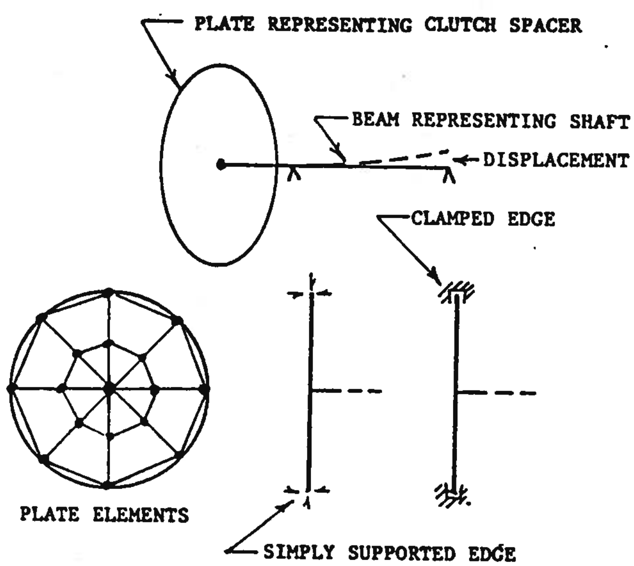


Figure 4. Finite Element Model of Clutch and Attached Shaft

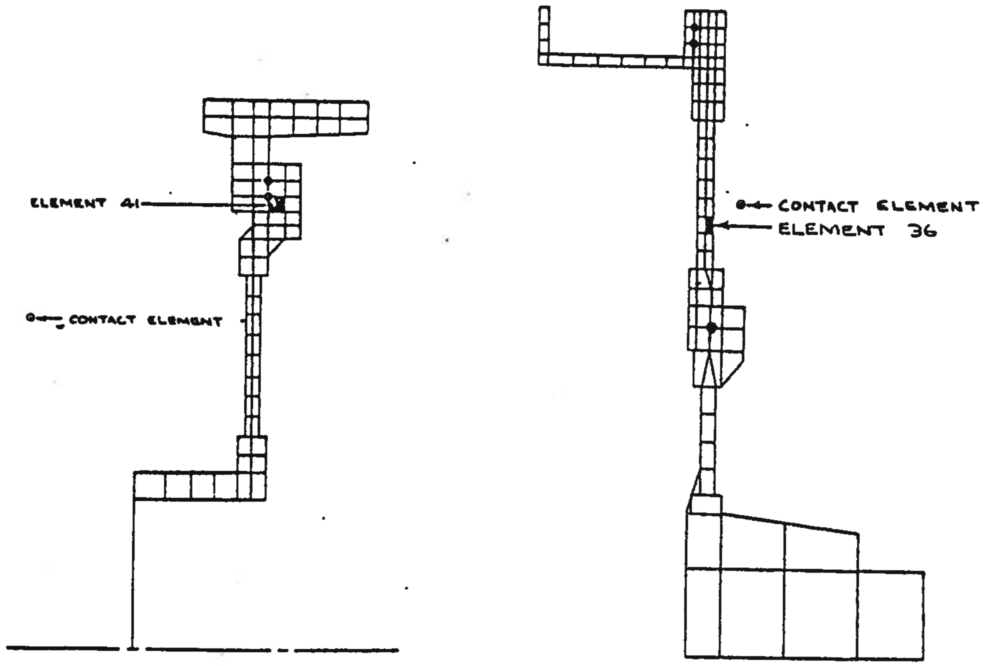


Figure 5. Finite Element Model of Pneumatic Clutch Drum and Spacer

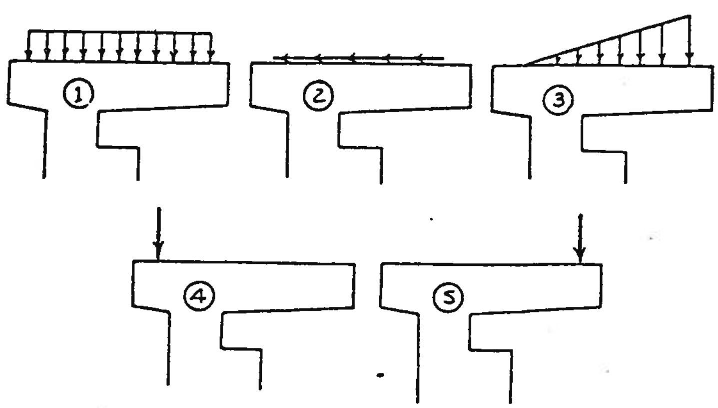


Figure 6. Load Cases for Clutch Analysis

SHAFTING ANALYSIS

Analysis by the Finite Element Technique

The calculations to establish alignment of the propulsion shafting are conveniently done, and with sufficient accuracy, by conventional beam-on-multiple-supports methods. ("Sufficient" here means that uncertainty in calculated results is less than the uncertainty in the actual alignment processes.) These methods have long ago been incorporated into computer programs that are widely used in ship design.

The finite element method is not fundamentally different from the conventional methods, since the elements are simply the segments of the beam that those methods analyze. Our work on the previous project (reference 1) confirmed this assertion by showing identical results from the competing methods. On the other hand, that project did produce shafting influence coefficents via a finite element model that included response of the bearing support structure to changes in bearing elevation, and that added sophistication of analysis is quite impractical by the conventional methods.

One outstanding advantage of the finite element method is its flexibility in introducing unconventional features (e.g. tilting of bearings, spring constants to represent flexible foundations, etc) into the analysis. A designer having a reasonable familiarity with this method, and access to a large and flexible computer tool like our MSAP and its handy preprocessor PREMSAP can quickly analyze shaft models of many types and degrees of complexity, whereas the user of the special-purpose shaft analysis program is restricted to that program's built-in capabilities. This virtue of the finite element method we have exploited in the shaft-related work done here.

Table 1 is shaft bearing load data recorded in 1981 aboard the American Steamship bulk carrier H LEE WHITE; its principal message is the shifting of load distribution between the two bull gear bearing as propulsive power changes. (Data is furnished in a private communication by Bay Shipbuilding, and is used with permission of American Steamship.) One may speculate on several causes for the change; a possible cause is the bending of the shaft by eccentric propeller thrust, perhaps independent of any deflections of gear casing, gear foundation, or hull structure. We have applied our wide-ranging MSAP tool to testing this possibility by looking at this type of shaft bending with several models of the bearing support action.

Four cases are indicated by the sketches of Figure 7. They are

1. Shaft with displacement restrained at the after two bearings, bull gear bearings represented by springs, and thrust bearing represented by an axial spring

2. Shaft on four bearings represented by springs, and thrust bearing represented by an axial spring

3. Shaft on four bearings represented by springs, and thrust bearing represented by a cantilever beam whose bending approximates that found in another phase of the project (see following section)

4. Shaft on four bearings represented by springs, thrust bearing represented by an axial spring, and aft bull gear bearing depressed by an artificial load to approximate bearing displacement by thrust acting on the gear casing.

An axial load representing propeller thrust at an estimated eccentric position acts in each case. Runs were made for each with zero thrust to allow the consequence of thrust to show. The general result is a decrease in the load on the aft bull gear bearing and an increase in the load on the forward bearing. The only counter result occurs in case 1. Since there is in actuality some clearance in all bearings, the fixity imposed on the two after bearings in our case 1. model cannot be exact, and one may conclude that it is likely to be the least accurate of the four cases. Modeling of the actual situation as represented by the H LEE WHITE data has not been established, and therefore the conclusion must be that better representation of bearing action is needed.

<u>Effect of Hull Deflections on Shaft Bearing Loads</u>

Figure 8 is reproduced from reference 1, the report of our hull deflection project. It lists our calculated hull deflections, due solely to propeller thrust, for "Ship A," a vessel essentially identical to the H LEE WHITE whose changes in bearing loads as functions of varying propulsive power are listed in Table 1. Reference 1 also lists our calculated shaft bearing influence coefficients (nearly the same in value as those used by the ship designer), and these are reproduced here as Table 2. We attempt therefore to use the influence coefficients and deflections to reproduce the measured changes of Table 1. The -0.0027 inch deflections shown in Figure 8 for the two bull gear bearings may be taken as relative to unchanged stern tube bearings. A conventional calculation of bearing load changes then produces an increase of 1112 lbf on the forward bullgear bearing, and a decrease of 1536 lbf on the aft bearing. Although the thrust value that produces this result is not necessarily the same as that producing the maximum change of bearing loads in Table 1, the fact that the calculation does not reproduce the actual occurence is clear -- the forward bearing is relieved by thrust, wheareas our calculation shows the opposite effect.

LOAD CONDITION	SHAFT RPM	REACTION (KIPS)	
		R_f	R_a
Ballast	45	16.1	5.0
"	67	11.4	10.3
"	78	10.5	11.3
"	89	8.1	14.5
"	98	4.8	20.0
"	111	8.4	13.7
Loaded	44	17.3	3.6
"	69	11.0	11.6
"	71	7.7	15.8
"	79	4.6	19.3
"	91	4.2	20.0
"	99	7.0	15.7
"	111	6.0	16.8

Table 1. Gear Bearing Vertical Reactions, H LEE WHITE (by private communication from Bay Shipbuilding)

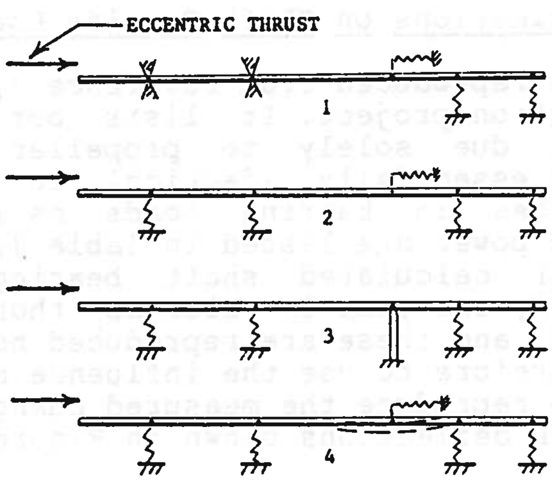
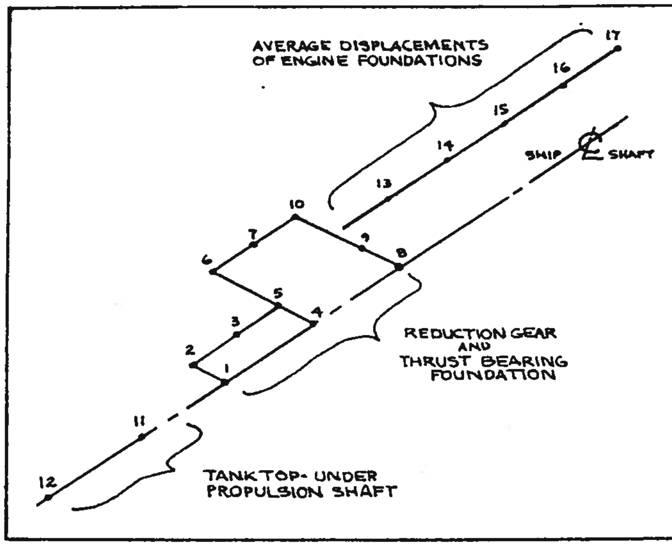


Figure 7. Models for Analysis of Propulsion Shaft Bent by Eccentric Thrust



NODE NO.	FR. NO.	VERTICAL DISPL.	LATERAL DISPL.	LONG'L. DISPL.
1	80	+.0039	0	+.0093
2	80	+.0040	+.0003	+.0086
3	79⅔	+.0035	-.0001	+.0090
4	79⅓	-.0027	0	+.0095
5	79⅓	-.0017	+.0001	+.0082
6	79⅓	+.0001	-.0004	+.0037
7	79	-.0007	-.0015	+.0034
8	78⅔	-.0027	0	+.0081
9	78⅔	-.0021	-.0003	+.0044
10	78⅔	-.0014	-.0008	+.0033
11	81	+.0023	0	+.0017
12	82	+.0012	0	+.0015
13	78	-.0019	-.0004	+.0021
14	77	-.0014	-.0002	+.0013
15	76	-.0008	-.0001	+.0009
16	75	-.0004	-.0000	+.0006
17	74⅓	-.0003	+.0002	+.0006

Figure 8. Displacement of Selected Points Caused by Propeller Thrust

	1	2	3	4
1	1548	-1960	522	-110
2	-1960	2529	-775	206
3	522	-775	453	-200
4	-110	206	-200	103

Table 2 Propulsion Shafting Influence Coefficients (from reference 1)

DEFLECTION OF REDUCTION GEAR STRUCTURE

The structure of a reduction gear casing is, of course, quite complex in its geometry. Except for very approximate or very local analyses, the only feasible method of structural analysis is the finite element.

We have constructed a model of 260 elements to represent the typical single-reduction gear set used on contemporary Great Lakes bulk carriers. (A cross section -- drawn as part of the modeling process -- is shown by Figure 9.) The gear transmits 3500 hp via each of two pinions, reducing speed from 900 rpm to 120 rpm. Our model is loaded by the corresponding thrust (estimated at 189,000 lbf) and moment at its output coupling, by the corresponding gear tooth force (estimated at 40,000 lbf) at the pinion bearings, and by deflections imposed on its base by the foundation.

The moment at the output coupling is imposed principally by the eccentricity of propeller thrust. Since we have no data on this eccentricity, our analyses were run with a range of values, but with very little difference in result. Gear tooth force on the bull gear bearings is effectively cancelled when both engines are driving, since the pinions lie in the plane of the bull gear shaft.

Deflections imposed by the foundation are the consequence of hull deflection, and the values we use come from our previous project (reference 1). In calculating those deflections, we used a finite element model that includes shaft, gear, and gear foundation as part of the ship structure; in consequence, the effect of the gear structure on the hull deflections was accounted for. The difference here is that we use a much more detailed model for the gear structure in order to find local deflections within it.

The deflections from reference 1 are reproduced in Table 3 and Figure 10. The figure is a plan view of the top plane of the gear foundation, with cross sections of the gear casing structure superimposed, and the nodes listed in the table located. All of these nodes lie in the foundation top plane. (In the case of deflections due to thrust we have applied a correction in magnitude since the reference 1 deflections are based on a thrust of only 50,000 lbf.)

The gear structure model used here includes 260 nodes, and the MSAP analysis routine prints the three

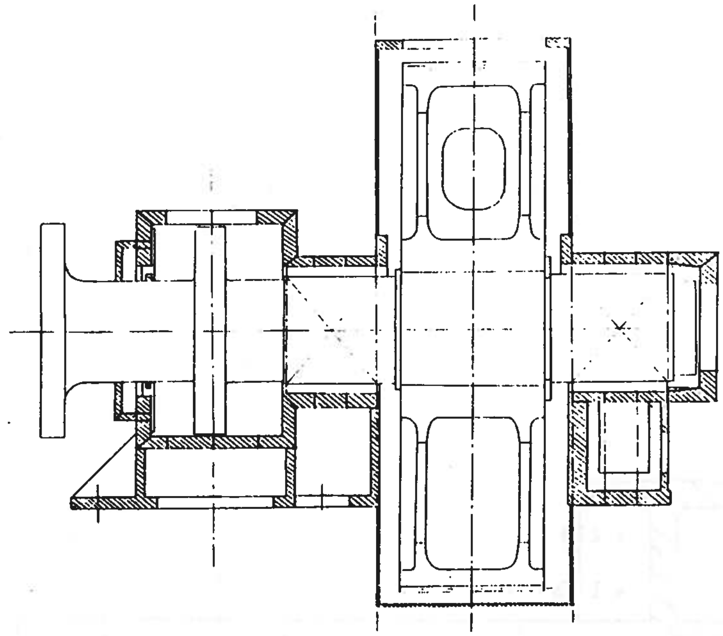


Figure 9. Cross Section of Typical Reduction Gear

Node Number	X	Y	Z
For Fully Loaded Ship			
030	0.00000725	0.008514	0.3269
031	0.0000218	0.008413	0.3270
081	-0.0000184	0.008522	0.3316
132	-0.0000270	0.00974	0.3318
148	0.000673	0.00990	0.3325
149	0.001370	0.0100	0.3333
165	0.001077	0.010225	0.3339
223	0.0000374	0.01217	0.3349
224	0.0000459	0.01278	0.3355
For Ballasted Ship			
030	0.0000235	0.007088	-0.2343
031	0.0000704	0.00704	-0.2342
081	0.000089	0.0070	-0.2134
132	0.0004526	0.007583	-0.2100
148	0.0003543	0.008014	-0.2055
149	0.000256	0.008445	-0.2010
165	0.000388	0.008572	-0.1957
223	0.000222	0.005153	-0.0790
224	0.0002482	0.004841	-0.1277
For Rated Propeller Thrust Only			
030	0.0000974	0.03494	0.01536
031	0.000292	0.03410	0.01414
081	0.000199	0.03356	-0.00569
132	-0.01362	0.01362	-0.000252
148	-0.004007	0.01332	-0.001348
149	-0.005390	0.013058	-0.002334
165	-0.01282	0.01284	-0.003342
223	-0.0006352	0.02408	-0.008219
224	-0.0007657	0.02239	-0.008374

Table.3 Calculated Hull Deflections at Mating Surface between Gear Casing and Its Foundation (adapted from reference 1)

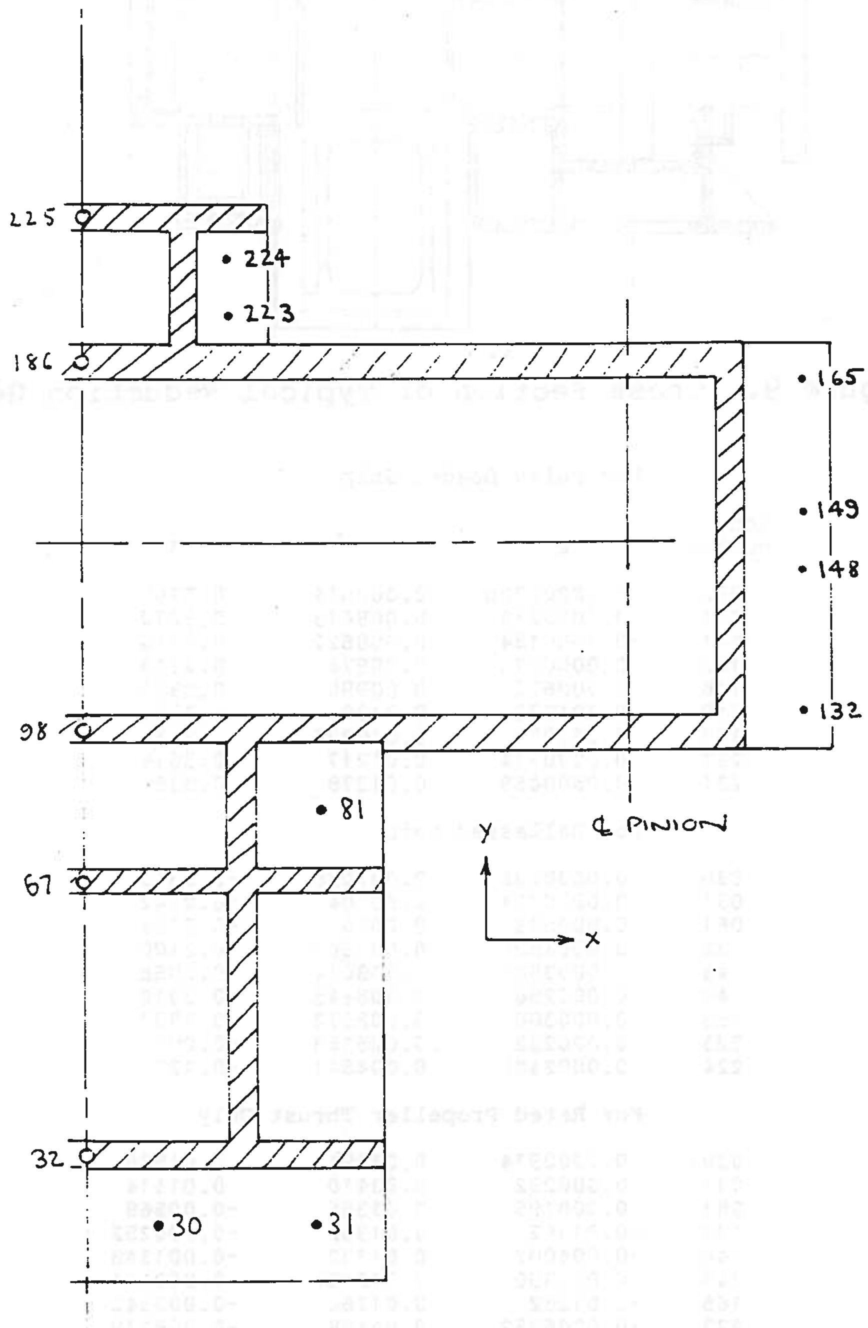


Figure 10. Locations of Nodes Listed in Table 3

deflections and three rotations for every one of them. In this report we refrain from that munificence to publish only the nodal deflections that show the deflections at or near the bull gear and pinion bearings (including thrust bearing). These are listed in Table 4; you may refer to Figure 11 to determine the location of the nodes numbered in the table.

It is obvious from Figures 10 and 11 that our model includes only one side of gear centerline, this following the observation that everything is symmetrical about this line. There is an exception to symmetry, of course, in the gear tooth force; one pinion drives up, the other drives down. Results here are for the down side. Deflections on the other side are of the same magnitude, but the vertical components are of opposite sign.

The question of greatest interest is whether there is significant deflection within the gear structure, i.e. in addition to the deflections imposed by the hull via the gear foundation. The answer is difficult to give with numerical precision because the foundation nodes do not coincide exactly with the nodes in the gear case structural model. Figure 12, however, allows an answer to be deduced from a brief inspection. The figure is a skeleton view of the gear case, much like Figure 11, with deflections for the thrust-only case labeled at significant points. At each such point a triplet of vectors (x vector omitted on the centerplane) shows the direction of the deflection, and each vector is labeled with the magnitude of the deflection in mils (i. e. 5 = 0.005 inches) taken from either Table 3 or Table 4. Node numbers are indicated within enclosurres -- rectangular for the hull-imposed deflections, circular for those calculated here.

Only the thrust-alone case is pictured in this way since examination of Tables 3 and 4 show that the other two cases (in which there is no thrust) have a comparatively small magnitude of deflections of the gear case relative to its foundation.

As expected, the gear structure in way of the thrust bearing housing deflects forward under the thrust loading; observe nodes 45, 80, 128, 218, and 239 in comparison to the imposed-deflection nodes 30, 81, and 223. The unexpected element in this figure is the deflection of node 175, located at the center of the pinion forward bearing journal. Note that an outboard deflection of 0.015 inches is shown there, quite in contrast to the 0.013 inch deflection in the opposite direction at the nearby node 165.

For Fully Loaded Ship

Node Number	X	Y	Z
032	0.0000	0.00852	0.3200
045	0.0000	0.000376	0.3190
067	0.0000	0.00868	0.3220
080	0.0000	0.000225	0.3220
098	0.0000	0.00865	0.3250
101	-0.000204	0.0102	0.3250
119	-0.00044	0.00384	0.3250
128	0.0000	0.00182	0.3250
175	-0.000537	0.00385	0.3320
186	0.0000	0.0122	0.3320
189	-0.00120	0.01030	0.3330
218	0.0000	0.00145	0.3320
225	0.0000	0.01220	0.3360
239	0.0000	0.00140	0.3360

For Ballasted Ship

Node Number	X	Y	Z
032	0.0000	0.00706	-0.2400
045	0.0000	-0.07360	-0.2420
067	0.0000	0.00740	-0.1960
080	0.0000	-0.07440	-0.1970
098	0.0000	0.00757	-0.1730
101	-0.000175	0.00813	-0.1770
119	0.0000	-0.0546	-0.1770
128	0.0000	-0.0597	-0.1720
175	-0.00577	-0.0546	-0.1070
186	0.0000	-0.00015	-0.1030
189	-0.00036	0.00858	-0.1020
218	0.0000	-0.0608	-0.0106
225	0.0000	0.0000	-0.0845
239	0.0000	-0.0583	-0.0819

For Rated Propeller Thrust Only

Node Number	X	Y	Z
032	0.0000	0.0349	0.0146
045	0.0000	0.0556	0.0147
067	0.0000	0.0346	0.00395
080	0.0000	0.0556	0.00352
098	0.0000	0.0346	-0.000816
101	-0.00250	0.0129	-0.000860
119	-0.00156	0.0302	-0.000580
128	0.0000	0.0509	-0.000326
175	-0.00165	0.0301	-0.0131
186	0.0000	0.0254	-0.00821
189	-0.00512	0.0130	-0.0169
218	-0.00512	0.0515	-0.00944
225	0.0000	0.0254	-0.0164
239	0.0000	0.0518	-0.0151

Table 4 Calculated Deflections at Significant Finite Element Nodes

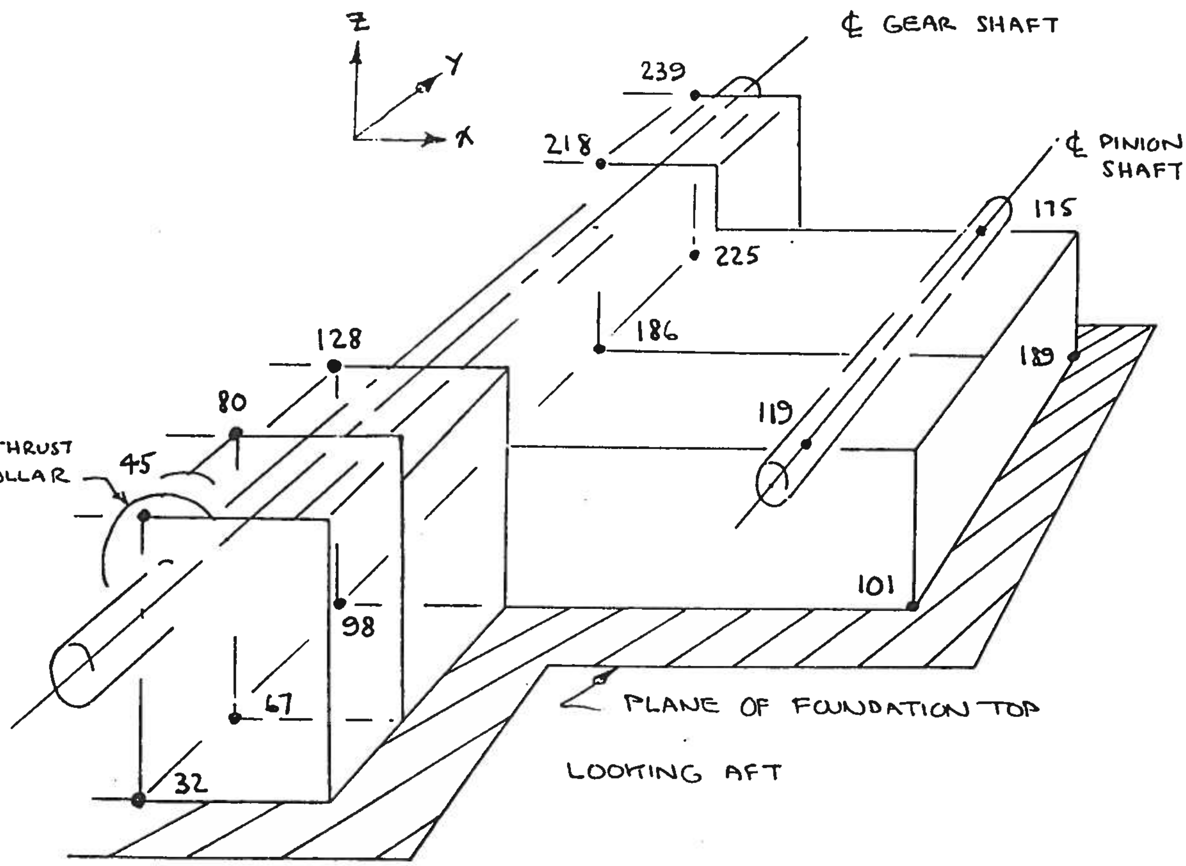


Figure 11. Skeleton Sketch of Half of Gear Casing to Show Location of Significant Finite Element Nodes

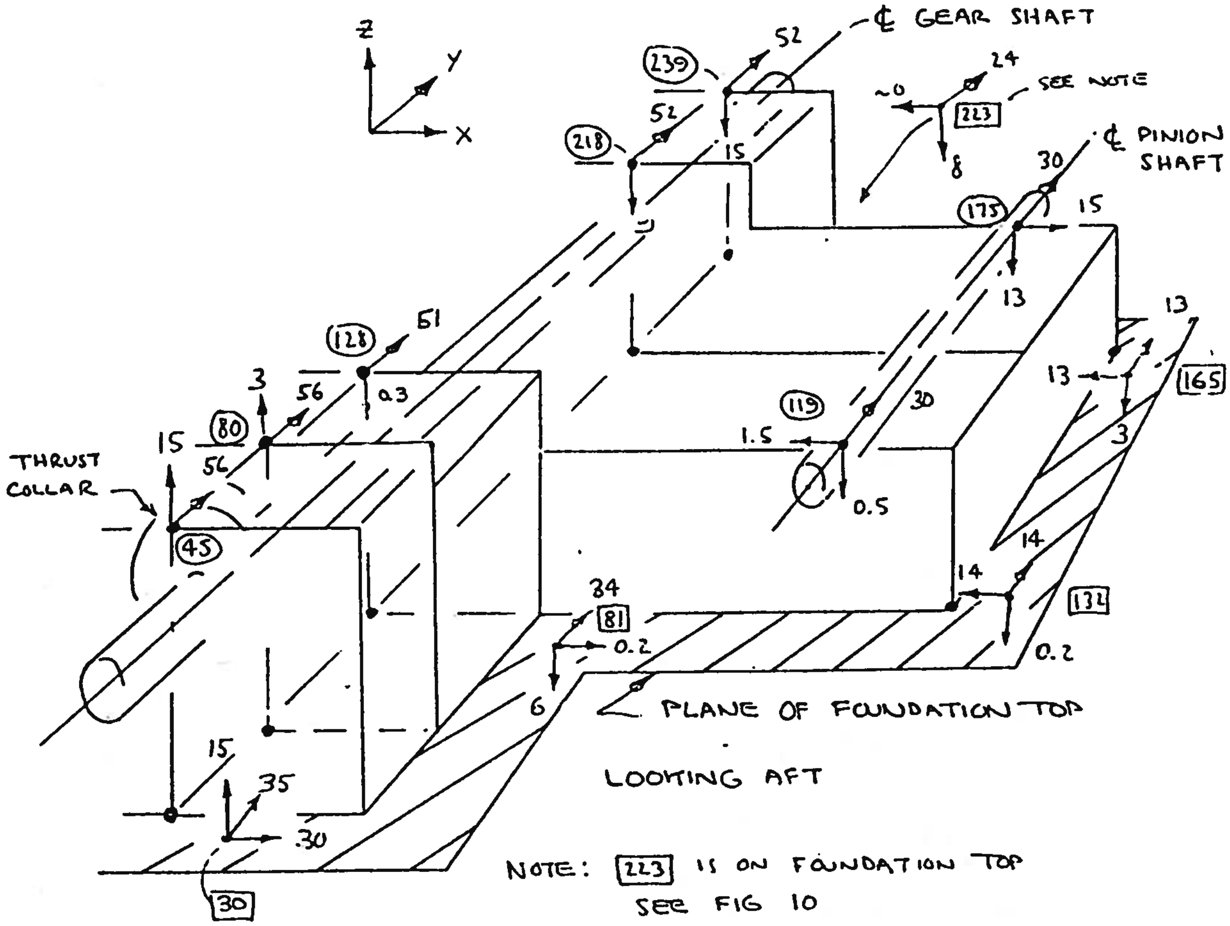


Figure 12. Figure 11 repeated with Approximate Nodal Displacements Shown

EVALUATION OF THE EFFECTS OF HULL DEFLECTIONS

Task 4 of our project calls for evaluation of the effect of the effects of hull deflection on the components that we analyze.

Our work on the pinion bending and torsional deflection does not involve hull bending. The analysis that we did make shows no evident need for discarding the traditional approach based on elementary beam theory.

Our work on clutch drums and spacers goes far beyond what is feasible with an approach other than finite element analysis, but we are unable to show any significant consequence of the hull deflections.

For the propulsion shaft, treated as a separate body loaded by eccentric thrust, our results are inconclusive. Although the finite element method allows a wide variety of cases to be explored, our results do not confirm actual bearing load measurements as listed in Table 1. Two explanations for the failure are evident. The first is the possibility that the observed effect is caused by the hull deflection, and not simple bending of the shaft, and this, of course, is the cause we are actually seeking. The other possible explanation lies in the clearance between shaft journals and the bearing surfaces. This clearance is 0.014 inch (diametral). The effect of the shaft moving within this clearance is difficult to model without good knowledge of the hydrodynamic "spring constant" within the clearance, and this knowledge was not available within the scope of our project. Nonetheless its significance is apparent from the results showing structural deflections of the same magnitude as this clearance.

Returning to consideration of the effect of hull deflections, we recall the calculation reported in an earlier section, a bearing load calculation using shaft influence coefficients and hull deflections taken from reference 1. The results of the calculation fail to confirm the measurements reported in Table 1. Now, however, we have the more detailed calculations whose principal results are given in the preceding section. An inspection of Figure 12, for example, indicates that the forward bull gear bearing is lowered with respect to the after by about 0.014 inch. A subsequent calculation with the influence coefficients from reference 1 shows that these deflections should result in an unloading of the forward bearing, just as the Table 1 results show. (Note again, though, that the difference in bearing deflections is equal to the bearing clearance.)

The major consequence of interest in gear structure deflection is the possible upset of the manufacturer's intended alignment between pinion teeth and bull gear teeth. Our deflection calculations, perhaps best exemplified by the numbers in Figure 12, show the pinion shaft moving outboard 0.015 inch at one end, 0.0015 inch inboard at the other (compare nodes 119 and 175 in Figure 12). The nodes in question are 30 inches apart, giving a misalignment of 0.00055 inch/inch. This is greater than the maximum advisable figure of 0.0002 inch/inch mentioned by Anderson and Zrodowski (reference 2), but the movement is "in plane" only; the teeth are not opening, but simply moving generally farther apart at one end.

A comparable look at the relative displacements in the vertical plane shows virtually no movement.

Our calculations show deflections of the gear casing structure that may be significant, but fail to show magnitudes that are unquestionably harmful to satisfactory gear tooth mesh.

CONCLUSIONS

Our work shows that improved analysis (i.e. application of the finite element method) in the ship designer's work on deflections of clutches and reduction gear pinions.

Our finite element analysis shows that the reduction gear structure does undergo deflections in addition to the movements imposed by the hull. The result is a possible calculational confirmation of a recorded change in bearing loads (Table 1), but on the other hand, no significant consequence to the tooth mesh is shown. However, our calculations of gear case deflections have not been confirmed by measurements on comparable actual hardware. The validity of our model is therefore not adequately established.

A significant result of our work on shafts and on the gear structure is that calculated deflections are of the same order as shaft bearing clearance. Our inability to include in the finite-element model a well-documented representation of the behavior of the shaft within the clearance reduces the validity of the results.

RECOMMENDATIONS

We have no recommendation to the propulsion design engineer for change in design practice. Rather, our work illuminates several areas of incomplete knowledge, and in consequence our recommendations are for additional research for later benefit atthe design level.

1. That deflections of reduction gear structure be investigated, with analysis such as that done in this project confirmed by measurements.

2. We have found that the deflection of shafts within their bearing clearances may be significant in analysis, but our present finite element techniques are not adequate to accommodate this factor. We therefore recommend that additional research be done on modeling the behavior of propulsion shafts, particularly to include the significance of clearances.

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