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ENCLOSURE 2

December 16, 1983

Mr. T. M. Novak
Assistant Director for Licensing
Division of Licensing
Office of Nuclear Reactor Regulation
Nuclear Regulatory Commission
Washington, DC 20555

Subject: Standby Diesel Generators at
Nuclear Power Plants

Reference: Mr. T. M. Novak's Letter of December 1, 1983

Dear Mr. Novak:

The Users' Group gave us a copy of the three-page list of nine questions at the November 30, 1983 meeting. This is the same list sent to us on November 29, 1983 by your Mr. R. Caruso. We are today sending the Users' Group answers to the nine questions. Since the content of the nine questions is essentially the same as the list submitted by the referenced letter, we are sending you a copy of our answers to the nine questions asked by the Users' Group.

We trust that these nine answers are responsive to the list submitted by the referenced letter. However, if there are any additional questions, please don't hesitate to call. We intend to cooperate fully with the NRC and the Users' Group to answer all your questions.

Very truly yours,

C. S. Mathews
Vice President and General Manager

CSM/WVD/pn

NRC QUESTIONS

Q #1 - Describe the history and evolution of crankshaft design of DSR-48 diesel generators.

A #1 - The DSR-48 diesel engine crankshaft was developed from the DSR-38 engine which has been in production since the early fifties.

The DSR-38 was developed from the "Q" engine which was in production since the thirties. The "Q" engine had a 10" diameter crankpin and 11" diameter main journal and was rated at 260, 327 and 360 rpm. The R-8 engine started with an 11" diameter crankpin and 11" diameter main journal (11" x 11") and changed to 11" x 13" and 12" x 13" during the course of evolution, from 300 rpm originally to 327, 360, 375, 400, 425 and 450 rpm. The first DSR-48 engine was built and shipped in 1969. It was rated for 400 RPM operation. The first 450 RPM DSR-48 engines were built in 1975.

Q #2(a) - What prompted you to change the size of the crankpin after the Shoreham engines were built?

A #2(a) - TDI changed the crankpin diameter to achieve higher torsional stiffness. This change to the engine was made to give broader capabilities as a driver for different applications, such as pump and marine drivers. Further, the change is a part of the evolutionary process. For example the crankpin of the "RV" had been changed from 12" to 13" the previous year for the same reasons and not because of a problem with the 12" pin shaft.

Q #2(b) - When was the decision made to change the crankpin size?

A #2(b) - The drawing for the 12" crankpin crankshaft no. 03-310-05-AD was dated 2/4/75.

Q #2(c) - Why was the crankpin fillet geometry changed?

A #2(c) - The "RV" crankshafts have a 3/4" fillet. When the change was made to the crankpin diameter of the "R" engine, TDI made the fillet radius change to again commonality in design between the R-48 and "RV". The commonality is desirable from a manufacturing standpoint.

Q #2(d) - When was LILCO informed of the change in crankpin size?

A #2(d) - Immediately following the crankshaft failure at LILCO. The requirement for a quick supply of new crankshafts dictated the 12" diameter pin shaft be supplied because it was the only shaft immediately available.

Q #3 - What is the TDI mechanism for informing its customers of problems of product improvements? Does TDI use a technical information letter approach or its equivalent?

A #3 - The TDI mechanism for informing it's customers of problems or product improvements is the Service Information Memo (SIM) program.

The SIM is to a Technical Information Letter with the additional advantage of an index system, which allows the collected SIMs to form a fourth volume of the Instruction Manuals.

Additionally, TDI informs nuclear plant customers of "potential defects" as required by Federal Law 10 CFR 21.

Q #4 & - In its report on the crankshaft failure, LILCO's consultant noted that
4(a) the forcing function used by TDI in its torional analysis changed significantly between 1975 and 1983.

A #4 & The torsional analysis for LILCO used the forcing functions which were
4(a) in the TDI computer program data base in 1974. We made two changes to the forcing functions values in 1975. The second change made in 1975 was used until 1977. In 1977, we made minor refinements to the forcing functions and these remain in use today.

Q #4(b) - Why did this change occur?

A #4(b) - The analytical results from the torsional analysis are verified by torsiograph tests. Since the intention of the analysis is to accurately predict the natural frequencies and stress levels of the diesel generator system, input data to the computor program is adjusted, so that the calculations result in agreement with the test results. The changes to the forcing functions are steps taken to match calculated or predicted values with those obtained from tests. Our current forcing functions predict slightly higher stress levels than measured.

Q #4(c) - What effect does this chagne have on any other components of DSR-48 engines?

A #4(c) - None. The changes to the forcing functions were made to get the computor assisted computation to accurately predict the actual behavior of the engine generator shaft mass elastic system.

Q #4(d) - What forcing functions were used in the design of other TDI engines (such as the DSRV-16-4s) in nuclear service?

A #4(d) - The following tabulation shows what forcing function groups were used for all the TDI engines for nuclear service.

CONTRACT NUMBER	CONTRACT NAME	TORSIONAL REPORT DATE	HARMONIC COEFFICIENT GROUP
74010	LILCO	7/18/74	1
75041	S.C.E.	6/27/75	1
75005	KUOSHENG	4/25/75	1
74046	C P & L	1975	2
74033	M P & L	9/15/75	3
75017	DUKE-CATAWBA		3
75051	C.E.I.	7/9/76	3
75084	WPPSS	8/16/76	3
75080	TVA-BELLEFONTE	4/27/76	3
76001	T.U.S.I.	1/5/76	3
77001	CONSUMERS	4/26/77	3
74039	GULF STATES	5/3/77	3
77024	TVA-STRIDE	8/16/77	4
76021	GEORGIA PWR.	8/1/78	4
78006	MAANSHAN	6/22/78	4
81015	S.M.U.D.	9/10/81	4

HARMONIC COEFFICIENT GROUP 1 1974 TO 1975
HARMONIC COEFFICIENT GROUP 2 1975
HARMONIC COEFFICIENT GROUP 3 1975 TO 1977
HARMONIC COEFFICIENT GROUP 4 1977 TO CURRENT

HARMONIC COEFFICIENT LISTING

PERIOD	74 - 75	75	75 - 77	77------
LISTING FROM	LILCO	C P & L	M P & L	STRIDE
HARMONIC	GROUP 1	GROUP 2	GROUP 3	GROUP 4
.5	11.00	90.88	97.00	155.45
1	20.62	89.78	94.34	94.21
1.5	19.00	94.88	100.70	129.21
2	24.06	45.43	42.53	42.61
2.5	20.20	62.38	65.61	71.51
3	19.97	14.84	16.57	16.52
3.5	16.70	38.91	40.61	42.72
4	13.30	29.04	30.25	27.62
4.5	9.85	12.48	12.73	12.72
5	7.30	9.21	9.39	9.38
5.5	5.65	7.01	7.14	7.14
6	4.18	5.55	5.68	5.68
6.5	3.29	4.39	4.49	4.49
7	2.66	3.60	3.69	3.68
7.5	2.23	2.98	3.05	3.04
8	1.87	2.46	2.52	2.52

8.5	1.61	2.20	2.26	2.26
9	1.42	1.92	1.97	1.97
9.5	1.25	1.50	1.53	1.52
10	1.11	1.25	1.27	1.27
10.5	1.00	1.13	1.14	1.14
11	.91	1.01	1.02	1.01
11.5	.82	.88	.89	.89
12	.74	.78	.79	.79

Q #4(e) - Have these forcing functions changed?

A #4(e) - The current Tn values have been in use since 1977.

Q #4(f) - Please describe the development of the forcing functions for each TDI diesel in nuclear service.

A #4(f) - The forcing functions are derived from Fourier analysis of the torque vs crank angle diagram for one cylinder. These forcing functions are subsequently adjusted to correlate the analytical results with test results as already noted.

Since all the TDI engines for nuclear service are rated at 225 bmep and 450 rpm, (except for S.C.E.) which has a lower rated RV-20-4, the forcing functions are similar.

Q #5(a) - What does TDI view as the reason for the Shoreham crankshaft failure?

A #5(a) - Site operating stresses approximately equal to the endurance limit caused high cycle (10. 06 to 10. 07) fatigue failure of the crankshaft.

Q #5(b) - What conclusions has TDI drawn from the LILCO failure report?

A #5(b) - The operating stresses in the 11 x 13 cranksaft were essentially equivalent to the endurance strength and results in high cycle (10 E 06 to 10 E 07) fatigue failure. The failure is effectively an unfortunate endurance limit test. Even a small reduction in stress (perhaps only 2 or 3 percent) would have resulted in unlimited life.

Since the 12" x 13" crankshaft is subjected to significantly reduced stresses it will result unquestionably in a shaft that will give unlimited life. In these matters we are in complete agreement with the LILCO/Failure Analysis Associates report. However, we do not feel that the FaAA analytical analysis, particularly the finite element model (Sec. 6 of report) is necessarily satisfactory. It fails to predict the actual state of stress measured by Stone & Webster (Sec. 4) and it fails to satisfactorily predict the crack location and direction. The crankshaft stress analysis is inadequate and therefore does not fully explain the reason for failure.

TDI is currently engaged in its own stress analysis program, which is expected to yield a more accurate analytical model and a clear understanding of the stresses which caused the failure.

Q #5(c) - What actions has TDI taken or does TDI plan to take for Shoreham and other plants as a result of the Shoreham crankshaft failure?

A #5(c) - TDI plans to continue its investigation into the reason(s) for the Shoreham crankshaft failure in accordance with the outline given in the discussion presented by Mr. Greg Beshouri (attached). The results of these investigations will be published at the appropriate time and made available to all interested parties.

Q #5(d) - Does TDI plan to prepare a report of its own regarding the Shoreham crankshaft failure?

A #5(d) - TDI will develop a formal report containing it's views on the reasons for the failures. Much of the report will be developed using understandings gained from the R&D studies outlined above.

Q #6(a) - Describe how TDI design calculations are reviewed and independently verified?

A #6(a) - Calculations performed by design engineers are reviewed, signed and dated by the Manager of Design Engineering.

Designs which rely on calculations in which assumptions cannot be verified are subject to experimental testing by the Research and Development group. In some instances the Manager of Applied Mechanics will also review the result. Some components are subjected to testing on a shaker table, if practical.

Q #6(b) - What detailed stress analysis of the crank web and pin were performed?

A #6(b) - No detailed analysis were done on the crankshaft other than the crankshaft was designed to American Bureau of Shipping Rules, as detailed in the attached excerpt from the rule book. TDI has successfully used such rules as a design standard for 45 years. The R-48 crankshaft was developed from the "Q" engine with 10" x 11" (10" diameter crankpin and 11" diameter main journal) to the first "R" engines with 11" x 11" crankshafts then 11" x 13" to the current 12" x 13" configuration.

Q #7 - Could the problem with the crankshaft have been detected during initial torsiograph testing at the factory?

A #7 - No. The total vibratory amplitude measured was only + or - .50 deg. which equates to a stress of 5314 psi. The portion attributed to the fourth order was + or - .43 deg. or 4570 psi, well within the 5000 psi allowed by DEMA for single order contribution. (The stress required to break the crankshaft was more on the order of + or - 30 to 35 ksi.)

Q #8(a) (i) -LILCO has also identified problems with failures of the diesel engine connecting rod bearings. We understand that they have provided you with a copy of their initial report on the subject. (a) What does TDI view as the reason for the Shoreham bearings failure?

A #8(a) - Four of the forty bearing shells were reported to be cracked, only one of which had a significant crack through the edge of the top shell. The small piece 4-7/16" long and 11/16" wide at the thickest point was jacked apart from the main body of the bearing shell for study. None of these shells had failed to the extent that the clearances were opened up nor did any of the shells result in damage to the crankpin. A photo- micrograph of this broken bearing showed porosity ranging from 0.01 to 0.03 in diameter. In addition, the material was found to be below standard for elongation. An examination of the fracture surface with scanning electron microscopy identified some of these voids as the apparent crack initiation locations. In compression the porosity would not pose a problem. However, the overhung bearing arrangement resulting from a 1/4" chamfer on the connecting rod as shown in Figure 1.1 (attached) in conjunction with the normal yawing of the crankshaft, put the I.D. of the bearing into tension. The surface porosity acted as stress intensifiers and with the poor material elongation characteristics, initiated a crack. This was clearly a material rather than design problem as evidenced by the fact that more than 300 cylinders of this connecting rod arrangement are in operation, many of which have operated for more than 25,000 hours without bearing problems.

Q #8(b) - What action has TDI taken to ensure that new bearing will not fail in a similar fashion?

A #8(b) - In regard to LILCO and other R-48 engines installed in emergency standby service, the crankshaft is fitted with a connecting rod which has a smaller 1/16", chamfer on the edges. Figure 1.2 (attached) shows the bearing is fully supported. Even though there may be some porosity in the bearing shell material, the shell is in compression and therefore minor porosity would not be detrimental.

Chemical and physical properites of casting lots or heats are tested to verify compliance with requirements. In addition, TDI does a visual inspection for porosity of each shell during manufacture.

Q #8(c) - What other TDI engines in nuclear service use similar bearing material?

A #8(c) - All of the nuclear and commercial engines which TDI manufactures contain bearings using identical B-850-T5 bearing material. This material is a 6% tin content aluminum alloy. We purchase castings from Aluminum Company of America and perform all machining and plating operations at the Oakland facility.

Q #8(d) - What action has TDI taken for other engines to preclude their failure in a similar fashion?

A #8(d) - The bearings in all other engines in nuclear service have connecting rod and bearing arrangements shown in Figure 1.2 and 1.3 (attached). Each provides full support for the bearing shell. The bearing shell is in compression, both from crush and operating forces, with no portion of the shell in tension. Therefore, if bearing material contains minor porosity, as all castings do, the loads present will not act with the stress intensifiers and result in cracks.

Q #8(e) - What controls has TDI provided on bearing material in the past?

A #8(e) - The purchase order for bearing material has required the supplier to furnish a Certified Material Test Report (CMTR). This was a requirement in 1974 and still is. The CMTR is reviewed for compliance to the material requirements. All bearings are inspected visually for porosity during the manufacturing process.

Q #8(f) - How did and does TDI ensure that bearing material meets its specifications?

A #8(f) - In 1975 TDI initiated it's own bearing material sample testing program to check checmical and Physical properties against specification and the CMTR supplied by the vendor. This program remains TDI's standard practice.

Q #8(g) - What other experience has TDI had with connecting rod bearing failures, of any kind, in any nuclear or non-nuclear installations?

A #8(g) - TDI customers have encountered occasional babbitt fatigue. It has the appearance of small worm holes in the surface of the babbitt. In addition several users have suffered the results of faulty reinstallation, dirt ingestion and abuse which have resulted in bearing failure.

Q #8(h) - What procedures does TDI use to ensure that bearings and journals are properly designed and manufactured?

A #8(h) - TDI has been designing , developing and building engines since before 1938. The intervening years have provided considerable experience and knowledge regarding what constitutes a properly designed crankshaft journal and mating bearing, such as L/D ratio, surface finish, babbitt thickness, etc. In addition, we work closely with the bearing material vendors regarding the bearing design. All of this information culmanates in a design that is translated into detail drawings for manufacturing. The QA department ensures conformance to the drawing requirements through TDI's 10CFR50B program. The bearing material vendor provides Certified Material Test Reports (CMTR's) for each casting heat which are review for conformance to the drawing requirements and are verified by TDI's own Chemical & Physical test for each casting heat.

Q #8(i) - Describe any problems you or any of your customers have encountered with the use or manufacture of aluminum bearings with babbitt overlays.

A #8(i) - Users have occasionally encountered babbit fatigue in the bearing overlay. This may occur if the tin content in the babbitt is too low, resulting in a weaker babbit. The composition of the babbitt is monitored quite closely. TDI has initiated a change in the babbitt composition to further improve the fatigue resistance. This calls for the inclusion of 2.75 - 3% copper in the S.A.E. - 19 babbitt. With the TDI bearing design, babbitt fatigue or even complete babbit overlay loss does not result in any sort of catastrophic bearing failure that might cause the engine to stop functioning properly. TDI has also occasionally encountered porosity and low elongation characteristics in the aluminum castings used to manufacture the bearing shells.

Q #9 - LILCO as also identified problems with cracks in almost all of the piston skirts at Shoreham.

A #9 - This statement is incorrect. There has been only one piston at LILCO which has been identified as having a crack. The examinations being conducted at the site are using an "eddy current" inspection process which TDI and it's Metallurgy Consultant considers not suitable for examination of cast nodular iron surfaces. This eddy current process has predicted linear indications in the piston skirts which in most cases may be nothing more than the grain boundaries within the nodular iron structure.

Q #9(a) - Describe the stress analysis and testing that has been done by TDI in the development of type AF, AN and AE pistons.

A #9(a) - AF, AN, and AE pistons have been subjected to many experimental test programs to reveal the patterns of stress and temperature existing in the assembly. The tests included studies of thermal distortion, effects of combustion pressure and inertia forces. Finite element analysis (FEM) was attempted on a crown, however the technique proved to be less than adequate.

Piston assemblies of the AN and AE type were successfully run for 687 hours in the experimental R5-V12 engine at 514 rpm and a power level of 937 BHP per cylinder to support the results of the static and analytical studies. Nuclear standby generator diesels are rated at 450 rpm and 609 BHP per cylinder. Therefore the test work subjected the pistons to considerably higher operating stresses than the pistons used in any standby engine.

Q #9(b) - Has TDI or any of its customers encountered similar or different problems with piston cracking?

A #9(b) - The crack reported by LILCO is the first such crack identified and reported on the modified "AF" style piston skirt which has been manufactured in accordance with design requirements. There are 252 modified "AF" piston skirts operating, which have accumulated in excess of 1,772,000 hours of successful operation.

The "AN" style piston has experienced several field failures, which have been attributed to high residual stresses not removed by a stress relief process. There have been no reported failures of the "AN" style piston which have been stress relieved and properly machined. There are 1374 "AN" style pistons opeataing which have accumulated in excess of 2,760,000 hours of successful operation.

The "AE" style piston is the latest TDI R-4 piston design and incorporates prior R-4 design and operating experience and new design knowledge we have gained through our R-5 engine test program. The "AE" piston design has been successfully tested in our R-5 test engine at 514 rpm and 302 BMEP and has acquired in excess of 7000 operating hours in a 16 cylinder 7000 kw engine in the field.

Q #9(c) - Has TDI modified its piston skirt design to improve stress levels in the area of the bolt holes?

A #9(c) - As part of a continuing program of product performance and reliability improvements, TDI has modified the piston skirt design to improve stress distribution in the area of the fastener holes and in the circumferential mid rib blend to the wrist pin boss.

Q #9(d) - How and when were these modifications made?

A #9(d) - Primarily as a result of the studies referred to in the answer to question 9a, TDI concluded that a more massive boss around the bolthole would better diffuse forces to the piston pin area. Calculations also verified that the protection afforded the fasteners against cyclic loading could be achieved with only 13 belleville washers instead of the original 26 washers.

On August 10, 1982, piston skirt 02-341-04-AE was released for production. It required that a change be made to the corebox in which the mold for the piston skirt interior is formed. This change provided the more massive bosses around the boltholes and precluded the manufacture of earlier designs.

CRANKSHAFT STRESS ANALYSIS PROGRAM

Greg Beshouri, Research Engineer

INTRODUCTION

With the failure of the 11" x 13" crankshafts in the LILCO DSR-48 (S/N 74010/12)engine, TDI initiated a stress analysis program (including physical testing and analytical modeling) with the objective of determining the stresses and their sources in an 8 throw 11" x 13" crankshaft in order to identify the actual causes of the failure of the LILCO shafts. In addition, this program is intended to provide a more sophisticated input for future crankshaft stress analysis and design.

PROCEDURE

Literature Review

Prior to the initiation of physical testing, an extensive review of the available literature was conducted. From this review we determined, as expected, that a crankshaft in service is subjected to a complex, dynamic state of combined stresses. The key to successful stress analysis is an understanding of the source of each stress component and how these individual components add into the combined stress state. The literature indicated the necessity of strain gage testing. The technical papers also were a good source of information on what other researchers had used in regard to gage type, length and location.

General Physical Test

As noted, the literature review confirmed the need for strain gage testing. At the beginning of our investigation, Stone and Webster (LILCO consultants) had already committed themselves to conducting dynamic strain gage testing,a path felt to be very difficult to follow because of the myriad of instrumentation problems associated with frequency modulated (FM) telemetry (a method of transmitting strain information via radio waves from the operating crankshaft). Therefore, we elected to perform static strain gage testing on an available engine at our facility in the hope it would complement the S&W Dynamic testing. This static testing was designed to provide information necessary to interpret and verify the feasibility of the dynamic test data.

The testing was done on a TDI Research and Development engine with a 11" x 13" crankshaft. The crankshaft is similar to but not identical to the 11" x 13" shaft which failed at LILCO. The crankshaft of this unit was statically loaded to simulate dynamic forces in the 5th through 8th throw of an R-48 engine rated at 225 psi BMEP at 450 RPM.

The crankshaft loads from gas pressure (less inertia), torque transmission and torsional vibration were first simulated independently. They were then added in several different combinations to determine the resulting stress. From this data a general solution was obtained by which it is possible to predict maximum crankshaft stress for any combination of bending and torsional stresses.

Test Apparatus

The crankshaft was subjected to torque in such a manner as to simulate torsional stresses from transmitted torque and from torsional vibration, and to bending forces simulating gas pressure (less inertia).

The necessary torque was generated by fitting cylinders No. 2 & 6 clearance volumes with spacers and O-ring seals (see Figures 1 & 2), and then pressuring them with oil with the two pistons located at 240 deg. and 120 deg. ATDC respectively.

Similarily, bending force was generated by sealing and pressurizing cylinder No. 3 with the piston at TDC and 10 degrees and 20 ATDC.

Stress Measurements

The stresses generated by torque and bending forces were measured by resistance type strain gages located on the No. 3 crankpin fillet and on the crankpin (see Figures 3 & 4). Rosettes B through E located in the fillets measured maximum strains and their principle direction. Rosettes A & F measured torsion and bending on the surface of the free part of the pin. Comparison of A & F (free part of the pin) with B through E (fillet), yields the stress concentration effect of the crankpin-fillet-web configurations.

Rosette H (only outer two gages used) located on the cylindrical surface of the No. 5 main journal verified the actual torque induced in the system.

All rosettes were rectangular three gage type of 0.125" (3 mm) effective length, manufactured by Micro-Measurements (P/N CEA-06-125UR-120).

Test Sequence

Pure torsion was first simulated by pressuring cylinders 2 & 6 only in 300 psi increments from 0 to 1200 psi yielding torques up to 2,360,000 in.lbf.

Pure bending at TDC was then simulated by pressuring cylinder No. 3 only, in 400 psi increments from 0 to 1600 psi, representing a maximum peak firing pressure in excess of 1900 psi, (Note that the primary and secondary inertia forces of the piston and connecting rod assembly oppose the firing pressures and have the equivalent effect of lowering the firing pressures by 377 psi when operating at 450 rpm.

Then, by pressuring cylinders 2, 3 and 6 appropriately, combined torsion and bending representing the actual stress state was simulated.

The pure bending and combined bending and torsion tests were repeated with cylinder No. 3 piston located at 10 and 20 equivalent degrees ATDC.

DATA ANALYSIS

The strain gage data were reduced to maximum and minimum principle stresses and principle directions.

From bending load and torque data, general mathematical expressions were derived for nominal stresses in the free part of the pin due to bending loads and torque. In addition, stress concentration factors were calculated for various fillet locations.

From the combined stress data, a general analytical technique (using Mohr's circle) for calculating combined stress due to a given bending load and torque was generated. It was then confirmed that this technique could be applied in reverse, i.e., given a certain combined stress state, the bending load and torque creating this stress could be calculated.

Using this technique the dynamic stress data taken by Stone & Webster on Unit 101, TDI S/N 74011, at LILCO were then broken down into components of dynamic bending load and dynamic torque creating the stress, resulting in a clear understanding of the dynamic state of stress components. Had the static testing not been conducted, it would not have been possible to satisfactorily decipher the dynamic stress data taken by S&W.

ON-GOING PROGRAM

Because of the failure of the LILCO crankshafts and a requirement to clearly understand the reasons for the failure and given the success to date of the several methods of crankshaft stress analysis applied to the 11" x 13" shaft, we have committed ourselves to an on-going crankshaft stress analysis program.

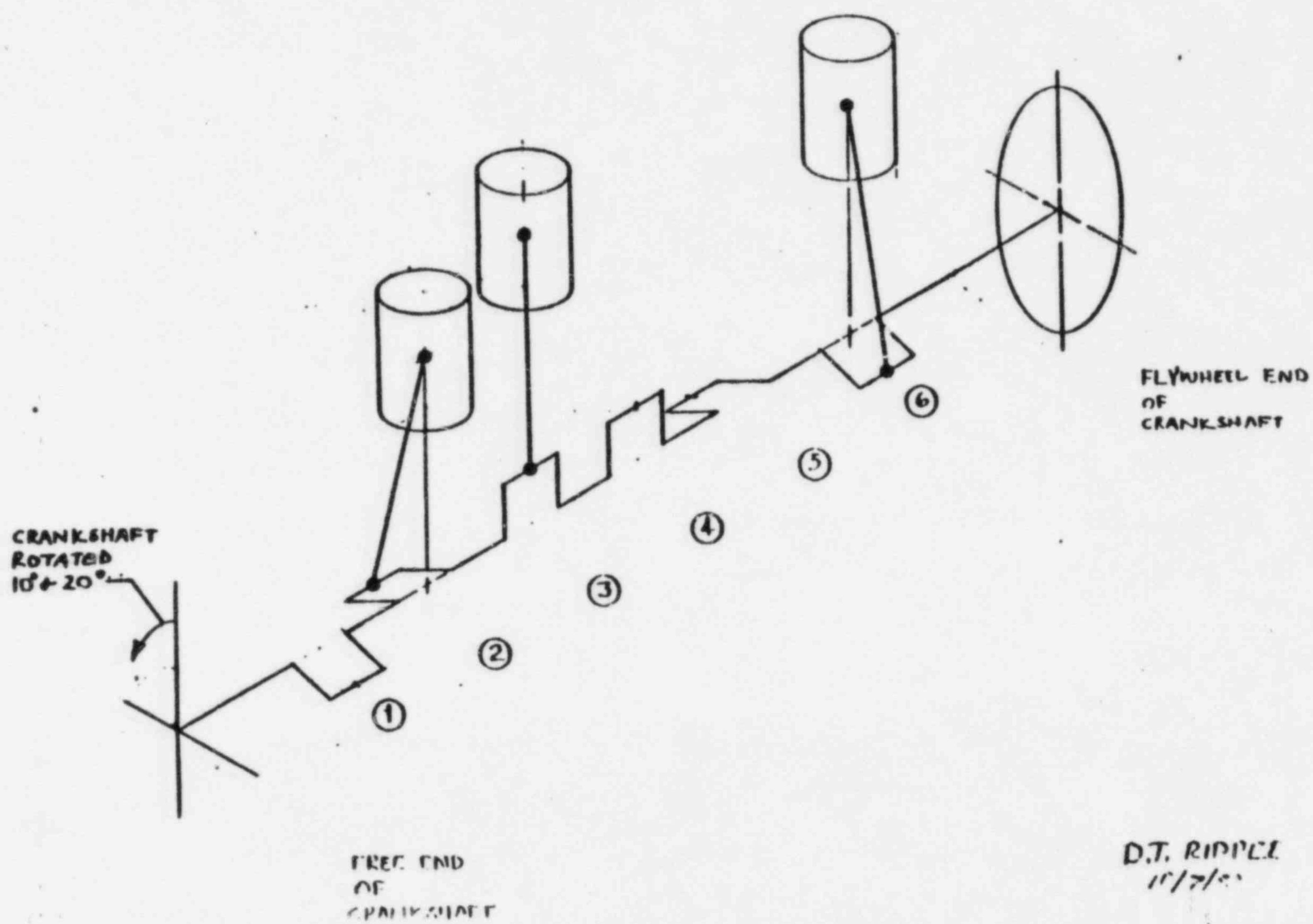
This program will proceed in two complimentary directions, analytical and experimental (Fig. 5).

First, static tests will be conducted on all crankshaft configurations to determine stress concentration factors in torsion and bending. Concurrently, a dynamic model which predicts torque and bending load will be generated. The stress concentration factors and dynamic bending load and torque calculations will then provide input for a second model which will calculate crankshaft stress vs crankangle. This calculation will then be verified by dynamic strain gage tests on selected crankshaft configurations. Once the stress calculation procedure is verified, exact maximum stress and exact operating factors of safety for any crankshaft configuration can be calculated.

In addition to providing design information for new engine programs, the calculation procedure will be used to confirm and refine the less sophisticated, more conservative stress calculation procedures currently used on R-4 series engines in nuclear service.

Obviously, the dynamic test on the 12" x 13" crankshaft, currently installed in DSR-48 engines, will be an important step in this program. Static testing must also be conducted on this shaft in order to properly decipher the dynamic data expected to accrue from tests planned for late this year.

FIGURE 1 - CRANK ORIENTATION FOR STATIC TEST



D.T. RIPPLE

FIGURE 2 - SEALING OF CYLINDERS # 2 3 & 6

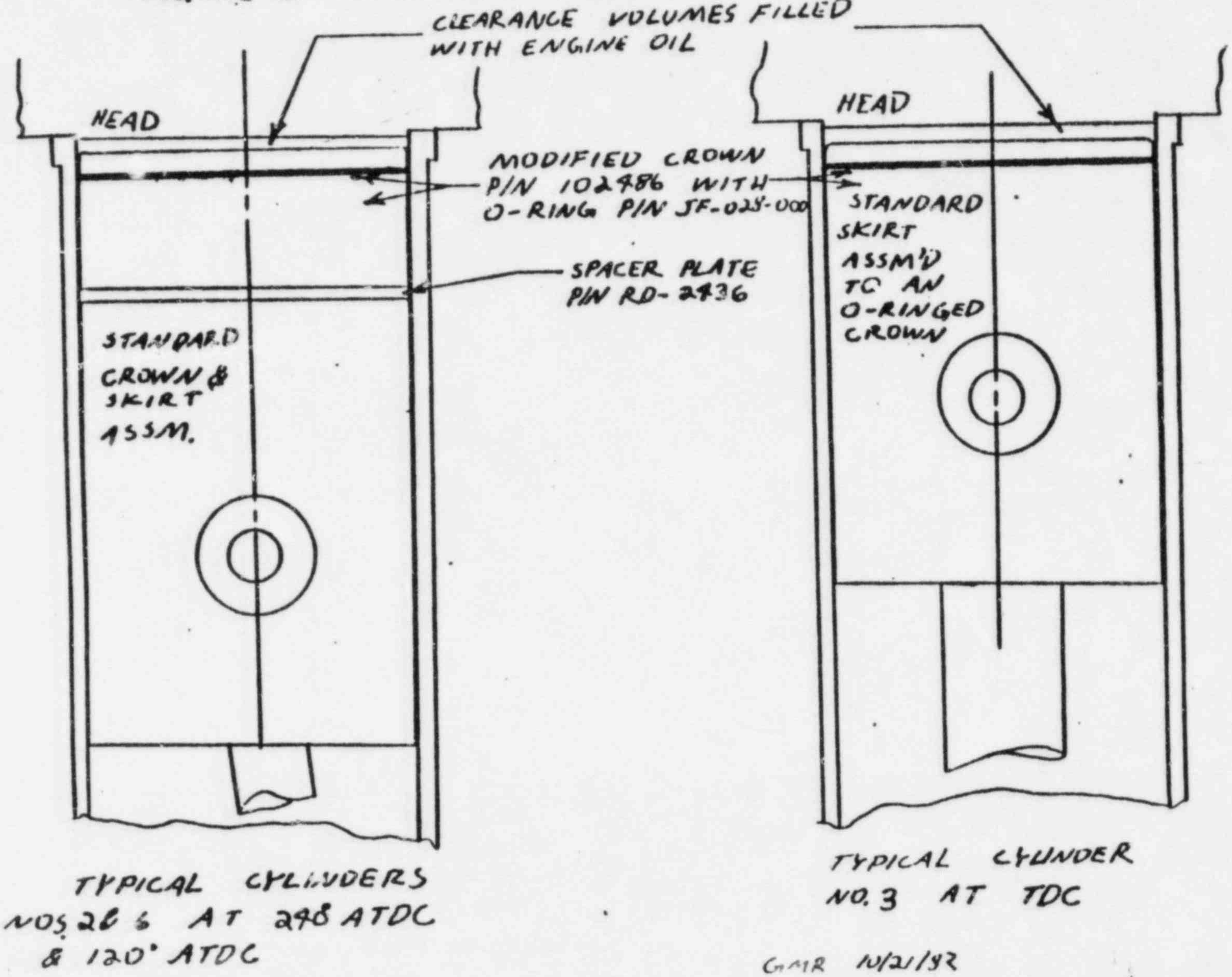
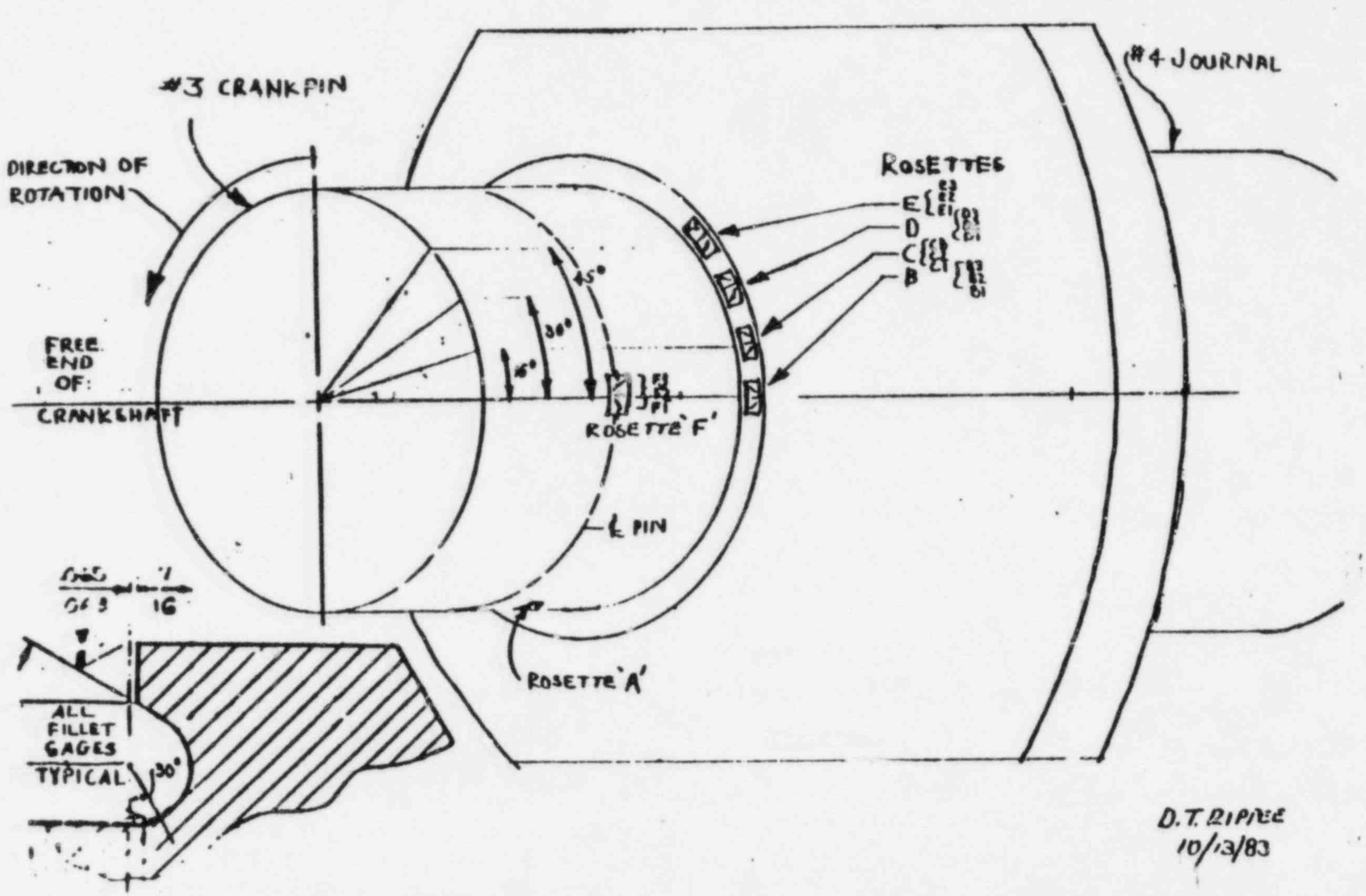


FIG. 2

FIGURE 3 - GAGE LOCATION NO. 3 CRANK PIN, VIEWED FROM UNDERSIDE OF PIN



D.T. RIPLEE
10/13/83

FIG. 3

FIGURE 4 - GAGE LOCATION NO.3 CRANKPIN & NO.5 MAIN.

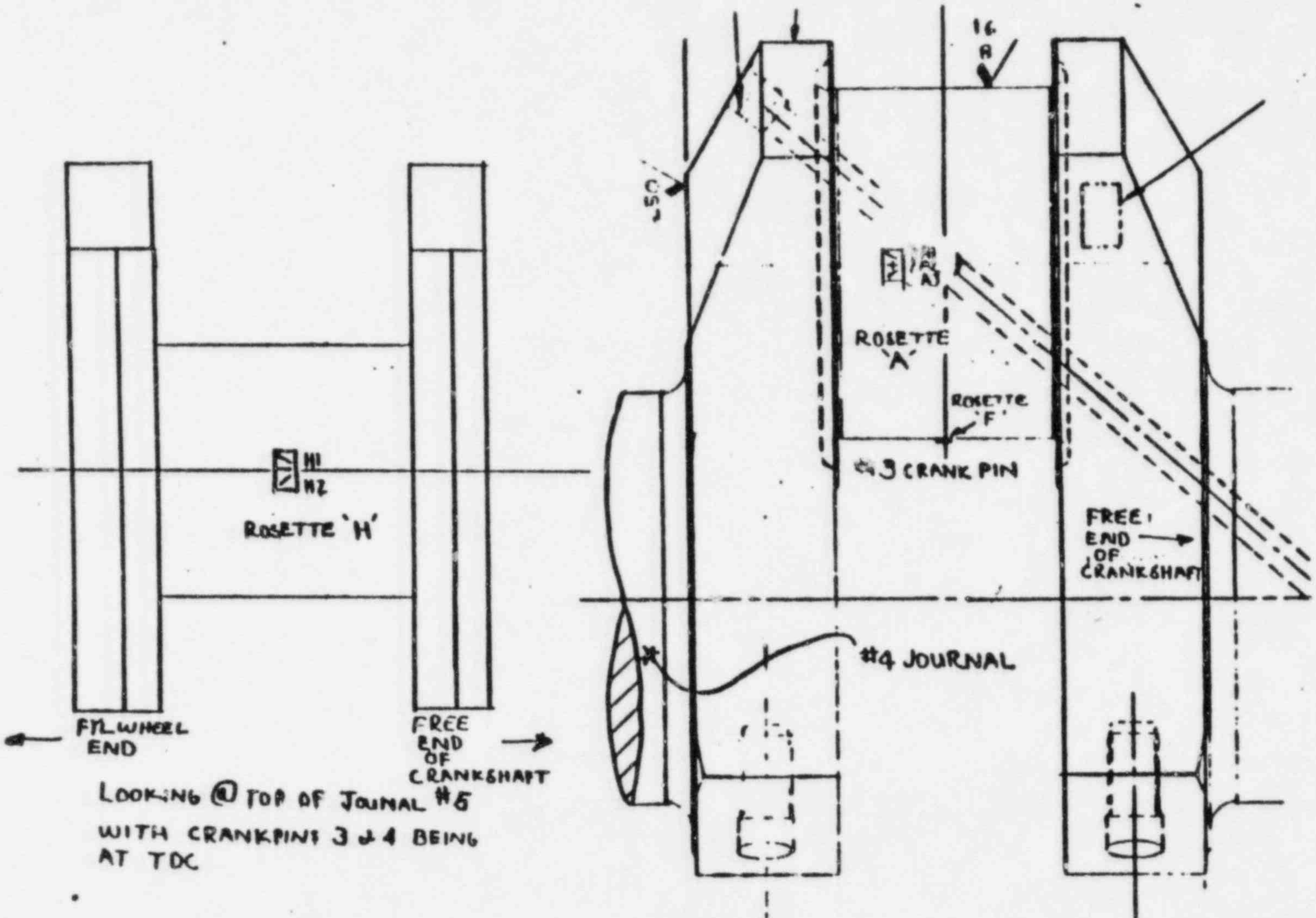


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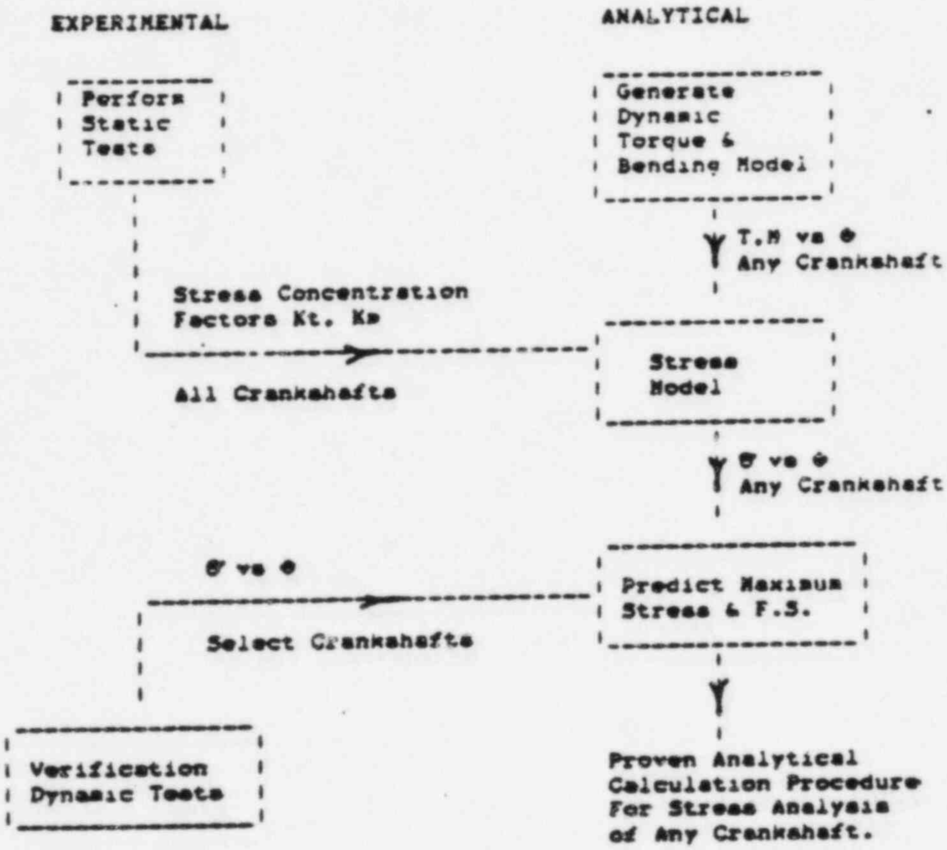
FIG. 4

CRANKSHAFT

STRESS ANALYSIS PROGRAM



GMB/wam
12/5/83

FIGURE 5 - FLOW CHART
CRANKSHAFT STRESS ANALYSIS

FIG. 5

BEARING COMPARISON

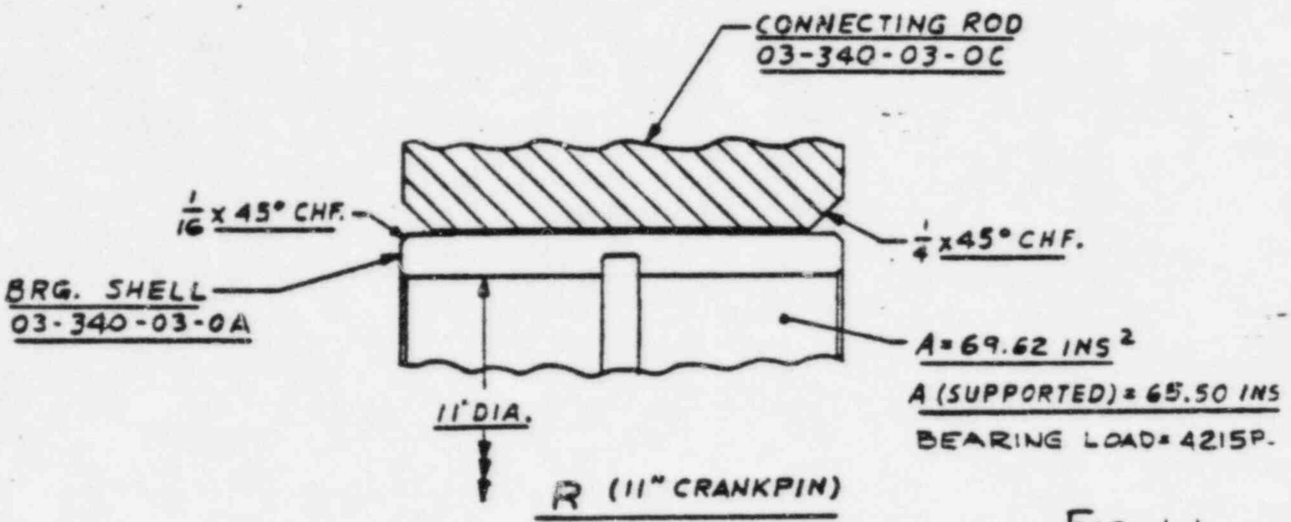


FIG 1.1

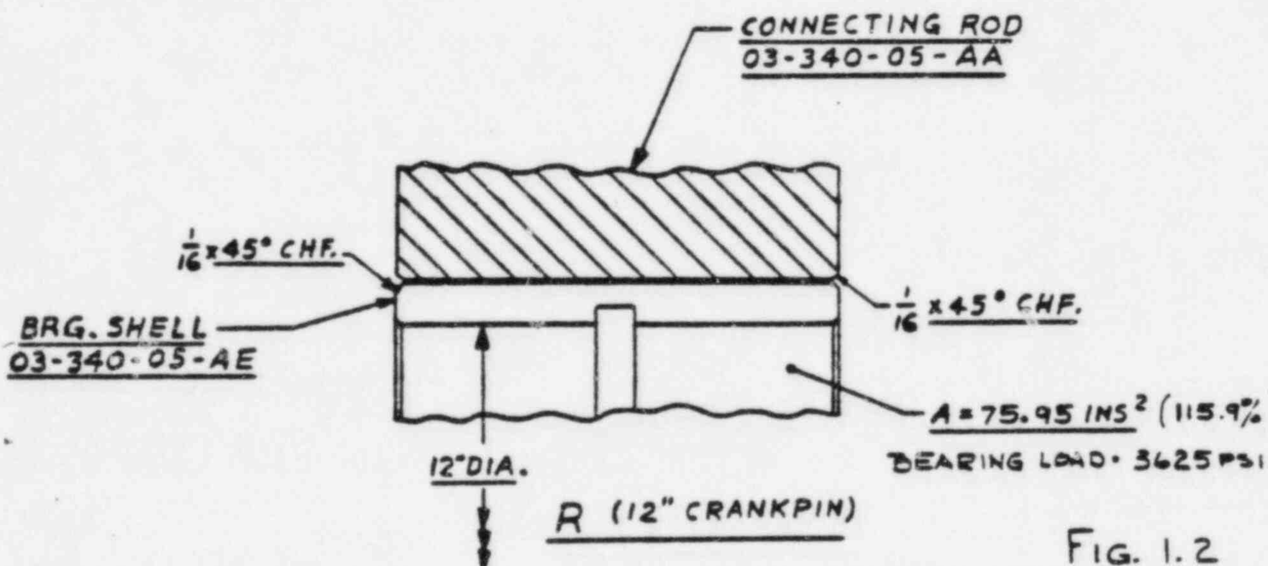


FIG. 1.2

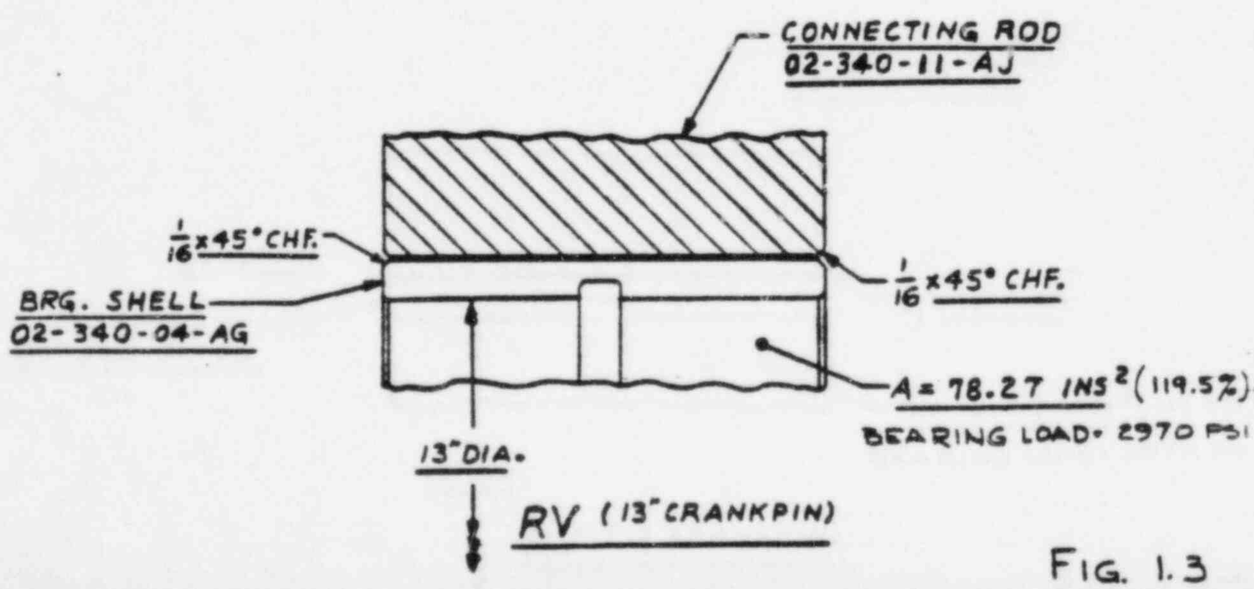


FIG. 1.3