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PLANETARY-GEAR-SUPPORT BEARING TEST RIG DESIGN

BY

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SKF INDUSTRIES, INCORPORATED

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16. Abstract
A test rig was designed to evaluate the performance of a spherical roller bearing with a geared outer ring operating under conditions similar to those of a planet bearing in a helicopter transmission. The configuration is an extension of the widely accepted four-square gearbox arrangement. It provides for testing of two bearings simultaneously with outer ring rotation, misalignment, diametrically opposed loading through the gear teeth, and under race lubrication.

Instrumentation permits the measurement of: inner and outer ring temperature, bearing drag torque, degree of misalignment, outer ring speed, cage speed, and applied load.

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FORWARD

This report documents work performed under U.S. Government Contract NAS3-23702 issued by the NASA-Lewis Research Center of Cleveland, Ohio under the administration of the Propulsion Systems Division.

The NASA Project Manager was Mr. H. H. Coe. The work was performed at SKF Industries, Inc., and Philadelphia Gear Corporation both in King of Prussia, Pennsylvania.

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1.0 SUMMARY

This program was an extension of the efforts initiated earlier under NASA Contracts NAS3-20824, NAS3-22807 aimed at establishing a comprehensive computer program for the analysis of the performance of spherical roller bearings in helicopter planetary transmissions. The work reported here consists of the design of a planetary bearing test rig, i.e. a test rig to evaluate the performance of a spherical roller bearing with a geared outer ring operating under conditions similar to those of a planet bearing in a helicopter transmission.

The program as initially structured consisted of six tasks starting with the rig design, proceeding through rig fabrication, installation and checkout, parametric testing of a selected spherical roller bearing with a geared outer ring and the comparison of the collected data with computer predictions generated under NASA Contract NAS3-22807. However, due to changing priorities and areas of concern with NASA, the program was terminated following the designing of the test rig.

Starting with the previously developed test rig concept, a design with a complete set of detail and assembly drawings was formalized. The test rig configuration is an extension of the widely accepted four-square gearbox arrangement used

extensively for the testing of gears. The rig design consists of separating each set of meshing gears in the four-square arrangement and inserting a geared outer ring test bearing between each set as idlers, thus reconstructing the four-square arrangement. This configuration simulates all of the primary planet bearing conditicns except the centrifugal effects of the carrier rotation.

The test rig design was the joint effort of SKF Industries and Philadelphia Gear Corporation. SKF held the project responsibility for the overall design activity including the establishment of the basic systems configuration and the interfacing of the various components, and the detail design of the test bearing modules. PGC conducted the design of the two gear boxes, drive gear configurations and the associated components.

The rig was designed specifically for testing a planet bearing used in the OH-58 helicopter transmission, SKF Bearing Number 462721, with a 31.75 mm (1.25 in.) bore. However, with modification to the rig gears and the stub shafts, a test bearing with a bore up to 50 mm can be incorporated. The rig was designed to run over a test bearing outer ring speed range from 0.1 to 0.6 million DN, 330 to 1980 rad/sec., (3,150 to 18,900 rpm) under loads producing

C/P values down to 3 and bearing misalignment up to 0.048 radians (2.7°) with under race lubrication applied to the test bearing.

Two copies of the test rig drawings were sent to the NASA project leader while the original drawings are stored at SKF Industries and the Philadelphia Gear Company.

2.0 INTRODUCTION

Standard and specially designed spherical roller bearings have long been used in a large number of industrial applications and a significant amount of knowledge is available about the performance characteristics of these bearings. As a result, the design philosophies and manufacturing requirements for these applications are well known and sophisticated computerized analysis tools exist to predict bearing performance in these applications.

Spherical roller bearings are now gaining widespread use as the planet gear support bearing in planetary transmissions for a number of aerospace applications such as helicopters. In these applications, the bearings are subjected to operating conditions which are significantly different from those experienced in standard industrial environments where spherical roller bearings have been applied so successfully in the past. These specialized conditions include outer ring rotation, the application of loads through the gear teeth integral with the bearing outer ring, centrifugal forces generated by the rotation of the gear carrier, substantial misalignment, the uses of synthetic lubricants, higher speeds, and higher temperature. The relatively large number of these planetary systems and the critical nature of these components in aircraft systems establish the need to obtain realistic information regarding the performance of spherical roller bearings under these unique conditions.

Specific efforts to eliminate those deficiencies in knowledge were initiated in 1978 under U.S. Government Contract NAS3-20824, directed by the NASA Lewis Research Center. The objective of this program was to create a specific computer code for the evaluation of spherical roller bearings running with inner ring rotation over a broad range of operating conditions. The referenced program also included an experimental portion where parametric tests were conducted on a specially instrumented high speed spherical roller bearing test rig to define the performance characteristics of the bearing design under a variety of speed, load, lubrication and thermal conditions for comparison with the analysis and definition of factors contained in the computer code. [1]*

The activity was continued in 1981 under Contract NAS3-22807. The objective of this latter program was to modify the spherical bearing analysis program to consider outer ring rotation and include the effects of significant amounts of misalignment. A number of parametric studies were then conducted using the expanded code to predict the operating characteristics of a specific bearing design under

* Numbers in brackets denote references at end of report.

planetary system operating conditions. However, the referenced project did not include experimental verification which is required to insure the accuracy of the computed predictions.

In anticipation of this need, the program did include a task to establish a conceptual design for a test rig which would simulate the primary planet bearing environment minus the centrifugal effects of the carrier rotation. [2]

In 1983 a third program initialed by the NASA Lewis Research Center was structured to formalize the rig design,to manufacture and check out the test rig, and to perform parametric testing to obtain experimental data. However, due to changing priorities and areas of concern within NASA only the design of the test rig was completed. The test rig design is the subject of this report.

3.0 TEST RIG DESIGN

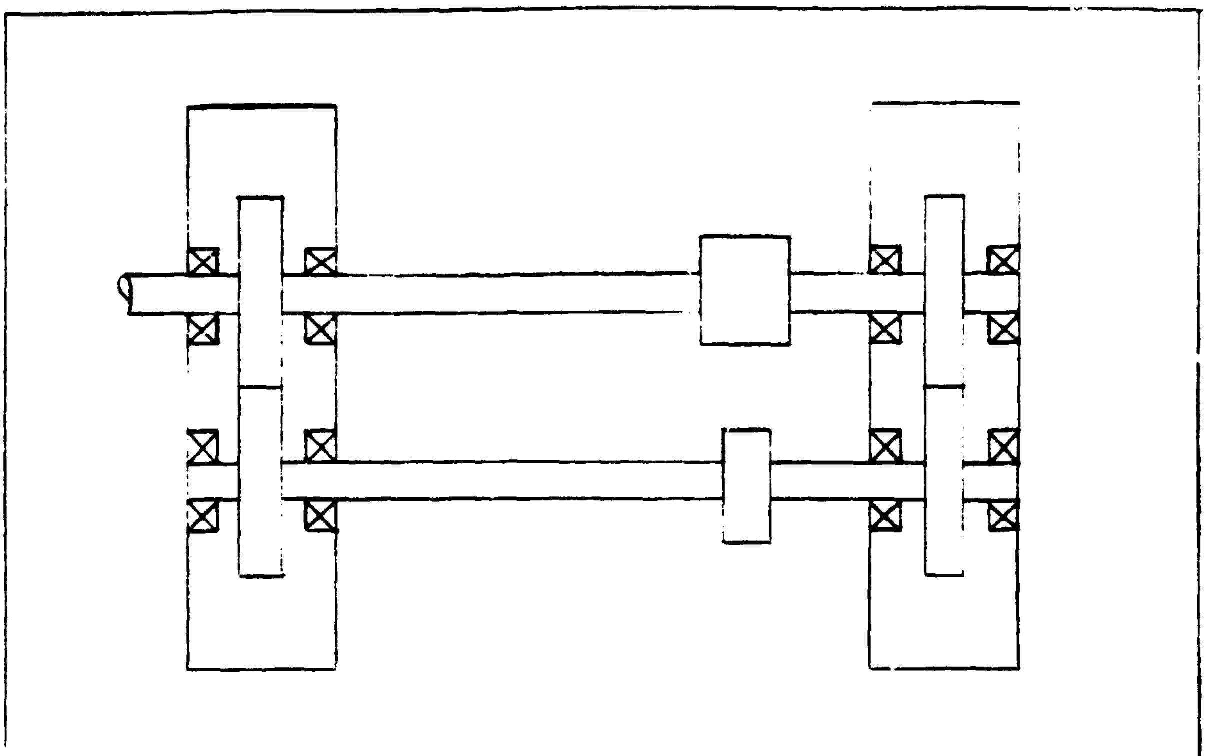
The test rig design is the result of the joint effort of SKF Industries and the Philadelphia Gear Company (PGC). SKF directed the overall design activity, specified the general configuration and interface condition, and designed the two test bearing modules. Philadelphia Gear conducted the design of the two gearboxes and analyzed the overall system dynamics.

3.1 Test Rig Configuration

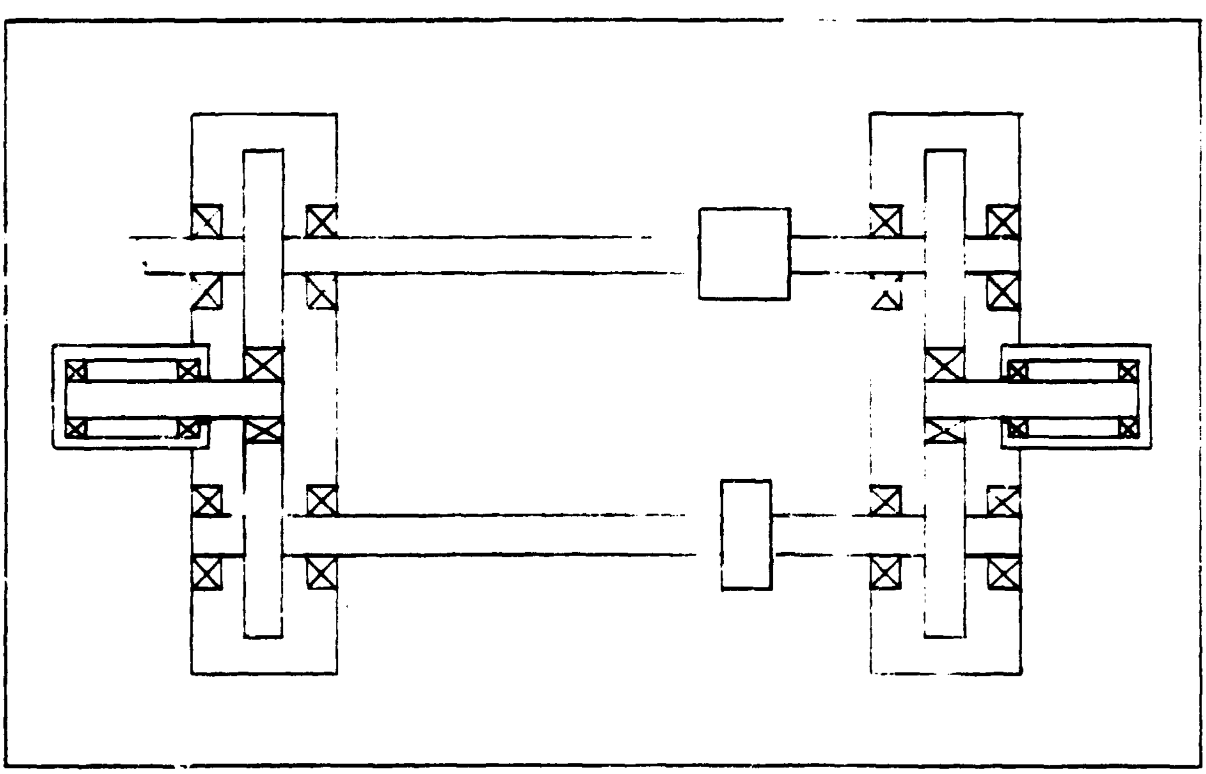
The test rig design was based on a previously conceived configuration which permitted the simulation of the major conditions imposed on a planet gear support bearing in a helicopter transmission. These conditions include outer ring rotation, loading produced from the torque applied to the flexible geared outer ring, misalignment, and under-race lubrication. It does not simulate the centrifugal loading created by the rotation of the bearing carrier.

The conceived configuration is a modification of the widely accepted four-square gearbox arrangement used extensively for the testing of gears and gear lubrication. Each set of primary drive gears in the four-square design is separated by the insertion of a geared outer ring spherical roller bearing, mounted on a stub shaft with its support bearings and housings as an idler. Thus, the four-square arrangement is maintained but now incorporates two test bearings with outer ring rotation, see Figure 1.

By using a special shaft coupling design, the primary test load (torque) can be locked into the closed loop. This torque produces the planet bearing load through the geared outer ring. This arrangement results in a large reduction in rig power consumption since the drive motor only needs to supply sufficient power to compensate for losses in the gears, test bearings and gear support bearings. The test bearing misalignment can be achieved by tilting the stub shaft, on which the inner ring of the test bearing is mounted, a controlled amount and an axial hole in the stub shaft can be used to supply under race lubrication. Thus all the major conditions can be applied to the test bearings using the described configuration.



STANDARD FOUR-SQUARE GEAR TEST CONFIGURATION



SCHEMATIC OF R . CONCEPT

FIGURE 1

3.2 Specified Operating Conditions and Measured Properties

The contract specified that the test rig would be designed to test spherical roller bearings with bore diameters ranging from 30 mm to 50 mm. In the associated analytical program, previously performed, a 31.75 mm bore, double row, spherical roller bearing was selected for analysis and a detailed analytical study performed.[2] Since it was the ultimate goal of this program to obtain test results for comparison with analytical results, the same bearing was selected and the rig design tailored to its configuration. However, the gearbox was sized to accommodate larger test bearings by varying the pitch diameter of the rig gears.

The bearing selected, SKF No. 462721 was a logical choice since it is presently in production and has been used for several years in the transmission of the Bell OH-58 helicopter.

The work statement specified that the rig be designed to operate in a speed range from 0.1 to 0.6 million DN and with loads varying from C/P values of 20 to 3. Where D is the bearing bore diameter in mm; N, the angular velocity in rpm; C, the dynamic capacity of the bearing; and P, the equivalent radial load applied.

The work statement also specified that the rig be capable of applying a test bearing misalignment from 0^{o} to a maximum value as limited by the bearing design.

Having selected the specific test bearing, these general expressions for speed, load and misalignment were converted into specific values. The test bearing outer ring speeds in rpm were calculated directly using the equation N = DN/D and are listed in Table 1. Also listed is the rig gear angular velocity established after its pitch diameter had been determined.

The loading experienced by a planetary bearing is illustrated in Figure 2. The total contact forces F, develop at diametrically opposed points on the gear pitch circle, due to the planet-sun and planet-ring meshes. The contact forces can be resolved into components at the neutral axis of the outer ring, as shown, resulting in a radial force R, a tangental force T, and a moment M due to the offset of the tooth contact point from the neutral axis.

The parameter C/P is used to calculate roller bearing life according to AFBMA standards using the equation $L_{10} = 16667/N\,(C/P)^{10/3}$ where L_{10} equals the bearing theoretical L_{10} life in hours.

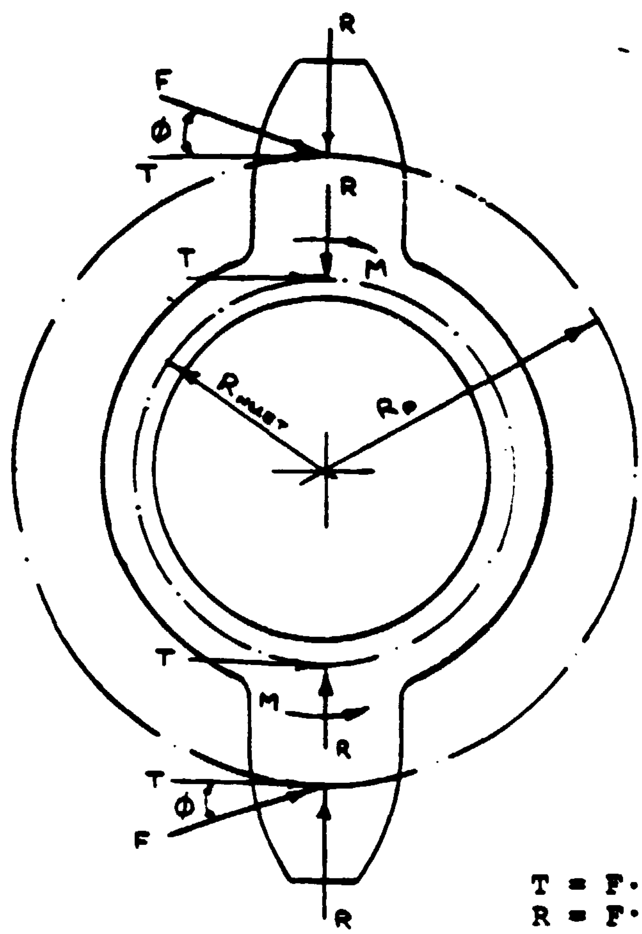
There is no straightforward way to consider the effects of R and M in determining the value of the equivalent applied load P. However, the presence of these loads obviously affect the roller load distribution and, therefore, the life of the bearing.

TABLE 1

SPECIFIED SPEEDS

TEST BEARING DN Value ($DN \times 10^{-6}$)	TEST BEARING Outer Ring Speed (rad/sec)	(rpm)	RIG GEAR Speed rad/sec	(rpm)
0.1	330	3,150	98	934
0.2	660	6,300	196	1,869
0.3	990	9,450	294	2,803
0.4	1,320	12,600	391	3,738
0.5	1,649	15,750	489	4,672
0.6	1,980	18,900	587	5,607

FIGURE 2 **PLANETARY BEARING LOADING**



$T = F \cdot \cos\phi$
$R = F \cdot \sin\phi$
$M = T(R_p - R_{NUET})$

To more accurately determine the load corresponding to the desired C/P values, an iterative procedure was employed using the SKF SPHERBEAN computer program. First, a value was selected for the tooth contact force F. The components R, T, and M were then calculated based on geometry. The SPHERBEAN program was executed to obtain the bearing L_{10} life. Using the life equation, the corresponding value of C/P was computed. The procedure was repeated, adjusting the value of F until the desired values of C/P were obtained.

The loads calculated using the above iteration procedure corresponding to the C/P specified values are summarized in Table 2. Also presented in the table are the corresponding torque values that must be locked into the closed loop to produce the desired loads. The horsepower value listed represents the driving power that would be needed when operating at the maximum speed if an open loop loading system was used.

The maximum permissible misalignment angle for the selected test bearing was calculated to be 0.048 rad (2.76°). This angle corresponds to the rotation of the inner ring relative to the outer ring that would cause the edge of the roller to touch the edge of the outer raceway surface.

The maximum values thus calculated for load, speed, and misalignment were utilized in establishing the rig design.

TABLE 2

SPECIFIED LOADS

TABLE 2

SPECIFIED LOADS

C/P	TANGENTIAL LOAD (N)	TANGENTIAL LOAD (LBS)	RADIAL LOAD (N)	RADIAL LOAD (LBS)	MOMENT LOAD (N-M)	MOMENT LOAD (IN-LBS)	TOOTH LOAD (N)	TOOTH LOAD (LBS)	TEST GEAR TORQUE (N-M)	TEST GEAR TORQUE (IN-LBS)	RIG GEAR TORQUE (N-M)	RIG GEAR TORQUE (IN-LBS)	POWER AT MAX. SPEED (KW)	POWER AT MAX. SPEED (HP)
3	11,027	2,479	5,142	1,156	71	624	12,165	2,735	548	4,849	1,837	16,260	1,078	1,445
5	6,392	1,437	2,980	670	41	362	7,050	1,585	318	2,811	1,065	9,425	624	837
6	5,151	1,158	2,403	540	33	292	5,685	1,278	256	2,265	858	7,595	503	675
8	3,608	811	1,681	378	23	204	3,981	895	179	1,586	601	5,319	253	473
9	3,109	699	1,450	326	20	176	3,430	771	154	1,367	518	4,584	304	407
10	2,718	611	1,268	285	17	154	2,998	674	125	1,195	453	4,007	265	356
20	1,076	242	503	113	7	61	1,188	267	53	473	179	1,587	105	141

In order to establish the operating performance characteristic of the test bearing under the various conditions, the following measurements are required:

1. Lubricant inlet temperature.
2. Lubricant flow rate to test bearing.
3. Inner ring temperature.
4. Outer ring temperature.
5. Outer ring speed.
6. Cage speed.
7. Applied load.
8. Bearing friction torque.

The rig was designed to permit the sensing and monitoring of these properties during operation.

3.3 Gearbox Design

The two gearbox assemblies are essentially the same. Each consists of a housing which is split in a horizontal plane at the center line of the connecting shafts; two shaft, gear and bearing sub-assemblies; and either labyrinth seals or blind caps to cover the shaft support bearings, see Figures 3 and 4.

The gears are press fit on the shaft prior to grinding to insure the desired concentricity between the two. Each gear is also keyed to its shaft to prevent relative motion under the heavy loads. Double slots and keys, 180° apart, are used to minimize the dynamic unbalance correction which would be required. The gears are located in the same ver-

FIGURE 3

GEARBOX ASSEMBLY

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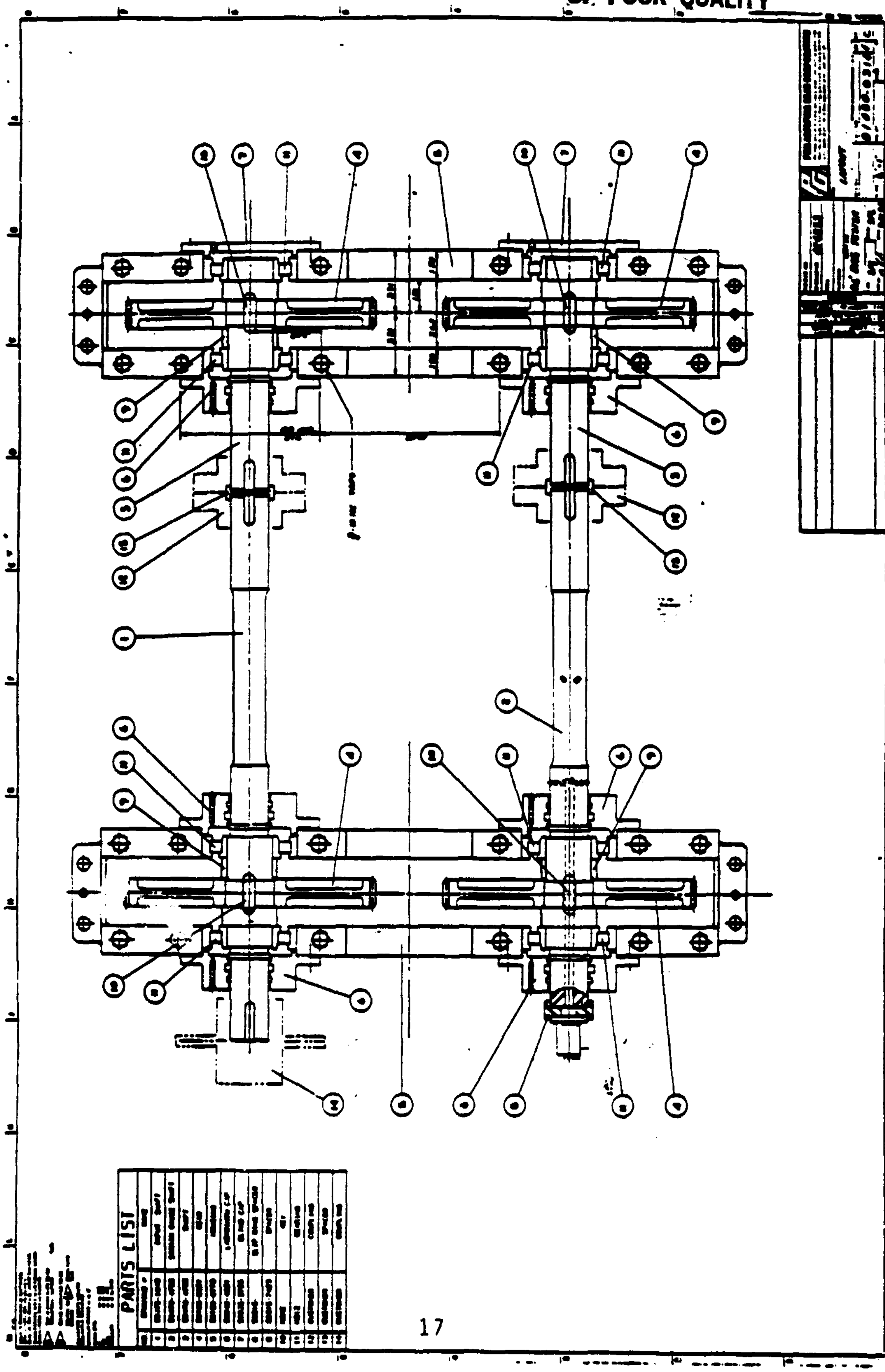
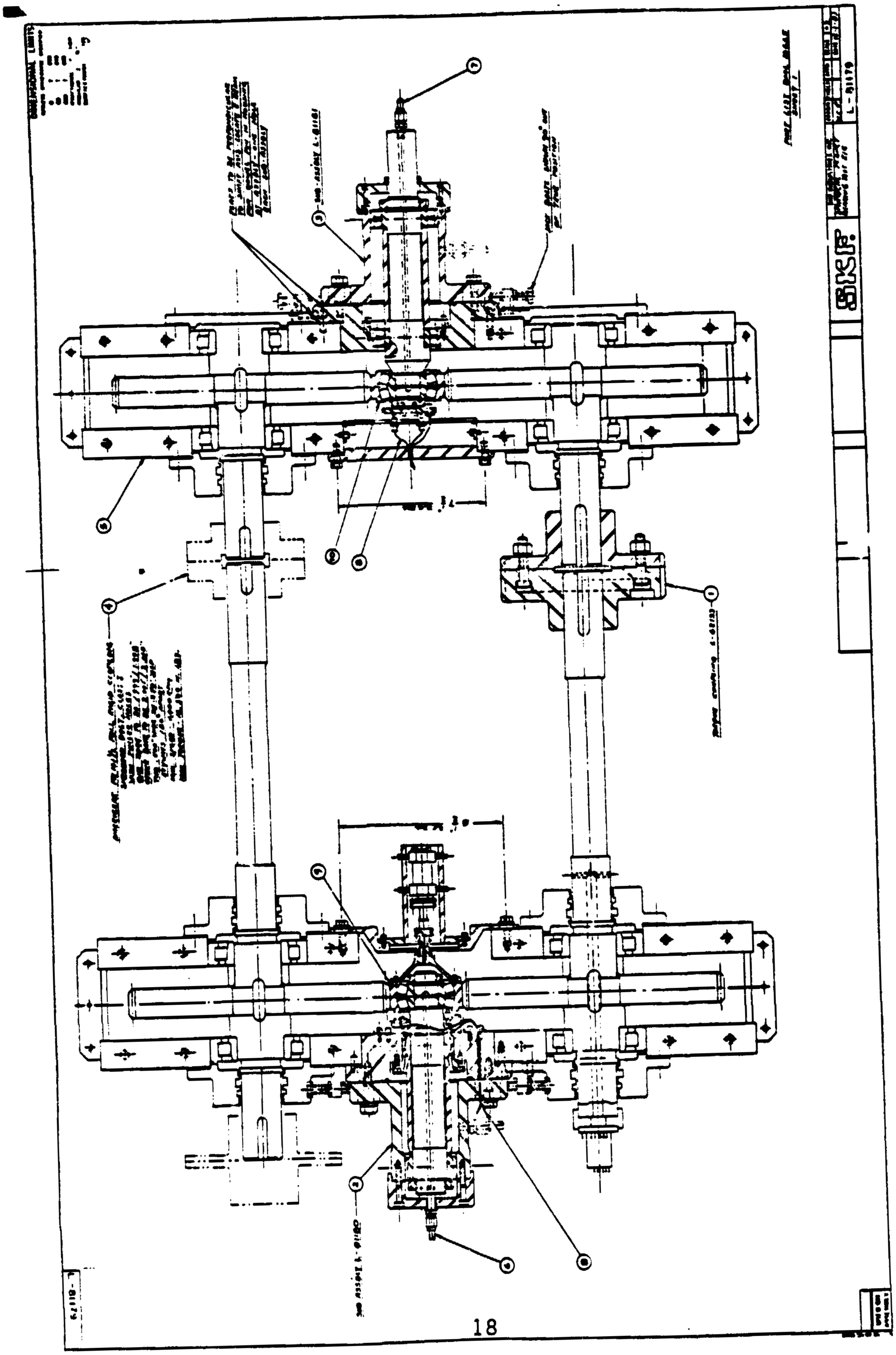


FIGURE 4 - TEST RIG LAYOUT DRAWING



tical plane at a specified center distance to allow the proper insertion of the test bearing between them. Special openings and mounting holes are machined into the housing to accept the stub shaft modules.

Each box has a full length detachable cover; two flanged lubricant drain pipes, one on each end; and a regenerative air breather. The cover provides for maximum access to permit gear inspections and adjustments to the gear lubrication tubes. Each seal or cap is rabbeted into the housing and contains holes to direct lubricant into the gear support bearings. The caps and seals, which retain the outer rings of the gear-shaft support bearings, are axially positioned relative to the bearings to accommodate thermal growth of the shafts during operation and permit separation of the shaft couplings during assembly, disassembly, and load application.

The connecting shafts on one side are attached by a standard coupling, Amerigear FR101 1/2. On the other side they are connected by a special-design torque adjusting coupler. As illustrated in Figure 5, one side of the torque coupling is held with a spanner wrench while the torque, which will produce the desired planet gear loading, is applied to the other side. The torque is locked in place by bolting the

FIGURE 5

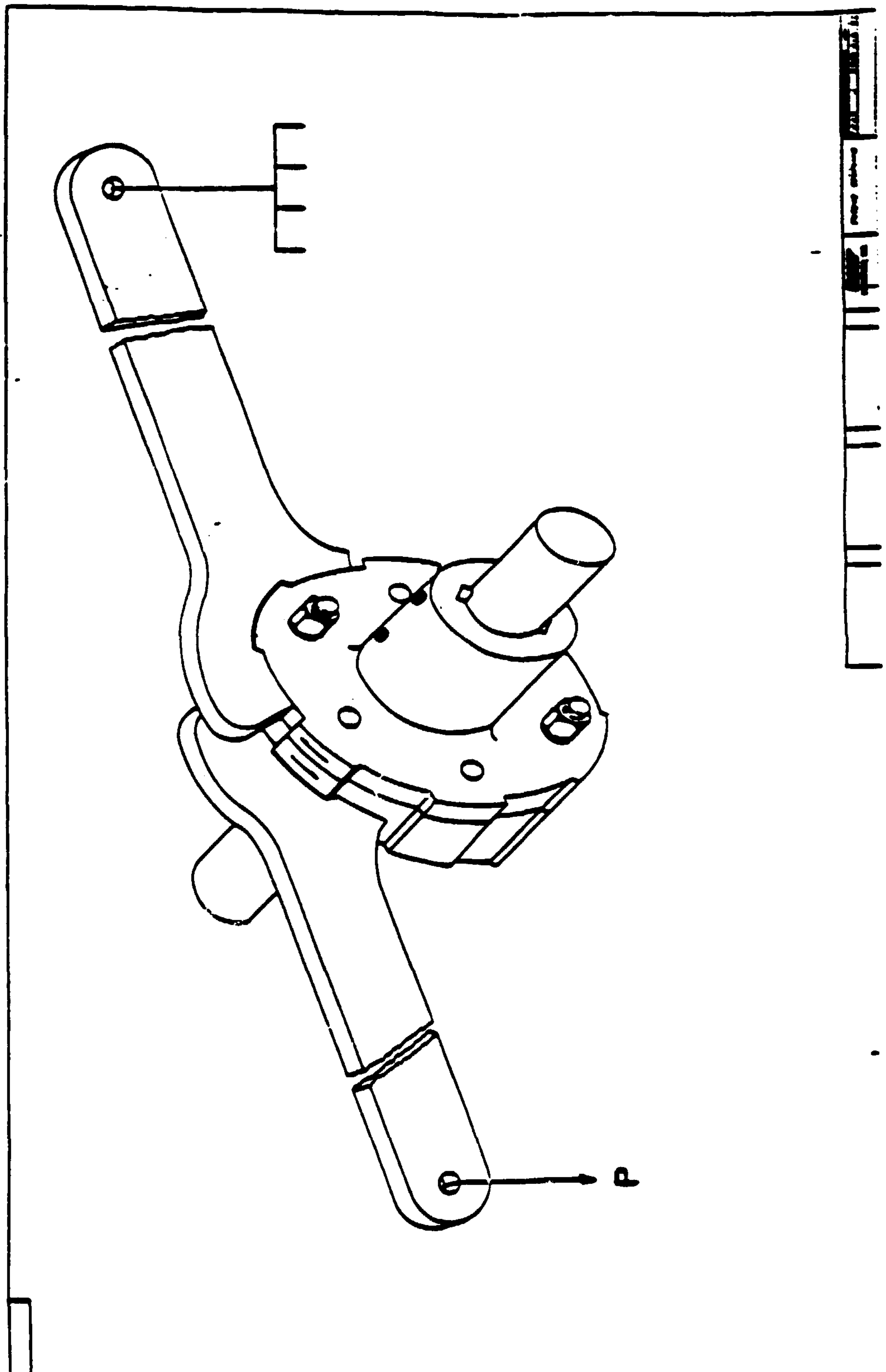


ILLUSTRATION OF TORQUE COUPLING

coupling halves together. T-headed bolts are used to prevent load slippage. Strain gages, attached to one shaft, with wire running down the center of the shaft to a slip ring assembly are used to sense the applied torque level.

Based on the test bearing gear geometry and the load and speed specifications, the rig gears were designed to have the following values:

Number of Teeth	118
Operating Pitch Diameter	336.93 mm (13.265")
Outside Diameter	343.33 mm (13.517")
Root Diameter	330.35 mm (13.006")
Addendum	5.16 mm (0.203")
Base Diameter	307.67 mm (12.113")
Face Width	31.75 mm (1.250")

A report prepared by PGC on the gearbox analysis is presented in Appendix 1. The report incluues the gear analysis, gear mesh efficiency and power loss, gearbox bearing analysis, and the torsional and lateral natural frequency analyses of the system including the drive motor.

The gear analysis using a velocity factor of 0.900 and a face distribution factor of 1.089, which reflects high quality gearing, shows that the contact stress and bending stress resulting from the maximum load condition are less than the allowable. The Dudley scoring index and the flash temperature calculations, assuming a 0.2 μm rms surface finish, indicate that tooth face scoring would not be expected.

Gear mesh power loss was calculated using both the EHD method and the empirical formula from Dudley's Gear Handbook. The power losses determined by the two methods were 9090 and 12,692 watts (12.19 and 17.02 horsepower) per mesh respectively. The calculated efficiencies were 0.9918 and 0.9884.

The gear shaft support bearings were selected by SKFI based on the loading that would result if a 50 mm bore test bearing was incorporated in the rig. Under these conditions the load bearings selected, a N312 cylindrical roller bearing, have a theoretical L_{10} life of 6000 hours. A theoretical L_{10} life of 37,000 hours results when the maximum loads are applied to the 31.75 mm bore test bearing.

The natural frequency analyses show that only one torsional and no lateral natural frequencies fall within the operating range. Since no torsional excitation forces are expected to exist in the closed loop, no vibration problems are anticipated.

3.4 Test Bearing Module Design

The test bearing modules are generally the same but differ in detail since some different performance properties are measured on each. The right hand module is instrumented to measure the bearing drag torque and the left hand module is instrumented to measure the rotating outer ring temperature.

Both modules are additionally instrumented to measure lubricant inlet temperature, inner ring temperature, cage speed, and misalignment.

Both Modules

The major consideration in designing the modules was to incorporate a configuration which would provide for an adjustable misalignment. The resulting configurations of the right and left modules are shown in Figure 6 and 7 respectively. The layout drawing of the combined modules and gearboxes is shown in Figure 4.

In order to produce a controlled misalignment, a stub shaft, with the test bearing mounted on the end, is supported by two rig bearings. Each rig bearing is mounted in separate yet relatively positioned housings. The two housings are doweled together during machining so the concentrically ground bearing seat arrangement can be restored by reinserting the dowel pin. The rear housing is attached to the

FIGURE 6 - RIGHT TEST BEARING MODULE

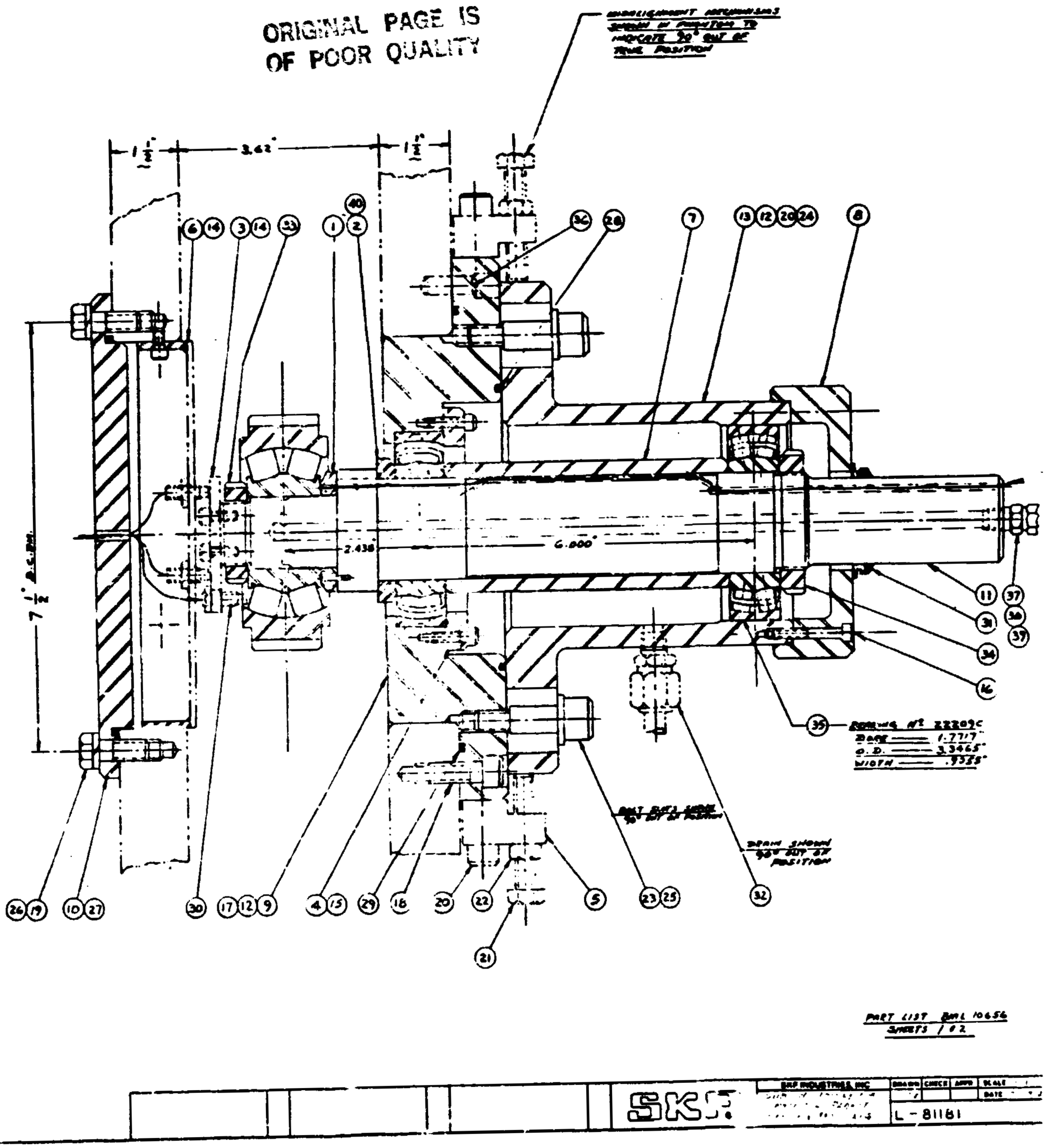


FIGURE 7 - LEFT TEST BEARING MODULE

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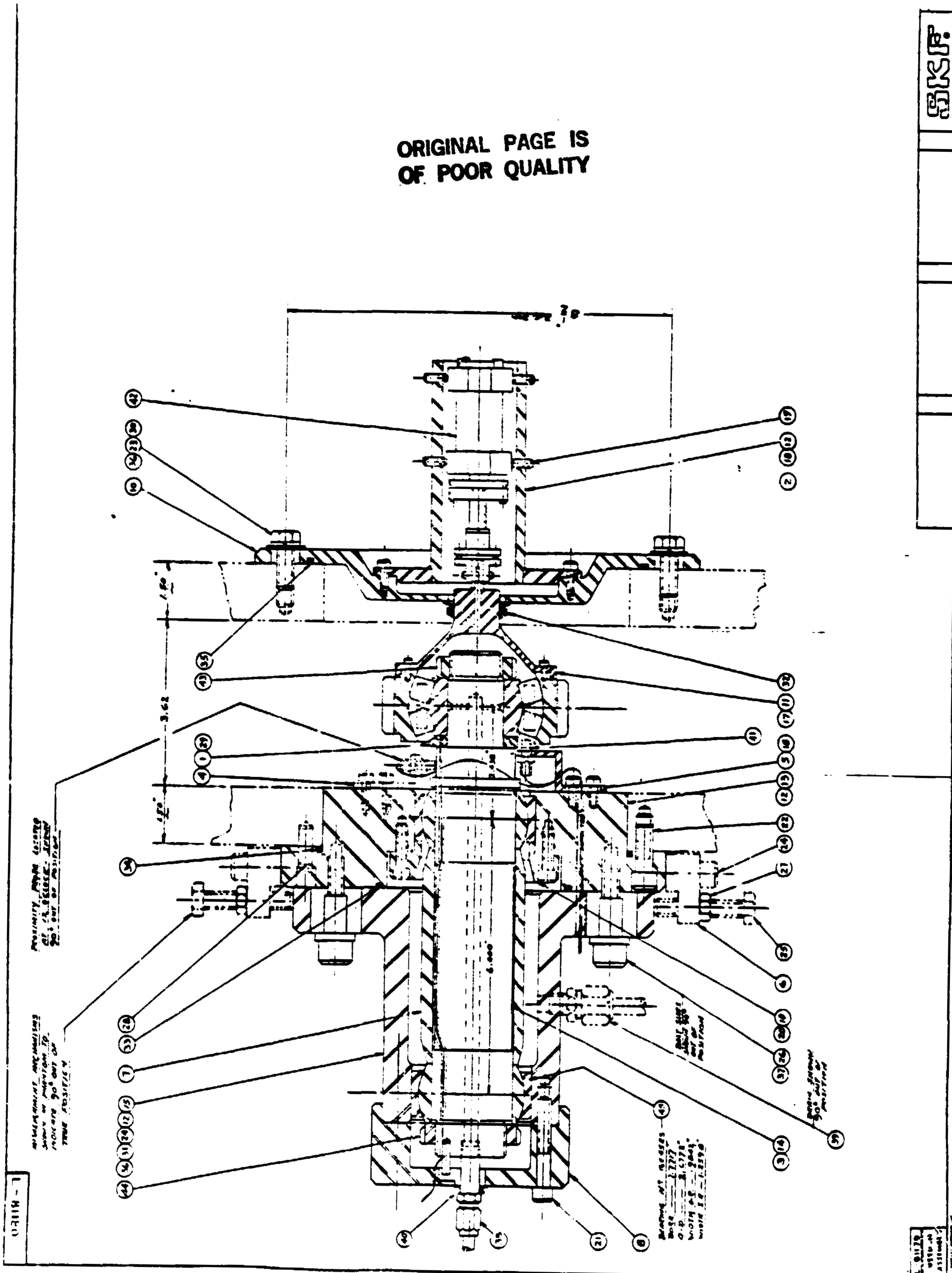

front housing by six shoulder bolts which extend through vertical slots. This slot and shoulder bolt combination permits only pure vertical motion between the two housings. This relative motion is produced by loosening the shouluer bolts and turning the adjusting screws located at both the top and bottom of the sub-assembly.

The front support bearing position is fixed relative to its housing and shaft and functions as the pivot when ti'ting the stub shaft. By correctly fitting the stub-shaft module into the gearbox, the proper gear mesh and perfect alignment of the test bearing is obtained.

By removing the dowel pin and producing a vertical displacement between the two housings, the stub shaft is tilted in a vertical plane. This motion causes a misalignment of the inner ring of the test bearing, which follows the tilt of the shaft, relative to the outer ring. The outer ring will move vertically and axially with the inner ring but its center line remains in a horizontal plane due to the restraint applied by the mating gear teeth.

The measurement of the relative displacement of the two housings from the pinned position can be converted into the angular misalignment of the test bearing. When this misalignment is being set, one of the inner loop couplings must be loose as the vertical motion of the outer ring causes a windup of the closed gear-train loop.

The tilting of the stub shaft produces a misalignment not only in the test bearing but also in the two support bearings, thus the need for self-aligning spherical bearings in these locations. The tilting also produces an axial motion of the rear support bearing; therefore, the outer ring of this bearing must have a loose fit in the housing.

An axial hole down the center of the stub shafts intersects with radial holes underneath the test bearing to permit under-race lubrication. The lubricant temperature is measured just prior to entering the shaft using J-type thermocouples. The two support bearings are grease lubricated to supplement the oil mist which is generated inside the gearbox. O-rings are used at the interfaces between housings, between the front housing and gearbox, and between the cover plate and gear box to prevent oil leakage. An oil drain is located in the rear housing to minimize oil build-up.

A J-type thermocouple, with the connecting wires extending down a groove in the stub shaft is used to measure the inner ring temperature.

Proximity probes are used to measure cage and outer ring speeds, and to monitor the misalignment during operation.

Right Module

The test bearing drag torque is measured in only the right module. To accommodate this measurement, spherical roller bearings are used to support the stub shaft which extends through a seal in the end cap. The roller bearings are used to minimize the resistance of the shaft to rotation due to friction in the test bearing. A strain gaged beam attached to the extended end of the stub shaft provides a good measurement of the test bearing drag torque.

In this module the proximity probe used to measure the cage speed is mounted on the shaft end plate and extends into the bearing where it senses the roller pass frequency. This value is readily converted into cage rotational speed. Since the probe moves with the shaft, the misalignment of the bearing has no effect on the position of the probe tip relative to the rollers.

Two proximity probes are used to monitor the misalignment of the bearing during operation. They are mounted on a stationary plate attached to the gearbox and sense the shaft end plate. The probes are located in a vertical plane equi-distant from the end plate and are equally spaced above and below the center line of the stub shaft when it is not tilted. Thus, they can be used to directly measure the misalignment adjustment or sense any change that may occur during operation.

Left Module

The test bearing outer ring temperature is measured in the left module via the use of a slip ring assembly. The slip ring is located in line with the stub shaft but on the opposite side of the gearbox. It is attached to an adjustable face plate which can be moved vertically to insure alignment with the outer ring under all test bearing positions. Whenever the test bearing alignment is changed the slip ring position must also be changed since an extension of the outer ring is attached to and drives the slip ring shaft through a flexible coupling.

The extension is cone shaped with all but two strips cut out of the side. The cutout area allows the lubricating oil to escape from the outer race. The strips, mechanically attached to the face of the outer ring, carry the T/C wires down to the apex where they pass through the coupling to the slip ring.

The proximity probes used to sense alignment and shaft speed are mounted on the support bearing housing. The alignment sensing probe is located directly above the shaft, twelve o'clock position, so it will detect position changes in the vertical direction. The cage speed sensor is located at the three o'clock position to minimize changes in its relative position to the rollers due to different degrees of misalignment.

Since the test bearing drag torque is not measured on the left module, the stub shaft is supported on plain spherical bearings which permit the required misalignment but have a greater load capacity. The stub shaft is kept from rotating by a restraining pin located in the end of the shaft and extending through a hole in the housing end cover.

Power loss calculation made for the maximum speed, load and misalignemnt condition show the following expected power losses.

		Watts	hp
Gear Support Bearing	- 477 watts x 8	3,818	5
Gear Machine	- 12677 watts x 4	50,708	68
Test Bearing	- 8202 x 2	16,404	22
	Total	70,930	95

Therefore, a drive motor with no less than 100 hp would be needed to drive the test rig.

4.0 RECOMMENDATIONS

As stated in the Introduction section, computer code SPHERBEAN was written under Contract NAS3-20824 to analyze the performance characteristics of spherical roller bearings. Under Contract NAS3-22807, the code was modified to incude analysis of a planet bearing operating in a helicopter transmission and specifically included operation with bearing misalignment and outer ring rotation. The code was then used to evaluate the performance of a specific bearing under a given set of conditions. However, no experimental data from misaligned and outer ring rotating bearings currently exists that can be used to compare with the predictions of the code; thus modifications to the code have not yet been verified.

Before the government can incorporate the modified code into its arsenal of analytical tools for evaluating and selecting bearings for aerospace applications, the predictions of the computer code must be verified. The design of the test rig here reported is the first step in obtaining the required experimental verification.

It is, therefore, recommended that the designed test rig be manufactured and experimental data accumulated as soon as practically feasible.

5.0 REFERENCES

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AT85T004

APPENDIX I

PHILADELPHIA GEAR CORPORATION

ENGINEERING CALCULATION REPORT

PHILADELPHIA GEAR CORPORATION

ENGINEERING CALCULATION REPORT

Unit Designation SKF TEST GEARBOX

P.G.C. Order Number 424833

Customer SKF

Prepared By K. ATWATER Date 1/12/84

Approved By E Nglinger Date 1/16/84

PHILADELPHIA GEAR CORPORATION ☐ KING OF PRUSSIA, PA 19406 ☐ (215) 265-3000 ☐ TELEX: 846321

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INTRODUCTION

This report contains the results of a comprehensive analysis of the SKF Test Gearbox, including gearing, gear horsepower loss, bearing analysis, torsional natural frequency analysis, lateral natural frequency analysis, and mass elastic data. The analyses were done at the maximum loading condition established by SKF.

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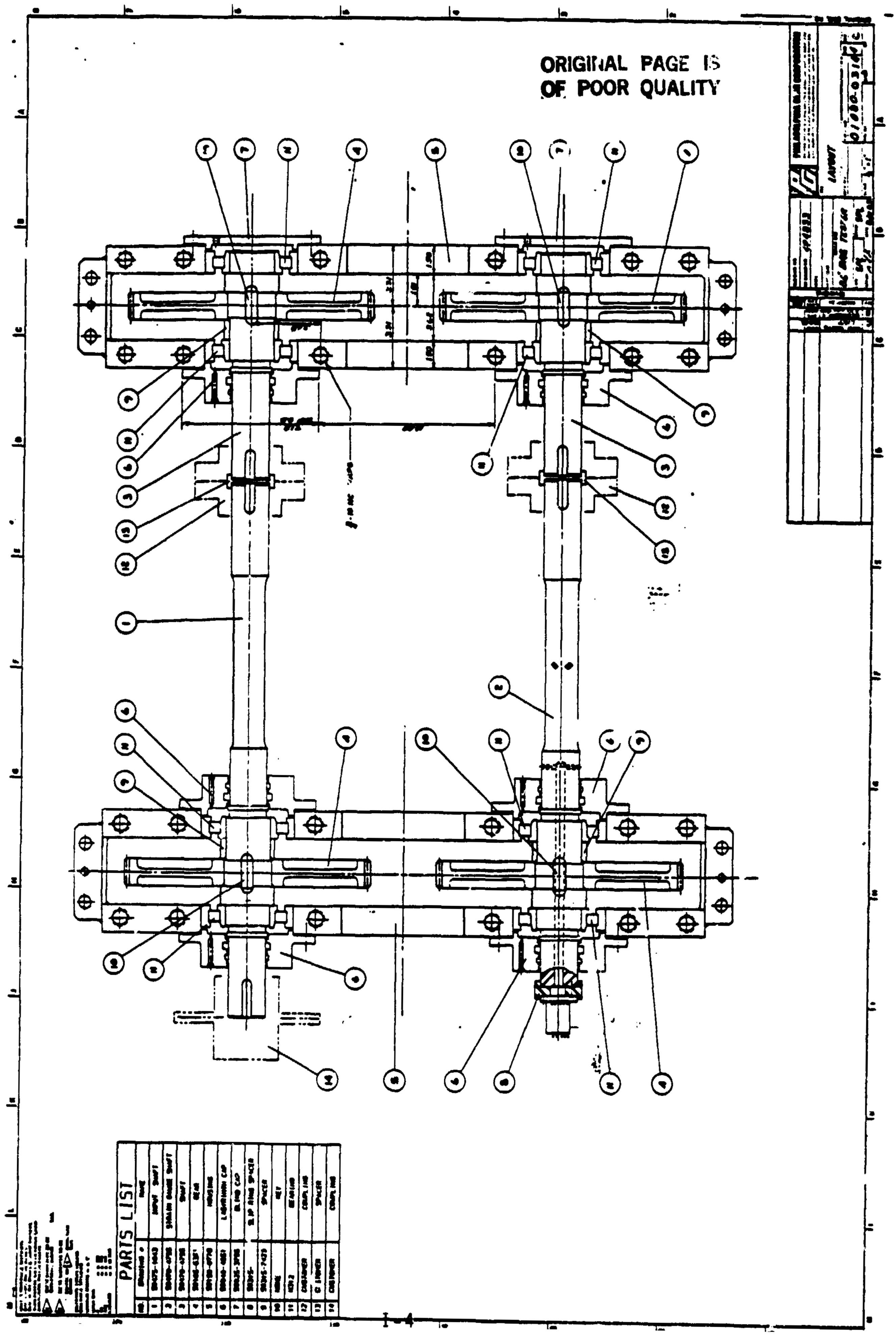

SKF TEST GEARBOX
P.G.C. ORDER NO. 424833

GEAR DESIGN

GEARING

The gearing designed for the SKF Test Stand is rated by AGMA Standard 218.01 at maximum torque and speed and 100 hours design life. A gear analysis summary, produced by the P.G.C. GEAR Program, may be found on the next page.

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*** PHILADELPHIA GEAR CORPORATION ********************************************** PHILADELPHIA GEAR CORPORATION ***
*                                      G E A R   A N A L Y S I S                                                   *
**************************************************************** TIME     9: 6:40 DATE   01/13/84* PAGE   1  *
*  424833    SKF TEST STAND                                    *   COMPUTER:PRIME 850
**************************************************************** ANALYST-K.ATWATER      APPROVED-............
GEARSET USED IN NO. 1 REDUCTION
--------------------------------------------------------------------------------------------------------------
                            PINION --- GEAR                                QUALITY NUMBER 12
NUMBER OF TEETH ....         35        118      CENTER DISTANCE....     8.600            OVERSIZE FACTOR ....   0.000
OPER.PITCH DIAMETER.      3.935     13.265      FACE WIDTH ........     1.250            ADDENDUM FACTOR ....   1.000
OUTSIDE DIA ........      4.134     13.517      GEAR RATIO ........     3.371 TO 1       WHOLE DEPTH FACTOR .   2.300
ROOT DIA ...........      3.623     13.006      DIAMETRAL PITCH-NOR     9.000            ACTION LINE LENGTH .   0.516
ADDENDUM ...........      0.122      0.203      DIAMETRAL PITCH-LN      9.000            CONTACT RATIO ...... 1.5999
OVERSIZE ...........      0.023      0.104      PRESSURE ANGLE-NOR.    22.500            OVERLAP RATIO ...... 0.0000
BASE DIA ...........      3.593     12.113      OPER.PRESS.ANG-LIN.    24.058            SPREAD FACTOR ...... 0.1100
LIMIT DIA ..........      3.733     13.092      HELIX ANGLE .......     0.000            I*((MG+1)/MG) FACT .   0.000
NCT AT PITCH DIA ...      0.184      0.251      HELIX ANGLE .......    0  0  0           I-FACTOR ...........   0.130
NCT AT OUTSIDE DIA .      0.071      0.071      OPER.PRESS.ANG NOR.    24.058            LOAD SHARING RATIO .   1.000
STRESS CONCT. FACTOR      1.979      2.084      NORM.BASE PITCH ...     0.322            F/D RATIO ..........   0.321
J-FACTOR ...........      0.492      0.577      CD-DEVIATION ......     0.100            FULL FILLET
NUMBER OF CYCLES ... 0.116+09   0.34E+08
--------------------------------------------------------------------------------------------------------------
18900.00 INPUT RPM       5605.93 OUTPUT RPM       19468.6 FT/MIN PITCH LINE VELOCITY      DESIGN HOURS =     100.
                                                  ASSUMED GENERAL COMMERCIAL APPLICATION IN LIFE FACTOR CALCULATION
--------------------------------------------------------------------------------------------------------------
CMC= 0.80000         CPF= 0.02812         CPM= 1.00000         CMA= 0.08336         CE= 1.00000
                                                               CMA CURVE NO. 3  (PRECISION ENCLOSED GEAR UNITS)
--------------------------------------------------------------------------------------------------------------
RATING PER AGMA 218.01                                               PINION CARBURIZED     GEAR CARBURIZED
     DURABILITY
          0.900 VELOCITY FACTOR           1.009 LOAD DISTRIBUTION FACTOR             C1=  2.00999     C2=    1.148

         2713.5 HP RATING                 225000. PSI ALLOWABLE CONTACT STRESS       C3=  1243.92     C4=    0.909
CA= 1.000  CR= 1.000  CI= 1.000  C = 1.000  CS(MIN)= 1.000      CP= 2300.0    CL(MIN) =     0.954     CH=    1.000
--------------------------------------------------------------------------------------------------------------
RATING PER AGMA 218.01
     STRENGTH                                                      KA= 1.000  KR=    1.000        KT= 1.000
          0.900 VELOCITY FACTOR           1.009 FACE DISTRIBUTION FACTOR             K1=  0.53118     K2=    1.148

                       HP RATING   ALLOWABLE BENDING STRESS                      K3          KL          KS
        PINION          1601.5          49000.                                40020.       0.980       1.000
        GEAR            2743.7          70000.                                70202.       1.003       1.000
--------------------------------------------------------------------------------------------------------------
TRANSMITTED HORSEPOWER =  1466.42     ACTUAL CONTACT STRESS  = 157739. (S.F.= 1.850 )   K-FACTOR        =     655.3
TORQUE (LB-IN)  PINION =     4890.                                                      DUDLEY SCORING-
                  GEAR  =    16486.     ACTUAL BENDING STRESS                                       INDEX =     23635.
TANGENTIAL FORCE P (LB)=  2405.6                    PINION   =  43970. (S.F.= 1.092 )   PVT AT PINION  =  1119092.
SEPARATING FORCE S (LB)=  1029.6                    GEAR     =  37521. (S.F.= 1.871 )          GEAR    =  2937760.
THRUST     FORCE T (LB)=     0.0

     213.   DEG FLASH TEMPERATURE BASED ON 120. DEG BLANK TEMP AND GEAR SET WITH TIP RELIEF     (AGMA 217.01)
       8.   RMS SURFACE FINISH  (S.F.=    1.10 )

   GEAR SET EFFICIENCY= 98.839 PERCENT     HORSEPOWER LOSS=  17.02
--------------------------------------------------------------------------------------------------------------
```

SKF TEST GEARBOX
P.G.C. ORDER NO. 424833

GEAR HORSEPOWER LOSS

Description

GEAR HORSEPOWER LOSS

Horsepower loss in the gears has been analyzed by thermal-elastohydrodynamic methods, taking into account sliding action of the gear teeth and oil film thickness. Philadelphia Gear has developed a computer program based on this method, and the results support the horsepower loss as calculated by the P.G.C. GEAR Program.

Power loss as calculated by the EHD method is 12.19 horsepower, and by the GEAR Program, based on an empirical formula from Dudley's Gear Handbook, is 17.02 horsepower.

-3-

TA

ORDER NUMBER	NORD= 424833
NUMBER OF TEETH OF THE PINION	Z1= 35.
NUMBER OF TEETH OF THE GEAR	Z2=118.
OUTSIDE DIAMETERS	DK1= 4.134
	DK2= 13.517
OVERSIZE FACTORS	XP1= .1220
	XP2= .2030
FACE WIDTH	BF= 1.25
DIAMETRAL PITCH-NORMAL	D.P.N= 9.000
PRESSURE ANGLE-NORMAL	ALFAN=22.50
HELIX ANGLE	BETA0= .00
TEETH CROWNINGS	DELTA-MAX-1= .00090
	DELTA-MAX-2= .00090
TEETH CROWNINGS START AT	YCRO= .313
LUBRICANT VISCOSITY ON 100 DEG F	ETA100= .86000-006
210 DEG F	ETA210= .10000-006
INLET OIL TEMPERATURE	T0=120.00
BULK TEMPERATURE RISE ACROSS THE MESH	DT= 93.02
BLANK TEMPERATURE	TB=120.00
TRANSMITTED HORSEPOWER	POWER= 1466.22
INPUT RPM	RPM1= 18900.000
TANGENTIAL FORCE	FTAN= .24860+004

RESULTS

MINIMUM OIL FILM THICKNESS	HMIN= .00001267
EHD POWER LOSS	PLOSS= 12.19
EFFICIENCY	ETA= .9918
FRICTION FACTOR	FRF= .0083

SKF TEST GEARBOX
P.G.C. ORDER NO. 424833

BEARING ANALYSIS

BEARING ANALYSIS

In accordance with correspondence from SKF dated July 28, 1983, the test gearbox has been designed using N312 cylindrical roller bearings. The bearings have been analyzed using the maximum loading supplied by SKF, and the L_{10} rating has been determined to be 37000 hours.

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*** PHILADELPHIA GEAR CORPORATION ********************************************** PHILADELPHIA GEAR CORPORATION ***
*                                                                                                              *
***************************************************************** TIME 11: 8:45 DATE 01/13/84*      PAGE   1 *
*  424833    SKF TEST STAND                                        * COMPUTER: PRIME  850                     *
***************************************************************** ANALYST-DLLIU           APPROVED-...........
    BEARING SPAN=    4.820 INCH

    BRG A AT + SIDE OF SHAFT AND BRG B AT - SIDE OF SHAFT
    LOCATION IS MEASURED FROM CENTERLINE OF BRG.SPAN



BEARINGS        BRR                   ---------- PART NUMBERS.ETC ---------
    A         23300.                 NJ312(2.3622*5.1181*1.2205)
    B         23300.                 NJ312(2.3622*5.1181*1.2205)

GEAR MEMBERS    PITCH    PRESS    HELIX    HAND
                 DIA      ANG      ANG    OF CUT    LOCATION    TYPE
   FIRST       13.265    24.1     0.000              -0.015     SPUR

    *****FIRST GEAR MEMBER IS DRIVING *****

SHAFT RPM= 5605.9  LOAD CONDITION--FIRST GEAR MESH= 1466.4 HP (TORQ.=  16486.3)

   BEARING LOADS
                       -----A -----          -------B-------       THRUST
  ROTATION
CW (VIEW A TO B)         1352.5                1369.7
CCW (VIEW A TO B)        1352.5                1369.7

   BEARING LIFE
                    ------------- A -------------      ------------ --- B -------------
    ROTATION        LB-10 HRS      EQUIV.RAD.LOAD       LB-10 HRS      EQUIV.RAD.LOAD
CW (VIEW A TO B)     39032.9                             37422.1
CCW (VIEW A TO B)    39033.0                             37422.1

    REFERENCE DATA

  GEAR MEMBERS
    FIRST    TANGENTIAL FORCE P=   2485.7 LBS
             SEPARATING FORCE S=   1109.7 LBS
             THRUST     FORCE T=      0.0 LBS

                    --- -----A - -- --              --------B -------
  ROTATION          X-PLANE  Y-PLANE  COMBINED      X-PLANE  Y-PLANE  COMBINED
                      LBS      LBS    DIR.(DEG)       LBS      LBS    DIR.(DEG)
CW (VIEW A TO B)     1235.0   -551.3    114.06       1250.7   -558.4    114.06
CCW (VIEW A TO B)   -1235.0   -551.3    245.94      -1250.7   -558.4    245.94
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SKF TEST GEARBOX
P.G.C. ORDER NO. 424833

TORSIONAL NATURAL FREQUENCY
ANALYSIS

Description

TORSIONAL FREQUENCIES OF TEST STAND

Prepared by	Date
Approved by	Date
Revision	Date

A torsional analysis of the SKF Drive was done in order to determine the torsional natural frequencies of the system. Two computer programs have been used in this analysis: UMEDAT was used in determining mass moment of inertia and stiffness of the system components, and VIBT utilized these results in a lumped mass model to determine torsional natural frequencies.

Figure 1 shows a model of the system components. The inertia and stiffness of the components are summarized in Table 1.

The model for the torsional frequencies is shown in Figure 2. The system consists principally of three large masses, the motor and two gearsets. Since the remaining components are small in comparison (the largest being just 10% of the size of one gearset), they are lumped with the masses of the major components, with minimal effect on the natural frequency calculation. The dashed lines in Figure 1 indicate the component groupings for the lumped mass model.

The lumped masses are connected by two stiffnesses, modeled by combining component stiffnesses as follows: (1) the stiffness between the motor and first gearbox is the series combination of the motor, coupling, and shaft stiffnesses; (2) the stiffness of each shaft pair and coupling between gearboxes is combined in series, and these two stiffnesses are then combined in parallel.

The results of the torsional natural frequency analysis may be found in Table 3, and the operating frequencies are listed in Table 2.

Description

SYSTEM MODEL

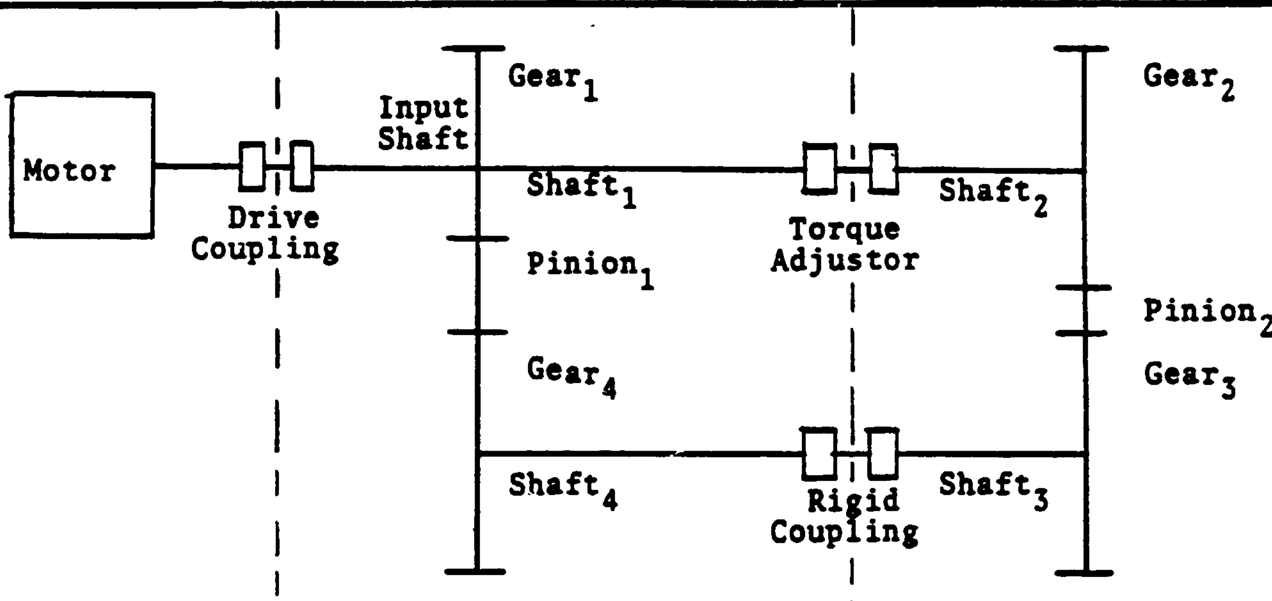

FIG. 1 ACTUAL SYSTEM

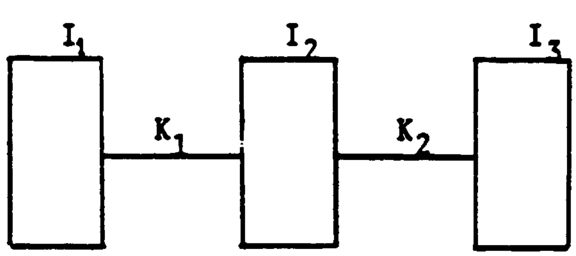

FIG. 2 SYSTEM MODEL

I_1 = Motor + 1/2 Drive Coupling + 1/2 Input Shaft & Motor Shaft

I_2 = 1/2 Drive Coupling + $Gear_1$ + $Gear_4$ + $Pinion_1$ + $Shaft_1$ + $Shaft_4$ + 1/2 Torque Adjustor + 1/2 Rigid Coupling + 1/2 Input Shaft

I_3 = 1/2 Torque Adjustor + $Shaft_2$ + $Shaft_3$ + $Gear_2$ + + $Pinion_2$ + $Gear_3$ + 1/2 Rigid Coupling & Slip Ring

K_1 = Motor Stiffness & Drive Coupling Stiffness & Input Shaft Stiffness and Motor Shaft Stiffness

K_2 = Shafts 1, 2, 3, 4 Stiffness & Torque Adjustor Stiffness & rigid coupling stiffness.

Description STIFFNESS AND MOMENT OF INERTIA OF SYSTEM COMPONENTS

TABLE 1

Item	Moment of Inertia Ref. L.S.S. (lb-in^2)	Stiffness (in.lb/rad)
Motor & Motor Shaft*	2906.5	7621572
Drive Cplg.*	5.8	420000
Input Portion of Shaft #1	8.0	3479554
Gears 1-4	583 (ea.)	∞
Rest of Shaft 1	11.0	1195442
Torque Adjustor*	287.5	750000
Shafts 3 & 4	16 (ea.)	1195442
Pinions 1 & 2	79.0 (ea.)	∞
Rigid Cplg.*	68.2	750000
Shaft 2	19	1194313
Misc. Moments of Inertia		
Slip Ring	1.3	

*SUPPLIED BY CUSTOMER

PHILADELPHIA GEAR CORPORATION
King of Prussia, PA 19406 ☐ (215) 265-3000 Telex 846321

Order No.

Description

STIFFNESS AND MOMENT OF INERTIA OF SYSTEM

Prepared by | Date

Approved by | Date

Revision | Date

$$I_1 = 2906.5 + 1/2\ (5.8) + 1/2\ (8)$$

$$= \underline{2913\ \text{LB-IN}^2}$$

$$I_2 = 1/2\ (5.8) + 1/2\ (8) + 2\ (583) + 79 + 11 + 16$$

$$+ 1/2\ (287.5) + 1/2\ (68.2)$$

$$= \underline{1458.\ \text{LB-IN}^2}$$

$$I_3 = 1/2\ (287.5) + 19 + 16 + 2\ (583) + 79 + 1/2\ (68.2)$$

$$+ 1.3$$

$$= \underline{1459\ \text{LB-IN}^2}$$

$$\frac{1}{K_1} = \frac{1}{7621572} + \frac{1}{420000.} + \frac{1}{3479554}$$

$$K_1 = \underline{357200\ \text{IN-LB/RAD}}$$

$$K_2 = \frac{1}{\frac{1}{1195442} + \frac{1}{1194313} + \frac{1}{750000}} + \frac{1}{\frac{1}{1195442} + \frac{1}{1195442} + \frac{1}{750000}}$$

$$= \underline{665170\ \text{IN-LB/RAD}}$$

Description
SYSTEM OPERATING AND SYSTEM TORSIONAL NATURAL FREQUENCIES SUMMARY

Prepared by Date
Approved by Date
Revision Date

DESCRIPTION	OPERATING FREQUENCY (Hz)
All Shafts	93.43
Helical Mesh	11025
Pinion Frequency	315

TABLE 2

DESCRIPTION	NATURAL FREQUENCY (Hz)
1ST	45.2
2ND	102.4

TABLE 3

SHAFTING SYSTEMS ANALYSIS

TORSIONAL NATURAL FREQUENCIES AND MODE SHAPES

SHAFTING SYSTEM IDENTIFICATION NUMBER 2

VT.T SKF 4 SQUARE TEST STAND 424333 DELIF

THE SHAFTING SYSTEM IS COMPOSED OF 4 SECTIONS, 3 TORSIONAL NATURAL FREQUENCIES WILL BE COMPUTED WITHIN AN ACCURACY OF PLUS OR MINUS 0.10 REVOLUTIONS PER MINUTE. IT WAS ESTIMATED THAT THE FIRST TORSIONAL NATURAL FREQUENCY WAS GREATER THAN 100. REVOLUTIONS PER MINUTE AND THAT NO TWO TORSIONAL NATURAL FREQUENCIES WERE CLOSER THAN 30. REVOLUTIONS PER MINUTE.
PROGRAM SOLVING FOR FREQUENCIES UP TO 100000. CPM
THE FOLLOWING DATA DESCRIBES THE SHAFTING SYSTEM.

THE SYSTEM DENSITY IS 0.000 LB/IN3, WITH A MODULUS OF 0.30000 08 PSI
THE SHEAR MODULUS IS 0.12000 08 PSI

SECTION	BRANCH	LENGTH (IN)	OUTER DIAMETER (IN)	INNER DIAMETER (IN)	LUMPED MASS (LBSC2/IN)	LUMPED INERTIA (LBIN*IN)	GEAR RATIO	LONG. FIXITY
1	0	0.71	1.07	0.0	0.0	0.00	1.00	0
2	0	1.00	10.35	0.0	0.0	2913.00	1.00	0
3	0	1.00	0.74	0.0	0.0	1458.00	1.00	0
4	0	1.00	0.87	0.0	0.0	1459.00	1.00	0

SECTION NUMBER	LOCAL STIFFNESS	RELATIVE COMPLIANCE	STATION NUMBER	RELATIVE INERTIA (LBININ)
			1	0.4909D-10
1	0.1178D 09	0.8488D-08		
			2	0.4267D-02
2	0.1000D 13	0.1000D-11		
			3	0.2913D 04
3	0.3572D 06	0.2800D-05		
			4	0.1458D 04
4	0.6652D 06	0.1503D-05		
			5	0.1459D 04

TORSIONAL NATURAL FREQUENCY
(REVOLUTIONS PER MINUTE)

MODE SHAPE

TORSIONAL NATURAL FREQUENCY (REVOLUTIONS PER MINUTE)	SECTION	ROTATION AT BEGINNING (RADIANS)	TORQUE AT BEGINNING (IN LB)	ROTATION AT END (RADIANS)	TORQUE AT END (IN LB)
2711.78					
	1	-0.772250 00	0.0	-0.772250 00	0.102220-13
	2	-0.772250 00	0.122620-13	-0.772250 00	0.103450-11
	3	-0.772250 00	0.100000 01	0.542210 00	0.100000 01
	4	0.542210 00	0.348280 00	0.100000 01	0.348280 00
6141.21					
	1	-0.129290 00	0.0	-0.129290 00	-0.663000-14
	2	-0.129290 00	-0.795370-14	-0.129290 00	-0.670950-12
	3	-0.129290 00	-0.648580 00	0.100000 01	-0.648580 00
	4	0.100000 01	0.100000 01	-0.741180 00	0.100000 01

MAXIMUM SPEED REACHED

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SKF TEST GEARBOX
P.G.C. ORDER NO. 424833

LATERAL NATURAL FREQUENCY
ANALYSIS

Description

LATERAL NATURAL FREQUENCIES
OF TEST STAND

The Philadelphia Gear Rotor Dynamics (ROTDYN) program has been used to analyze lateral vibration characteristics of the Test Stand shafting. An eigenvalue problem is formulated by assuming small excursions from the equilibrium configuration of the rotor/bearing system. ROTDYN employs an extension of the Prohl critical speed method, modified to account for elastic rotor mountings and for the influence of distributed shaft mass, and was run neglecting damping consideration. The program yields all the natural frequencies of the system and their mode shapes.

ROTDYN INPUT DATA EXPLANATION

Shaft Element	Shaft is divided into elements, which are numbered in ascending order, starting from one end of the shaft.
Element Stations	Station number of the two ends of a shaft element.
Element Type	Describes the shaft element type as a solid or hollow, tapered or uniformed.
Material Type	Material type no. of the element. In this program, each shaft element is assumed to be made of a single material. The properties for each material are specified in material cards, which include material type number, Young's Modulus (PSI), Shear Modulus (PSI), and Density (lb/in^3)
Length (in)	Length of element
O.D. (in)	Outside diameter of the shaft element end.
I.D. (in.)	Inside diameter of the same shaft element end.
O.D. (in.)	Outside diameter of the other shaft element end.

INPUT SHAFT, SHAFT 1, SHAFT 2

NUMBER OF STATIONS..........	26	NUMBER OF DISCS............	5	NUMBER OF BEARINGS.........	4
NUMBER OF MATERIALS........	1	NUMBER OF SHAFT ELMTS......	25	NUMBER OF BRANCH ELMTS.....	0
GRAVITY LOADING............	NO	GRAVITY...................	386.4	SPEED.....................	5605.0

* MATERIAL PROPERTIES *

MATERIAL TYPE NO.	MODULUS OF ELASTICITY	SHEAR MODULUS	DENSITY	MATERIAL DAMPING
1	.30000000+008	.11500000+008	.73240164-003	.00000000

* SHAFT ELEMENT DATA *

SHAFT ELEMENT	ELEMENT STATIONS		ELEMENT TYPE	MATERIAL TYPE	LENGTH	O.D. (LEFT)	I.D. (LEFT)	O.D. (RIGHT)	I.D. (RIGHT)
1	1	2	1	1	.700000+000	.200000+001	.000000	.200000+001	.000000
2	2	3	1	1	.357000+001	.200000+001	.000000	.200000+001	.000000
3	3	4	1		.630000+000	.200000+001	.000000	.200000+001	.000000
4	4	5	1	1	.600000+000	.236300+001	.000000	.236300+001	.000000
5	5	6	1	[illegible]	.590000+000	.236300+001	.000000	.236300+001	.000000
6	6	7	1	1	.119000+001	.284000+001	.000000	.284000+001	.000000
7	7	8	1	1	.610000+000	.250000+001	.000000	.250000+001	.000000
8	8	9	1	1	.610000+000	.250000+001	.000000	.250000+001	.000000
9	9	10	1	1	.181000+001	.236000+001	.000000	.236000+001	.000000
10	10	11	1	1	.600000+000	.236300+001	.000000	.236300+001	.000000
11	11	12	1	1	.630000+000	.200000+001	.000000	.200000+001	.000000
12	12	13	1	1	.343000+001	.200000+001	.000000	.200000+001	.000000
13	13	14	1	1	.110000+002	.175000+001	.000000	.175000+001	.000000
14	14	15	1	1	.240000+001	.200000+001	.000000	.200000+001	.000000
15	15	16	1	1	.600000+000	.200000+001	.000000	.200000+001	.000000
16	16	17	1	1	.600000+000	.200000+001	.000000	.200000+001	.000000
17	17	18	1	1	.527500+001	.200000+001	.000000	.200000+001	.000000
18	18	19	1	1	.630000+000	.200000+001	.000000	.200000+001	.000000
19	19	20	1	1	.600000+000	.236300+001	.000000	.236300+001	.000000
20	20	21	1	1	.181000+001	.236300+001	.000000	.236300+001	.000000
21	21	22	1	1	.610000+000	.250000+001	.000000	.250000+001	.000000
22	22	23	1	1	.610000+000	.250000+001	.000000	.250000+001	.000000
23	23	24	1	1	.119000[illegible]	.284000+001	.000000	.284000+001	.000000
24	24	25	1	1	.5900[illegible]000	.236[illegible]00+001	.000000	.236300+001	.000000
25	25	26	1	1	.6300[illegible]+000	.236300+001	.000000	.236300+001	.000000

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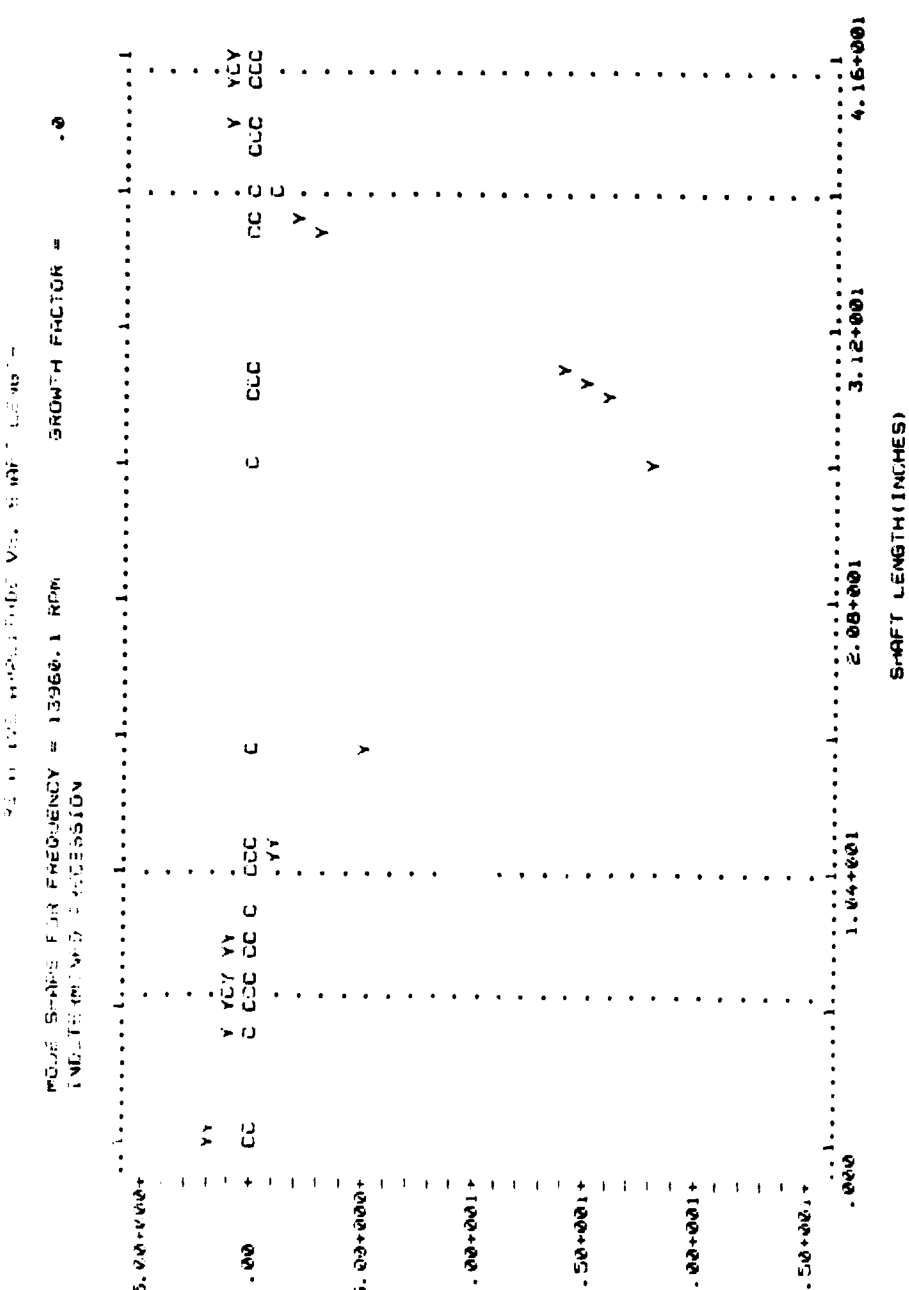
* ROTOR PARAMETERS *

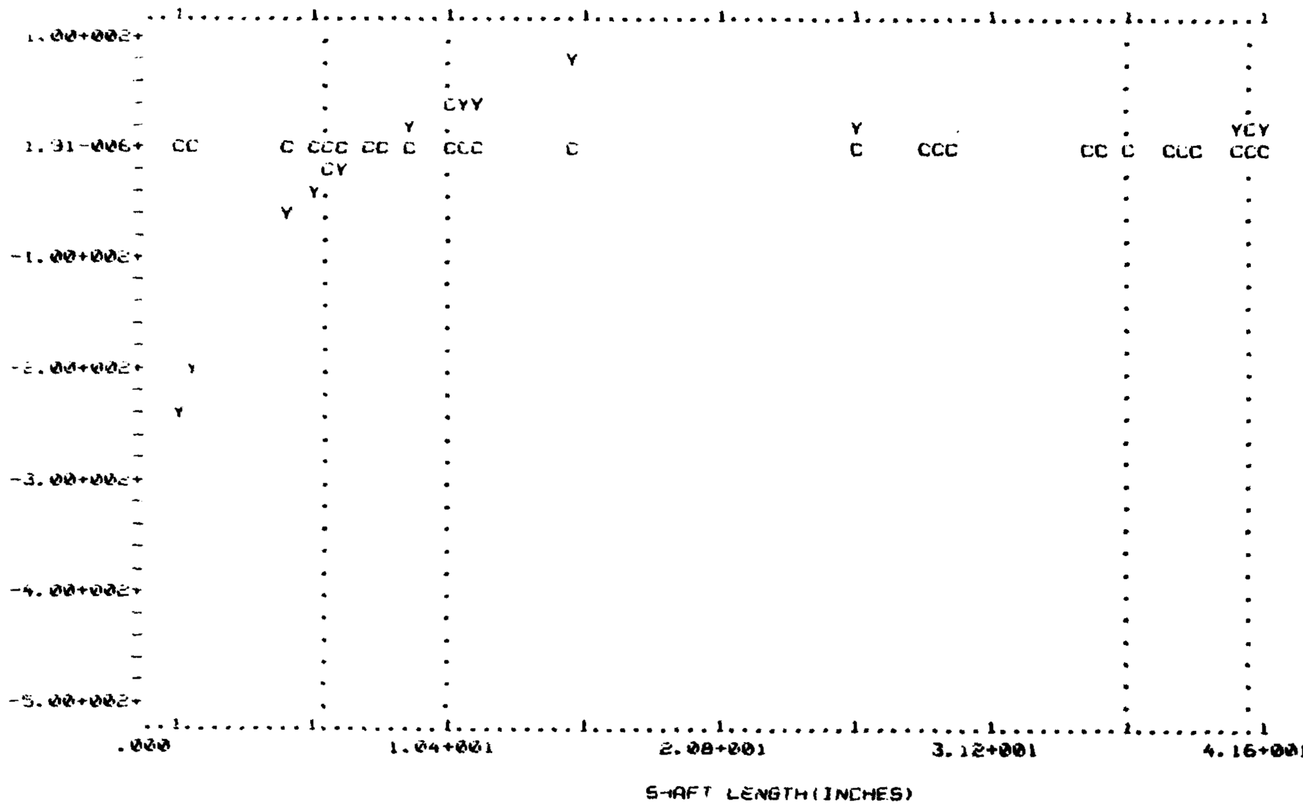
WEIGHT OF SHAFT.....	40.61689	WEIGHT OF DISCS......	89.15000	WEIGHT OF ROTOR..	129.76688
SHAFT LENGTH........	41.61500	LOCATION OF C.G......	24.06581		

DISC NUMBER	STATION NUMBER	POLAR MOMENT OF INERTIA	TRANSVERSE MOMENT OF INERTIA	MASS	RADIUS OF C.G.	ANGULAR POSITION OF C.G.	WEIGHT
1	2	.000000	.000000	.582298-002	.000000	.000000	2.25
2	8	.000000	.000000	.603002-001	.000000	.000000	23.30
3	15	.000000	.000000	.439959-001	.000000	.000000	17.00
4	17	.000000	.000000	.603002-001	.000000	.000000	17.00
5	22	.000000	.000000	.603002-001	.000000	.000000	23.30


MODE SHAPE FOR FREQUENCY = 13960.1 RPM
GROWTH FACTOR = .0
SHAFT LENGTH(INCHES)
5.00+000
.00
-5.00+000
-1.00+001
-1.50+001
-2.00+001
-2.50+001
.000
1.04+001
2.08+001
3.12+001
4.16+001

RELATIVE AMPLITUDE VS. SHAFT LENGTH

MODE SHAPE FOR FREQUENCY = 44303.3 RPM GROWTH FACTOR = .0
INDETERMINED PRECESSION



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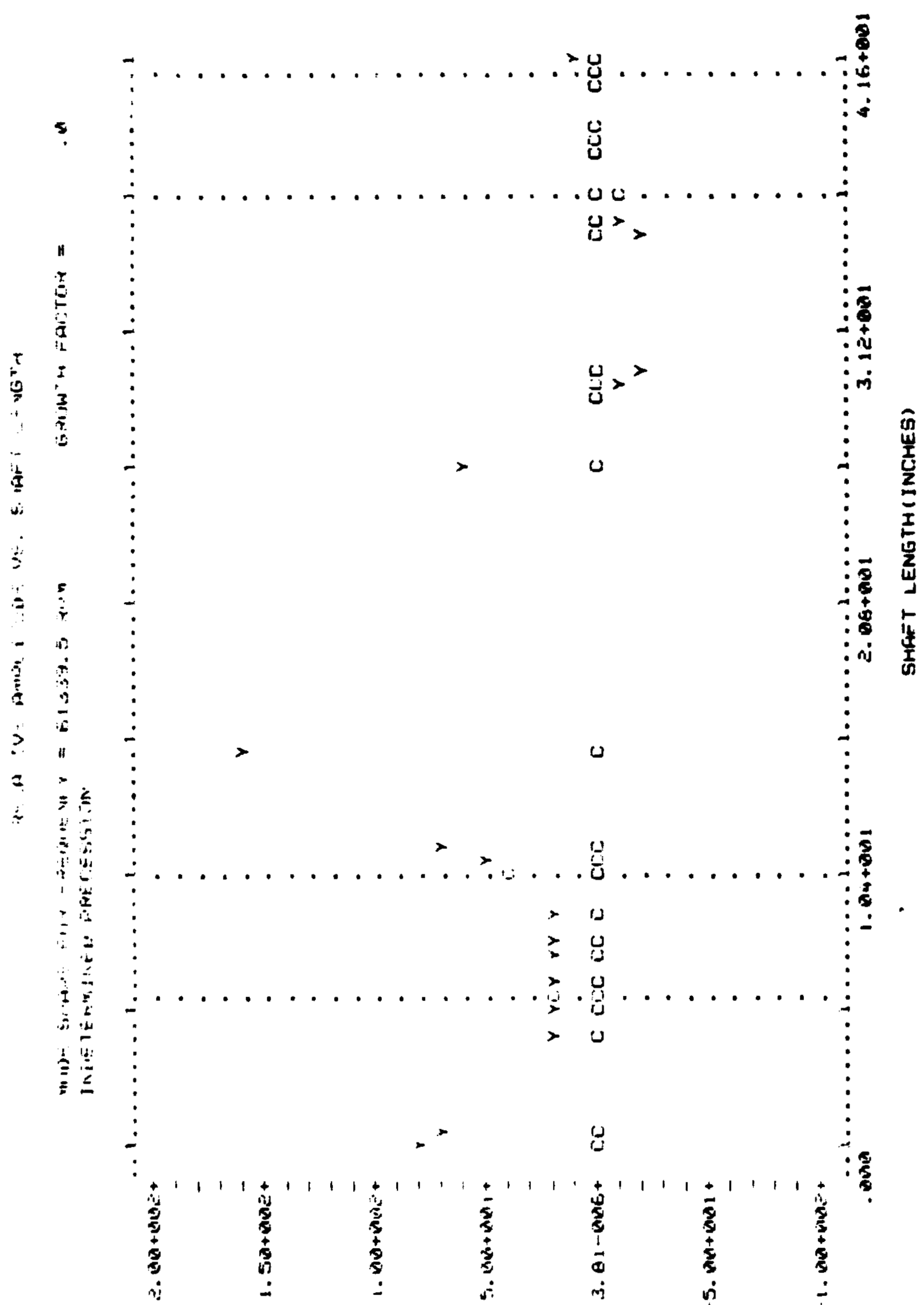

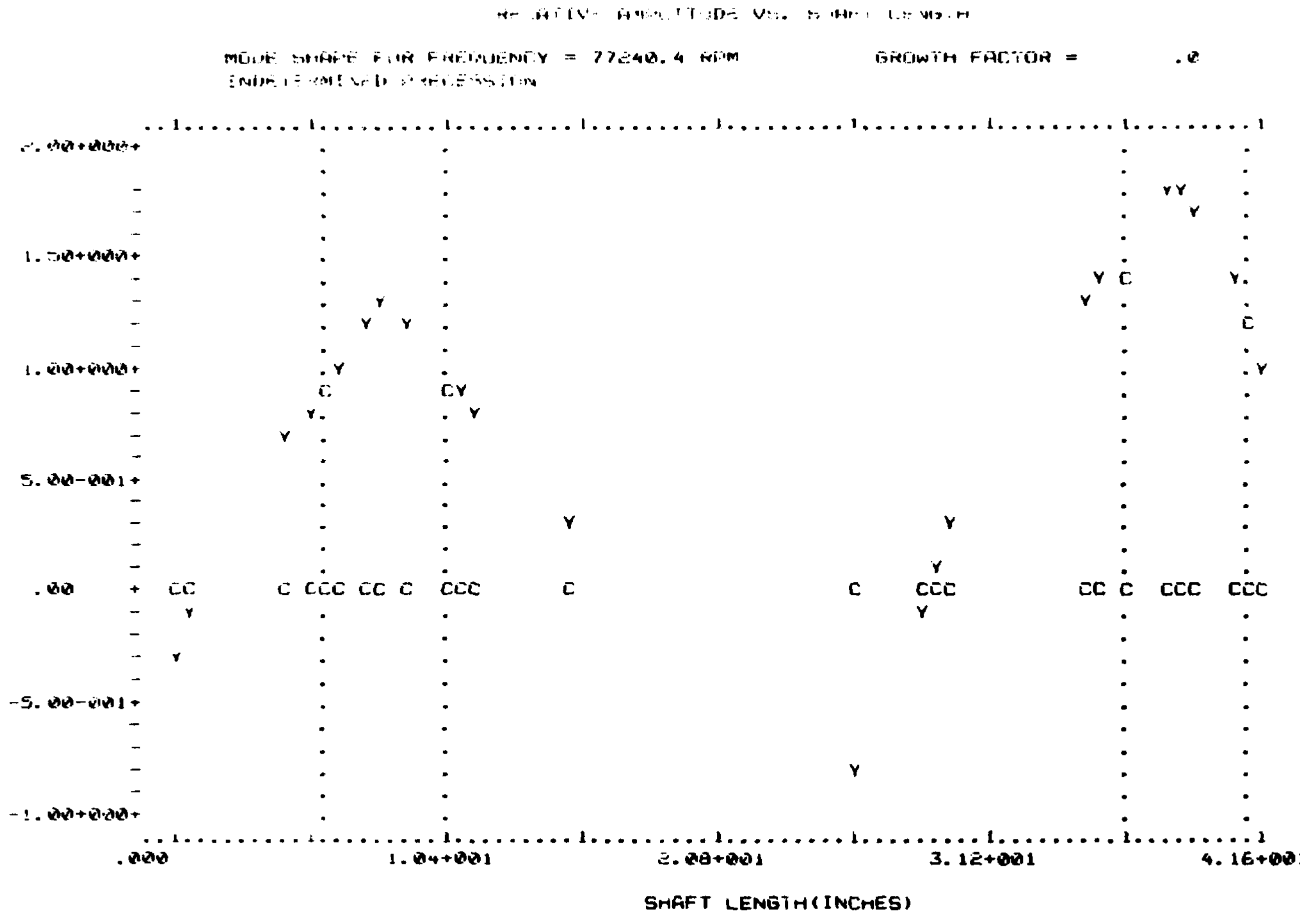
MODE SHAPE FOR FREQUENCY = 77240.4 RPM
GROWTH FACTOR = .0
2.00+000
1.50+000
1.00+000
5.00-001
.00
-5.00-001
-1.00+000
.000
1.04+001
2.08+001
3.12+001
4.16+001
SHAFT LENGTH(INCHES)

424B-3 TEST STAND **SHAFT 3, SHAFT 4**

NUMBER OF STATIONS..........	26	NUMBER OF DISCS............	5	NUMBER OF BEARINGS.........	4
NUMBER OF MATERIALS........	1	NUMBER OF SHAFT ELMTS......	25	NUMBER OF BRANCH ELMTS.....	0
GRAVITY LOADING............	NO	GRAVITY....................	386.4	SPEED......................	5605.0

* MATERIAL PROPERTIES *

MATERIAL TYPE NO.	MODULUS OF ELASTICITY	SHEAR MODULUS	DENSITY	MATERIAL DAMPING
1	.30000000+008	.11500000+008	.73240164-003	.00000000

* SHAFT ELEMENT DATA *

SHAFT ELEMENT	ELEMENT STATIONS		ELEMENT TYPE	MATERIAL TYPE	LENGTH	O.D. (LEFT)	I.D. (LEFT)	O.D. (RIGHT)	I.D. (RIGHT)
1	1	2	1	1	.150000+001	.125000+001	.000000	.125000+001	.000000
2	2	3	1	1	.500000+000	.750000+000	.000000	.750000+000	.000000
3	3	4	1	1	.313000+001	.200000+001	.000000	.200000+001	.000000
4	4	5	1	1	.600000+000	.236300+001	.000000	.236300+001	.000000
5	5	6	1	1	.590000+000	.236300+001	.000000	.236300+001	.000000
6	6	7	1	1	.119000+001	.284000+001	.000000	.284000+001	.000000
7	7	8	1	1	.610000+000	.250000+001	.000000	.250000+001	.000000
8	8	9	1	1	.610000+000	.250000+001	.000000	.250000+001	.000000
9	9	10	1	1	.181000+001	.236000+001	.000000	.236000+001	.000000
10	10	11	1	1	.600000+000	.236300+001	.000000	.236300+001	.000000
11	11	12	1	1	.630000+000	.200000+001	.000000	.200000+001	.000000
12	12	13	1	1	.343000+001	.200000+001	.000000	.200000+001	.000000
13	13	14	1	1	.110000+002	.175000+001	.000000	.175000+001	.000000
14	14	15	1	1	.240000+001	.200000+001	.000000	.200000+001	.000000
15	15	16	1	1	.600000+000	.200000+001	.000000	.200000+001	.000000
16	16	17	1	1	.600000+000	.200000+001	.000000	.200000+001	.000000
17	17	18	1	1	.527500+001	.200000+001	.000000	.200000+001	.000000
18	18	19	1	1	.630000+000	.200000+001	.000000	.200000+001	.000000
19	19	20	1	1	.600000+000	.236300+001	.000000	.236300+001	.000000
20	20	21	1	1	.181000+001	.236300+001	.000000	.236300+001	.000000
21	21	22	1	1	.610000+000	.250000+001	.000000	.250000+001	.000000
22	22	23	1	1	.610000+000	.250000+001	.000000	.250000+001	.000000
23	23	24	1	1	.119000+001	.284000+001	.000000	.284000+001	.000000
24	24	25	1	1	.590000+000	.236300+001	.000000	.236300+001	.000000
25	25	26	1	1	.630000+000	.236300+001	.000000	.236300+001	.000000

• ROTOR PARAMETERS •

WEIGHT OF SHAFT.....	39.53778	WEIGHT OF DISCS......	66.60001	WEIGHT OF ROTOR..	106.13779
SHAFT LENGTH........	41.74500	LOCATION OF C.G......	23.93245		

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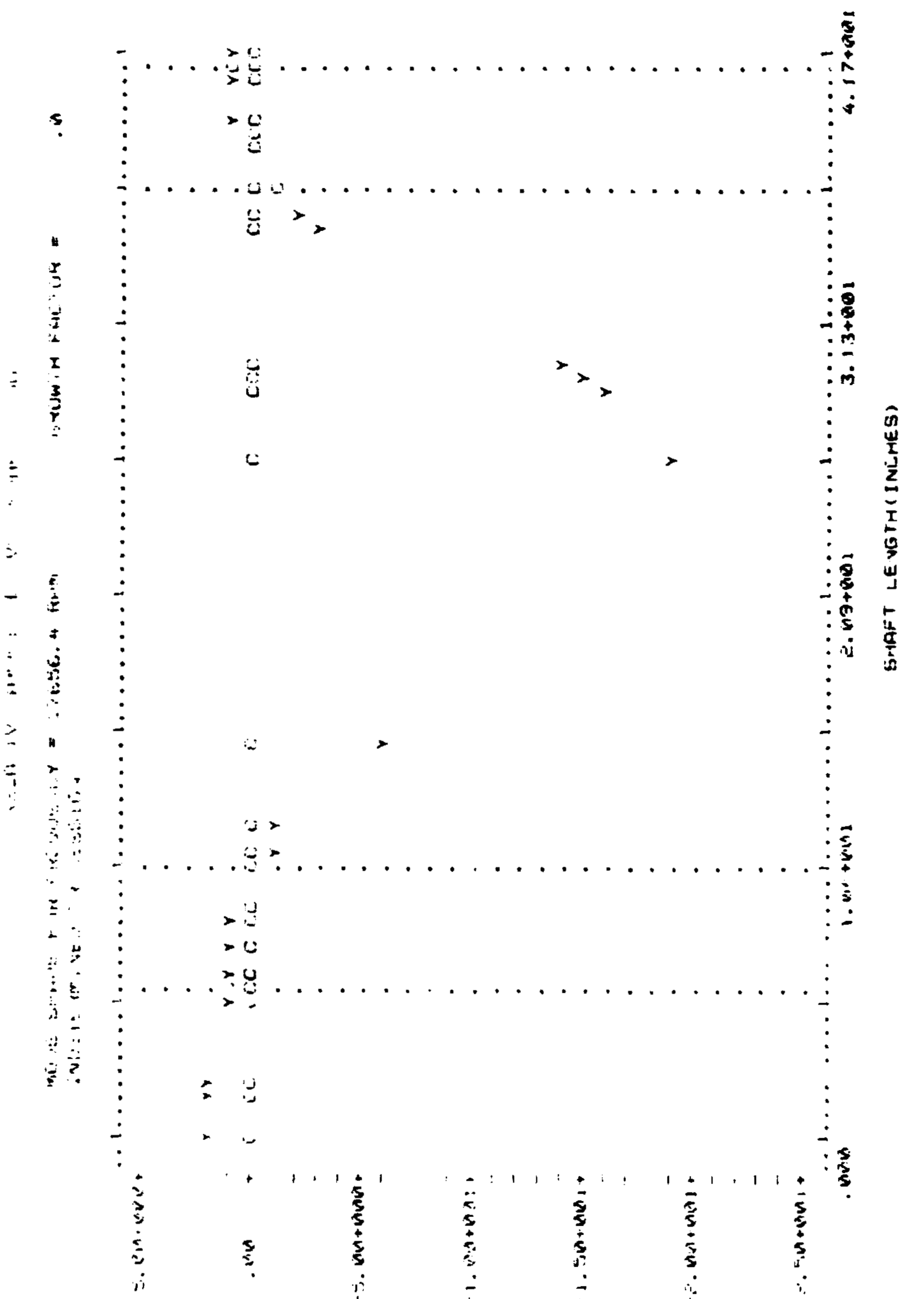
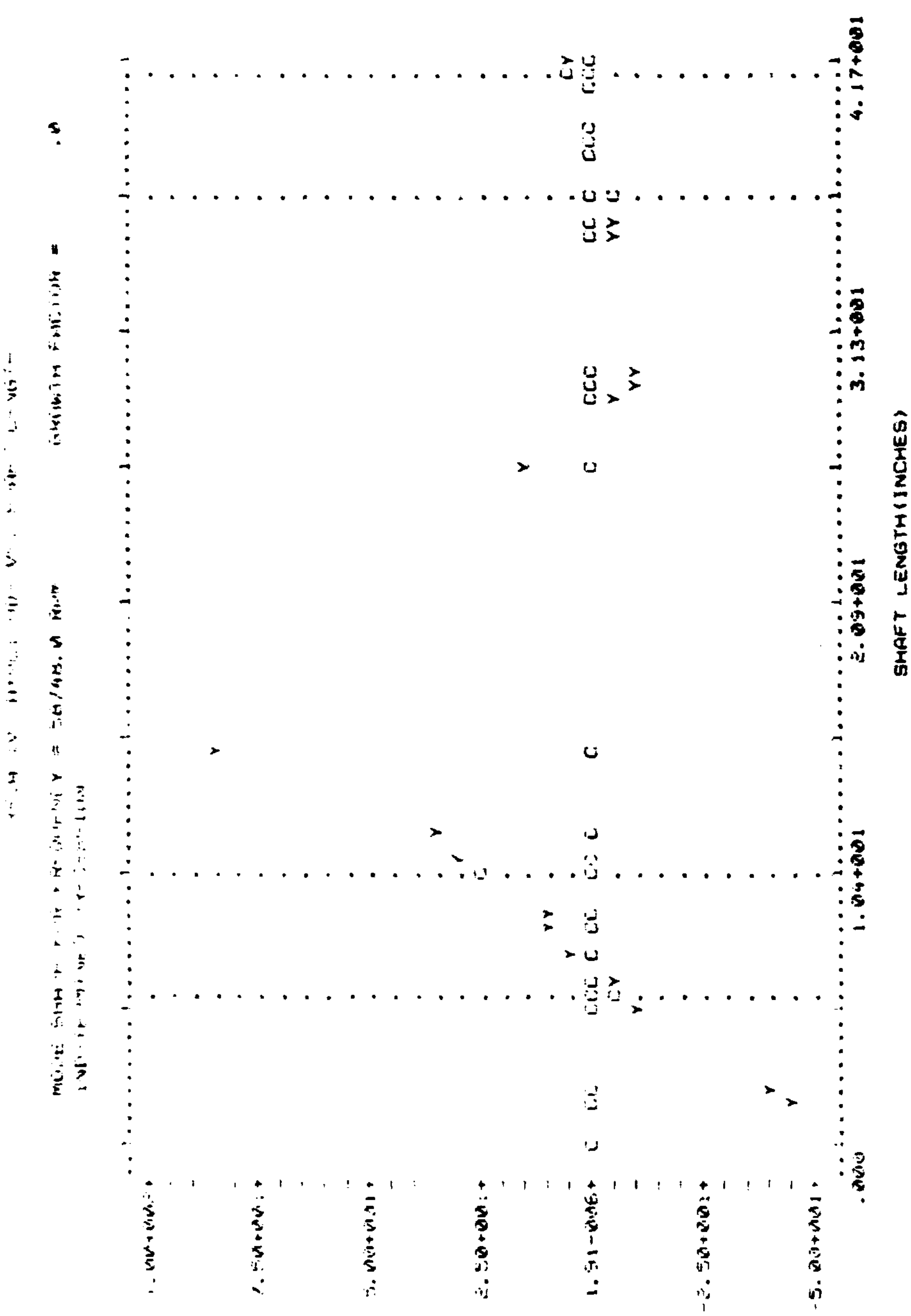
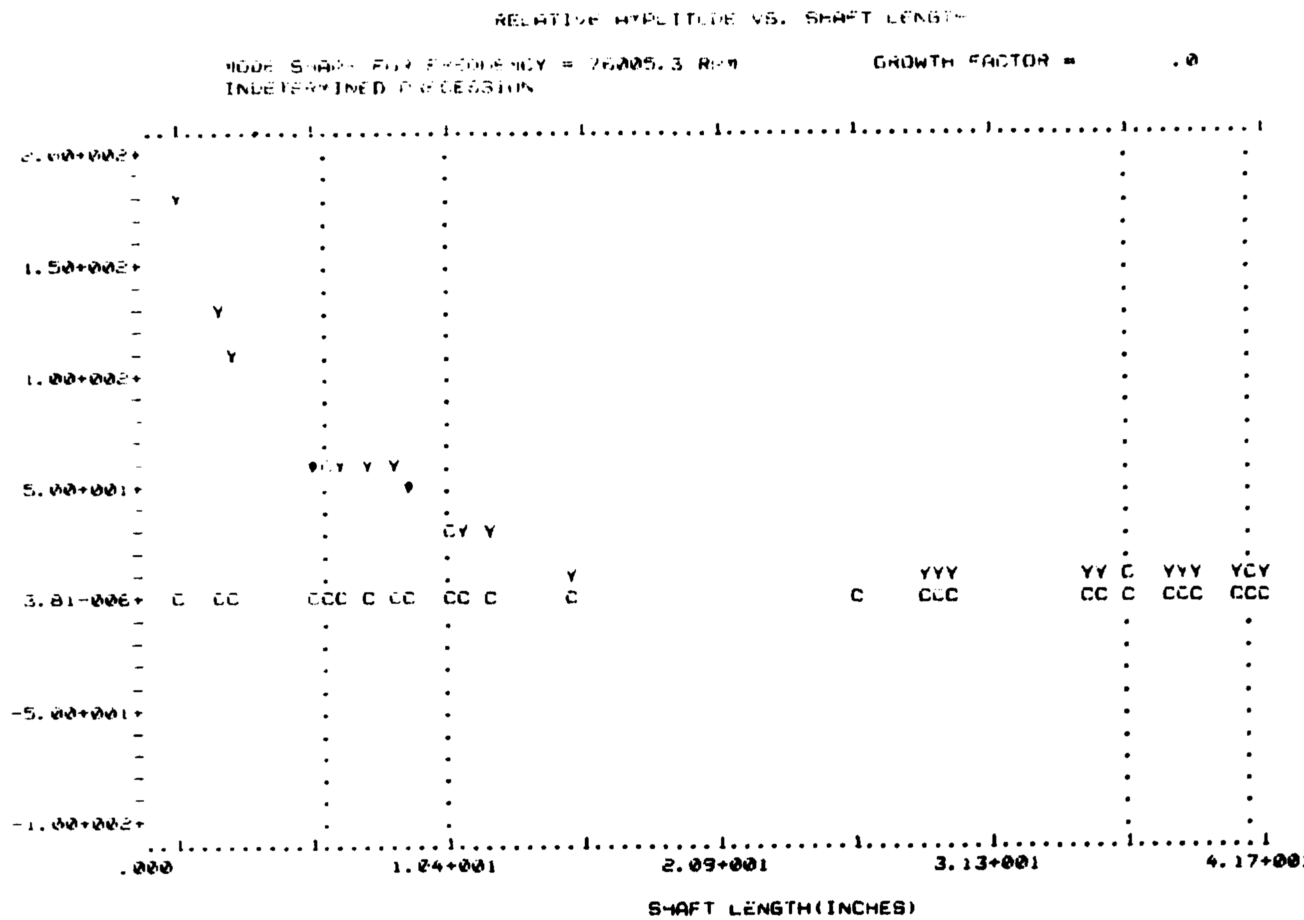
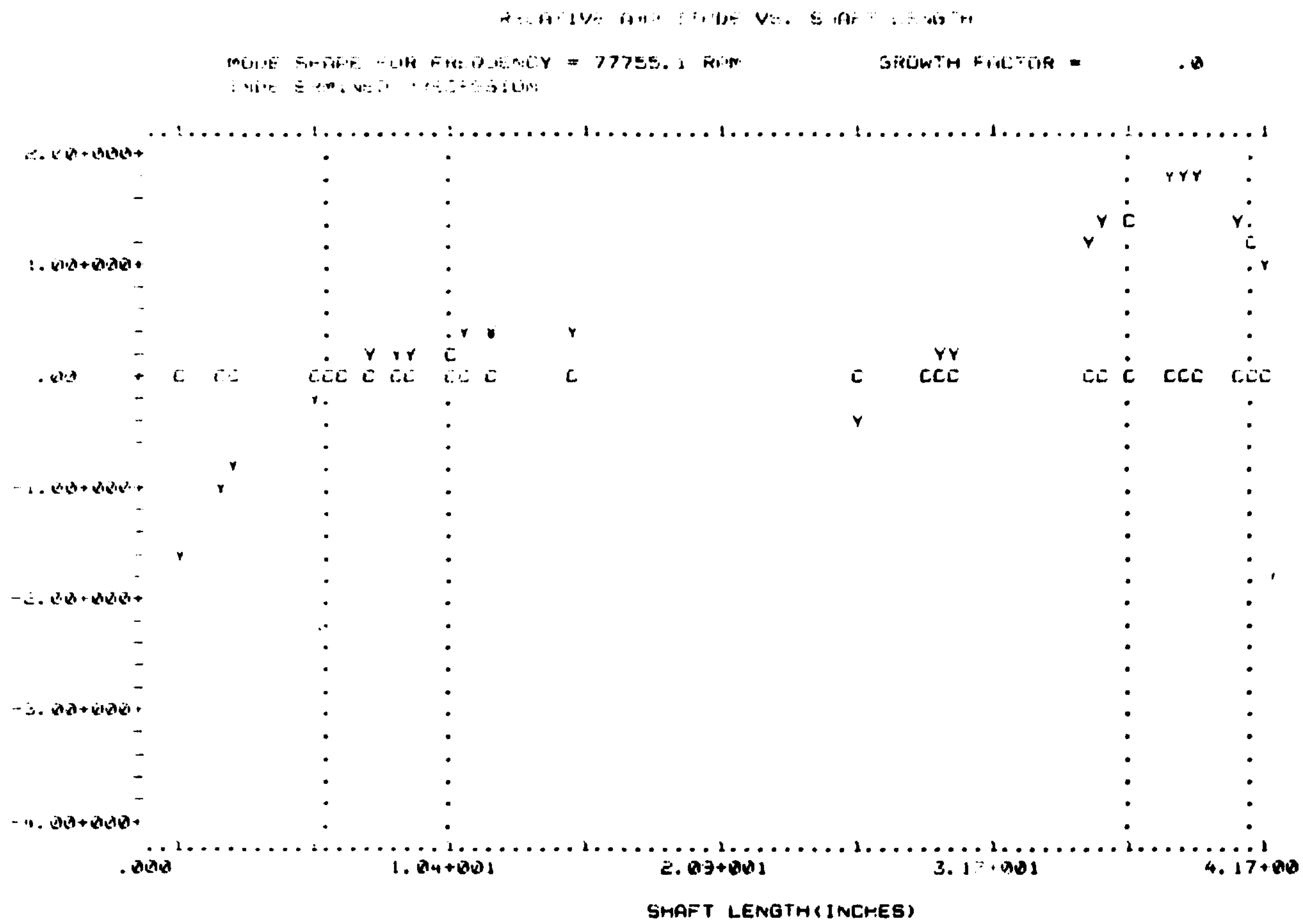
OF POOR QUALITY

DISC NUMBER	STATION NUMBER	POLAR MOMENT OF INERTIA	TRANSVERSE MOMENT OF INERTIA	MASS	RADIUS OF C.G.	ANGULAR POSITION OF C.G.	WEIGHT
1	2	.000000	.000000	.258[illegible]-007	.000000	.000000	.00
2	8	.000000	.000000	.603002-001	.000000	.000000	23.30
3	15	.000000	.000000	.258799-001	.000000	.000000	10.00
4	17	.000000	.000000	.258799-001	.000000	.000000	10.00
5	22	.000000	.000000	.603002-001	.000000	.000000	23.30


SHAFT LENGTH(INCHES)
.000
1.04+001
2.09+001
3.13+001
4.17+001
.00
-5.00+000
-1.00+001
-1.50+001
-2.00+001
-2.50+001


SHAFT LENGTH(INCHES)
.000
1.04+001
2.09+001
3.13+001
4.17+001
-5.00+001
-2.50+001
1.91-006
2.50+001


RELATIVE AMPLITUDE VS. SHAFT LENGTH
MODE SHAPE FOR FREQUENCY = 76005.3 RPM
GROWTH FACTOR = .0
INDETERMINED PRECESSION
2.00+002
1.50+002
1.00+002
5.00+001
3.81-006
-5.00+001
-1.00+002
.000
1.04+001
2.09+001
3.13+001
4.17+001
SHAFT LENGTH(INCHES)


GROWTH FACTOR = .0
.000
1.04+001
2.09+001
4.17+001
SHAFT LENGTH(INCHES)
1.00+000
.00
-1.00+000
-2.00+000
-3.00+000
-4.00+000

SKF TEST GEARBOX
P.G.C. ORDER NO. 424833

MASS ELASTIC DATA

Description

MOTOR AND COUPLING MASS ELASTIC DATA

MOTOR & MOTOR SHAFT:	I = .627 Lb-Ft - Sec^2
	K = 635000 Lb-Ft/Rad.
DRIVE COUPLING:	I = .00125 Lb-Ft - Sec^2
	K = 35000 Lb-Ft/Rad.
SHAFT COUPLING:	I = .0147 Lb-Ft - Sec^2
	K = 62500 Lb-Ft/Rad.
TORQUE ADJUSTOR:	I = .062 Lb-Ft - Sec^2
	K = 62500 Lb-Ft/Rad.

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UNIT MASS ELASTIC DATA COMPUTATION

424833 P.6 BRG. TESTER — TIME 14:30:12 DATE 12/20/83 — PAGE 1

COMPUTER: UNIVAC 1100

ANALYST-DELID — APPROVED-............

INPUT SHAFT

SECTION NO. FOR BEGIN STIFF CALC	2
SECTION NO. FOR END STIFF CALC	8
MATERIAL DENSITY	.28340
MATERIAL SHEAR MODULUS	11500000.

SECTION NO.	SECTION O.D.	SECTION I.D.	SECTION LENGTH	SECTION O.D. FOR WT.	SECTION WEIGHT	SECTION WR**2
1	2.000	.000	.667	2.000	.59	.3
2	2.000	.000	1.333	2.000	1.19	.6
3	2.000	.000	2.380	2.000	2.12	1.1
4	2.250	.000	.250	2.250	.28	.2
5	2.000	.000	.380	2.000	.34	.2
6	2.637	.000	1.190	3.000	2.38	2.7
7	2.840	.000	1.190	2.840	2.14	2.2
8	2.500	.000	.625	2.500	.87	.7
9	2.500	.000	.595	2.500	.83	.6
10	2.363	.000	2.410	3.000	4.83	5.4
11	2.000	.000	.380	2.000	.34	.2
12	2.250	.000	.250	2.250	.28	.2
13	2.000	.000	2.745	2.000	2.44	1.2
14	1.875	.000	6.900	1.875	5.40	2.4
15	2.000	.000	1.417	2.000	1.26	.6
16	2.000	.000	.583	2.000	.52	.3

WEIGHT	26.	LBS
WR**2	19.	LB*IN2
STIFFNESS	3479555.	IN*LBS/RAD
SHAFT LENGTH	23.29500	IN
CENTER OF GRAVITY	10.6629	INCHES FROM BEGINNING OF SECTION ONE
EFF. LENGTH	5.191509	INCHES BASED ON A DIAMETER OF 2.0000 IN
WR**2 RELATIVE TO L.S. SHAFT	19.	LB*IN2
STIFFNESS RELATIVE TO L.S. SHAFT	3479555.	IN*LBS/RAD
SHAFT SPEED RELATIVE TO L.S. SHAFT	1.0000	

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*** PHILADELPHIA GEAR CORPORATION ************ ******  ********************** PHILADELPHIA GEAR CORPORATION ***
*UNIT MASS ELASTIC DATA COMPUTATIONS*
*************************************************************************** TIME 14:30:12 DATE 12/20/83*      PAGE   2 *
*  424833     8.6 BRG. TESTER                                             * COMPUTER: UNIVAC 1100                       *
*************************************************************************** ANALYST-DELIO          APPROVED-............
*  INPUT SHAFT                                                            *
***************************************************************************
```

SECTION NO. FOR BEGIN STIFF CALC 9
SECTION NO. FOR END STIFF CALC 15
MATERIAL DENSITY .28340
MATERIAL SHEAR MODULUS 11500000.

SECTION NO.	SECTION O.D.	SECTION I.D.	SECTION LENGTH	SECTION O.D. FOR WT.	SECTION WEIGHT	SECTION WR**2
1	2.000	.000	.667	2.000	.59	.3
2	2.000	.000	1.333	2.000	1.19	.6
3	2.000	.000	2.380	2.000	2.12	1.1
4	2.250	.000	.250	2.250	.28	.2
5	2.000	.000	.380	2.000	.34	.2
6	2.637	.000	1.190	3.000	2.38	2.7
7	2.840	.000	1.190	2.640	2.14	2.2
8	2.500	.000	.625	2.500	.87	.7
9	2.500	.000	.595	2.500	.83	.6
10	2.363	.000	2.410	3.000	4.83	5.4
11	2.000	.000	.380	2.000	.34	.2
12	2.250	.000	.250	2.250	.28	.2
13	2.000	.000	2.745	2.000	2.44	1.2
14	1.875	.000	6.900	1.875	5.40	2.4
15	2.000	.000	1.417	2.000	1.26	.6
16	2.000	.000	.583	2.000	.52	.3

WEIGHT	26.	LBS
WR**2	19.	LB*IN2
STIFFNESS	1195442.	IN*LBS/RAD
SHAFT LENGTH	23.29500	IN
CENTER OF GRAVITY	10.6629	INCHES FROM BEGINNING OF SECTION ONE
EFF. LENGTH	36.891724	INCHES BASED ON A DIAMETER OF 2.5000 IN
WR**2 RELATIVE TO L.S. SHAFT	19.	LB*IN2
STIFFNESS RELATIVE TO L.S. SHAFT	1195442.	IN*LBS/RAD
SHAFT SPEED RELATIVE TO L.S. SHAFT	1.0000	

* 424833 8.6 BRG. TESTER * COMPUTER: UNIVAC 1100 *
ANALYST-DELID APPROVED-............
* LOWER SHAFT *

SECTION NO. FOR BEGIN STIFF CALC 8
SECTION NO. FOR END STIFF CALC 15
MATERIAL DENSITY .28340
MATERIAL SHEAR MODULUS 11500000.

SECTION NO.	SECTION O.D.	SECTION I.D.	SECTION LENGTH	SECTION O.D. FOR WT.	SECTION WEIGHT	SECTION WR**2
1	2.625	.750	.625	2.625	.88	.8
2	2.000	.500	2.500	2.000	2.09	1.1
3	2.250	.500	.250	2.250	.27	.2
4	2.000	.500	.380	2.000	.32	.2
5	2.363	.500	1.190	3.000	2.32	2.7
6	2.840	.500	1.190	2.840	2.07	2.2
7	2.500	.500	.625	2.500	.83	.7
8	2.500	.500	.595	2.500	.79	.6
9	2.363	.500	2.410	3.000	4.69	5.4
10	2.000	.500	.380	2.000	.32	.2
11	2.250	.500	.250	2.250	.27	.2
12	2.000	.500	2.430	2.000	2.03	1.1
13	2.000	.000	.315	2.000	.28	.1
14	1.875	.000	6.900	1.875	5.40	2.4
15	2.000	.000	1.417	2.000	1.26	.6
16	2.000	.000	.583	2.000	.52	.3

WEIGHT	24.	LBS
WR**2	19.	LB*IN2
STIFFNESS	1194313.	IN*LBS/RAD
SHAFT LENGTH	22.04000	IN
CENTER OF GRAVITY	9.8225	INCHES FROM BEGINNING OF SECTION ONE
EFF. LENGTH	36.926578	INCHES BASED ON A DIAMETER OF 2.5000 IN
WR**2 RELATIVE TO L.S. SHAFT	19.	LB*IN2
STIFFNESS RELATIVE TO L.S. SHAFT	1194313.	IN*LBS/RAD
SHAFT SPEED RELATIVE TO L.S. SHAFT	1.0000	

SECTION NO. FOR BEGIN STIFF CALC 2
SECTION NO. FOR END STIFF CALC 8
MATERIAL DENSITY .28340
MATERIAL SHEAR MODULUS 11500000.

SECTION NO.	SECTION O.D.	SECTION I.D.	SECTION LENGTH	SECTION O.D. FOR WT.	SECTION WEIGHT	SECTION WR**2
1	2.000	.000	.583	2.000	.52	.3
2	2.000	.000	1.417	2.000	1.26	.6
3	1.875	.000	6.900	1.875	5.40	2.4
4	2.000	.000	2.745	2.000	2.44	1.2
5	2.250	.000	.250	2.250	.28	.2
6	2.000	.000	.380	2.000	.34	.2
7	2.363	.000	2.410	3.000	4.83	5.4
8	2.500	.000	.595	2.500	.83	.6
9	2.500	.000	.625	2.500	.87	.7
10	2.840	.000	1.190	2.840	2.14	2.2
11	2.363	.000	1.220	3.000	2.44	2.7

WEIGHT	21.	LBS
WR**2	16.	LB*IN2
STIFFNESS	1195442.	IN*LBS/RAD
SHAFT LENGTH	18.31500	IN
CENTER OF GRAVITY	10.9266	INCHES FROM BEGINNING OF SECTION ONE
EFF. LENGTH	15.110850	INCHES BASED ON A DIAMETER OF 2.0000 IN
WR**2 RELATIVE TO L.S. SHAFT	16.	LB*IN2
STIFFNESS RELATIVE TO L.S. SHAFT	1195442.	IN*LBS/RAD
SHAFT SPEED RELATIVE TO L.S. SHAFT	1.0000	

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*** PHILADELPHIA GEAR CORPORATION ************************************************* PHILADELPHIA GEAR CORPORATION ***
*UNIT MASS ELASTIC DATA COMPUTATIONS*
****************************************************************** TIME 14:30:12 DATE 12/20/83*      PAGE   5 *
*  424933     8.6 BRG. TESTER                                      * COMPUTER: UNIVAC 1100                   *
****************************************************************** ANALYST-DELIO          APPROVED-............
*  GEARS                                                           *
******************************************************************

     MATERIAL DENSITY                         .28340
     MATERIAL SHEAR MODULUS                11500000.
```

SECTION NO.	SECTION O.D.	SECTION I.D.	SECTION LENGTH	SECTION O.D. FOR WT.	SECTION WEIGHT	SECTION WR**2
1	13.265	12.183	1.250	13.265	7.66	310.7
2	12.183	2.500	.438	12.183	13.84	267.7
3	3.875	2.500	.813	3.875	1.59	4.2

```
     WEIGHT                                      23.   LBS
     WR**2                                      583.   LB*IN2
     SHAFT LENGTH                           2.50000   IN
     CENTER OF GRAVITY                       1.2317   INCHES FROM BEGINNING OF SECTION ONE
     WR**2 RELATIVE TO L.S. SHAFT               583.   LB*IN2
     SHAFT SPEED RELATIVE TO L.S. SHAFT      1.0000
```

*** PHILADELPHIA GEAR CORPORATION ** PHILADELPHIA GEAR CORPORATION ***
UNIT MASS ELASTIC DATA COMPUTATIONS

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********************************************************************* TIME 12:47:00 DATE 11/14/83*      PAGE   4 *
*  424833    0.6 BRG TESTER                                         * COMPUTER: UNIVAC 1100                    *
********************************************************************* ANALYST-INPUT SHAF     APPROVED-...........
*  PINIONS                                                          *
*********************************************************************
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MATERIAL DENSITY .28340
MATERIAL SHEAR MODULUS 11500000.

SECTION NO.	SECTION O.D.	SECTION I.D.	SECTION LENGTH	SECTION O.D. FOR WT.	SECTION WEIGHT	SECTION WR**2
1	3.935	2.500	1.250	3.935	2.57	7.0

WEIGHT	3.	LBS
WR**2	7.	LB*IN2
SHAFT LENGTH	1.25000	IN
CENTER OF GRAVITY	.6250	INCHES FROM BEGINNING OF SECTION ONE
WR**2 RELATIVE TO L.S. SHAFT	79.	LB*IN2
SHAFT SPEED RELATIVE TO L.S. SHAFT	3.3710	