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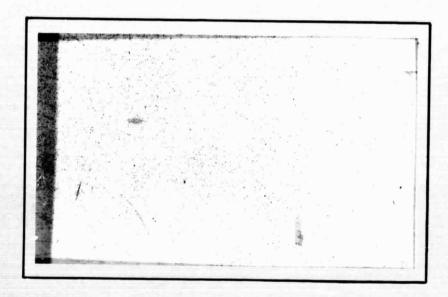
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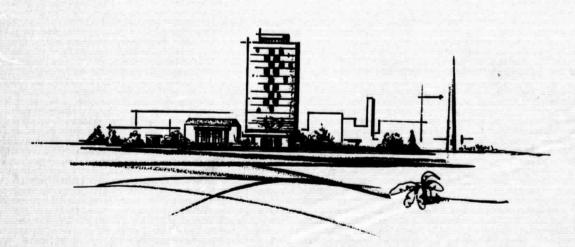
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SUMMARY REPORT

on

ANALYSIS OF BRINELLING FAILURE OF BEARINGS FROM VIBRATION TESTED PLV FANS

to

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION GEORGE C. MARSHALL SPACE FLIGHT CENTER

by

W. A. Glaeser, S. K. Batra, and R. H. Prause

June 8, 1970

BATTELLE MEMORIAL INSTITUTE Columbus Laboratories 505 King Avenue Columbus, Ohio 43201

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SUMMARY REPORT

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SUMMARY

Brinelling damage has been identified in PLV fan bearings as the cause for noisy running after exposure to a random-vibration environment.

Assuming a maximum Hertz stress of 460,000 psi to produce brinelling, the maximum axial load for the fan bearing was calculated as 43 pounds. Dynamic analysis of the rotor-bearing-spring system revealed that with one preload spring, the maximum predicted axial load would be 297 pounds. Reduction in maximum bearing loads under vibration conditions can be achieved by reducing the rotor natural frequency and by increasing the damping through using a soft spring system. This means multiple springs at both ends of the shaft. When a system of five springs at each end of the rotor was considered, the bearing load predicted was 58 pounds. Actual vibration tests at Marshall Space Flight Center using the 5-spring preload configuration have resulted in reduction of bearing damage. Use of Belleville springs for this particular rotor-bearing configuration was found undesirable because the collapse load of the springs would be easily exceeded during the anticipated operating conditions.

A bearing load-capacity computer program was written so that, given the bearing parameters, inner race diameter, ball diameter, diametral clearance, total curvature of the race groove and number of balls, the limiting axial load can be determined using maximum Hertz stress and over-riding of the race land as the failure criteria.

STATEMENT OF THE PROBLEM

PLV fans subjected to acceptance vibration tests have developed rough running, noisy bearings. Brinelling of the bearing races was suspected as the cause of rough running. If brinelling has been the cause, then overloading of the bearings from inertial loads has occurred and can only be alleviated by increasing the size of the bearing or reducing the peak inertial loads. If fretting or "false brinelling" has been the cause of rough races, then load capacity of the bearing will have no significance in the serverity of the effect. Brinelling is a plastic deformation process; fretting is a time-dependent wear process and the latter is a function of rubbing amplitude and lubrication.

THE MODE OF ANALYSIS

An analysis was performed on the rotor-bearing system to determine the following:

- (1) The nature of damage to fan bearings (whether it is brinelling or fretting).
- (2) The maximum bearing loads resulting from test random vibration environment.
- (3) Hertz stresses developed in the bearings and displacement of balls in races using the load values obtained in Number (2).
- (4) The influence of bearing preload and preload spring configuration on bearing contact stress levels.

Analysis of Failed Bearings

Samples of both new bearings and bearings from noisy fans were analyzed in the Battelle Lubrication Mechanics Laboratory. Bearings were disassembled and the race surfaces examined by stereoptican microscope. The race surface topography was measured with a Talyrond roundness instrument.

Microscopy revealed classic brinelling on the inner and outer races of bearings taken from tested fans. The following bearings were examined:

(1) <u>Used bearing</u> with much of the grease gone. Severe brinelling over-running the lip of the ball groove. brinelling on one side of the race groove.

- (2) New bearing filled with grease. All surfaces free of defects.
- (3) <u>Used bearing</u> with much of the grease gone. Mild brinelling on inner and outer races.
- (4) <u>Used bearing</u> with some residual grease. One set of brinell marks on inner race and barely visible on outer race.
- (5) <u>Used bearing</u> full of grease. Very light brinell marks on inner and outer races.

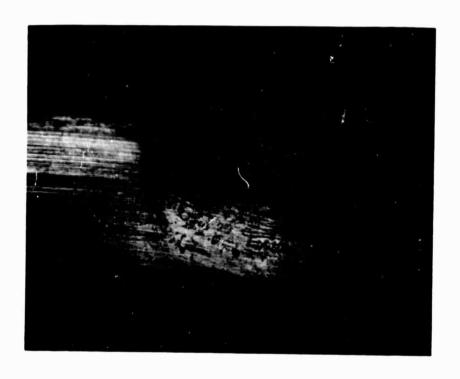
The extent of ball indentation in the severely brinelled bearing is shown in the photomicrograph in Figure 1.

Talyrond measurements were made on several circumferential positions of the inner races of the bearings. The traces were made with the stylus riding (1) on the bottom of the ball groove, (2) on the side of the ball groove, and (3) on the race lands. Typical traces of brinelled bearings are shown in Figures 2, 3, and 4. The location of the tracing stylus is shown in the drawing at the top of each tracing. For instance, the trace in Figure 2a was made on the race land.

Figure 2 shows an example of heavy brinelling. The bearing was so overloaded axially that the balls were driven up the sides of the ball groove and indented the lip causing a mounding up on the race land surface. The mounding up can be seen in Figure 2a. Each mound represents the position of one ball in the bearing. All eight balls have produced indentations. The height of the mounds averaged 0.00025 inch. The indentations in the thrust side of the ball groove can be seen in Figure 3. The trace indicates one set of severe brinell marks together with at least two other sets of lighter brinell marks. Apparently this bearing was subjected to several separate conditions of axial vibration, one of them being most severe. Average depth of the severe indentations was about 0.00025 inch. Width of the indentations at the surface averaged about 0.03 inch. (The proportions of the indentations are distorted on the Talyrond traces because radial magnification is much higher than circumferential.)

The brinell marks did not extend into the bottom of the race groove as shown in the trace in Figure 2b. The trace is smooth, and slightly egg-shaped,

The second secon



75 X

FIGURE 1. SEVERE BRINELLING DAMAGE ON INNER RACE
OF FAN BEARING. THE PHOTOGRAPH SHOWS
ONE BALL INCENTATION EXTENDING OVER
THE RACE GROOVE LIP

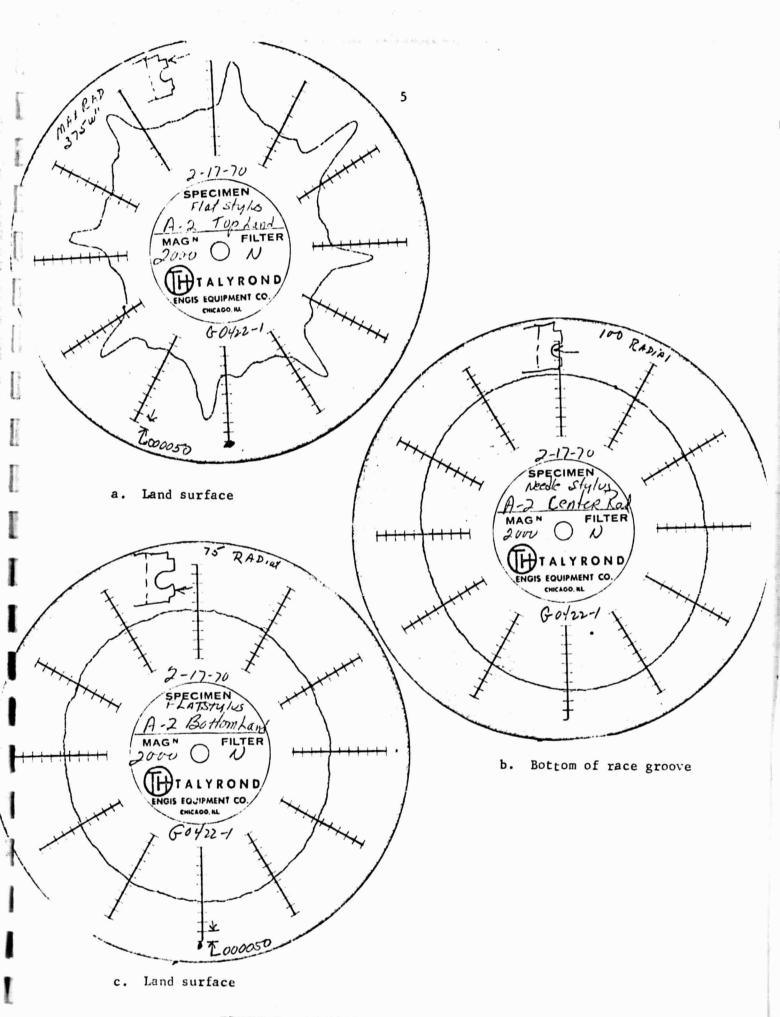


FIGURE 2. TALYROND TRACES OF SEVERELY BRINELLED FAN BEARING INNER RACE

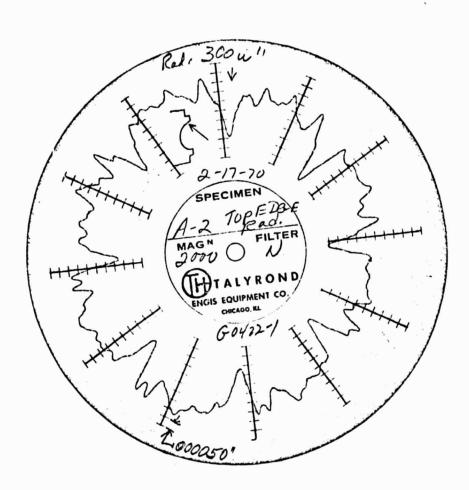
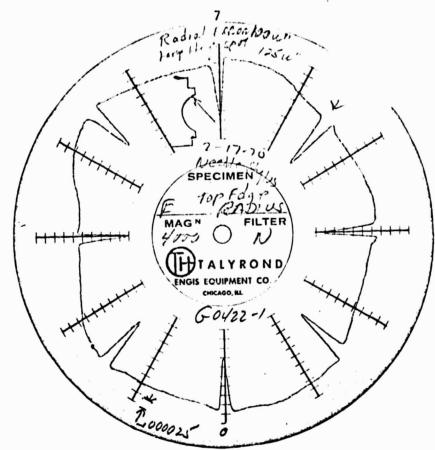
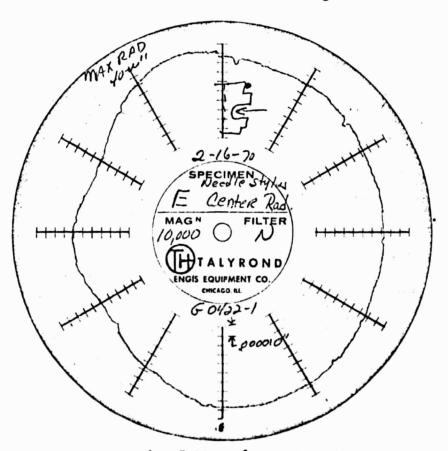


FIGURE 3. TALYROND TRACE OF SEVERELY BRINELLED FAN BEARING INNER RACE TAKEN ON THE THRUST SIDE OF THE RACE GROOVE



,a. Thrust side of race groove



b. Bottom of race groove

FIGURE 4. TALYROND TRACES OF LIGHTLY BRINELLED FAN BEARING INNER RACE

indicating ovality in the race geometry. It was concluded, therefore, that brinelling damage was associated with axial loads and not radial loads.

An example of light brinelling damage is shown in Figure 4. These marks were barely visible under the microscope. Maximum depth of the indentations are about 0.0001 inch. Note that faint indications of light dents show up in the trace made of the bottom of the ball groove. This indicates that the balls were not displaced as far from their no-load position as they were in the bearing exhibiting heavy brinelling. In addition, the trace shows only one set of brinell marks in this bearing. If the bearing was subjected to more than one vibration condition only one was severe enough to produce damage.

It has been concluded from the examination of failed bearings that the damage is true brinelling resulting from inertial overloads and that the conformity and radial play conditions in these bearings allow sufficient relative motion of rolling elements under axial load so that ball over-riding of the race groove lip is possible under heavy enough load.

VIBRATION ANALYSIS

An important objective of this program was to establish an analytical method for estimating maximum bearing loads when the PLV fan was subjected to a random vibration environment. Calculated bearing loads could then be used to compare the predicted results with the experience from tests where bearings have failed, and to evaluate proposed modifications to the bearing support system in order to select a modification most promising for further testing.

Equations for predicting the maximum expected rotor displacements and bearing loads have been derived and a detailed development is included in Appendix A, with numerical examples to demonstrate correct application. These equations are based on certain simplifying assumptions regarding the shape of the power spectrum of the vibration test specifications, as well as the use of a linear spring representation for the shaft bearings and preload springs, which are actually quite nonlinear. Even so, it is believed that this idealized model is a useful design tool.

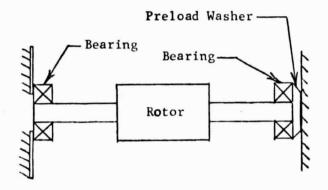
Dynamic Analysis Model

Figure 5a is a sketch of the original configuration of the fan rotor and bearings using one wave washer to obtain a 6 pounds axial preload. Figure 5b shows springs representing the flexibility of the bearings and preload washer, and this is transformed to the equivalent single-degree-of-freedom model shown in Figure 5c. The washer is very flexible relative to the bearings. Therefore, the only contribution of the washer is to establish sufficient preload so that for small motions about the shaft equilibrium position, the total effective stiffness is that of the one bearing that is preloaded against the rigid housing.

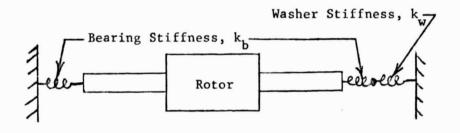
When the shaft deflection exceeds the 0.005 inch preload deflection, the bearing will be unloaded, and the effective stiffness will be that of the wave washer. Figure 6 shows the load-deflection curves for several different preloads to demonstrate how the bearing stiffness is the important parameter in determining the effective axial stiffness. These curves also show the net external force limitations imposed by the 43 pounds maximum axial load capability of the bearings. There will be no significant change in these curves if multiple washers are used to replace the single washer so long as they are all on one end of the shaft and the total preload force is the same.

If, however, washers are installed at both ends of the shaft, Figure 7 shows that the stiffness of the flexible washers will be the determining factor of the total effective stiffness. Figure 8 shows several load-deflection curves for five Belleville springs stacked in parallel at each end of the shaft. The load-deflection characteristics obtained from MSFC Drawing SK20-5072 indicate these springs have a maximum load capability of about 14.5 pounds and Figure 8 shows this collapse load.

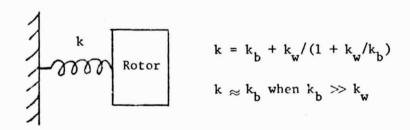
The bearing considered in the analysis was Barden SR4SS, One-quarter-inch bore, angular contact, currently in use. Assuming a limiting maximum Hertz stress of 460,000 psi, the maximum axial load capacity of the bearing was determined as 43 pounds using the computer program described in the next section.



a. Bearing Support Configuration

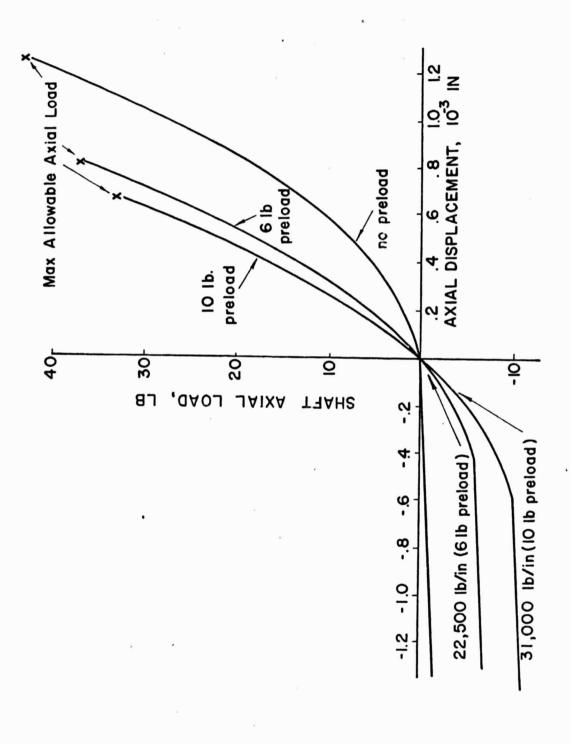


b. Spring Model of Bearing Supports



c. Simplified Model for Dynamic Analysis

FIGURE 5. DEVELOPMENT OF MODEL REPRESENTING THE AXIAL DYNAMICS
OF THE PLV FAN ROTOR ASSEMBLY WHEN PRELOADED BY FLEXIBLE
WASHERS AT ONE END



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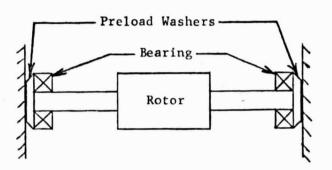
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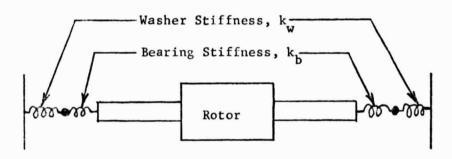
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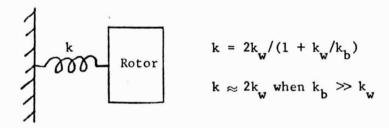
FIGURE 6. AXIAL LOAD-DEFLECTION CURVES FOR FAN SHAFT PRELOADED BY ONE WAVE WASHER AT ONE END



a. Bearing Support Configuration



b. Spring Model of Bearing Supports



c. Simplified Model for Dynamic Analysis

FIGURE 7. DEVELOPMENT OF MODEL REPRESENTING THE AXIAL DYNAMICS OF THE PLV FAN ROTOR ASSEMBLY WHEN PRELOADED BY FLEXIBLE WASHERS AT BOTH ENDS

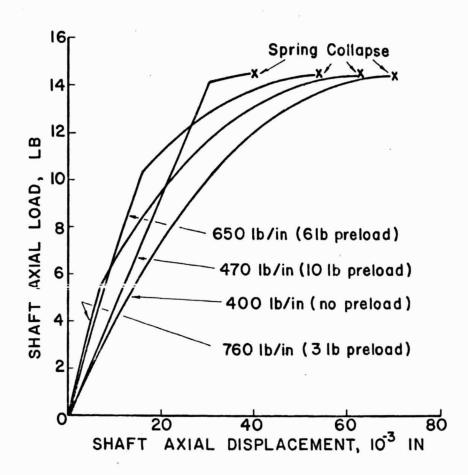


FIGURE 8. AXIAL LOAD-DEFLECTION CURVES FOR FAN SHAFT WITH FIVE BELLEVILLE SPRINGS AT EACH END STACKED IN PARALLEL

Experimental Measurements

In order to predict the maximum expected bearing loads using the equations derived in Appendix A, it is necessary to know the natural frequency and the resonant amplification factor Q of the rotor in the axial direction. The natural frequency can be predicted analytically, at least within the linear system approximations, but the Q of the system must be measured.

In order to measure both the natural frequency and effective damping (Q), the fan housing was clamped rigidly in a heavy vise and a soft rubber mallet was used to tap the rotor and produce a transient vibration. A Kistler Model 802-A piezoelectric accelerometer mounted on the fan impeller was used with a Kistler Model 568 charge amplifier, and the vibration signal was displayed on a Tektronix Type 502 oscilloscope. A Polaroid camera was used to record the transient vibration and the photographs were analyzed to determine the natural frequency and damping.

Table 1 summarizes the measured frequencies and Q factors for three bearing support configurations. Equations A-13 and A-15 in Appendix A were used to calculate the maximum expected bearing loads and rotor displacements. Although the calculated values of bearing loads for the original fan configuration (Configuration I) are probably higher than would be actually measured because of the neglected nonlinearities, it is evident that the bearing loads from the vibration tests are considerably higher than the 43 pounds maximum allowable load. The observed reduction in natural frequency at higher vibration amplitudes was caused by the rotor motion exceeding the washer preload so that the bearings were unloaded for part of each vibration cycle. The photographs of the transient vibration indicated that the preload was exceeded at an amplitude of about 0.004 inches (± 16 g's), which agrees closely with the design specifications for the wave washer.

With one Belleville spring installed at each end of the shaft, the calculated and measured natural frequencies were in close agreement and the damping was increased considerably (lower Q). However, the vibration environment is most severe in the frequency range of 60 to 150 Hz (see Figure A-2) so the predicted bearing loads are excessive and the Belleville springs would collapse.

SUMMARY OF MEASURED ROTOR DYNAMIC CHARACTERISTICS AND PREDICTED BEARING LOADS FOR 100 PERCENT LEVEL VIBRATION TEST SPECIFICATIONS TABLE 1.

11	ı	15	
Maximum Predicted Rotor Displacement, inch	0.0099 0.0087	0.123	060.0
Maximum Predicted Bearing Load, 1b (3)	297(1) 205(1)	118(1,2)	58(1,2)
8	7 7 7 7	::	m
Frequency, Hz Measured	494 (±10 g's) 440 (±20 g's) 385 (±80 g's)	125	100
Rotor Natural Frequency, Hz Theoretical Measured	427 (4)	138	s 73
Configuration	One Wave Washer at one end 6 lb design preload	II. One Belleville Spring each end Approximately 10 lb preload	Five Belleville Springs each end Approximately 6 lb preload
	ï	11.	III.

Exceeds maximum allowable bearing load. £36£

Exceeds Belleville spring collapse load. 100 percent Level Vibration Test Specification (MSFC Memo. S&E-ASTN-EME-69-243). A 10 lb preload gives a 500 Hz theoretical natural frequency.

With five Belleville springs at each end (Configuration 3) the damping was increased considerably. The predicted loads were reduced sufficiently to suggest that the bearings might be capable of surviving the vibration tests, but the Belleville springs would be expected to collapse. It does not appear practical to use Belleville springs with a 14.5 pounds collapse load with bearings which have a 43 pounds allowable load.

Methods for Modifying Bearing Load Levels

Figure A-2 shows that the most severe vibration excitation is in the 60-540 Hz frequency range with reduced levels extending to the 20 Hz low frequency limit and to the 2000 Hz high frequency limit. In order to reduce the fan bearing loads it is obvious from Figure A-2 that to have reduced excititing forces either the natural frequency should be reduced considerably from the 494 Hz resonance of the original design configuration, or the system should be stiffened to increase its natural frequency. One advantage of reducing the natural frequency by supporting the rotor with flexible springs is that this type of modification will increase the damping and reduce the resonant amplification factor Q. If the natural frequency is increased, the damping will be quite low (high Q) as indicated by the measurements, and it is quite difficult to introduce additional damping in a high frequency system.

If it is assumed that Q=5 is reasonable for a softly-sprung bearing mount, the maximum allowable bearing load of 43 pounds can be used to calculate a reasonable frequency suitable for a design goal. Results of this calculation show that the rotor axial natural frequency must be reduced below 55 Hz for the predicted bearing loads to be less than the maximum allowable. For a safety factor of 2, the natural frequency should be below 43 Hz, and the springs must permit the maximum expected rotor displacement of \pm 0.19 inch.

These large relative displacements present practical problems because the radial clearance between the bearing O.D. and the bearing housing should be kept small for satisfactory fan operation. However, the bearings must be quite free axially to obtain the required low natural frequency and large displacements. Slight misalignment or excessive friction could bind the bearings so that while a "soft" support system appears theoretically satisfactory, extreme care in manufacturing and assembly would be required to obtain high reliability using the current blaring mount configuration.

The alternative solution of rigidly mounting the bearings to obtain a high natural frequency eliminates the requirement of providing the axial motion. However, it would probably be necessary to increase both the bearing size and number of bearings. For example, if the highest measured value of Q = 44 is assumed, the natural frequency must be greater than 2000 Hz if the load on a single bearing is reduced below 43 pounds. If a pair of larger preloaded bearings are used with the rating of each bearing doubled to 86 pounds and the bearing stiffness is increased proportionately (actually the bearing stiffness may increase by a greater factor than the load rating), then the rotor natural frequency would be doubled (≈ 1000 Hz). The total predicted bearing force would be 164 pounds or 82 pounds per bearing, which would be acceptable. The maximum rotor displacement would be only 0.0013 inch. The limitation with this solution is that the bearing housing structure must be rigid relative to the bearings, and it is difficult to design a bearing system that will actually have a resonance as high as 1000 Hz without a severe weight penalty.

After examining the alternatives of either a soft, low-frequency bearing system or a stiff, high-frequency system, it does not appear that the present PLV fan configuration can be easily modified in order to pass the vibration tests and operate reliably. It is recommended that if the fan design is revised, the bearings should be soft-mounted in the axial direction with the flexible element attached rigidly to the bearing outer case to eliminate any sliding elements. Damping should either be obtained with the flexible element, such as the hysteresis loss in an elastomer, or a damping device introducing friction could be attached independent of the bearing supports.

Angular Contact Bearing Design Analysis

Essential features of the angular contact bearing design are reasonably well understood. The mathematical analysis of the geometrical parameters influencing the load capacity of the bearing is well detailed in the books by Harris² and Jones³. In estimating the load capacity of the bearing in the axial and/or radial direction the hydrodynamic effects of lubrication

² Tedric A. Harris, "Rolling Bearing Analysis", 1966, John Wiley & Sons, Inc.

A. B. Jones, "Analysis of Stresses and Deflection", Volume 1 and 2, Copyright 1946, New Departure, Division of General Motors Corp.

are generally ignored; the design criteria are based on the Hertz theory of dry-static contact between the rolling elements.

The formulations pertaining to thrust load, applied to single row angular contact bearing, have been employed in the analysis of the PLV for bearings. The appropriate mathematical equations have been adopted from Harris's book. Using thes expressions a computer code was written for the G.E. Time Sharing System (Mark I); this is included and explained in Appendix B. This program was written with the intent of providing a useful tool for analyzing the influence of various geometric parameters on the load-deflection curve of PLV fan bearing. For instance for the bearings currently in use, assuming 57 percent conformity and diametral clearance of 0.00065 inch the value of the axial deflections were computed (computer output I) for various static loads. The load deflection curve is plotted in Figure 9; this curve was used in the dynamic analysis of the previous section.

To illustrate the use of the computer program several additional runs (computer outputs II-V) were made in which ball race conformity and the diametral clearance (contact angle) were selectively varied. To understand the output of the program reference may be made to the illustration in Figure 10. Here α designates the contact angle and θ_0 (θ_i for the inner race) is the minimum angle the outer race must subtend, as shown, in order to avoid the riding of the ball over the land. Ho (Hi for the inner race) designates the minimum land height the outer race must posses in order to prevent ball over riding. All this is true for a given thrust load. The program computes these quantities as well as the maximum Hertz stress for both the inner and

inner race diameter, di = 0.3401 inch (See Figure 10)
ball diameter, D = 0.0937 inch
number of balls = 8
diametral clearance = 0.00065 inch
B, or (fi+fo-1) = 0.14

where fi = inner race conformity fo = outer race conformity

$$\alpha = \cos^{-1} \left(\frac{P_d}{2A}\right)$$
 where P_d = diametral clearance P_d = fo + fi -1

The critical dimensions used in the analysis for the SR4SS bearing are as follows:

Diametral clearance and contact angle are related in the following way:

where fo = outer race groove radius/ ball diameter fi = inner race groove radius/ball diameter.

COMPUTER OUTPUT I

IN PARTS	PLV BEARING MAXIMUM LOAD	CAPACITY
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THRUST LOAD, AXL. DSPL.

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	INNER	• 1337E+06	· 29 0 0E+ 00	• 39 13E-02
	OUTER	• 1108E+06	· 28 74E+00	• 39 13E-62
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	OUTER	· 2129 E+06	• 361 4E+ 66	. 6055E-02
8 . 2224	4.97466E-04			
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	OUTER	• 2439 E+ 06	· 38 61E+00	• 69 0 IE- 02
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COMPUTER OUTPUT I (CONTINUED)

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33. 7027	.0611			
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	OUTER	. 349 1 .+ 66	• 4758 E+ 00	• 1641E-01
37.3662	.0011			Total Control
	INNER	. 4353 E+ 06	. 49 49 E+ 00	. 1125E-01
	OUTER	. 3619 E+ 06	• 48 64E+ 00	- 103 7E-01
18.1313	.0012		The state of the s	
	INNER	. 4500E+06	. 5057E+00	1173E-01
	OUTER	· 3723E+06	• 49 69 E+ @Ø	. 1134E-61
47. (11)	• 6613		A STATE OF THE STA	
1,72 112, (LNNER	. 1639 E+ 06	• 5164E+00	· 1222E-01

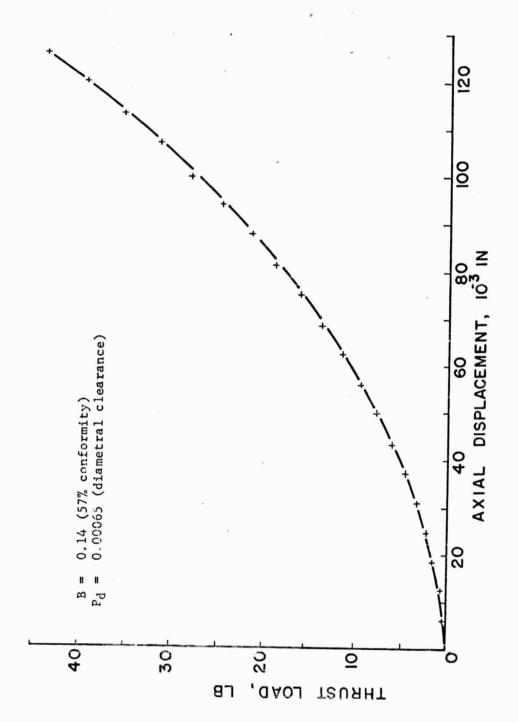


FIGURE 9. LOAD DEFLECTION CURVE FOR PLV FAN BEARING

COMPUTER OUTPUT II

AXLOD 16:07 CY FRI 04/24/70

IN PART2

? 0.340075,0.09375,0.0001,0.04,8,0.0046,460000. INN.RACE DIA.,BALL DIA.,DIA. CLEARANCE, B , NO. OFBALLS AND INCREMENT IN ALFA , MAX HERTZ LIMIT

• 3401 • 69 37 1 • 60000E - 64

• 04

8

•0046 460000•00 SM•ABJ 2•1909 120441•83

THRUST LOAD, AKL. DSPL.

R	ACE . HERTZ F	RESSURE . MIN.	ANGLE AND LAND	UELOUE
@3@3	1.74965E-05	,,,,,,,,,,,,,,,,,,,,,,,,,,,,,,,,,,,,,,,	ATTOLE MID LAND	HEI GHI
	INNER	• 38 12E+Ø5	• 2026E+ 00	. 10 10 5-00
	OUTER	- 3123E+05	·2015E+00	• 19 18 E-02 • 189 6E-02
· 2792	7. Ø1533E-Ø5			10706-02
	INNER	• 779 ØE+ Ø5	• 2525E+ 00	00 705 00
	OUTER	- 638 2E+ 05	·2501E+00	· 29 72E- 02
• 739 2	1 · 23078 E- 04		120012.00	• 29 1 7E- 02
	INNER	• 1052E+06	- 29 11E+00	20 445 00
	OUTER	•8 619 E+05	• 28 79 E+ 00	• 39 44E-02
1 • 4336			120 17 21 00	• 38 58 E- 02
		• 128 3E+06	• 3259 E+ 00	· 49 35E-02
	OUTER	• 1051E+06	- 3220E+00	• 48 18 E- Ø2
2.3985				
	INNER	• 149 1E+ 06	• 3588 E+ 00	• 59 69 E- 02
	OUTER	• 1222E+06	- 3542E+00	• 58 19 E- 02
3.6761	2-83686E-04		00 0 45, 00	20525-00
	INNER	• 168 6E+06	• 39 Ø4E+ ØØ	• 7053E-02
	OUTER	• 138 1E+06	• 38 52E+ 00	• 68 69 E-02
5.3131	3.379 12E-04	10715:0/	- 4212E+00	-8 19 3E-02
	INNER	• 18 71E+06	.4154E+00	. 79 7 4E- 02
	OUTER 3.92522E-04	•1533E+06	E. R. H. W. 185	
7.3607	INNER	. 2050E+06	. 451 4E+ 00	.9 39 1E-02
	OUTER	. 168 0E+06	. 4451E+00	.9134E-02
	4. 47542E- 04			
9.8737	INNER	. 2224E+06	. 48 13E+00	. 1065E-01
	OUTER	. 1822E+06	. 4744E+00	. 1035E-01
12.9116	5. 02998 E- 04	. 239 4E+ Ø6	. 5108 E+00	. 1 19 7E-01
	INNER	· 1961E+06	.5034E+00	· 1163E-01
	OUTER	., .,		

COMPUTER OUTPUT II (CONTINUED)

14.0517	5.21585E-04		1 1 2	**
- 78000, 1-20, 3911, 240, 571	INNER	. 245ØE+Ø6	• 5206E+00	-1242E-01
	OUTER	. 2007E+06	.5130E+00	-1207E-01
15.2595	5. 40224E-04			
	INNER	. 2506E+06	• 5304E+00	- 1288 E-01
	OUTER	· 2053E+06	. 5226E+00	- 1251E-01
16.5377	5.58917E-04			
	INNER	. 2562E+06	5402E+00	-1335E-01
	OUTER	· 2099 E+ 06	•5322E+00	• 129 7E-01
20.8201	6. 15326E-04			
	INNER	•2727E+06	• 569 4E+00	-1479 E-01
	OUTER	· 2234E+06	• 5609 E+00	• 1436E-@1
25.8313	6. 72252E-04			
	INNER	· 289 1E+06	• 598 5E+ ØØ	.1630E-01
200 00000	OUTER	• 2368 E+ Ø6	• 589 5E+ 00	- 158 2E-01
31.6488	7. 29 72 6E- Ø4			
	INNER	· 3053E+06	• 6276E+00	. 178 7E-01
	OUTER	. 2501E+06	• 618 ØE+ ØØ	1734E-01
38 • 3552	7.87778 E- Ø4		45.455.44	
	INNER	- 3215E+06	• 6567E+ 00	• 19 50E-01
	OUTER	· 2634E+06	• 6465E+ 00	• 1892E-01
46.0354	8 • 46438 E- 04			
	INNER	· 3376E+06	• 68 58 E+ 00	.2120E-01
F / B06	OUTER	·2766E+06	• 6751E+00	.2056E-01
54.792	9 • 05739 E- 04			
	INNER	• 3537E+06	• 7150E+00	· 229 6E-01
	OUTER	• 2898 E+06	• 7036E+00	.2227E-01
64. 7157	9 · 65715E-04	0.400 0.04		
	INNER	• 3698 E+ Ø6	• 7442E+ 00	•2478E-01
75.9155	OUTER • 001	- 3030E+06	• 7323E+00	·2403E-01
1309133	INNER	• 38 59 E+ Ø6	• 7735E+00	-2667E-01
	OUTER	• 3162E+06	• 7610E+00	• 258 6E-01
60 6000	• 0011	• 31022+00	• 10102.00	*250 02 01
88 • 5039	INNER	. 4020E+06	-8 029 E+ 00	· 28 63E-01
	OUTER	. 329 4E+ 06	. 7898 E+ ØØ	. 2775E-01
102 - 6009	• 0012			
102.000	INNER	. 418 2E+ 06	·8325E+00	.3065E-01
	TIMINE			

USED 60.00 UNITS

COMPUTER OUTPUT III

-1178

8

INN. RACE DIA., BALL DIA., DIA. CLEARANCE, B, NO. OFBALLS AND INCREMENT I N ALFA. MAX HERTZ LIMIT • 09 37 1 • 00000E - 04

.3401

.0046 460000.00 SM. ABJ

2.7668 428927.06

THRUST LOAD, AXL. DSPL.

LOAD ML.	05/6			
RACE .	HERTZ PRESSUR	E . MIN. ANG	F AND LAND	HFIGHT
· 0287 5.10			ac into critto i	
INN		+ 05 .12	63E+00	• 7465E-03
OUT				• 7333E- Ø3
· Ø8 79 1 · Ø2			000.00	10002 20
INN		+ 05 • 14	23E+00	•9 477E-03
	ER • 6559 E			•9263E-03
• 1746 1• 53			012.00	• / E 00 E - E 0
INN		+05 -15	60E+00	- 1138 F- 02
	ER •8126	7 17		1109 E-02
	512E-04		402.00	
INN		C+06 • 168	3 5E+ 00	• 1327E- Ø2
	ER •9 49 ØE			• 129 1E-02
	768 E- 04		0.2.00	
INN		+06 • 18	02F+00	• 1519 E- 02
				• 1475E-02
	08 1E- 04		TOE. DO	
INN		+06 • 19	15F+00	.1714E-02
OUT				• 1662E-02
	452E-04		00.00	· rooll bl
INN		+06 .20	24E+00	. 19 1 4E- 02
OUT				• 18 55E- Ø2
	88 4E- Ø4			10000
INN		+06 •21	31E÷00	.2120E-02
OUT				• 2052E-02
	378 E-04			
INN		+06 •22	35E+00	- 2332E-02
OUT	ER - 1502E			. 2255E-02
1.7209 5.12	9 38 E- 04			
INN	ER • 1934E	+06 .23	37E+00	. 2549 E- 02
OUT	ER • 1599 E	+06 +229	88 E+00	. 2464E-02
2.1115 5.64	1565E-04			
INN	ER • 2049 E	+06 .24	38 E+00	.2773E-02
OUT	ER • 169 4E	+06 .23	7E+00	• 2679 E- 02
2.5548 6.16	262E-04	901		
INN	ER .21618	+06 •25	38 E+00	.3004E-02
OUT	ER • 178 6E	+06 .249	4E+00	. 29 Ø1E-Ø2
3. 0546 6.68	031E-04			
INN	ER .22718	+96 •26	37E+00	. 3241E-02
OUT	ER • 18 77E	+06 •259	1E+00	-3128 E-02
3 • 61 48 7 • 19	874E-04			
INN		+06 .27	35E+00	• 348 4E-02
- OUT	ER • 19 67E	+06 .26	3 6E+ ØØ	- 3362E-Ø2
4.2397 7.71	79 4E-04		*	
INN		+06 •28	32E+00	• 3735E-02
OUT	ER . 2055E	+06 .278	3 1 E+ 00	-3603E-02
	179 4E- 04			
INN			29 E+ ØØ	• 3992E-02
OUT		+06 •28	76E+00	• 38 49 E- Ø2
	875E-04			
INN				4256E-02
0U T	ER • 2227	+06 •29	69 E+ 00	• 4103E-02

COMPUTER OUTPUT III (CONTINUED)

			III (CONTINUED)	
6.5445	9 . 28 040E-04	K.		
			• 3120E+00	- 452 6F- 02
	OUTER	· 2312E+06	• 3063E+00	• 4363E-02
7.4712			000002.00	43032-02
,,,,,,	INNER		• 3215E+00	. 48 0 4E- 02
		• 239 6E+ Ø6	• 3156E+ ØØ	· 4629 E - 02
8 • 48 5	.001			
			· 3310E+00	
		. 248 ØE+ Ø 6	• 32 48 E+ 00	· 49 Ø 3E - Ø 2
9 • 59 09	.0011			
	INNER	. 3099 E+ 06	. 3404E+00	. 538 ØE- Ø2
		· 2562E+ Ø6		
10.7939			100-112-20	101011
100 17 07			- 3498 E+ 00	E (92 E - 00
10 0000		• 2645E+ Ø6	• 3433E+00	• 5 4 69 E - 02
12.0993				
	INNER	• 329 7E+ Ø6	• 359 2E+ ØØ	
	OUTER	· 2726E+06	• 3524E+00	.5762E-02
13.5124	.0012			
			• 368 6E+ 00	• 629 6E- Ø2
	OUTER	- 28 Ø8 E+ Ø6	.3616E+00	· 6062E-02
15.0388			100102.22	ODOZE DZ
134 2330			2770 5 . 66	//155 00
	OUTED	• 349 4E+ 06	• 3779 E+ 00	
	OUTER		- 3707E+00	• 6368 E- Ø2
16 • 68 41				
	INNER	• 359 1E+ Ø6		
		• 29 69 E+ Ø6	• 3798 E+ ØØ	.668 2E-02
18 • 4542	.0014			
	INNER	• 3688 E+ Ø6	• 39 65E+ 00	.7275E-02
		- 3049 E+ 06	• 3889 E+ ØØ	.7002E-02
20.355	.0015			
			• 4058 E+ 00	. 7615E= 00
		• 3129 E+ Ø6	• 398 ØE+ ØØ	• 7328 E- 02
22 • 39 27	OUTER	• 3127 ET EO	• 376 PET DE	• 1320 E- 02
22. 3721				
			• 4151E+00	
-		• 3209 E+ Ø6	· 4071E+00	• 7662E-02
24.5735	-0016			
	INNER	· 39 77E+ Ø6	• 4244E+ 00	·8317E-02
	OUTER	• 3288 E+ Ø6	• 4162E+00	·8002E-02
26.9041	.0016			
	INNER	• 4073E+ Ø6	• 4337E+ ØØ	.8 678 E- Ø2
	OUTER	• 3367E+ Ø6	• 4252E+00	•8 349 E- Ø2
29 • 39 08	• 0017	• 00072.00	*4232E. 00	*0 347 L- DZ
27 • 37 63		41 (0 Ft 0 (4 400 5 . 00	0.0.425-00
		• 41 69 E+ Ø6		•9 Ø47E- Ø2
		• 3447E+ Ø6	• 4343E+ 00	•8 703E-02
32.0407	.0017			
		· 4264E+ Ø6	• 4522E+00	•9 422E-02
	OUTER	• 3525E+Ø6	• 4433E+ 00	•9063E-02
34.8605	• 0018			
	INNER	. 4359 E+ Ø6	. 461 4E+ 00	.98 Ø5E- Ø2
	OUTER	. 3604E+06	. 4524E+00	.9 430E-02
37.8574	• 0018			
31.0314	INNER	. 4454E+ 06	. 4707E+00	- 1019 E-01
44 6600	OUTER	• 368 3E+ Ø6	• 461 4E+ 00	•9804E-02
41.0388	• 0019			
		• 45 49 E+ Ø6	• 4799 E+ 00	• 1059 E-01
		. 3761E+06	• 4705E+00	• 1018 E-01
44. 412	.0619			transfer of the same
	INNER	. 4644E+Ø6	• 489 2E+ 00	- 1099 E-01

COMPUTER OUTPUT IV

INN. RACE DIA., BALL DIA., DIA. CLEARANCE, B , NO. OFBALLS AND INCREMENT IN ALFA , MAX HERTZ LIMIT

• 3401 • 093 • 0046 460000• 00 • 09 37 8 • 00000E - 04 • 04

2 • 19 1 120439 • 37

THRUST LOAD, AXL. DSPL.

T LOAD,	AXL.	DSPL.					
	DACE .	UFDTT	PRESSURE .	MTAL	ANG E A	ND I AND	UELOLE
		3546E- 05		MI IA	ANGLE A	ND LAND	HEIGHI
• 4317		IER			6 200 F4	00	100.05-01
		TER			• 5288 E+		• 129 ØE-Ø1 • 128 ØE-Ø1
1.0455		001E-0			• 3255 ET	66	• 120 BE-BI
1 . 2.433		IER			. 540454	00	-1435E-01
		ER			• 5578 E+		• 1421E-01
2 • 3326		378 E- ØS			• 33 16 64	6 B	• 14216-01
240020	INN		•1157E+06		- 58 AR F+	99	.1558E-Ø1
		TER	•9 477E+05		• 58 13E+		.1540E-01
3.6605		69 3E- ØS			. 30 102	D D	13466-61
0.0003		IER .			- 6061F+	aa	-1670E-01
		ER	• 1098 E+ 06		.6021E+		.1648E-01
5.214		961E-05					
	INN		. 1504E+06		• 6257E+	00	.1776E-01
	OU?		- 1232E+06		.6211E+		-1751E-01
6.9849		1520E- 0					
	INN	IER	. 1653E+06		. 6439 E+	00	- 18 77E-01
	out		- 1354E+06		. 6388 E+		- 18 49 E-01
8 . 9 69 2	1.37	442E-0	4				
	INN	IER	. 179 1E+ 06		.6612E+	00	- 19 76E-01
	OU?	TER	- 1468 E+ 06		.6557E+	00	- 19 44E-01
11-1656	1.57	464E-0	4				
	INN	IER	. 1922E+06		. 6778E+	00	.2072E-01
	OU:	TER	• 1574E+ Ø6		.6719E+	00	-2038E-01
13.574	1.77	158 7E- 0					
	INN						.2167E-01
	001				• 68 75E+	00	.2129E-01
16. 19 59							
	INN						. 2261E-01
	ou		• 1772E+06		• 7026E+	00	- 5556E-01
19.0335		146E-0					
		NER					-2354E-01
00.0204		TER			• 71 73E+	00.	-2310E-01
22.0396		58 4E- 0					01115 01
		VER	• 238 6E+ Ø6		• 739 1E+		.2446E-01
25.3679			• 19 55E+06		• 731 7E+	000	.2399E-01
23.3617		I I SUE- D	• 249 2E+ 06		• 7535E+	00	05005-01
		TER	• 2042E+06		• 7535E+		•2538E-01 •2488E-01
28 - 8 722		78 7E- Ø			• 1435 ET	66	• 2450 E-01
23.01.22		VER	• 259 5E+ Ø6		. 767754	00	. 9630F=01
			•2126E+06		• 759 6E+		.2577E-01
32.6069		0555E-0			13702	00	· 23/12-01
011000	INN		•2696E+06		. 78 1 7F+	aa	.2721E-01
		TER			. 7732E+		.2666E-01
36.5766		437E-0					
	INN				• 79 54E+	00	-28 13E-01
			. 2289 E+ Ø6				
40.7866		2434E-0					
		ER			-8 09 0E+	00	. 29 0 4E-01
	OU:	TER	. 2368 E+ Ø6				
45.2419		3549 E- 0					
	INN		• 298 4E+ Ø6		.8225E+	00	. 299 6E-01
	0111	CCD	OAAFE. OC		0 12154	0.0	00 000 01

COMPUTER OUTPUT IV (CONTINUED)

49 • 9 48 3	3-8 478 2E-04			
	INNER	- 3076E+06	•8 358 E+00	-3088E-01
	OUTER	.2520E+06	-8261E+00	.3021E-01
54-9115				,
	INNER	. 3167E+06	.8 489 E+ 00	-318 ØE-01
	OUTER	. 259 5E+ Ø6	•8 389 E+ ØØ	.3110E-01
60.1376	4.27615E-04			
	INNER	· 3256E+06	•8 620E+00	. 3273E-01
	OUTER	· 2668 E+ Ø6	•8517E+00	. 3200E-01
65 6329	4. 49217E-04		/	
	INNER	· 3344E+06	-875@E+@@	. 3366E-01
	OUTER	.2740E+06	·8644E+00	. 3289 E-01
71 - 4039	4.709 47E-04			
	INNER	· 3432E+ 06	•88 79 E+ 20	. 3459 E-01
	OUTER	• 28 12E+ 66	•8 769 E+00	. 338 ØE- Ø1
77.4574				
	INNER	. 3518E+06	•9 007E+ 00	.3552E-01
	OUTER	· 288 2E+ 06	•889 4E+00	. 3470E-01
83.8003				
	INNER	• 3603E+06	•9134E+00	.3646E-01
	OUTER	• 29 52E+ Ø6	•9 019 E+ 00	.3561E-01
90.4398				
	INNER	• 368 7E+ Ø6	•9261E+00	-3741E-01
07 22 24	OUTER	. 3021E+06	•9142E+00	-3653E-@1
97.3331	5.59177E-04	02205.07	0.02.25.40	03.045.04
	INNER OUTER	• 3770E+06 • 3089E+06	•9 38 7E+ 00	• 38 36E-01
104.638	5-81573E-04	• 3009 E+ 10	•9266E+00	• 3744E-01
164.030	INNER	• 38 5 3E+ 06	•9513E+00	- 39 31E-01
	OUTER	• 3157E+ 86	•9 388 E+ 00	• 38 3 7E- 01
112.2121	6. 04109 E- 04	• SISTET DO	• 7 36 6 E + WE	• 30 3 /E- 61
	INNER	• 39 35E+ 06	•9 638 E+00	- 4027E-01
	OUTER	• 3224E+66	•9510E+00	• 39 30E-01
120.1135	6-26787E-04	OLL AL. BO	173102.22	• 47 002 01
	INNER	. 4016E+06	•9 763E+00	. 4124E-01
	OUTER	. 329 ØE+ Ø6	•9632E+00	. 4023E-01
128 - 3505	6. 49 61 0E-04			
	INNER	. 409 7E+06	.988 7E+00	. 4221E-01
	OUTER	. 3357E+06	.9 754E+00	. 4117E-01
136.9313	6.72579 E-04			
	INNER	. 4177E+06	- 1001E+01	. 4319 E-01
	OUTER	. 3422E+86	•9875E+00	- 4212E-01
145.8647	6.95698E-04			
	INNER	· 4256E+06	• 1014E+01	. 4417E-01
	OUTER	· 3488 E+06	•9996E+00	- 4307E-01
155 • 159 5	7. 189 68 E- 04			
	INNER	• 4336E+06	• 1026E+01	. 4516E-01
	OUTER	• 3552E÷ Ø6	• 1012E+01	· 4402E-01
164-8249	7. 4239 3E-04			
	INNER	. 4415E+06	• 1038 E+01	.4616E-01
	OUTER	.3617E+06	-1024E+01	. 4499E-01
174.8701	7. 659 74E- 04	440.00.04	10515.51	
	INNER	• 449 3E+ 06	• 1051E+01	. 4716E-01
105 0040	OUTER	• 368 1E+06	• 1036E+01	• 459 6E-@1
185.3047	7.89714E-04 INNER	. A571F+04	10405+01	49 1 25 - 61
	OUTER	• 4571E+06 • 3745E+06	• 1063E+01 • 1048 E+01	• 48 1 7E - 01 • 469 3E - 01
196-1385	8 - 13617E-04	• 3143E+06	• 1040 E-01	• 407 36-61
., 0-1000	INNER	. 4649 E+06	• 1075E+01	. 49 18 E- Ø1
	Time En			> 10 5- 01

COMPUTER OUTPUT V

AXLOD 15:27 CY FRI 04/24/70

IN PART2

? 0.340075, 0.09375, 0.00080, 0.1178, 8, 0.0046

INN. RACE DIA., BALL DIA., DIA. CLEARANCE, B, NO. OFBALLS AND INCREMENT IN ALFA

.3401

• 09 37 8 • 00000E-04 • 1178 8

. 0046

SM. ABJ

2.7669 428916.72

THRUST LOAD, AXL. DSPL.

• 3767	RACE . HERTZ 5-27779 E-05	PRESSURE, MAX.	ANGLE AND LAND	HEI GHT
		•9 367E+05	- 319 3E+00	-4740E-02
		• 7746E+ 05	· 3174E+00	. 468 4E- 02
1.0984	1-0569 3E-04			146042 52
		• 1331E+06	. 3428 F+ 00	.5455E-02
		• 1101E+06	.3401E+00	•537ØE-Ø2
2.0796	1 - 58 747E- 04			100162 62
		• 1638 E+06	-3621E+00	. 603 0E-02
	The state of the s	• 1354E+06	• 3588 E+ ØØ	• 59 69 E- Ø2
3.2984	2 · 119 45E-04			
		• 1900E+06	. 379 3E+00	.6663E-02
		• 1571E+06	- 3754E+00	. 6529 E- Ø2
4. 7473	2 · 6528 7E-04			
		-2134E+06	. 39 51E+00	.7224E-02
		• 1765E+06	. 39 08 E+00	- 7067E-02
6. 4246	3. 18 778 E- 04	1		
	INNER	- 2349 E+06	- 4100E+00	.7772E-02
		• 19 42E+06	. 4052E+00	. 759 3E- Ø2
8 • 3323	3.72420E-04			
	INNER	. 2549 E+06	. 4243E+00	·8312E-02
	OUTER	. 2108 E+06	. 419 1E+00	·8112E-02
10-4741	4.26216E-04	,		
	INNER	• 2738 E+06	. 438 ØE+ ØØ	-88 48 E- 02
	OUTER	· 2264E+06	· 4323E+00	-8 62 6E- Ø2
12.8552	4.8 0169 E-04			
	INNER	· 29 17E+Ø6	. 4512E+00	-9 38 2E-02
	OUTER	· 2412E+06	. 4452E+00	•9 139 E-02

COMPUTER OUTPUT V (CONTINUED)

15. 48 18	5.34282E-04			
	INNER	. 3089 E+06	. 4641E+00	.9917E-02
	OUTER	.2554E+Ø6	. 4578 E+00	.9 652E-02
18 . 361				
	INNER	- 3255E+06	• 4767E+00	• 1045E-01
	OUTER	.2691E+06	• 4700E+00	-1017E-01
21.5004	6.43000E-04			
	INNER	.3415E+06	· 489 1E+00	. 1099 E-01
	OUTER	· 23 2 4E+ Ø6	· 48 20E+00	-1068 E-01
24.9231	6.97611E-04			
	INNER	.3570E+06	.5012E+00	•1153E-01
	OUTER	· 29 52E+ Ø6	• 49 38 E+ 00	.1120E-01
28 • 5927	7. 5239 4E- 04			
	INNER	. 3722E+06	.5131E+00	. 1207E-01
	OUTER	· 3077E+06	• 5055E+00	· 1172E-01
32.5632	8 • 07353E-04			
	INNER	• 38 69 E+ Ø6	• 52 49 E+ 00	.1262E-01
	OUTER	· 3200E+06	• 51 69 E+ ØØ	-1225E-Ø1
36.8291	8 . 6249 DE-04			
16	INNER		•5366E+00	• 1317E-01
	OUTER	• 3319 E+ 06	• 528 3E+00	• 1278 E-01
41.40	9 - 178 09 E- 04			
	INNER	· 4156E+06	. 548 1E+00	• 1373E-Ø1
	OUTER	• 3437E+06	• 539 5E+ ØØ	• 1331E-01
46.2861	9 • 73313E-04			
	INNER	· 429 5E+06	• 559 5E+ ØØ	• 1 429 E-01
_	OUTER	• 3552E+Ø6	• 5506E+00	- 138 5E-01
51.4977				
	INNER	· 4433E+06	• 5703 E+00	
	OUTER	• 3665E+06	• 5 61 6E+ ØØ	-1440E-01
57.0454				
	INNER		• 58 20E+00	
	OUTER	• 3777E+ Ø6	•5725E+00	• 1 49 5E-01
62.9403				
	INNER	. 4701E+06	• 59 31 E+ 00	. 1601E-01

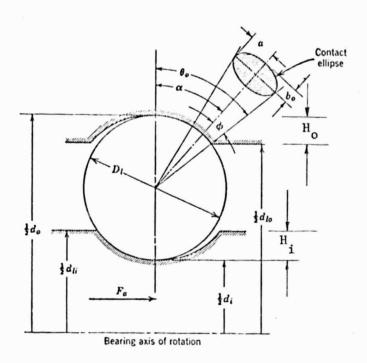


FIGURE 10. BALL RACEWAY CONTACT UNDER LIMITING THRUST LOAD

outer race, at the specified increments in α . It continues to do so until the maximum Hertz pressure at the inner race (or outer race) exceeds the limiting value specified through input, at which point it terminates further calculations. The termination of calculations is also affected if the required value of θ_0 (or θ_i) exceeds $\pi/2$.

In the computer output II and III it may be observed that for a limiting Hertz stress of 460,000 psi the load capacity of the bearing is reduced from approximately 102 pounds to 44 pounds as the parameter B, or (fo + fi - 1) is increased from 0.04 to 0.1178, keeping all other geometrical parameters constant. Increasing B reduces ball-race conformity or the closeness of fit between ball and race groove.

Computer outputs IV and V for the same calculations are repeated for an 8-fold increase in diametral clearance. These results indicate that the reduction in conformity level (or increase in B) reduces the load carrying capacity of the bearing, whereas an increase in diametral clearance (which increases contact angle) increases the load capacity with a limit imposed by race land height. Thus, by changing the contact angle with the present size bearing, an increase in load capacity is possible without changing bearing size. It must be kept in mind, however, that increasing contact angle will also increase ball spin and heat generation in the bearing. This effect will tend to reduce the endurance life of the bearing in terms of lubricant degradation and ball wear.

RECOMMENDED PROCEDURE FOR CHOICE OF BEARINGS IN SIMILAR FUTURE APPLICATIONS

The analysis of the present PLV fan bearings has demonstrated the importance of the dynamical considerations in the choice of bearings for a similar application. At the same time, one is forced to recognize that it is not possible to outline a simple step-by-step procedure that would end in the selection of a bearing satisfying all the design requirements. As is typical of such problems the designer must consider the influence of various parameters on the outcome before arriving at the final optimum design. The next few paragraphs present a plan of analysis to arrive at such an optimum design.

It is assumed that the mass of the rotor is known. It is also assumed that Power Spectral Density (PSD) of the vibration environment to be imposed on the rotor, is known. It is desired to determine the maximum axial load that would be borne by the bearing under fairly severe conditions. Based on this estimate, the geometry of the angular contact bearing would be selected so as to avoid the brinelling of the race as well as balls over-riding the race land. To facilitate the choice of the bearing it is necessary that this load-limit be as small as possible.

Equation (A-14) relates the axial load on the bearing with the stiffness of the system, its damping, and the maximum expected amplitude of vibration on the bearing-rotor assembly. The expected amplitude of vibration, for the white-noise approximation, is in turn related to the maximum value of the PSD (W_0) resonant frequency (f_n) and the magnification factor (Q) as given by (A-13). If the axial displacement of the rotor must be kept at an absolute minimum (from the viewpoint of design limitation) then the resonant frequency of the system must be chosen past the higher end of the PSD spectrum to minimize the value of W_0 . This requires that stiffness of the system be high which would in turn require that the support system must be made very rigid. The latter requirement may be difficult to realize if the stipulated resonant frequency is of the order of 2000 HZ and over; in this case a compromise will have to be made, such as, allowing for larger amplitude-limit and perhaps even larger loads on the bearing. If, however, the required resonant frequency can be obtained with a reasonably rigid support system the reduction of the load-limit on the bearing would be easily achieved in the high-frequency end of the PSD spectrum.

On the other hand, if the axial displacement is not the limiting quantity, the load-limit on the bearing can be reduced by going to the low-frequency-end of the PSD spectrum. Below a certain level of the resonant frequency the value of W_O will decrease. The low resonant frequency can be designed by suspending the rotor-bearing system in a soft suspension system (e.g. five springs in the present PLV fan bearings) such that the preloaded bearing acts essentially as a rigid member. For the suspension system use can be made of thick elastomeric packing material which would provide high damping factor as well. It is, of course, necessary that the "collapse-

load" of the soft suspension system be at least as high as the maximum load capacity of the bearing.

Once the load limit of the bearing has been arrived at, in the above manner, the design parameter for the angular contact bearing can be established by the computer program described in the preceding section. To guard against brinelling it is recommended that maximum Hertz pressure be kept below 460,000 psi under all conditions. This would then provide a limiting criteria for the geometrical parameters of the bearings to be selected.

CONCLUSIONS AND RECOMMENDATIONS FOR FUTURE WORK

The following conclusions have been drawn from the analysis of the PLV fan bearing problem:

- (1) The bearing failure mode is vibration induced race brinelling, the damaging loads occurring in the axial direction.
- (2) Maximum bearing load can be reduced by decreasing the natural frequency of the rotor-bearing-preload spring system. This can be accomplished by mounting preload springs in series at both bearing supports on the rotor.
- (3) A computer program, written to calculate maximum load capacity based on brinelling mode of failure, has demonstrated that bearing load capacity can be increased by altering contact angle and ball-race conformity.
- (4) A system has been established for selection of ball bearings for similar rotor support problems in vibration environment (assuming the Power Spectral Density is given).

FUTURE WORK

It was found that the criterion for brinelling damage in ball bearings is based on an arbitrary maximum Hertz stress level. This level has been arrived at in bearing technology by static load tests on ball-race configurations. For applications where minimum bearing size is required a more accurate criterion should be established. We recommend that selected rolling contact bearings be subjected to vibration evaluations in which the maximum bearing load as determined from the analysis in this summary be varied over a range selected to cover nonbrinelling and brinelling levels. The extent of damage would then be evaluated on the basis of noise level during rotation under steady state load.

APPENDIX A

ANALYSIS OF FAN BEARING LOADS FROM RANDOM VIBRATION

APPENDIX A

ANALYSIS OF FAN BEARING LOADS FROM RANDOM VIBRATIONS

The dynamic response of the fan rotor assembly in the axial direction can be estimated by using the linear single-degree-of-freedom model shown in Figure A-1.

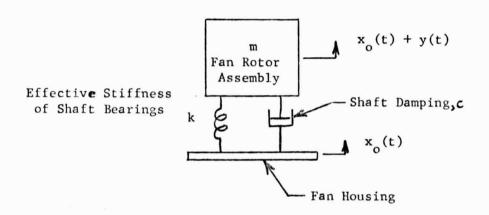


FIGURE A-1. SINGLE-DEGREE-OF-FREEDOM MODEL FOR DYNAMIC ANALYSIS

The axial motion of the rotor relative to its housing is of primary interest and the equation of motion is

$$m\ddot{y} + c\dot{y} + ky = -m\ddot{x}_{0}, \qquad (A-1)$$

which is often written in the form

$$\ddot{y} + \left(\frac{w_n}{Q}\right) \dot{y} + w_n^2 y = -\ddot{x}_0. \tag{A-2}$$

where

$$\omega_{\rm p} = (k/m)^{0.5}$$
 natural frequency (A-3)

$$Q = m \omega_n / c \tag{A-4}$$

y = axial displacement of the fan rotor relative to fan housing. m = fan rotor assembly mass

k = effective axial stiffness of the rotor bearing supports

 $x_0 = axial displacement of the fan housing.$

The solution of Equation A-2 for random vibration can be expressed as (see Reference A-1 or other standard texts)

$$W_{y}(f) = |H(f)|^{2}W_{o}(f), \qquad (A-5)$$

$$|H(f)|^2 = \frac{1/(4\pi^2)^2}{(f_n^2 - f^2)^2 + (f_n^2 f^2)/Q^2},$$
 (A-6)

where

f = excitation frequency, Hz

 $f_n = rotor natural frequency (<math>w_n/2\pi$), Hz

 $W_{v}(f)$ = displacement response, power spectral density

W (f) = acceleration excitation, power spectal density,

This gives the response as a function of frequency. In order to be useful, an estimate of the maximum expected displacement of the rotor relative to its housing is needed. This can be obtained by first calculating the root mean square (rms) of the displacement response from

$$y_{rms} = \sqrt{y^2} = \left[\int_0^\infty W_y(f)df\right]^{\frac{1}{2}}$$
, (A-7)

and then making a statistical approximation that the maximum expected rotor displacement, y_{max} , will be no greater than three times the rms displacement.

$$y_{\text{max}} = e(y_{\text{rms}}). \tag{A-8}$$

Reference A-1. Crandall, S. H., and Mark, W. D., "Random Vibration in Mechanical Systems", Academic Press, 1963, pp 77-80

Therefore, the problem is reduced to evaluating the integral in Equation A-7 using the relations in Equations A-5 and A-6 and the power spectral density (PSD) of the vibration environment for the fan housing shown in Figure A-2. However, in order to avoid evaluating a complicated integral, the result is usually approximated by replacing the complex spectrum shown in Figure A-2 by a constant spectrum for white-noise with the amplitude determined by the amplitude of the real PSD at the natural frequency of rotor. The validity of this approximation is based on the knowledge that a low-damped single-degree-of-freedom system is highly responsive to excitation at frequencies very close to its natural frequency and relatively insensitive to excitation at frequencies much lower or higher than its natural frequency. Thus, the response can be predicted quite accurately by using a constant value of the excitation spectrum so long as this is an accurate level in the resonant bandwidth of the rotor. This resonant bandwidth depends on the damping and can be checked by

$$f_1 = f_n \left(1 - \frac{1}{2Q} \right),$$
 (A-9)

$$f_2 = f_n \left(1 + \frac{1}{20} \right),$$
 (A-10)

where f and f define the frequency bandwidth of interest centered at the rotor natural frequency $\mathbf{f}_{\rm n}$.

Using the approximation of replacing a complex PSD by a constant level white-noise spectrum with the excitation levels identical at the rotor natural frequency makes it possible to use a standard result from vibration texts (see Reference A-1). Equation A-7 then becomes

$$y_{\rm rms} = \left[\frac{W_0 g^2 Q}{32\pi^3 f_n^3} \right]^{\frac{1}{12}}$$
 (A-11)



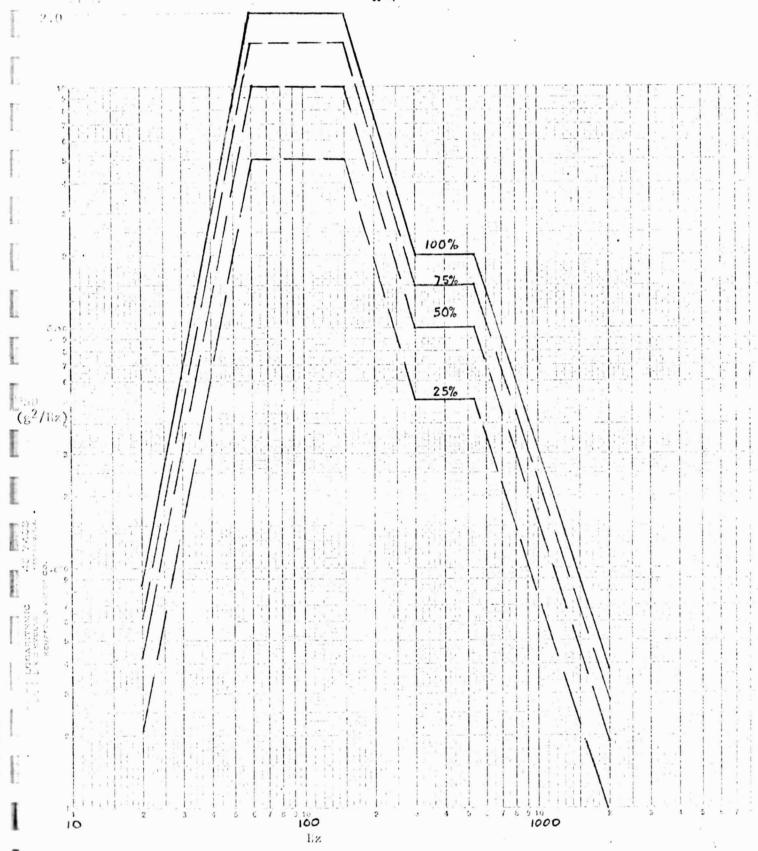


FIGURE A-2. POWER SPECTRAL DENSITY OF VIBRATION ENVIRONMENT FOR PLV FAN (REFERENCE MSFC MEMORANDUM S&E-ASTN-EME-09-243)

$$y_{\rm rms} = 12.26 \left[\frac{W_{\rm o}^{\rm Q}}{f_{\rm n}^{\rm 3}} \right]^{\frac{1}{2}}$$
 (A-12)

and

$$y_{\text{max}} = 36.78 \left[\frac{W_0}{f_0^3} \right]^{\frac{1}{2}}$$
 (A-13)

where

 $W_{o}(f_{n})$ = test PSD at rotor natural frequency, g's/Hz g = acceleration of gravity, (386 in./sec²) f_{n} = rotor natural frequency, Hz Q = rotor amplification factor.

The maximum force which will be transmitted between the housing and the otor through the bearings $\mathbf{F}_{\mathbf{b}}$ is given by

$$F_b = k \sqrt{1 + \frac{1}{0^2} (y_{max})},$$
 (A-14)

or

$$F_b = (2\pi)^2 \text{ mf}_n^2 \sqrt{1 + \frac{1}{0^2} (y_{\text{max}})}$$
 (A-15)

Example Calculation

To illustrate the use of Equations A-13 and A-15 for estimating the maximum expected axial displacements of the rotor and the bearing forces, the following parameters for the original fan configuration using one or more wave washers have been selected.

 $f_n = 494$ Hz (from experimental measurements) Q = 44 (from experimental measurements m = 1.2/386 = 0.00311 lb sec^2/in .

$$W_0 = 0.2 \text{ g}^{3}\text{Hz}$$
 (100% level at 494 Hz from Figure A-2)
 $k = (2\pi)^{2}\text{m}$ $f_n^2 = (2\pi)^{2}$ (0.0031) $(494)^{2} = 29.962 \text{ lb/in.}$

$$y_{\text{max}} = 36.78 \left[\frac{W_0 Q}{f_n^3} \right]^{\frac{1}{2}} = 36.78 \left[\frac{0.2(44)}{(494)^3} \right]^{\frac{1}{2}} = 0.0099 \text{ inch}$$

$$F_b = k \sqrt{1 + \frac{1}{Q^2}} (y_{max}) = 29,962 \sqrt{1 + \frac{1}{44^2}} (0.0099) = 297 \text{ lb.}$$

Since this force would be transmitted through a single bearing, it substantially exceeds the predicted allowable bearing force of 43 pounds.

To check the validity of the "white-noise" approximation for this analysis, the frequency bandwidth for the rotor can be calculated using Equations (A-9) and (A-10).

$$f_1 = f_1 (1 - \frac{1}{20}) = 494 (1 - \frac{1}{88}) = 488 \text{ Hz},$$

$$f_2 = f_n (1 + \frac{1}{20}) = 494 (1 + \frac{1}{88}) - 500 \text{ Hz}.$$

Because of the very low damping (high Q), the resonant bandwidth is very narrow and the "white-noise" approximation should be entirely adequate. The principle source of difference between analytical predictions of maximum bearing forces and the actual bearing forces which will occur during tests will be the nonlinearities of the bearings and preload spring. Measurements show that when the bearings are preloaded with any type of soft spring at one end of the shaft, the rotor natural frequency will be reduced and the damping will increase (lower Q) as the excitation amplitude is increased.

APPENDIX B

ANALYSIS OF THE ANGULAR CONTACT BALL BEARING FOR AXIAL THRUST LOAD

APPENDIX B

ANALYSIS OF THE ANGULAR CONTACT BALL BEARING FOR AXIAL THRUST LOAD

This analysis was essentially derived from the book by Harris. Equations (2.6) through (2.10) and (2.25), (2.26), (5.30), (5.42), (6.26), (6.33), and (9.32) through (9.35) yield all the necessary information to perform this analysis. The appended computer code AXLOD utilizes these relationships. The program requires the inner race diameter (DI) (inches) ball diameter) (D) (inches) diametral clearance (PD) (inches) total curvature of the bearing (B=fi+fo-1), number of balls in the bearing (NB), increment in the contact angle (DLF) (radians) at which load, deflection and other parameters are sought, and the limit of the maximum Hertz pressure (PLM) (psi) beyond which the program should be terminated, as input parameters. The output consists of thrust load (lbs) and axial displacement (in.), the Hertz stress (psi), the corresponding minimum angle (radians) and land height (in.) required at the inner and outer race. These output quantities are calculated at intervals determined by the parameter (DLF). For PD = 0.00065 (in.) and B=0.14 the load-deflection calculated by this program agree with the corresponding quantities provided by Barden Corp. for the PLV fan bearings. In the method developed by Harris the analysis of angular contact ball bearings is dependent upon a certain constant derived by Jones which is usually read from a plot. In the computer code described here, Jones' assumptions about his constant are accepted but the value of the constant is computed within the program. It is assumed that the bearings consist of steel with Young's modulous of 30 x 106 psi and Poissons ration of 0.3.

PROGRAM AXLOD

```
CY FRI 04/24/10
AXLOD
          15: 46
       DIMENSION SMCR(2), FRO(2), CDEL(2), FHRZ(2), ASTR(2)
100
      COMMON FR.XE,XK
110
120
      EXTERNAL FUNCI
      PI = 3.1415926536
130
       INPUT, DI , D, PD, B, NB, DLF, PLM
140
      PRINT," INN. RACE DIA., BALL DIA., DIA. CLEARANCE, B , NO. OF
150
160 +BALLS AND INCREMENT IN ALFA, MAX HERTZ LIMIT "
170
       PRINT, DI , D, PD, B, NB, DLF, PLM
       A = B*D ; ALO=ACOS(1.-PD/2./A) ; ENDP=2.*A*SIN(ALO)
180
190
       CMP=(B+1.)/2.; RCRD=CMP+D
200
       SMCR(1) = 4./D+2./DI-1./RCRD
       FRO( 1) = (2./DI+1./RCRD)/SMCR( 1)
210
250
       RO=PD/2.+D+DI/2.
230
       SMCR(2) = 4./D-1./RO-1./RCRD
240
       FRO(2) = (1./RCRD-1./RO)/SMCR(2)
250
       BL=0.005 ; BU=0.99995 ; ERP=0.00001 ; N=20
       DO 4 KOUNT = 1.2
260
       FR = FRO(KOUNT)
270
280
       EMD = YNEST(BL, BU, ERR, FUNCI, N, I, NTRY)
       IF (I-0) 5, 5, 6
290
300
      6 PRINT," MODULUS, DID NOT CONVERGE, KOUNT, I, NTRY", KOUNT, I, NTRY
310
      5 CONTINUE
       AKAP = 1./EMD
320
330
       ASTR(KOUNT) = (2.* AKAP* AKAP*XE/PI)**(1./3.)
340
       BSTR = (2.*XE/AKAP/PI)**(1./3.)
350
       CDEL(KOUN T) = ( SMCR(KOUN T) * EM D* EM D* D/X E) **(1./3.) *XK
       FHRZ(KOUNT)=(D*SMCR(KOUNT))**(2./3.)*100000./4.327/BSTR
360
362 +
                      /ASTR(KOUNT)
370
      4 CONTINUE
       SM = CDEL(1) + CDEL(2)
38 Ø
39 0
       ABJ = (B*1.E+06/SM/7.4858795)**1.5 ; ABJ3=ABJ**(1./3.)
       PRINT ," SM, ABJ", SM, ABJ, +
400
       PRINT , " THRUST LOAD, AXL. DSPL.", +
410
420
       PRINT,"
                             RACE , HERTZ PRESSURE , MIN, ANGLE AND LAND
425 + HEI GHT "
       FTH R=NB+D+D+ABJ
430
       AL = ALO
440
450
       DO 10 J=1,100
       AL = AL+DLF ; FCOS=SQRT( COS(ALO)/COS(AL)-1.)
460
470
       THRST = FTHR*SIN(AL)*FCOS**3
       DSPL = B*D*SIN L-ALO)/COS(AL)
480
       PRINT, THRST, DSFL
485
       DC 15 K = 1.2
49 A
       PHRZ = FHRZ(K)*FCOS*ABJ3
500
       THT = 0.009*ABJ3*ASTR(K)*FCOS/((D*SMCR(K))**(1.73.))
510
       THT = ASIN(THT) + AL
515
520
       HL = (1 - COS(THT))*D
       IF(K-2) 16, 17, 17
530
      16 PRINT 20, PHRZ, THT, HL
540
       20 FORMAT (15X," INNER", 3E15.4)
550
560
       GO TO 14
570
     17 PRINT 21, PHRZ, TH T, HL
      21 FORMAT( 15X, " OUTER", 3E15. 4)
58 Ø
590
      14 IF(PHRZ-PLM) 15, 25, 25
600
       IF ( THT-PI/2.) 15,25,25
610
      15 CONTINUE
```

620

630

640

10 CONTINUE

25 STOP ; END

SUSE PART2

11 A(1)=A(2)

1520

```
READY .
LI STNH
         FUNCTION FUNCICE)
1000
         COMMON FRO, XE, XK
1010
1020
         ZZ = SQRT (1.-Z*Z)
1030
         AKAP = 1.72 ; XE = DLE(22) ; XK = DLK(22)
1040
         FUNC1 = FRO-((AKAP*AKAP+1.)*XE-2.*XK)/XE/(AKAP*AKAP-1.)
         RETURN ; END
1050
         FUNCTION ASIN(X)
1060
         ASIN = ATAN(X/SQRT(1.-X*X))
1070
1080
         RETURN : END
         FUNCTION ACOS(X)
1110
         ACOS = ATAN(SORT(1.-X*X)/X)
1120
1130
         RETURN ; END
1140
            FUNCTION YNEST(BL, BU, E, FUNC, N, I, NTRY)
1150
            DIMENSION A(3), B(3)
1160
            NTRY = 0
            NZ=0
1170
1180
            B( 1) = BL
1190
            B( 3) = BU
1200
            I = 0
1210
         35 A( 1) = FUNC( B( 1) )
1220
              IF(ABSF(A(1))-E) 333, 333, 334
        333 YNEST=B( 1)
1230
1240
            RETURN
1250
        334 A(3)=FUNC(B(3))
              IF( ABSF( A( 3) ) - E) 335, 335, 336
1260
        335 YNEST= B(3)
1270
            RETURN
1280
        336 P=A(1) +A(3)
1290
              IF(P) 25, 24, 24
1300
         24 YNEST= 0.
1310
1320
            I=2
            PRINT 1, B(1), A(1), B(3), A(3)
1330
         I FORMAT (" ROOT NOT NESTED, BL=",E15.8,5X,"LFUNC=",E15.8/
1340
1350
          +5X, "BU=", E15.8, 5X, "UFUNC=", E15.8)
1360
            RETURN
         25 DO 5 J= 1.N
1370
            NTRY=J
1380
            B(2)=(B(1)+B(3))/2.
1390
1400
            A(2) = FUNC(B(2))
1410
              IF(ABSF(A(2))-E)26,26,27
         26 YNEST=B(2)
1420
1430
            RETURN
1440
         27 BP=RTF(B, A)
1450
            AP=FUNC(BP)
1460
               IF(ABSF(AP)-E)9,9,10
1470
         10 P=A(2)+A(3)
1480
               IF(P) 11, 12, 12
         12 A(3) = A(2)
1 49 0
1500
            B(3) = B(2)
            GO TO 13
1510
```

```
1530
            B(1)=B(2)
1540
         13 P=AP+A(3)
1550
               IF(P) 15, 15, 16
         16 A(3) = AP
1560
1570
            B( 3) = BP
            GO TO 5
1580
1590
         15 A(1) = AP
1600
            B(1) = BP
1610
          5 CONTINUE
1620
            I = 1
1630
          9 YNEST=BP
            RETURN
1640
1650
            END
1660
            FUNCTION RTF(X,Y)
1670
            DIMENSION X(3),Y(3)
168 @
            A=(Y(3)+Y(1)-2.*Y(2))/2.
1690
            B=(Y(3)-Y(1))/2.
1700
            C=Y(2)
            CHK=(ABSF(B)+ABSF(C))+1.E-4
1710
1720
               IF(ABSF(A) - CHK) 20
1730
            D= SQRTF( B*B- 4. * A*C)
            XM=(-B-D)/2./A
1740
1750
               IF(ABSF(XM)-1.)21,21,25
         25 XM=(-B+D)/2./A
1760
1770
            GO TO 21
1780
         20 XM=-(C/B+A+C+C/B/B/B)
1790
         21 RTF=X(2)+(X(3)+X(2))*XM
1866
            RETURN
1810
            END
1820
            FUNCTION ELK(X)
            IF(X*X-.5) 10, 10, 20
1830
         10 S=1.
18 40
1850
            6=1.
1860
            DO 3 N=1, 100
            U=N
1870
1880
            Q= Q+(U-.5)+X/U
1890
            D= 0* 0
1900
            S= S+ D
1910
            IF( ABS( D) - 1 . E- 13) 2, 3, 3
1920
          3 CONTINUE
            PRINT 5
1930
1940
          5 FORMAT (1X, 3HDIV)
          2 DLK= S* 3.14159265358979/2.
1950
1960
            RETURN
         20 Y1=1.-X*X
1970
            Y=SORT(Y1)
1980
            V= 1 .
1990
            B= 0.
2000
             S= ALOG( 4./Y)
2010
            DO 6 M=1, 100
8080
            U=M
2030
            V=V*(.5-U)/U+Y
2040
2050
            B= B+ 1./U/( 2.*U-1.)
            D= V* V*( ALOG( 4./Y) - B)
2060
2070
             S= S+ D
2080
            IF( ABS( D/S) - 1. E-13) 7, 6, 6
2090
          6 CONTINUE
2100
            PRINT 5
2110
          7 ELK=S
2120
            RETURN
             EN D
2130
```

```
FUNCTION ELECX)
2140
2150
            IF(X + X - . 5) 10, 10, 20
2160
         10 S=1.
2170
            V= 1 .
            DO 2 N=1,100
2180
2190
            U=N
            V= V*( • 5- U)/U*X
2200
            D= V* V/( 1 -- 2 - * U)
2210
            S= S+ D
5556
2230
            IF(ABS(D)-1.E-13)3,2,2
2240
          2 CONTINUE
2250
            PRINT 5
          5 FORMATCIX, 3HDIV)
2260
2270
          3 ELE= S* 3. 14159265358979/2.
2280
            RETURN
2290
         20 Y1=1.-X*X
2300
            IF(Y1)
                       22,22,21
2310
         22 ELE= 1.
2320
            RETURN
2330
         21 Y=SQRT(Y1)
2340
            S= 1.+(2.* ALOG( 4./Y)-1.)*Y*Y/4.
2350
            V=Y/2.
2360
            C= 1 .
2370
            DO 6 M=1, 100
2380
            U=M
2390
            V= V* (U- . 5) /U+Y
2400
            C=C+1./U/(2.*U-1.)+1./(U+1.)/(2.*U+1.)
            D= V* V*(2.*U+1.)/(U+1.)*(2.*ALOG(4./Y)-C)
2410
2420
            S= S+ D
            IF(ABS(D)-1.E-13) 7, 6, 6
2430
          6 CONTINUE
2440
            PRINT 5
2450
2460
          7 ELE=S
2470
            RETURN
2480
            END
```

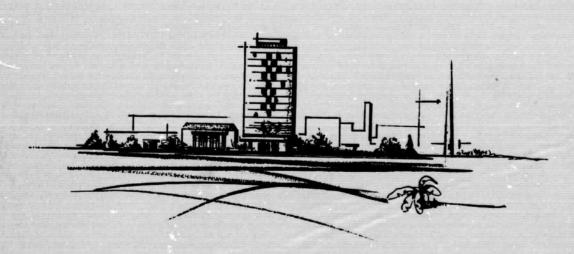
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SUMMARY REPORT

on

ANALYSIS OF BRINELLING FAILURE OF BEARINGS FROM VIBRATION TESTED PLV FANS

to

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION GEORGE C. MARSHALL SPACE FLIGHT CENTER

by

W. A. Glaeser, S. K. Batra, and R. H. Prause

June 8, 1970

BATTELLE MEMORIAL INSTITUTE Columbus Laboratories 505 King Avenue Columbus, Ohio 43201

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PROGRAM AXLOD

SUMMARY REPORT

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ANALYSIS OF BRINELLING FAILURE OF BEARINGS FROM VIBRATION TESTED PLV FANS

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NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

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BATTELLE MEMORIAL INSTITUTE Columbus Laboratories

SUMMARY

Brinelling damage has been identified in PLV fan bearings as the cause for noisy running after exposure to a random-vibration environment.

Assuming a maximum Hertz stress of 460,000 psi to produce brinelling, the maximum axial load for the fan bearing was calculated as 43 pounds. Dynamic analysis of the rotor-bearing-spring system revealed that with one preload spring, the maximum predicted axial load would be 297 pounds. Reduction in maximum bearing loads under vibration conditions can be achieved by reducing the rotor natural frequency and by increasing the damping through using a soft spring system. This means multiple springs at both ends of the shaft. When a system of five springs at each end of the rotor was considered, the bearing load predicted was 58 pounds. Actual vibration tests at Marshall Space Flight Center using the 5-spring preload configuration have resulted in reduction of bearing damage. Use of Belleville springs for this particular rotor-bearing configuration was found undesirable because the collapse load of the springs would be easily exceeded during the anticipated operating conditions.

A bearing load-capacity computer program was written so that, given the bearing parameters, inner race diameter, ball diameter, diametral clearance, total curvature of the race groove and number of balls, the limiting axial load can be determined using maximum Hertz stress and over-riding of the race land as the failure criteria.

STATEMENT OF THE PROBLEM

PLV fans subjected to acceptance vibration tests have developed rough running, noisy bearings. Brinelling of the bearing races was suspected as the cause of rough running. If brinelling has been the cause, then overloading of the bearings from inertial loads has occurred and can only be alleviated by increasing the size of the bearing or reducing the peak inertial loads. If fretting or "false brinelling" has been the cause of rough races, then load capacity of the bearing will have no significance in the serverity of the effect. Brinelling is a plastic deformation process; fretting is a time-dependent wear process and the latter is a function of rubbing amplitude and lubrication.

THE MODE OF ANALYSIS

An analysis was performed on the rotor-bearing system to determine the following:

- (1) The nature of damage to fan bearings (whether it is brinelling or fretting).
- (2) The maximum bearing loads resulting from test random vibration environment.
- (3) Hertz stresses developed in the bearings and displacement of balls in races using the load values obtained in Number (2).
- (4) The influence of bearing preload and preload spring configuration on bearing contact stress levels.

Analysis of Failed Bearings

Samples of both new bearings and bearings from noisy fans were analyzed in the Battelle Lubrication Mechanics Laboratory. Bearings were disassembled and the race surfaces examined by stereoptican microscope. The race surface topography was measured with a Talyrond roundness instrument.

Microscopy revealed classic brinelling on the inner and outer races of bearings taken from tested fans. The following bearings were examined:

(1) <u>Used bearing</u> with much of the grease gone. Severe brinelling over-running the lip of the ball groove. brinelling on one side of the race groove.

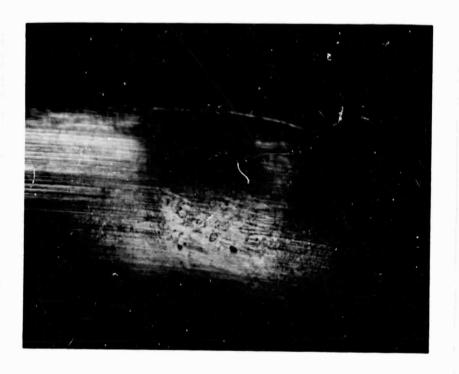
- (2) New bearing filled with grease. All surfaces free of defects.
- (3) Used bearing with much of the grease gone. Mild brinelling on inner and outer races.
- (4) <u>Used bearing</u> with some residual grease. One set of brinell marks on inner race and barely visible on outer race.
- (5) <u>Used bearing</u> full of grease. Very light brinell marks on inner and outer races.

The extent of ball indentation in the severely brinelled bearing is shown in the photomicrograph in Figure 1.

Talyrond measurements were made on several circumferential positions of the inner races of the bearings. The traces were made with the stylus riding (1) on the bottom of the ball groove, (2) on the side of the ball groove, and (3) on the race lands. Typical traces of brinelled bearings are shown in Figures 2, 3, and 4. The location of the tracing stylus is shown in the drawing at the top of each tracing. For instance, the trace in Figure 2a was made on the race land.

Figure 2 shows an example of heavy brinelling. The bearing was so overloaded axially that the balls were driven up the sides of the ball groove and indented the lip causing a mounding up on the race land surface. The mounding up can be seen in Figure 2a. Each mound represents the position of one ball in the bearing. All eight balls have produced indentations. The height of the mounds averaged 0.00025 inch. The indentations in the thrust side of the ball groove can be seen in Figure 3. The trace indicates one set of severe brinell marks together with at least two other sets of lighter brinell marks. Apparently this bearing was subjected to several separate conditions of axial vibration, one of them being most severe. Average depth of the severe indentations was about 0.00025 inch. Width of the indentations at the surface averaged about 0.03 inch. (The proportions of the indentations are distorted on the Talyrond traces because radial magnification is much higher than circumferential.)

The brinell marks did not extend into the bottom of the race groove as shown in the trace in Figure 2b. The trace is smooth, and slightly egg-shaped,



75 X

FIGURE 1. SEVERE BRINELLING DAMAGE ON INNER RACE
OF FAN BEARING. THE PHOTOGRAPH SHOWS
ONE BALL INDENTATION EXTENDING OVER
THE RACE GROOVE LIP

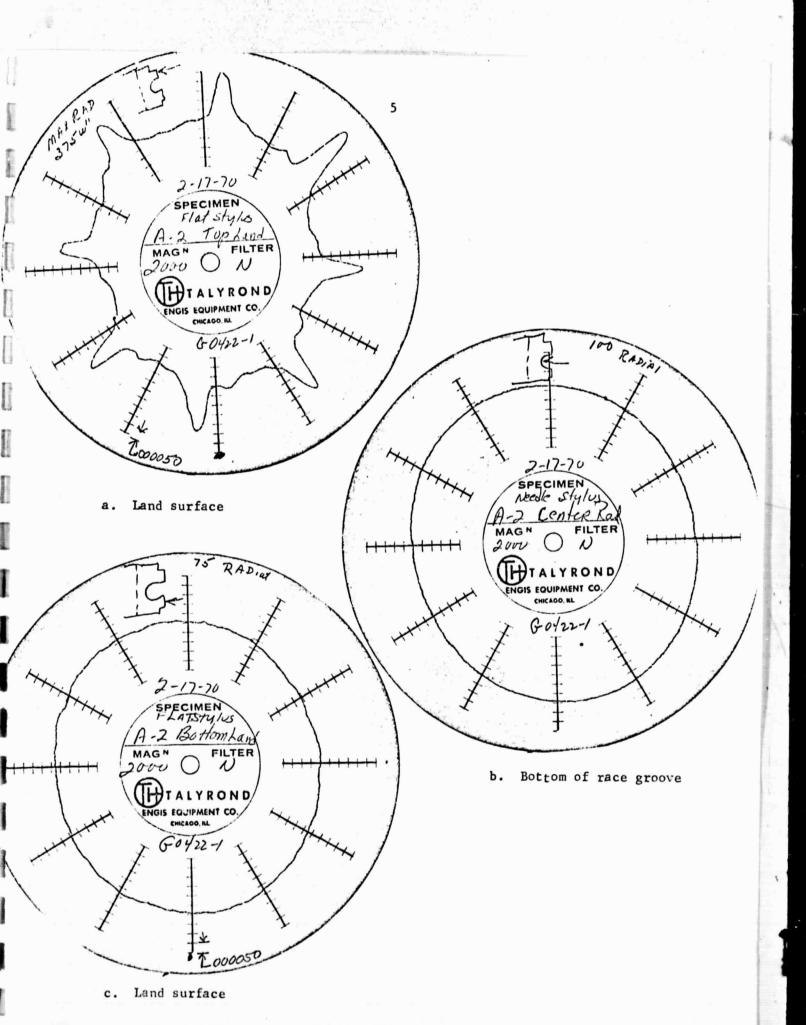


FIGURE 2. TALYROND TRACES OF SEVERELY BRINELLED FAN BEARING INNER RACE

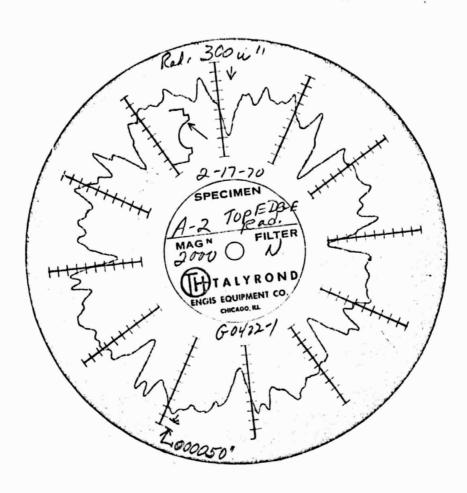
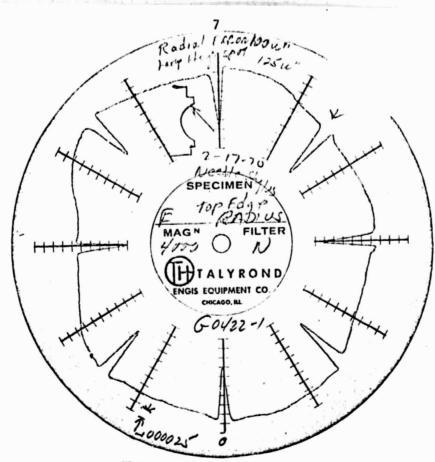
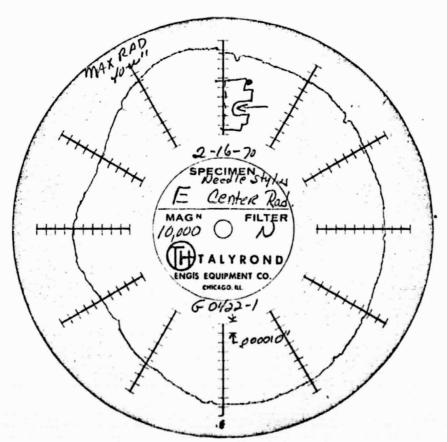


FIGURE 3. TALYROND TRACE OF SEVERELY BRINELLED FAN BEARING INNER RACE TAKEN ON THE THRUST SIDE OF THE RACE GROOVE



.a. Thrust side of race groove



b. Bottom of race groove

FIGURE 4. TALYROND TRACES OF LIGHTLY BRINELLED FAN BEARING INNER RACE

indicating ovality in the race geometry. It was concluded, therefore, that brinelling damage was associated with axial loads and not radial loads.

An example of light brinelling damage is shown in Figure 4. These marks were barely visible under the microscope. Maximum depth of the indentations are about 0.0001 inch. Note that faint indications of light dents show up in the trace made of the bottom of the ball groove. This indicates that the balls were not displaced as far from their no-load position as they were in the bearing exhibiting heavy brinelling. In addition, the trace shows only one set of brinell marks in this bearing. If the bearing was subjected to more than one vibration condition only one was severe enough to produce damage.

It has been concluded from the examination of failed bearings that the damage is true brinelling resulting from inertial overloads and that the conformity and radial play conditions in these bearings allow sufficient relative motion of rolling elements under axial load so that ball over-riding of the race groove lip is possible under heavy enough load.

VIBRATION ANALYSIS

An important objective of this program was to establish an analytical method for estimating maximum bearing loads when the PLV fan was subjected to a random vibration environment. Calculated bearing loads could then be used to compare the predicted results with the experience from tests where bearings have failed, and to evaluate proposed modifications to the bearing support system in order to select a modification most promising for further testing.

Equations for predicting the maximum expected rotor displacements and bearing loads have been derived and a detailed development is included in Appendix A, with numerical examples to demonstrate correct application. These equations are based on certain simplifying assumptions regarding the shape of the power spectrum of the vibration test specifications, as well as the use of a linear spring representation for the shaft bearings and preload springs, which are actually quite nonlinear. Even so, it is believed that this idealized model is a useful design tool.

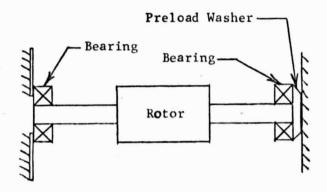
Dynamic Analysis Model

Figure 5a is a sketch of the original configuration of the fan rotor and bearings using one wave washer to obtain a 6 pounds axial preload. Figure 5b shows springs representing the flexibility of the bearings and preload washer, and this is transformed to the equivalent single-degree-of-freedom model shown in Figure 5c. The washer is very flexible relative to the bearings. Therefore, the only contribution of the washer is to establish sufficient preload so that for small motions about the shaft equilibrium position, the total effective stiffness is that of the one bearing that is preloaded against the rigid housing.

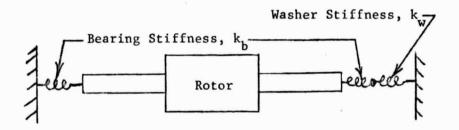
When the shaft deflection exceeds the 0.005 inch preload deflection, the bearing will be unloaded, and the effective stiffness will be that of the wave washer. Figure 6 shows the load-deflection curves for several different preloads to demonstrate how the bearing stiffness is the important parameter in determining the effective axial stiffness. These curves also show the net external force limitations imposed by the 43 pounds maximum axial load capability of the bearings. There will be no significant change in these curves if multiple washers are used to replace the single washer so long as they are all on one end of the shaft and the total preload force is the same.

If, however, washers are installed at both ends of the shaft, Figure 7 shows that the stiffness of the flexible washers will be the determining factor of the total effective stiffness. Figure 8 shows several load-deflection curves for five Belleville springs stacked in parallel at each end of the shaft. The load-deflection characteristics obtained from MSFC Drawing SK20-5072 indicate these springs have a maximum load capability of about 14.5 pounds and Figure 8 shows this collapse load.

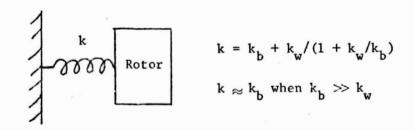
¹ The bearing considered in the analysis was Barden SR4SS, One-quarter-inch bore, angular contact, currently in use. Assuming a limiting maximum Hertz stress of 460,000 psi, the maximum axial load capacity of the bearing was determined as 43 pounds using the computer program described in the next section.



a. Bearing Support Configuration



b. Spring Model of Bearing Supports



c. Simplified Model for Dynamic Analysis

FIGURE 5. DEVELOPMENT OF MODEL REPRESENTING THE AXIAL DYNAMICS OF THE PLV FAN ROTOR ASSEMBLY WHEN PRELOADED BY FLEXIBLE WASHERS AT ONE END

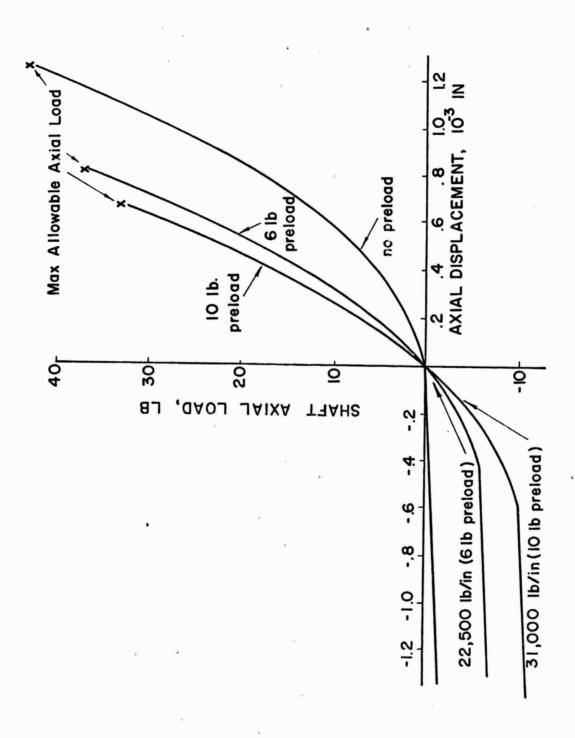
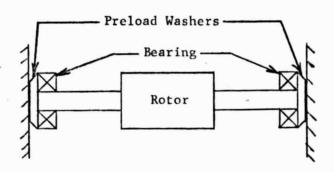
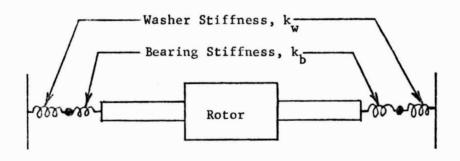


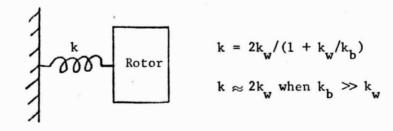
FIGURE 6. AXIAL LOAD-DEFLECTION CURVES FOR FAN SHAFT PRELOADED BY ONE WAVE WASHER AT ONE END



a. Bearing Support Configuration



b. Spring Model of Bearing Supports



c. Simplified Model for Dynamic Analysis

FIGURE 7. DEVELOPMENT OF MODEL REPRESENTING THE AXIAL DYNAMICS OF THE PLV FAN ROTOR ASSEMBLY WHEN PRELOADED BY FLEXIBLE WASHERS AT BOTH ENDS

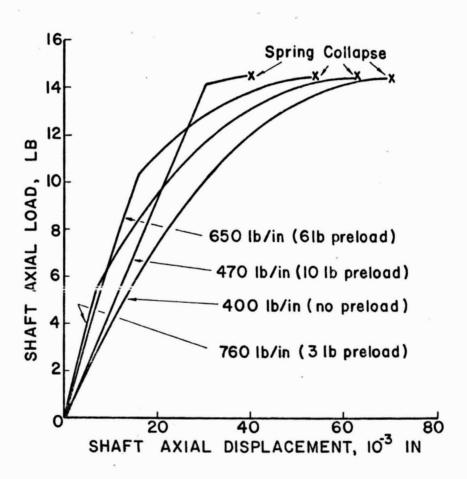


FIGURE 8. AXIAL LOAD-DEFLECTION CURVES FOR FAN SHAFT WITH FIVE BELLEVILLE SPRINGS AT EACH END STACKED IN PARALLEL

Experimental Measurements

In order to predict the maximum expected bearing loads using the equations derived in Appendix A, it is necessary to know the natural frequency and the resonant amplification factor Q of the rotor in the axial direction. The natural frequency can be predicted analytically, at least within the linear system approximations, but the Q of the system must be measured.

In order to measure both the natural frequency and effective damping (Q), the fan housing was clamped rigidly in a heavy vise and a soft rubber mallet was used to tap the rotor and produce a transient vibration. A Kistler Model 802-A piezoelectric accelerometer mounted on the fan impeller was used with a Kistler Model 568 charge amplifier, and the vibration signal was displayed on a Tektronix Type 502 oscilloscope. A Polaroid camera was used to record the transient vibration and the photographs were analyzed to determine the natural frequency and damping.

Table 1 summarizes the measured frequencies and Q factors for three bearing support configurations. Equations A-13 and A-15 in Appendix A were used to calculate the maximum expected bearing loads and rotor displacements. Although the calculated values of bearing loads for the original fan configuration (Configuration I) are probably higher than would be actually measured because of the neglected nonlinearities, it is evident that the bearing loads from the vibration tests are considerably higher than the 43 pounds maximum allowable load. The observed reduction in natural frequency at higher vibration amplitudes was caused by the rotor motion exceeding the washer preload so that the bearings were unloaded for part of each vibration cycle. The photographs of the transient vibration indicated that the preload was exceeded at an amplitude of about 0.004 inches (± 16 g's), which agrees closely with the design specifications for the wave washer.

With one Belleville spring installed at each end of the shaft, the calculated and measured natural frequencies were in close agreement and the damping was increased considerably (lower Q). However, the vibration environment is most severe in the frequency range of 60 to 150 Hz (see Figure A-2) so the predicted bearing loads are excessive and the Belleville springs would collapse.

SUMMARY OF MEASURED ROTOR DYNAMIC CHARACTERISTICS AND PREDICTED BEARING LOADS FOR 100 PERCENT LEVEL VIBRATION TEST SPECIFICATIONS TABLE 1.

	15	
0.0099	0.123	060°0
297(1) 205(1)	118(1,2)	58 (1,2)
44 24	:	m
494 (±10 g's) 440 (±20 g's) 385 (±80 g's)	125	100
427 (4)	138	s 73
e Wave Washer at e end 1b design preload	One Belleville Spring each end Approximately 10 lb preload	Five Belleville Springs each end Approximately 6 lb preload
6.99	Pr Pr	Fi App
	494(±10 g's) 44 297(1) 440(±20 g's) 24 205(1) 385(±80 g's)	427(4) 494(±10 g's) 44 297(1) 440(±20 g's) 24 205(1) 385(±80 g's) ing 138 125 11 118(1,2)

Exceeds maximum allowable bearing load.

Exceeds Belleville spring collapse load.

100 percent Level Vibration Test Specification (MSFC Memo. S&E-ASTN-EME-69-243). A 10 lb preload gives a 500 Hz theoretical natural frequency. £363

With five Belleville springs at each end (Configuration 3) the damping was increased considerably. The predicted loads were reduced sufficiently to suggest that the bearings might be capable of surviving the vibration tests, but the Belleville springs would be expected to collapse. It does not appear practical to use Belleville springs with a 14.5 pounds collapse load with bearings which have a 43 pounds allowable load.

Methods for Modifying Bearing Load Levels

Figure A-2 shows that the most severe vibration excitation is in the 60-540 Hz frequency range with reduced levels extending to the 20 Hz low frequency limit and to the 2000 Hz high frequency limit. In order to reduce the fan bearing loads it is obvious from Figure A-2 that to have reduced excititing forces either the natural frequency should be reduced considerably from the 494 Hz resonance of the original design configuration, or the system should be stiffened to increase its natural frequency. One advantage of reducing the natural frequency by supporting the rotor with flexible springs is that this type of modification will increase the damping and reduce the resonant amplification factor Q. If the natural frequency is increased, the damping will be quite low (high Q) as indicated by the measurements, and it is quite difficult to introduce additional damping in a high frequency system.

If it is assumed that Q = 5 is reasonable for a softly-sprung bearing mount, the maximum allowable bearing load of 43 pounds can be used to calculate a reasonable frequency suitable for a design goal. Results of this calculation show that the rotor axial natural frequency must be reduced below 55 Hz for the predicted bearing loads to be less than the maximum allowable. For a safety factor of 2, the natural frequency should be below 43 Hz, and the springs must permit the maximum expected rotor displacement of \pm 0.19 inch.

These large relative displacements present practical problems because the radial clearance between the bearing O.D. and the bearing housing should be kept small for satisfactory fan operation. However, the bearings must be quite free axially to obtain the required low natural frequency and large displacements. Slight misalignment or excessive friction could bind the bearings so that while a "soft" support system appears theoretically satisfactory, extreme care in manufacturing and assembly would be required to obtain high reliability using the current blaring mount configuration.

The alternative solution of rigidly mounting the bearings to obtain a high natural frequency eliminates the requirement of providing the axial motion. However, it would probably be necessary to increase both the bearing size and number of bearings. For example, if the highest measured value of Q = 44is assumed, the natural frequency must be greater than 2000 Hz if the load on a single bearing is reduced below 43 pounds. If a pair of larger preloaded bearings are used with the rating of each bearing doubled to 86 pounds and the bearing stiffness is increased proportionately (actually the bearing stiffness may increase by a greater factor than the load rating), then the rotor natural frequency would be doubled (\$\approx 1000 Hz). The total predicted bearing force would be 164 pounds or 82 pounds per bearing, which would be acceptable. The maximum rotor displacement would be only 0.0013 inch. The limitation with this solution is that the bearing housing structure must be rigid relative to the bearings, and it is difficult to design a bearing system that will actually have a resonance as high as 1000 Hz without a severe weight penalty.

After examining the alternatives of either a soft, low-frequency bearing system or a stiff, high-frequency system, it does not appear that the present PLV fan configuration can be easily modified in order to pass the vibration tests and operate reliably. It is recommended that if the fan design is revised, the bearings should be soft-mounted in the axial direction with the flexible element attached rigidly to the bearing outer case to eliminate any sliding elements. Damping should either be obtained with the flexible element, such as the hysteresis loss in an elastomer, or a damping device introducing friction could be attached independent of the bearing supports.

Angular Contact Bearing Design Analysis

Essential features of the angular contact bearing design are reasonably well understood. The mathematical analysis of the geometrical parameters influencing the load capacity of the bearing is well detailed in the books by Harris² and Jones³. In estimating the load capacity of the bearing in the axial and/or radial direction the hydrodynamic effects of lubrication

² Tedric A. Harris, "Rolling Bearing Analysis", 1966, John Wiley & Sons, Inc.

A. B. Jones, "Analysis of Stresses and Deflection", Volume 1 and 2, Copyright 1946, New Departure, Division of General Motors Corp.

are generally ignored; the design criteria are based on the Hertz theory of dry-static contact between the rolling elements.

The formulations pertaining to thrust load, applied to single row angular contact bearing, have been employed in the analysis of the PLV for bearings. The appropriate mathematical equations have been adopted from Harris's book. Using these expressions a computer code was written for the G.E. Time Sharing System (Mark I); this is included and explained in Appendix B. This program was written with the intent of providing a useful tool for analyzing the influence of various geometric parameters on the load-deflection curve of PLV fan bearing. For instance for the bearings currently in use, assuming 57 percent conformity and diametral clearance of 0.00065 inch the value of the axial deflections were computed (computer output I) for various static loads.4 The load deflection curve is plotted in Figure 9; this curve was used in the dynamic analysis of the previous section.

To illustrate the use of the computer program several additional runs (computer outputs II-V) were made in which ball race conformity and the diametral clearance (contact angle) were selectively varied. To understand the output of the program reference may be made to the illustration in Figure 10. Here α designates the contact angle and θ_0 (θ_i for the inner race) is the minimum angle the outer race must subtend, as shown, in order to avoid the riding of the ball over the land. Ho (H, for the inner race) designates the minimum land height the outer race must posses in order to prevent ball over riding. All this is true for a given thrust load. The program computes these quantities as well as the maximum Hertz stress for both the inner and

inner race diameter, di = 0.3401 inch (See Figure 10) = 0.0937 inchball diameter, D number of balls = 0.00065 inchdiametral clearance

B, or (fi+fo-1) = 0.14where fi = inner race conformity

fo = outer race conformity

A - B X ball diameter $\alpha = \cos^{-1} \left(\frac{d}{2A}\right)$ where $P_d = \text{diametral clearance}$ B = fo + fi - 1

where fo = outer race groove radius/ ball diameter fi = inner race groove radius/ball diameter.

The critical dimensions used in the analysis for the SR4SS bearing are as follows:

Diametral clearance and contact angle are related in the following way:

COMPUTER OUTPUT I

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114 1 111 16	FLV DEARING PRAIRIUM DUAD CAPACITY

? 6.346075, 6.69375, 6.66665, 6.14,8, 6.66453, 466666. INN. HACE DIA., BALL DIA., DIA. CLEARANCE, É , NO. OFBALLS AND INCREMENT IN ALFA

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INNER		OUTER	• 158 3E+ 06	• 3263E+ 66	. 4769 E-02
OUTER	3.6725	3-19857E-04			
4.99 51 3.72246E-04 INNER		INNER	· 2148 F+ 66	• 3339 E+00	.5331E-02
INNER		OUTER	. 178 DE+ 66	.3347E+80	. 5203E-02
OUTER	4.9951	3.72246E-04			
1 NN ER		INNEK	• 2366E+ 66	· 3529 E+ 66	.5777E-12
INNER		OUTER	. 1960E+06	- 348 3E+66	.5631E-02
0UTER	6.51	4.34782E-04			
1		INNER	. 2569 E+ 66	- 3663E+00	. 6219 E- 02
INNER .2761E+06 .3792E+00 .6661E-02 OUTER .2238E+06 .3739E+00 .6478E-02 10.1383 5.60304E-04 INNER .29 44E+06 .39 18 E+00 .7103E-02 OUTER .2439 E+06 .38 61E+00 .69 01E-02 12.2649 6.2329 7E-04 INNER .3120E+06 .4640E+00 .7547E-02 OUTER .258 4E+06 .39 3 0E+00 .732.7E-02 14.6098 6.86450E-04 INNER .3289 E+06 .4159 E+00 .7755E-02 17.181 7.49 765E-04 INNER .3453E+06 .4877E+00 .8444E-02 OUTER .2860E+06 .4810E+00 .818 7E-02 19.9871 8.13246E-04 INNER .3612E+06 .439 2E+00 .8899 E-02		OUTER	. 2129 E+06	. 3614E+00	. 6655E-02
OUTER	8 . 2224	4.97466E-04			
OUTER		INNER	.2761E+06	. 379 2E+00	. 6661E-62
INNER .29 44E+06 .39 18 E+00 .7103E-02 OUTER .2439 E+06 .38 61 E+00 .69 01 E-02 12.2649 .6.2329 7E-04 INNER .3120E+06 .4040E+00 .7547E-02 OUTER .258 4E+06 .39 3 0E+00 .732.7E-02 14.6098 .6.86450E-04 INNER .3289 E+06 .4159 E+00 .799 4E-02 OUTER .2725E+06 .409 6E+00 .7755E-02 17.181 7.49 765E-04 INNER .3453E+06 .4277E+00 .8444E-02 OUTER .2860E+06 .4210E+00 .816 7E-02 19.9871 8.13246E-04 INNER .3612E+06 .439 2E+00 .8899 E-02		OUTER	. 2238 E+ 66	• 3739 E+ 00	. 6478 E- #2
OUTER	10.1383	5.60304E-04			
12. 2649		INNER	· 29 44E+ 06	• 39 18 E+ 00	.7163E-62
INNER .3120E+66 .7640E+00 .7547E-02 OUTER .2584E+06 .3980E+00 .7327E-02 14.6698 6.86450E-04 INNER .3289E+06 .4159E+00 .7994E-02 OUTER .2725E+06 .4696E+00 .7755E-02 17.181 7.49765E-04 INNER .3453E+06 .4877E+00 .8444E-02 OUTER .2860E+06 .4810E+00 .8187E-02 19.9871 8.13246E-04 INNER .3612E+06 .4392E+00 .8899E-02		OUTER	. 2439 E+06	· 38 61E+00	. 69 0 1E-02
OUTER	12.2649	-6.2329 7E-04			
14.6698 6.86450E-04 INNER .3259 E+ 06 .4159 E+ 00 .799 4E-02 OUTER .2725E+ 06 .469 6E+ 00 .7755E-02 17.181 7.49765E-04 INNER .3453E+ 06 .4277E+ 00 .8444E-02 OUTER .2860E+ 06 .4210E+ 00 .8167E-02 19.9871 8.13246E-04 INNER .3612E+ 06 .439 2E+ 00 .8899 E-02		INNER	· 3120E+66	. / 640E+00	. 7547E-02
INNER .3289 E+ 06 .4159 E+ 00 .799 4E- 02 OUTER .2725 E+ 06 .409 6E+ 00 .7755 E- 02 17. 18 1 7. 49 765 E- 64 INNER .3453 E+ 06 .42 77 E+ 00 .8 444 E- 02 OUTER .28 60 E+ 06 .42 10 E+ 00 .8 18 7 E- 02 19. 98 71 8. 132 46 E- 04 INNER .3612 E+ 06 .439 2E+ 06 .8899 E- 02		OUTER	· 258 4E+ 06	- 393 ØE+ ØØ	. 7327E-02
OUTER	14.6698	6.86450E-04			
17. 18 1 7. 49 765E- 64 INNER .3453E+ 66 .48 77E+ 60 .8 444E- 62 OUTER .28 60E+ 66 .48 10E+ 60 .8 18 7E- 62 19.98 71 8. 13246E- 64 INNER .3612E+ 66 .439 2E+ 66 .8899 E- 62		INNER	- 3259 E+ 06	. 4159 E+ 66	. 799 4E- 02
INNER		OUTER	• 2725E+#6	. 469 6E+ 60	.7755E-02
INNER	17.131				
19.9871 8.13246E-04 INNER .3612E+06 .4392E+00 .8899E-02		INNER	. 3453E+@6	. 4877E+00	-8 444E- 02
19.9871 8.13246E-04 INNER .3612E+06 .4392E+00 .8899E-02					
INNER . 3612E+06 . 4392E+00 .8899E-02	19.9871				All and Aleger
				. 439 2E+00	-8899 E-02
	all said				the second secon

COMPUTER OUTPUT I (CONTINUED)

23.1371	8 . 7639 6E-	64	A TOTAL CONTRACTOR AND A STREET, A STREET,	or the property of the second
	INNER	. 3767E+06	. 4506E+00	•9 358 E- @2
	OUTER	.3121E+06	. 4433E+ 60	.9063E-02
26.3019	9.40718E-	04		
	INNER	• 39 19 E+ 06	. 4619 E+ 00	.98 22E-02
	OUTER	· 3247E+66	. 4543E+00 ·	.9508E-02
29.9161	• 6.6	1	•	
	INNER	· 40 63 E+ 6 6	• 4736E+66	• 1029 E-01
	OUTER	. 3370E+66	• 4651E+ ØØ	.9958E-02
33.7/27	• 66	11		IK.
	INNER	. 4214 +16	48 40E+ 00	. 1077E-01
	OUTER	. 3491 + 16	• 4758 E+ 00	.1641E-61
37. 1619	• 6 6	11		*
	INNER	• 4353 E+ 66	· 49 49 E+ 00	. 1125E-61
	OUTER	. 361(E+06	• 48 64E+ 00	· 168 7E-61
12.1113	• 66	12		
	INNER	. 450 CE+ 06	• 5057E+00	· 1173E-01
	OUTER		• 49 69 E+ @ Ø	. 1134E-61
47. (() .	• 6.k	13		
	LNNER	• 1639 E+ 06	• 5164E+00	.1222E-61

COMPUTER OUTPUT III

-1178

INN. RACE DI A., BALL DI A., DI A. CLEARANCE, B., NO. OFBALLS AND INCREMENT I N ALFA. MAX HERTZ LIMIT

> . 3401 - 0046 460000-00

• 09 37 1 • 00000E-04

2.7668 428927.06

RACE , HERTZ PRESSURE , MIN, ANGLE AND LAND HEIGHT .0287 5.10551e-05 INNER					•
INNER			PRESSURE . MI	N. ANGLE AND LAND	HEI GHT
0UTER	• Ø28 7	5-10551E-05			
0UTER .458 3E+05 .1252E+00 .7333E-03 .08 79		INNER	• 5543E+05	• 1263E+00	• 7465E-03
INNER		OUTER	 458 3E+ Ø5 	• 1252E+00	• 7333E- Ø3
INNER	· Ø8 79	1.02157E-04		×	
0UTER .6559E+05 .1407E+00 .9263E-03 .1746 1.53309E-04	,		• 79 33E+ 05	• 1 423E+00	.9 477E-03
1746 1.53399 E-04		OUTER			
INNER 0829 E+05 1540E+00 1138 E-02 2898 2.04512E-04 INNER 0UTER 9490E+05 1540E+00 1129 E-02 2898 2.04512E-04 INNER 0UTER 9490E+05 1661E+00 1291E-02 285768 E-04 INNER 1298 E+06 1661E+00 1291E-02 285768 E-04 INNER 1298 E+06 1802E+00 1519 E-02 285768 E-04 INNER 1298 E+06 1802E+00 1519 E-02 285768 E-04 INNER 1437E+06 1915E+00 1714E-02 2829 5 3.58 452E-04 INNER 1569 E+06 202 4E+00 1914E-02 2829 5 3.58 452E-04 INNER 1569 E+06 202 4E+00 1914E-02 2829 5 3.58 452E-04 INNER 1569 E+06 203 4E+00 1855E-02 285768 E-04 285768 E-04	• 1746				***************************************
0UTER				-1560F+00	-1138F-02
148 148	4.				
INNER 0148 E+ 06 0168 5E+ 00 0127 E- 02 • 4356 2.55768 E- 04 INNER 1298 E+ 06 01661 E+ 00 0129 1E- 02 • 6146 3.0708 1E- 04 INNER 1073 E+ 06 01776 E+ 00 01475 E- 02 • 6146 3.0708 1E- 04 INNER 1188 E+ 06 0188 6E+ 00 01662 E- 02 • 8295 3.58 452 E- 04 INNER 1188 E+ 06 0188 6E+ 00 01662 E- 02 • 8295 3.58 452 E- 04 INNER 1569 E+ 06 0202 4E+ 00 019 14E- 02 OUTER 1297 E+ 06 0199 2E+ 00 01855 E- 02 1.0833 4.098 84 E- 04 INNER 1695 E+ 06 0209 6E+ 00 0205 E- 02 1.3793 4.61378 E- 04 INNER 1816 E+ 06 0299 6E+ 00 0205 E- 02 1.7209 5.12938 E- 04 INNER 1934 E+ 06 0237 E+ 00 0255 E- 02 1.7209 5.12938 E- 04 INNER 294 E+ 06 0237 E+ 00 0255 E- 02 2.1115 5.64565 E- 04 INNER 2049 E+ 06 0237 E+ 00 0267 E- 02 0UTER 1694 E+ 06 0237 E+ 00 0267 E- 02 3.0546 6.68031 E- 04 INNER 2216 E+ 06 0253 E+ 00 0304 E- 02 OUTER 178 6E+ 06 0253 E+ 00 0304 E- 02 0UTER 2773 E- 04 INNER 2216 E+ 06 0253 E+ 00 0304 E- 02 0UTER 2773 E- 04 INNER 2216 E+ 06 0253 E+ 00 0324 E- 02 0UTER 2775 E- 04 INNER 2271 E+ 06 0253 E+ 00 0324 E- 02 3.0546 6.68031 E- 04 INNER 2271 E+ 06 0253 E+ 00 0324 E- 02 0UTER 1877 E+ 06 0259 IE+ 00 0324 E- 02 4.2397 7.7179 4E- 04 INNER 2379 E+ 06 0273 E+ 00 0348 E- 02 4.2397 7.7179 4E- 04 INNER 2455 E+ 06 0268 E+ 00 0368 E- 02 4.9333 8.2379 4E- 04 INNER 2455 E+ 06 0268 E+ 00 0368 E- 02 4.9333 8.2379 4E- 04 INNER 2455 E+ 06 0268 E+ 00 0368 E- 02 4.9333 8.2379 4E- 04 INNER 259 0E+ 06 0268 E+ 00 0368 E- 02 5.70 8.758 75E- 04 INNER 269 4E+ 06 0305 E+ 00 038 49 E- 02	- 2898				11076-02
OUTER	- 2070			1 48 5F4 00	1207F-00
. 4356					
INNER 0173E+06 .1802E+00 .1519E-02 .6146 3.0703 1E-04 INNER .1437E+06 .1915E+00 .1714E-02 .8295 3.58452E-04 INNER .1569E+06 .2024E+00 .1914E-02 .00TER .1569E+06 .2024E+00 .1914E-02 .10833 4.0988 4E-04 INNER .1695E+06 .2131E+00 .2120E-02 .10833 4.0988 4E-04 INNER .1695E+06 .2131E+00 .2052E-02 1.3793 4.61378E-04 INNER .1816E+06 .2235E+00 .2252E-02 1.7209 5.1233E-04 INNER .1934E+06 .2337E+00 .2255E-02 1.7209 5.1233E-04 INNER .1934E+06 .2337E+00 .2549E-02 2.1115 5.64565E-04 INNER .2049E+06 .2337E+00 .2679E-02 2.5546 6.16262E-04 INNER .2049E+06 .233E+00 .2679E-02 2.5546 6.16262E-04 INNER .2161E+06 .2538E+00 .3004E-02 0UTER .1786E+06 .2438E+00 .2679E-02 3.0546 6.66031E-04 INNER .2161E+06 .2538E+00 .3004E-02 0UTER .1786E+06 .2494E+00 .3241E-02 3.0546 6.66031E-04 INNER .2271E+06 .2637E+00 .3241E-02 0UTER .1877E+06 .2637E+00 .3241E-02 3.0546 6.66031E-04 INNER .2271E+06 .2637E+00 .3362E-02 4.2397 7.71794E-04 INNER .2271E+06 .2637E+00 .3362E-02 4.2397 7.71794E-04 INNER .2485E+06 .2832E+00 .3735E-02 4.9333 8.23794E-04 INNER .2485E+06 .2832E+00 .3735E-02 4.9333 8.23794E-04 INNER .2694E+06 .2832E+00 .3849E-02	. 4256			• 10015+00	• 124 IE-05
0UTER	• 4336			15.005.50	
10 10 10 10 10 10 10 10					
INNER			• 1073E+06	• 1776E+ 00	• 1475E-02
OUTER	• 6146				
*** *** *** *** *** *** *** *** *** **					
INNER 1269 E+06 199 2E+00 19 1 4E-02 OUTER 129 7E+06 199 2E+00 18 55E-02 1.08 33 4.09 88 4E-04 INNER 169 5E+06 213 1E+00 2052E-02 1.379 3 4.61378 E-04 INNER 18 16E+06 2235E+00 2332E-02 1.7209 5.129 38 E-04 INNER 1934E+06 2337E+00 2255E-02 1.7209 5.129 38 E-04 INNER 1934E+06 2337E+00 2549 E-02 2.1115 5.64565E-04 INNER 2049 E+06 2298 E+00 26773E-02 2.5546 6.16262E-04 INNER 2161E+06 2538 E+00 3004E-02 3.0546 6.68 031E-04 INNER 2161E+06 249 4E+00 29 01E-02 3.0546 6.68 031E-04 INNER 2271E+06 2637E+00 3241E-02 3.0546 7.198 74E-04 INNER 2379 E+06 268 6E+00 3362E-02 4.239 7 77179 4E-04 INNER 2379 E+06 268 6E+00 3735E-02 4.239 7 7179 4E-04 INNER 248 5E+06 28 32E+00 3735E-02 4.9333 8.2379 4E-04 INNER 248 5E+06 28 32E+00 3735E-02 0 UTER 2055E+06 28 32E+00 3735E-02 0 UTER 2055E+06 28 32E+00 379 2E-02 5.70 8.758 75E-04 INNER 259 0E+06 28 76E+00 38 49 E-02			• 1 188 E+ 06	• 188 6E+ ØØ	• 1662E-Ø2
OUTER	•8 29 5				
1.08333					the second second
INNER 0169 5E+06 02131E+00 0252E-02 1.3793				• 199 2E+ 00	• 18 55E- Ø2
OUTER 1401E+06 209 6E+00 2052E-02 1.379 3 4.61378E-04	1 • 08 33	4.0988 4E-04			
1.3793 4.61378E-04		INNER	• 169 5E+06	•2131E+00	.2120E-02
INNER		OUTER	- 1401E+06	• 209 6E+ ØØ	-2052E-02
OUTER	1.3793	4. 61378 E- 04			
1. 72 09 5. 129 38 E - 04		INNER	• 18 16E+06	• 2235E+ 00	.2332E-02
1. 72 09 5. 129 38 E - 04		OUTER	- 1502E+06	- 2198 E+ 00	.2255E-02
OUTER	1.7209	5 · 129 38 E- 04			The second second second
OUTER		INNER	• 19 34E+ 06	-2337E+00	. 2549 E- 02
2.1115 5.64565E-04					
INNER .2049 E+ 06 .2438 E+ 00 .2773E- 02 OUTER .169 4E+ 06 .239 7E+ 00 .2679 E- 02 2.5548 6.16262E- 04	2.1115				70 40 AD
OUTER				- 2438 F+ 00	-9773F-09
2.5548 6.16262E-04 INNER .2161E+06 .2538E+00 .3004E-02 OUTER .1786E+06 .2494E+00 .2901E-02 3.0546 6.68031E-04 INNER .2271E+06 .2637E+00 .3241E-02 OUTER .1877E+06 .2591E+00 .3128E-02 3.6148 7.19874E-04 INNER .2379E+06 .2735E+00 .3484E-02 OUTER .1967E+06 .2686E+00 .3362E-02 4.2397 7.71794E-04 INNER .2485E+06 .2832E+00 .3735E-02 OUTER .2055E+06 .2781E+00 .3603E-02 4.9333 8.23794E-04 INNER .2590E+06 .2929E+00 .3992E-02 OUTER .2141E+06 .2876E+00 .3849E-02 5.70 8.75875E-04 INNER .2694E+06 .3025E+00 .4256E-02					
INNER .2161E+06 .2538E+00 .3004E-02 OUTER .1786E+06 .2494E+00 .2901E-02 3.0546 6.68031E-04 INNER .2271E+06 .2637E+00 .3241E-02 OUTER .1877E+06 .2591E+00 .3128E-02 3.6148 7.19874E-04 INNER .2379E+06 .2735E+00 .3484E-02 OUTER .1967E+06 .2686E+00 .3362E-02 4.2397 7.71794E-04 INNER .2485E+06 .2832E+00 .3735E-02 OUTER .2055E+06 .2781E+00 .3603E-02 4.9333 8.23794E-04 INNER .2590E+06 .2929E+00 .3992E-02 OUTER .2141E+06 .2876E+00 .3849E-02 5.70 8.75875E-04 INNER .2694E+06 .3025E+00 .4256E-02	2.5548			V207 12-00	*E017E-DE
OUTER	2005-0			0538 54 66	200 AF 00
3. 0546 6.68 031E-04 INNER .2271E+06 .2637E+00 .3241E-02 OUTER .18 77E+06 .259 1E+00 .3128 E-02 3. 6148 7. 198 74E-04 INNER .2379 E+06 .2735E+00 .348 4E-02 OUTER .19 67E+06 .268 6E+00 .3362E-02 4. 239 7 7. 7179 4E-04 INNER .248 5E+06 .28 32E+00 .3735E-02 OUTER .2055E+06 .278 1E+00 .3603E-02 4. 9 333 8. 2379 4E-04 INNER .259 0E+06 .29 29 E+00 .399 2E-02 OUTER .2141E+06 .28 76E+00 .38 49 E-02 5. 70 8. 758 75E-04 INNER .269 4E+06 .3025E+00 .425 6E-02					
INNER .2271E+06 .2637E+00 .3241E-02 OUTER .1877E+06 .2591E+00 .3128E-02 3.6148 7.19874E-04	3-0546			• 247 427 00	• 27 01E-02
OUTER	00 00 00			042751.00	20 415- 60
3.6148 7.19874E-04 INNER .2379 E+06 .2735E+00 .348 4E-02 OUTER .1967E+06 .268 6E+00 .3362E-02 4.2397 7.7179 4E-04 INNER .248 5E+06 .28 32E+00 .3735E-02 OUTER .2055E+06 .278 1E+00 .3603E-02 4.9333 8.2379 4E-04 INNER .259 0E+06 .29 29 E+00 .399 2E-02 OUTER .2141E+06 .28 76E+00 .38 49 E-02 5.70 8.758 75E-04 INNER .269 4E+06 .3025E+00 .425 6E-02					
INNER	2 (1 10		• 10 1 1E+ NO	• 524 I E+ 66	• 3158 F-05
OUTER	3.0145		00000.04		
4.239 7 7.7179 4E-04 INNER .248 5E+06 .28 32E+00 .3735E-02 OUTER .2055E+06 .278 1E+00 .3603E-02 4.9 333 8.2379 4E-04 INNER .259 0E+06 .29 29 E+00 .399 2E-02 OUTER .2141E+06 .28 76E+00 .38 49 E-02 5.70 8.758 75E-04 INNER .269 4E+06 .3025E+00 .425 6E-02					
INNER			• 19 67E+ 06	• 268 6E+ ØØ	• 3362E-02
OUTER	4. 239 7			and the second second second	
4.9333 8.2379 4E-04 INNER .259 0E+06 .29 29 E+00 .399 2E-02 OUTER .2141E+06 .28 76E+00 .38 49 E-02 5.70 8.758 75E-04 INNER .269 4E+06 .3025E+00 .425 6E-02					The second second second
INNER			· 2055E+06	- 278 1 E+ 00	•3603E-02
OUTER •2141E+06 •2876E+00 •3849E-02 5•70 8•75875E-04 INNER •2694E+06 •3025E+00 •4256E-02	4.9333				
5.70 8.75875E-04 INNER .2694E+06 .3025E+00 .4256E-02					-3992E-02
INNER . 269 4E+ 06 . 3025E+ 00 . 4256E- 02			-2141E+06	• 28 76E+ 00	. 38 49 E- Ø2
	5.70		1 100	Water to the King of	
OUTER • 2227E+06 • 29 69 E+00 • 4103E-02				- 3025E+ 00	. 4256E-02
		OUTER	- 2227E+06	• 29 69 E+ 00	.4103E-02

COMPUTER OUTPUT III (CONTINUED)

			1 111 (GOMITMOLD)	
6.5445	9 • 28 040E-04		Control of the China control of the	Mark of the Artife gar
	INNER	• 279 6E+ Ø6	• 3120E+00	• 452 6E- Ø≥
	OUTER	·2312E+06	• 3063E+ 00	-4363E-02
7.4712			2	
	INNER	• 2898 E+ 06	• 3215E+ 00	• 48 0 4E - 02
	OUTER	• 239 6E+ Ø6	• 31 56E+ ØØ	. 4629 E-02
8 • 48 5	.001			
•	INNER		-3310E+00	• 5089 E- 02
	OUTER	. 248 ØE+ Ø6	• 32 48 E+ 00	• 49 Ø3E- Ø2
9 • 59 09				
	INNER	. 3099 E+ Ø6	• 3404E+00	• 538 ØE- Ø2
	OUTER	· 2562E+06	- 3341E+ 00	•518 2E-02
10.7939				
	INNER	• 3199 E+ Ø6	• 3498 E+ 00	. 5678 E-02
	OUTER	• 2645E+ Ø6	• 3433E+00	• 5469 E- Ø2
12.0993				
	INNER	• 329 7E+ Ø6	• 359 2E+ ØØ	• 598 4E-02
	OUTER	· 2726E+06	• 3524E+00	.5762E-02
13.5124	.0012			
	INNER	- 339 6E+ 06	• 368 6E+ ØØ	• 629 6E- 02
	OUTER	- 28 Ø8 E+ Ø 6	-3616E+00	. 6062E-02
15.0388	.0013		,	
	INNER		• 3779 E+ 00	. 6615E-02
	OUTER	- 2888 E+ Ø6	• 3707E+00	- 6368 E- 02
16.6841	.0013			
	INNER		• 38 72E+ 00	• 69 42E-02
V	OUTER	• 29 69 E+ Ø6	• 3798 E+ 00	• 668 2E-02
18 - 4542	.0014	Service Control of the Control of th	001702.22	- 000 EL - DE
		- 3688 E+ Ø6	• 39 65E+ 00	-7275E-02
	OUTER	• 3049 E+ 06	• 3889 E+ 00	• 7002E-02
20.355	•0015		130072700	• 10025-02
22.000	INNER		• 4058 E+ 00	.7615E-02
	OUTER		• 398 ØE+ ØØ	• 7328 E-02
22.3927			•370 02+00	• 1320 E-02
22.0727		- 388 1E+ Ø6	. 4151E+00	• 79 62E-02
	OUTER	- 300 IE+ 00	• 4071E+ 00	• 7662E-02
24.5735	•0016	• 5267 2 • 60	• 40112+00	• 10025-02
2403733	INNER	• 39 77E+06	- 4244E+ 00	-8 31 7E- 02
	OUTER	• 3288 E+06		•8 002E-02
26.9041	• 0016K	• 3200 ET 00	• 41 62E+ 00	•0 ABSE- 85
20.7841	INNER	. A072E+0/	A2025+ 66	0 (70 5 00
	OUTER	• 4073E+06		•8 678 E- Ø2
29 • 39 08		• 3367E+ 06	• 4252E+00	•8 349 E- 02
27 • 37 05	• 0017	A1 (0 E) 0 (A 400 E 4 00	0.0425 00
F	INNER	• 41 69 E+ 06	• 4429 E+ 00	•9 047E-02
20 0 40=	OUTER	-3447E+06	• 4343E+ 00	•8 703E-02
32.0407	-0017		*******	0.4000 00
	INNER	• 4264E+Ø6	• 4522E+00	•9 422E-02
	OUTER	- 3525E+06	• 4433E+ 00	•9063E-02
34.8605	• 0018			A-1-1-1-1-1-1-1-1-1-1-1-1-1-1-1-1-1-1-1
	INNER	• 4359 E+ Ø6	• 461 4E+ 00	•98 Ø5E- Ø2
	OUTER	• 3604E+06	• 4524E+00	•9 430E-02
37.8574	• 0018			
1	INNER	• 4454E+06	· 4707E+00	• 1019 E-01
	OUTER	• 368 3E+ 06	• 461 4E+ 00	-98 04E-02
41.0388	.0019		The state of the s	Strain Strain
	INNER	. 45 49 E+ 06	+4799 E+00	-1059 E-01
	OUTER	- 3761E+06	• 4705E+ 00	- 1018 E-01
44. 412	- 0619		and the state of t	Apple of the second
	INNER	. 4644E+06	- 489 2E+ 00	-1099E-01
		and the control of the control of the		The second secon

COMPUTER OUTPUT IV

INN. RACE DIA., BALL DIA., DIA. CLEARANCE, B , NO. OFBALLS AND INCREMENT IN ALFA , MAX HERTZ LIMIT

• 3401 • 6937 8 • 00000E - 04 • 0046 460000 • 00

SM, ABJ 2.191 120439.37

	ACE , HERTZ 1	PRESSURE , MIN,	ANGLE AND LAND	HEIGHT
• 4517	*****	• 6632E+Ø5	- 5308 E+00	100.00
	OUTER	• 5 43 4E+ 05		• 129 ØE- Ø1 • 128 ØE- Ø1
1 0455		• 3434E+ 63	• 5288 E+ 00	• 129 BE- BI
1.2455		0.4405.05	F / O / F : O O	
	INNER	•9 412E+05	• 5606E+00	• 1 435E-01
0.0007	OUTER	•7711E+05	• 5578 E+ ØØ	• 1 42 1E-01
2.3326		4.555.04		
	INNER	• 1157E+ Ø6	• 58 48 E+ 00	. 1558 E-Ø1
	OUTER	•9 477E+05	• 58 13E+00	· 1540E-01
3. 6605		10100.01	10110.00	
	INNER	• 1340E+06	• 6061E+00	·1670E-01
	OUTER	• 1098 E+ 06	.6021E+00	• 1648 E-01
5.214				
	INNER	• 1504E+06	• 6257E+00	•1776E-01
	OUTER	• 1232E+06	•6211E+00	• 1751E-01
6.9849			V-1222-12	
	INNER	· 1653E+06	• 6439 E+00	• 18 77E- Ø1
	OUTER	• 1354E+06	• 6388 E+ 00	• 18 49 E- 01
8 • 9 69 2		*		
	INNER	• 179 1E+06	·6612E+00	• 19 76E-01
	OUTER	• 1 468 E+ Ø6	• 6557E+00	• 19 44E-01
11-1656	1.57464E-04			
	INNER		• 6778 E+00	.2072E-01
	OUTER	• 1574E+06	• 6719 E+00	-2038 E-01
13.574	1.77587E-04			
	INNER	·2045E+06	• 69 38 E+00	-2167E-01
14 1050	OUTER	• 1676E+06	• 68 75E+00	•2129E-01
16. 1959	1.9 78 14E-04		#00 0B - 00	00445 04
	INNER	•2163E+06	• 709 3E+00	.2261E-01
10 0005	OUTER	• 1772E+06	• 7026E+00	- 5556E-01
19 • 0335	2. 18 146E-04	00775.01	20 405 400	00545-01
	INNER	•2277E+06	• 7243E+00 • 7173E+00	•2354E-01
22.0396		• 18 65E+ Ø6	• 11 13E+00.	-2310E-01
22.6076	INNER	• 238 6E+ Ø6	• 739 1 E+ 00	04445-01
	OUTER		• 731 7E+00	.2446E-01 .2399E-01
25.3679	2.59 130E-04		• 131 1ET 00	· 2377E-61
23.0017	INNER	• 249 2E+ 06	• 7535E+00	-2538 E-Ø1
	OUTER	·2042E+06	• 7458 E+00	·2488 E-01
28 • 8 722	2.79 78 7E-04		• 1430 67 66	• 2 400 E-01
LOVOTEL	INNER	• 259 5E+ Ø6	• 7677E+00	-2630E-01
	OUTER	•2126E+06	• 759 6E+00	•2577E-01
32.6069	3.00555E-04	-21202100	• 737 627 88	• E311E-01
024 0007	INNER	- 269 6E+06	• 78 1 7E+ 00	-2721E-01
	OUTER	· 2209 E+06	• 7732E+00	.2666E-01
36.5766	3-21437E-04		• 11322400	. 2000E-DI
3043100	INNER	• 279 4E+Ø6	• 79 54E+00	-28 13E-01
	OUTER	• 2289 E+ Ø6	• 78 67E+00	.2754E-01
40.7866	3. 42434E-04		- 10012100	JE 1346-01
1000	INNER	- 2889 E+ 06	-8 09 0E+00	. 29 04E-01
	OUTER	•2368 E+06	-8000E+00	· 28 43E-01
45.2419	3.63549 E-04			- 20 402 01
	INNER	• 298 4E+ 06	-8225E+00	- 299 6E-01
	OUTER	. 2445F+ 86	-8131F+00.	- 29 305-81

COMPUTER OUTPUT IV (CONTINUED)

49 • 9 48 3	3-8 478 2E-04		and a property of the second	placement in the
	INNER	. 3076E+06	•8 358 E+ 00	-3088E-01
	OUTER	. 2520E+06	·8261E+00	.3021E-01
54.9115	4.06137E-04			,
	INNER	.3167E+06	.8 489 E+ 00	-318 ØE-Ø1
	OUTER	. 259 5E+ Ø6	•8 389 E+ 00	-3110E-01
60.1376	4.27615E-04			
	INNER	• 3256E+06	•8 620E+00	- 3273E-01
	OUTER	• 2668 E+ 06	•8517E+00	- 3200E-01
65 6329	4. 49217E-04		/	
	INNER	· 3344E+06	•8 750E+00	- 3366E-01
	OUTER	. 2740E+06	•8644E+00	- 3289 E-01
71.4039	4.709 47E-04			
	INNER	. 3432E+06	•88 79 E+ 20	• 3459 E-01
	OUTER	-2812E+66	•8 769 E+00	• 338 ØE- Ø1
77. 4574	4.92806E-04			
	INNER	- 3518 E+ 06	•9 007E+00	-3552E-01
	OUTER	· 288 2E+ 06	•889 4E+00	. 3470E-01
83.8003	5. 1479 5E- 04			
	INNER	-3603E+06	•9134E+00	-3646E-01
	OUTER	· 29 52E+ Ø6	•9 019 E+ 00	-3561E-01
90.4398	5.369 19 E- Ø4			
	INNER	.3687E+06	•9261E+00	· 3741E-01
	OUTER	.3021E+06	.9142E+00	- 3653E-@1
97.3831	5.59177E-04			
	INNER	. 3770E+06	•9 38 7E+00	• 38 3 6E- @1
	OUTER	. 3089 E+ Ø 6	•9266E+00	.3744E-01
104.638	5.8 1573E-04			
	INNER	• 38 5 3E+ 06	•9513E+00	• 39 31E-01
	OUTER	•3157E+06	•9 388 E+00	• 38 3 7E- Ø1
112.2121	6.04109 E-04			
	INNER	• 39 35E+ 06	•9 638 E+00	• 402 7E- 01
*	OUTER	· 3224E+06	•9510E+00	- 39 30E-01
120-1135	6.26787E-04			
	INNER	· 4016E+06	•9 763E+00	· 4124E-01
	OUTER	• 329 ØE+ Ø6	•9632E+00	· 4023E-01
128 • 3505	6.49610E-04			
	INNER	· 409 7E+ 96	•9887E+00	-4221E-01
	OUTER	• 3357E+06	•9 754E+00	· 4117E-01
136.9313	6.72579 E- Ø4			
	INNER	• 4177E+06	• 1001E+01	• 4319 E-01
	OUTER	• 3422E+06	•98 75E+00	• 4212E-01
145.8647	6.95698E-04			2.0
	INNER	• 4256E+06	• 1014E+01	• 4417E-01
	OUTER	• 3488 E+ 66	•9996E+00	• 4307E-01
155 • 1595	7- 189 68 E- 04			
	INNER	· 4336E+06	• 1026E+01	• 4516E-01
	OUTER	•3552E+06	• 1012E+01	• 4402E-01
164-8249	7. 4239 3E- Ø4			
	INNER	• 4415E+06	• 1038 E+01	• 4616E-01
	OUTER	•3617E+06	- 1024E+01	• 4499 E-01
174.8701	7- 659 74E- 04	440.00.00	1	
	INNER	• 449 3E+ 06	• 1051E+01	• 4716E-01
10.5	OUTER	• 368 1E+06	• 1036E+01	• 459 6E- @1
185.3047	7.89714E-04			
	INNER	• 4571E+06	• 1063E+01	• 48 1 7E-01
104 1000	OUTER	• 3745E+06	• 1048 E+01	• 469 3E-01
196-1385	8 - 13617E-04			
	INNER	. 4649 E+06	• 1075E+01	. 49 18 E- Ø1

COMPUTER OUTPUT V

AXLOD 15:27 CY FRI 04/24/70

IN PART2

? 0.340075, 0.09375, 0.00080, 0.1178, 8, 0.0046

INN. RACE DI A., BALL DI A., DI A. CLEARANCE, B , NO. OFBALLS AND INCREMENT IN ALFA

. 3401

• 09 37 8 • 00000E-04 • 1178

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2.7669 428916.72

	RACE . HERTZ	PRESSURE, MAX.	ANGLE AND LAND	HEIGHT
• 3767	5-27779 E-05			
	INNER	•9 367E+05	- 319 3E+00	-4740E-02
	OUTER	• 7746E+ 05	.3174E+00	- 468 4E- 02
1.0984	1 · 0569 3E - 04			
	INNER	• 1331E+06	- 3428 E+ 00	.5455E-02
	OUTER	• 1101E+06	.3401E+00	.537ØE-Ø2
2.0796	1 • 58 747E-04			
	INNER	• 1638 E+ 06	-3621E+00	. 608 ØE- Ø2
	OUTER	- 1354E+06	- 3588 E+00	. 59 69 E- Ø2
3.2984	2.119 45E-04			
	INNER	• 19 00E+06	- 379 3E+00	.6663E-02
,	OUTER	• 1571E+06	- 3754E+ 00	• 6529 E- Ø2
4.7473	2.65287E-04			
	INNER	·2134E+06	• 39 51 E+ ØØ	.7224E-02
	OUTER.	• 1765E+06	• 39 Ø8 E+ Ø Ø	• 7067E-02
6.4246	3. 18 778 E- 04			
	INNER	- 2349 E+06	- 4100E+00	• 7772E- 02
	OUTER	• 19 42E+06 .	• 4052E+00	• 759 3E- Ø2
8 • 3323	3-72420E-04			
	INNER	• 2549 E+ Ø6	- 4243E+00	·8312E-Ø2
	OUTER	-2108E+06	. 419 1E+00	·8112E-02
10-4741	4-26216E-04	111		
	INNER	- 2738 E+06	- 438 ØE+ ØØ	-88 48 E- Ø2
	OUTER	· 2264E+06	· 4323E+00	•8 62 6E- Ø2
12.8552	4-8 0169 E-04			
	INNER	• 29 17E+Ø6	. 4512E+00	•9 38 2E-02
	OUTER	-2412E+06	• 4452E+ ØØ	•9 139 E- 02

COMPUTER OUTPUT V (CONTINUED)

15. 48 18	5 - 3 428 2E- 04		× 100	
	INNER	• 3089 E+ 06	.4641E+00	.9917E-02
	OUTER	• 2554E+Ø6	. 4578 E+00	•9 652E-02
18 - 361	5.88558E-04			
	INNER	• 3255E+ Ø6	• 4767E+ 00	• 1045E-01
	OUTER	-2691E+06	• 4700E+ 00	•1017E-01
21.5004	6.43000E-04			
	INNER	- 3415E+06	• 489 1 E+ 00	· 1099 E-01
	OUTER	• 23 2 4E+ Ø6	• 48 20E+ 00	• 1068 E-01
24.9831	6-97611E-04			*
	INNER	• 3570E+06	• 5012E+ 00	•1153E-01
	OUTER	· 29 52E+06	• 49 38 E+ 00	·1120E-01
28 • 59 27	7. 5239 4E- 04			
	INNER	. 3722E+06	•5131E+00	• 1207E-01
	OUTER	• 3077E+06	• 5055E+ 00	•1172E-01
32. 5632	8 · 07353E-04		Y	
	INNER	• 38 69 E+ Ø6	• 52 49 E+ 00	· 1262E-01
*	OUTER	- 3200E+06	• 51 69 E+ 00	-1225E-Ø1
36.8 29 1	8 · 6249 ØE- 04			
	INNER	. 401 4E+ 06	•5366E+00	• 1317E-01
	OUTER	- 3319 E+ 06	• 528 3E+ 00	• 1278 E-01
41.40	9 • 178 09 E- 04	3		
	INNER	• 4156E+06	.5481E+00	• 1373E-Ø1
	OUTER	• 3437E+06	• 539 5E+ 00	· 1331E-01
46.2861	9 • 73313E-04			
	INNER	• 429 5E+06	• 559 5E+ ØØ	• 1 429 E- 01
_	OUTER	• 3552E+Ø6	• 5506E+00	- 138 5E-01
51.4977	-001			
	INNER	• 4433E+06	• 5703 E+00	• 148 6E-01
	OUTER	• 3665E+06	•5616E+00	-1440E-01
57-0454				
	INNER	• 4568 E+06	• 58 20E+ 00	· 1543E-01
	OUTER	• 3777E+ Ø6	•5725E+00	• 1 49 5E-01
62.9 403	.0011			
	INNER	. 4701E+86	• 59 31 E+ 00	. 1601E-01

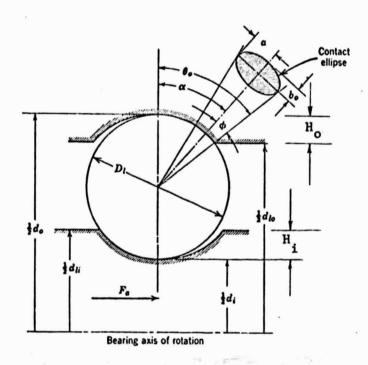


FIGURE 10. BALL RACEWAY CONTACT UNDER LIMITING THRUST LOAD

outer race, at the specified increments in α . It continues to do so until the maximum Hertz pressure at the inner race (or outer race) exceeds the limiting value specified through input, at which point it terminates further calculations. The termination of calculations is also affected if the required value of θ_0 (or θ_1) exceeds $\pi/2$.

In the computer output II and III it may be observed that for a limiting Hertz stress of 460,000 psi the load capacity of the bearing is reduced from approximately 102 pounds to 44 pounds as the parameter B, or (fo + fi - 1) is increased from 0.04 to 0.1178, keeping all other geometrical parameters constant. Increasing B reduces ball-race conformity or the closeness of fit between ball and race groove.

Computer outputs IV and V for the same calculations are repeated for an 8-fold increase in diametral clearance. These results indicate that the reduction in conformity level (or increase in B) reduces the load carrying capacity of the bearing, whereas an increase in diametral clearance (which increases contact angle) increases the load capacity with a limit imposed by race land height. Thus, by changing the contact angle with the present size bearing, an increase in load capacity is possible without changing bearing size. It must be kept in mind, however, that increasing contact angle will also increase ball spin and heat generation in the bearing. This effect will tend to reduce the endurance life of the bearing in terms of lubricant degradation and ball wear.

RECOMMENDED PROCEDURE FOR CHOICE OF BEARINGS IN SIMILAR FUTURE APPLICATIONS

The analysis of the present PLV fan bearings has demonstrated the importance of the dynamical considerations in the choice of bearings for a similar application. At the same time, one is forced to recognize that it is not possible to outline a simple step-by-step procedure that would end in the selection of a bearing satisfying all the design requirements. As is typical of such problems the designer must consider the influence of various parameters on the outcome before arriving at the final optimum design. The next few paragraphs present a plan of analysis to arrive at such an optimum design.

It is assumed that the mass of the rotor is known. It is also assumed that Power Spectral Density (PSD) of the vibration environment to be imposed on the rotor, is known. It is desired to determine the maximum axial load that would be borne by the bearing under fairly severe conditions. Based on this estimate, the geometry of the angular contact bearing would be selected so as to avoid the brinelling of the race as well as balls over-riding the race land. To facilitate the choice of the bearing it is necessary that this load-limit be as small as possible.

Equation (A-14) relates the axial load on the bearing with the stiffness of the system, its damping, and the maximum expected amplitude of vibration on the bearing-rotor assembly. The expected amplitude of vibration, for the white-noise approximation, is in turn related to the maximum value of the PSD (W_0) resonant frequency (f_n) and the magnification factor (Q) as given by (A-13). If the axial displacement of the rotor must be kept at an absolute minimum (from the viewpoint of design limitation) then the resonant frequency of the system must be chosen past the higher end of the PSD spectrum to minimize the value of W. This requires that stiffness of the system be high which would in turn require that the support system must be made very rigid. The latter requirement may be difficult to realize if the stipulated resonant frequency is of the order of 2000 HZ and over; in this case a compromise will have to be made, such as, allowing for larger amplitude-limit and perhaps even larger loads on the bearing. If, however, the required resonant frequency can be obtained with a reasonably rigid support system the reduction of the load-limit on the bearing would be easily achieved in the high-frequency end of the PSD spectrum.

On the other hand, if the axial displacement is not the limiting quantity, the load-limit on the bearing can be reduced by going to the low-frequency-end of the PSD spectrum. Below a certain level of the resonant frequency the value of W_o will decrease. The low resonant frequency can be designed by suspending the rotor-bearing system in a soft suspension system (e.g. five springs in the present PLV fan bearings) such that the preloaded bearing acts essentially as a rigid member. For the suspension system use can be made of thick elastomeric packing material which would provide high damping factor as well. It is, of course, necessary that the "collapse-

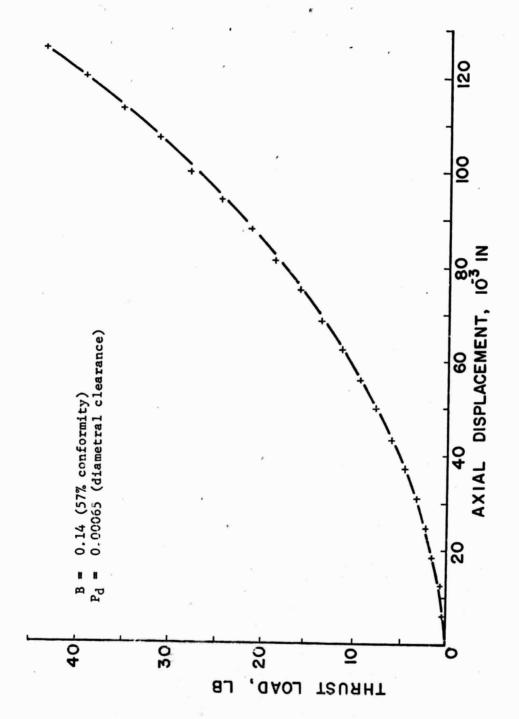


FIGURE 9. LOAD DEFLECTION CURVE FOR PLV FAN BEARING

COMPUTER OUTPUT II

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IN PART2

? 0.340075, 0.09375, 0.0001, 0.04,8, 0.0046, 460000. INN.RACE DIA., BALL DIA., DIA. CLEARANCE, B, NO. OFBALLS AND INCREMENT IN ALFA, MAX HERTZ LIMIT .3401 .0937 1.00000E-04 .04 8

•0046 460000•00

SM. ABJ 2.1909 120441.83

R	ACE . HERTZ	PRESSURE .	MIN. ANGLE AND LANE	UETOUT
• 0303	1.749 65E- 05		THE HAD CHAI	HEIGHI
	INNER	. 38 12E+05	• 2026E+ 00	10.100.00
	OUTER	- 3123E+ Ø5	•2015E+00	• 19 18 E-02
. 2792	7- 01533E-05		-2013E+00	• 189 6E- 02
	INNER	• 779 ØE+ Ø5	05055.00	
	OUTER	• 638 2E+ 05	• 2525E+ 00	· 29 72E- 02
• 7392	1-23078 E-04	*030 2 2 4 0 3	•2501E+00	• 29 1 7E- 02
	INNER	- 1052E+ 06	and the state of the	
	OUTER		•29 11E+00	• 39 44E- 02
1 - 4336	Annual Control of the	•8 619 E+ 05	• 28 79 E+ 00	• 38 58 E- 02
	INNER	• 128 3E+ Ø6	• 3259 E+ ØØ	- 49 35E-02
mage projection	OUTER	. 1051E+06	• 3220E+00	• 48 18 E- 02
2.3985	2-29821E-04		02202.00	V-10 10 E DE
	INNER	. 1 49 1E+ 06	- 3588 E+ 00	. 59 69 E- 02
	OUTER	. 1222E+ 06	- 3542E+ 00	- 58 19 E- 02
3.6761	2-83686E-04			
	INNER	· 168 6E+ Ø6	• 39 Ø 4E+ ØØ	.7053E-02
	OUTER	. 138 1E+06	• 38 52E+ 00	• 68 69 E- 02
5.3131	3.379 12E- 04			
	INNER	. 1871E+Ø6	- 4212E+00	•8 19 3E-02
	OUTER	.1533E+06	• 4154E+00	• 79 7 4E- 02
7.3607	3-92522E-04		45 45 400	.9 39 1E-02
,,,,,,,,,,,,,,,,,,,,,,,,,,,,,,,,,,,,,,,	INNER	.2050E+06	. 451 4E+ 00	.9 134E-02
	OUTER	. 168 ØE+ Ø6	. 4451E+00	
9 . 8 7 3 7	4. 47542E- 04	4		.1065E-01
9.0 101	INNER	. 2224E+ Ø6	. 48 13E+00	.1035E-01
	OUTER	. 1822E+06	. 4744E+00	. 16225-61
12.9116	5. 02998 E- 0	4		. 1 19 7E-01
12.7110	INNER	. 239 4E+ Ø6	. 5108 E+00	.1163E-01
	OUTER	. 1961E+06	•5034E+00	• 1102F- DI

COMPUTER OUTPUT II (CONTINUED)

14.0517	5-21585E-04	F (F) (F) (F)		•
	INNER	· 2450E+06	• 5206E+00	• 1242E-01
	OUTER	· 2007E+06	•5130E+00	-1207E-01
15.2595	5. 40224E-04			
	INNER	• 2506E+06	• 5304E+00	• 1288 E-01
	OUTER	· 2053E+06	• 5226E+ 00	. 1251E-01
16.5377	5.58917E-04			
	INNER	.2562E+06	\$ 5402E+00	.1335E-01
	OUTER	- 2099 E+ 06	•5322E+00	• 129 7E-01
20.8201	6- 15326E-04			-
	INNER	.2727E+06	• 569 4E+ 00	-1479 E-01
	OUTER	· 2234E+ 06	• 5609 E+00	. 1436E-@1
25.8313	6. 72252E-04			
	INNER	- 289 1E+ 06	• 598 5E+ 00	-1630E-01
	OUTER	-2368E+06	• 589 5E+ 00	-1582E-01
31 • 6488	7. 29 726E- 04			• • • • • • • • • • • • • • • • • • • •
•	INNER	- 3053E+06	• 6276E+00	• 178 7E- Ø1
	OUTER	. 2501E+06	• 618 ØE+ ØØ	1734E-01
38 • 3552	7-87778 E- 04			717042 01
	INNER	· 3215E+ Ø6	• 6567E+ 00	• 19 50E-01
	OUTER	· 2634E+06	• 6465E+ 00	• 1892E-01
46.0334	8 · 46438 E- Ø4		*	
	INNER	- 3376E+Ø6	• 68 58 E+ 00	.2120E-01
	OUTER	. 2766E+06	• 6751E+00	·2056E-01
54.792	9 - 05739 E- 04			TOOOL DI
	INNER	- 3537E+06	• 7150E+00	-229 6E-01
	OUTER	- 2898 E+ 06	• 7036E+00	-2227E-01
64.7157	9 . 65715E-04			
	INNER	- 3698 E+ Ø6	• 7442E+ ØØ	-2478 E-01
	OUTER	. 3030E+06	• 7323E+00	-2403E-01
75.9155	.001			
	INNER	• 38 59 E+ Ø6	• 7735E+ 00	.2667E-01
	OUTER	· 3162E+06	• 7610E+00	. 258 6E-01
88 - 5039	.0011			
	INNER	. 4020E+06	-8 029 E+ 00	• 28 63E-01
	OUTER	. 329 4E+ 06	• 7898 E+ 00	. 2775E-01
102-6009	.0012			
	INNER	. 418 2E+ Ø6	-8 325E+00	-3065E-01

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load" of the soft suspension system be at least as high as the maximum load capacity of the bearing.

Once the load limit of the bearing has been arrived at, in the above manner, the design parameter for the angular contact bearing can be established by the computer program described in the preceding section. To guard against brinelling it is recommended that maximum Hertz pressure be kept below 460,000 psi under all conditions. This would then provide a limiting criteria for the geometrical parameters of the bearings to be selected.

CONCLUSIONS AND RECOMMENDATIONS FOR FUTURE WORK

The following conclusions have been drawn from the analysis of the PLV fan bearing problem:

- (1) The bearing failure mode is vibration induced race brinelling, the damaging loads occurring in the axial direction.
- (2) Maximum bearing load can be reduced by decreasing the natural frequency of the rotor-bearing-preload spring system. This can be accomplished by mounting preload springs in series at both bearing supports on the rotor.
- (3) A computer program, written to calculate maximum load capacity based on brinelling mode of failure, has demonstrated that bearing load capacity can be increased by altering contact angle and ball-race conformity.
- (4) A system has been established for selection of ball bearings for similar rotor support problems in vibration environment (assuming the Power Spectral Density is given).

FUTURE WORK

It was found that the criterion for brinelling damage in ball bearings is based on an arbitrary maximum Hertz stress level. This level has been arrived at in bearing technology by static load tests on ball-race configurations. For applications where minimum bearing size is required a more accurate criterion should be established. We recommend that selected rolling contact bearings be subjected to vibration evaluations in which the maximum bearing load as determined from the analysis in this summary be varied over a range selected to cover nonbrinelling and brinelling levels. The extent of damage would then be evaluated on the basis of noise level during rotation under steady state load.

APPENDIX A

ANALYSIS OF FAN BEARING LOADS FROM RANDOM VIBRATION