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SEG-TDR-64-4

AN EXPERIMENTAL INVESTIGATION OF THE FATIGUE LIFE AND LIMIT LOAD CHARACTERISTICS OF NEEDLE ROLLER BEARINGS UNDER OSCILLATING LOAD CONDITIONS

Technical Documentary Report No. SEG-TDR-64-4

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Air Force Systems Command
Wright-Patterson Air Force Base, Ohio

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(Prepared under Contract No. AF 33(616)-3589 by Franklin Institute Laboratories, Philadelphia, Pennsylvania; Authors: J. S. Tawresey and W. W. Shugarts, Jr.)

NOTICES

FOREWORD

The work reported herein was performed by The Franklin Institute Laboratories, Philadelphia, Pennsylvania, for the Research and Technology Division (RTD), Air Force Systems Command, Wright-Patterson Air Force Base (WPAFB), Ohio, and the U. S. Army Transportation Research Command under contract AF 33 (616)-3589. The work was performed over the period March 1956 to December 1959 and from December 1960 to September 1963. During the earlier period the work was monitored by Messrs. C. F. Rice, O. Z. Brenning, and N. J. Guarino of the Propeller Laboratory (later designated Propulsion Laboratory). The work was monitored during the later period by Capt. G. E. Severs of the Directorate of Systems Engineering and most recently by Messrs. Hollis A. Cochran, Emil J. Thomas and Rodney L. Clark (Project Engineer) of the V/STOL Propulsion Branch, Propulsion and Flight Vehicle Power Division, RTD. Since December 1960 liaison with the U. S. Army Transportation Research Command has been carried on by Lt. H. D. Bowman and later Lt. R. Tegenkamp of the Army Liaison Office at WPAFB. More recently the liaison consisted of direct contact with Messrs. N. Daniel and L. M. Bartone (Project Engineer) of the Systems and Equipment Division, U. S. Army Transportation Research Command, Ft. Eustis, Va.

At The Franklin Institute Laboratories the project was under the leadership of Messrs. S. Abramovitz and later A. M. Loeb until January 1959. Since that time Mr. J. S. Tawresey has been Project Engineer. During the life of the project many people contributed to the performance of the work. Among these are Messrs. A. W. Caldwell, D. D. Fuller, R. C. Herrick, J. G. Hinkle, F. E. Kramberger, E. B. Sciulli, N. E. Sindlinger, R. T. Smith, C. H. Stevenson and J. Stone. This report carries the FIL designation Final Report F-A1913.

ABSTRACT

This investigation was made to determine, by test of full scale bearings, the factors influencing life of a bearing in cyclic oscillation, as encountered in helicopter rotor hinges.

A field survey was made to guide selection of an appropriate size and type of bearing for test (caged, needle roller), and to establish the principal operating variables. A machine was designed, a prototype unit was built and tested, followed by the construction of seven additional machines. Four test bearings were run simultaneously in each machine, under essentially identical conditions.

Each test bearing was subjected to fixed conditions of load, speed, amplitude of oscillation, and lubrication (grease or oil). It was run continuously until evidence developed of incipient failure in the form of spalling or flaking breakout of the raceways. The elapsed time in test prior to failure, marks the life of the bearing. The well known and characteristic deviation of individual bearing lives, within a group of bearings, was evident in all test runs Approximately 750 bearings were tested in 25 different test runs, comprising a systematic combination of operating variables.

The basis of an oscillating load capacity rating for the test bearing is presented and forms the basis for life prediction. A grease lubricant exhibited longer B-10 life with this test bearing than an oil lubricant under the controlled conditions of these laboratory tests. Effects on the B-10 life of both the speed and the amplitude of oscillation were revealed in these tests. Attempts at logical explanation of these effects were unproductive. The potential increase of life performance by pre-stressing the bearing was demonstrated.

PUBLICATION REVIEW

This technical documentary report has been reviewed and is approved.

James G. Barrett

JAMES G. BARRETT
Chief, Propulsion & Flight Vehicle Power Division
Directorate of Defense & Transport Systems Engineering

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INTRODUCTION

The purpose of this investigation was to determine the degree to which the life performance of a needle type roller bearing is affected by the circumstances of cyclic oscillation under load. This condition is of direct concern in the design of hinges for helicopter blades and rotores. Experience has shown that service life of such bearings is influenced by factors otherthan those that control life performance in the cause of rotating bearings.

The investigation was initiated by conducting an industry survey, and by disucssion with designers and builders of helicopters and with bearing manufacturers. Information so obtained served to highlight important phases of the problem and resulted in the formulation of a test program to be followed.[1,2*]

Failure to discover an available, suitable type of test equipment, led to the decision to undertake design and construction of a machine specifically adapted to the life testing of full scale bearings, under conditions duplicating those that are actually encountered in service.

Certain novel features were proposed in this connection, such as the use of hydrostatic bearings to transmit heavy loads in the absence of rotation, and to support such loads with a minimal degree of friction. These characteristics would enable provision to be made for the monitoring of frictional torque in the test bearing during progress of the test. It seemed reasonable to assume that this functional property of the test bearing would serve as a sensitive indication of failure.

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*Superscript numerals refer to references at the end of this report.
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A design was prepared embodying these features, and a prototype machine was built and operated. During this period unforeseen difficulties were encountered and some delay experienced in successful operation of the machine. However, the decision was then made to build seven additional machines conforming to the basic design of the prototype unit, but including appropriate changes in detail and construction. Opportunity was taken to effect economy in fabrication of the machines by the use of semi-production methods and controls.

In planning the program of investigation it was recognized that an extensive series of tests was necessary in order to define adequately the effects of the principal variables involved. Further, the innate variability in fatigue life would require the testing of a substantial number of bearings in each category. Tests were conducted on a typical hinge pin bearing size, under controlled conditions simulating those encountered in helicopter flight. Results of these tests are intended to provide a basis for future design that will insure attainment of a predetermined level of performance.

It was originally proposed to test three types of bearings:

(1) Caged-type needle roller bearing

(2) Tapered roller bearing

(3) Spherical roller bearing

Also to determine the variation of life in terms of:

(1) Radial load

(2) Speed in cycles per minute

(3) Amplitude of oscillation

(4) a. Oil lubrication

b. Grease lubrication

The test program as formulated and pursued in the life-testing phase covered the following conditions:

Load

(1) 20,000 lb

(2) 14,000 lb

(3) 10,000 lb

Speed

(1) 520 Cycles per minute

(2) 260 Cycles per minute

(3) 130 Cycles per minute

Amplitude of oscillation

(1) $\pm 9°$ Angle

(2) $\pm 5°$ Angle

(3) $\pm 1°$ Angle

Lubrication

(1) Oil - MIL - L - 25336, Sinclair L-743

(2) Grease - MIL - G - 25537, Aeroshell 14

During progress of the initial life tests, further investigation was made of the potential use of the tapered and spherical roller types of bearing. Suitable inquiry disclosed that there was currently no use being made of such bearings, and therefore it was decided to omit both tapered and spherical roller bearings from the test program.

Additional tests were authorized for the caged-type needle roller bearing including those in the low speed range of 130 cycles per minute. Likewise the extent of limit load testing was increased to provide opportunity for further investigation of the effects of work hardening on bearing life.

The particular bearing selected for use in the test program is essentially that used in both vertical and horizontal hinges of the Vertol H21, an aircraft type in use by the armed services and in commercial transport. The bearing is a reasonable mean size, in the range of bearings in operational equipment, and was considered to be typical in application. (See Figure 1.)

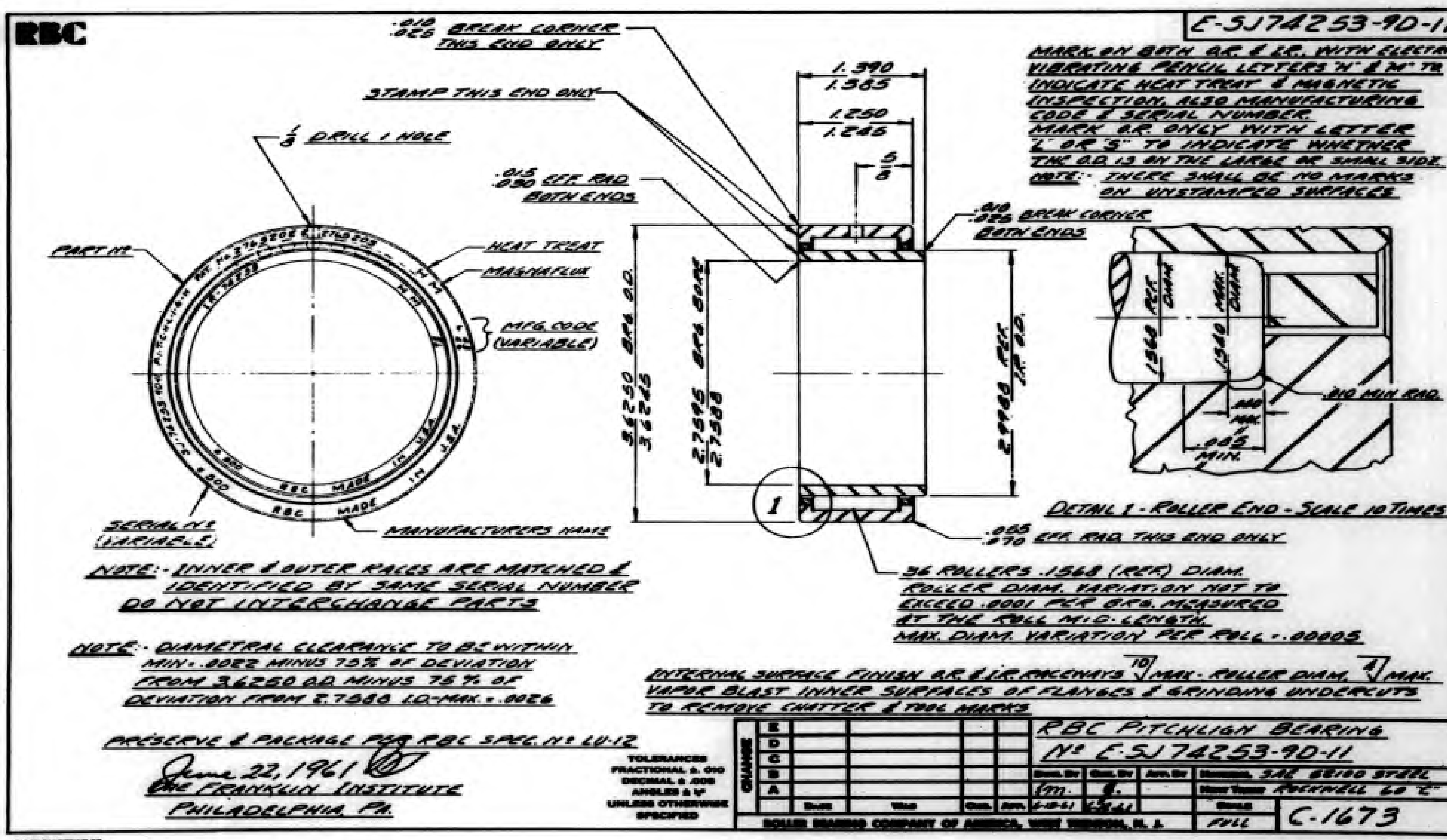

Fig. 1 - DRAWING OF TEST BEARING

TEST RESULTS

A complete log of the tests performed under this program may be found in Appendix D, Tables D-34 and D-35. Upon completion of all tests under any one set of conditions, the data for the individual tests were tabulated in order of the number of hours accumulated up to the time of removal. A table for each set of conditions may be found in Appendix D, Tables D-1 to D-33. Tables I, II and III list all bearings tested and serve as an index to Appendix D and as a summary of the test program. Each bearing was classified as a failure or non-failure, in accordance with the presence or absence of flaking breakout. The contact pattern impressed on the raceways and rollers during progress of the test shows a range of chromatic values varying from light amber to a brilliant shade of blue, and no doubt betrays the presence of a thin surface oxide. Photographs of six typical results are shown in Figure 2 with bearing serial numbers noted.

In order to present the results of any particular test series in a generalized and usable form, it is desirable to express the cumulative percentage of failure as a function of the time in service. This transformation is readily accomplished by considering sub-intervals in the sequence of time and calculating for each an average percentage failure rate. In some cases the sub-interval may be one test. The average rate for the interval is determined as the ratio of the number of bearings showing fatigue failure during the interval to the total number of bearings exposed to failure during the same interval. The cumulative percentage of failure is obtained by numerical integration of the successive interval rates, and this provides a generalized summary of the test results.

Extensive study and analysis of the cumulative percentage failure rates, derived for the many test runs, revealed a consistent pattern of variation. It was found that the cumulative failure is essentially a linear function of time (see insert in Table IV); with one straight line characterizing the relationship below the median life, and a different line applying in the region

Table I

LIFE TEST SERIES

Speed cpm	Angle	Lubricant	20,000 lb Load			14,000 lb load			10,000 lb. Load		
			Test Symbol	No. of Bearings	Appendix Table	Test Symbol	No.of Bearings	Appendix Table	Test Symbol	No. of Bearings	Appendix Table
520	±9°	Oil	A	24	D-1	B	48	D-2	G	40	D-7
		Grease	a	18	D-15	b	26	D-16			
	±5°	Oil	E	83	D-5	F	19	D-6	H	31	D-8
		Grease	e	4	D-19						
	±1°	Oil	C	5	D-3	D	12	D-4			
		Grease	c	9	D-17	d	4	D-18			
260	±9°	Oil	A'	34	D-9	B'	43	D-10			
		Grease	a'	5	D-20	b'	24	D-21			
	±5°	Oil	E'	18	D-13	F'	14	D-14			
		Grease									
	±1°	Oil	C'	13	D-11	D'	13	D-12			
		Grease				d'	4	D-22			
130	±5°	Oil	E"	8	D-28	F"	4	D-29			
		Grease	e"	4	D-30	f"	4	D-31			

Table II

LIMIT LOAD TEST SERIES

Number of Cycles of Prestress *	Prestress 30,000 lb.			Prestress 25,000 lb.		
	Test Symbol	No. of Bearings	Appendix Table	Test Symbol	No. of Bearings	Appendix Table
80,000	L3	24	D-25	L4	16	D-26
40,000				L2	40	D-24
20,000	L1	49	D-23			
5,000	L5	24	D-27			

* 10,000 cycles corresponds to 1% of B10 life as previously observed in "E" test. The bearings subsequently life tested under conditions of the "E" test.

Table III

SUPPLEMENTARY TEST SERIES

Load lb	Speed cpm	Angle	Lubricant	Test Symbol	No. of Bearings	Remarks
20,000	520 260	$\pm$ 5°	Oil	EE'	6	Each speed for alternate 24 hour periods.
14,000	520 260	$\pm$ 5	Oil	FF'	20	Each speed for alternate 24 hour periods
20,000	520	$\pm$5°	None	FC	7	Fretting corrosion
				DR	25	Dummy replacement bearings used for replacement in limit load life tests
				A°	9	Bearings used in proof testing machines and preliminary runs.

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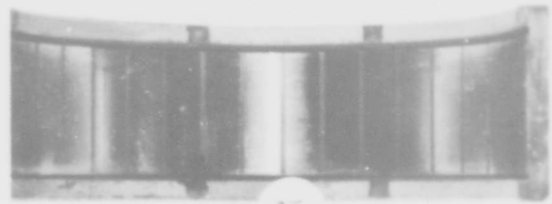
a. Unfailed Inner Race
14,000 lb Load, ±1° Angle



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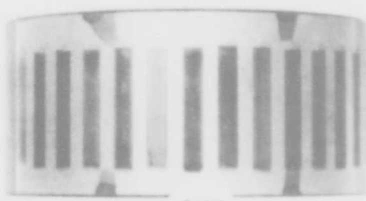
b. Unfailed Outer Race
14,000 lb Load, ±1° Angle



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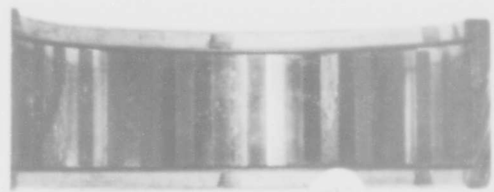
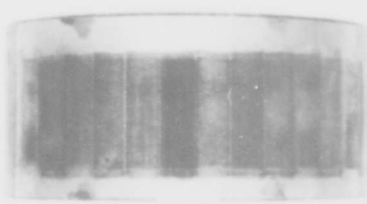
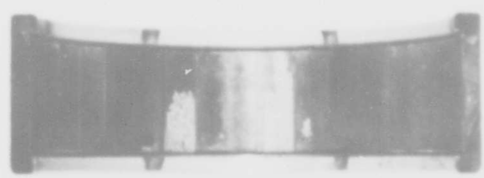
c. Unfailed Inner Race
20,000 lb Load, ±5° Angle



d. Failed Outer Race
20,000 lb Load, ±5° Angle



e. Unfailed Inner Race
20,000 lb Load, ±9° Angle



f. Failed Outer Race
20,000 lb Load, ±9° Angle

Fig. 2 - PHOTOGRAPHS OF TYPICAL TESTED BEARINGS

above the median life. This convenient relationship facilitates application of the least-square method in deriving two functional expressions from the experimental data. Intuitively it would seem that this relationship should be continuous, and it may well be that a continuous function could be made to fit the test data.

It is of interest to note that the procedure as outlined utilizes all of the information obtained in test, including the observed lives of both failed and non-failed bearings. This is important because testing could not always be continued until the actual development of fatigue failure. A number of bearings were removed prior to failure, as the result of terminating a test run and replacing all bearings. Others were removed because of suspected failure that was not confirmed by subsequent examination.

Much able and proficient effort by others has been directed to the study of procedures suited to the presentation and reduction of data obtained in the endurance testing of bearings. Many of these are well known and have served a very useful purpose in bringing advanced statistical concepts into actual use to simplify the problem of evaluating bearing life. The most common procedure in recent years is to arrive at a straight line that best represents the data on a plot in which the abscissa is a logarithmic scale of time and the ordinate is a special scale based on the Weibull probability distribution. However, it is recognized that there are instances in which the simple linear plot of the Weibull distribution fails to define adequately the actual life distribution beyond a limited range of values. Such an instance has occurred here.

Since interest is centered on the B-10 life, or 10% failure point, the linear relationship of hours of test vs percent failure for each series of tests up to approximately the median or 50% point was calculated. Appendix E presents a complete sample calculation for one test. Table IV shows a tabulation of these calculated relationships (slope and intercept) for each set of conditions wherein enough data were available to warrant such a calculation. The linear relationship, of different slope, for the data beyond this point is not included here. Table IV also includes a tabulation of the B-10 life (in cycles) that will be discussed later.

TABLE IV

SLOPE AND INTERCEPT DATA

LIFE TEST SERIES

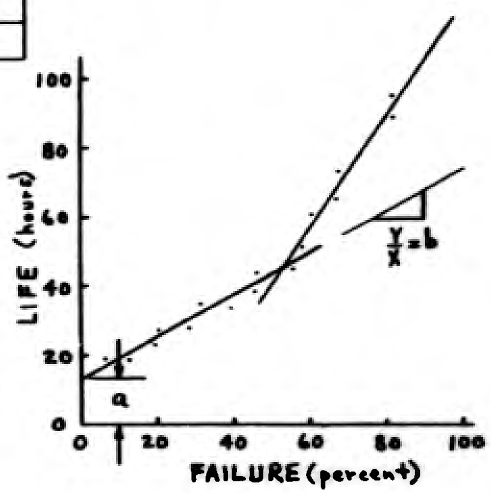
Speed cpm	Angle	Lubricant	20,000 lb Load Test	Slope b	Intercept a	B-10 Life 10^6 cycles	14,000 lb Load Test	Slope b	Intercept a	B-10 Life 10^6 cycles	10,000 lb Load Test	Slope b	Intercept a	B-10 Life 10^6 cycles
520	± 9°	Oil	A	1.26	18.9	0.983	B	4.01	77.7	3.675	G	14.72	522.5	20.89
		Grease	a	2.92	24.8	1.685	b	18.58	107.4	9.148				
	± 5°	Oil	E	1.76	10.1	0.864	F	7.75	133.4	6.580	H	23.30	379.8	19.12
260	± 9°	Oil	A'	1.56	29.5	0.704	B'	7.34	96.0	2.643				
		Grease					b'	18.99	177.9	5.738				
	± 5°	Oil	E'	3.64	43.7	1.250	F'	25.30	15.9	4.195				
	± 1°	Oil	C'	8.39	27.3	1.735	D'	10.19	303.4	6.323				
130	± 5°	Oil	E"	9.96	120.0	1.709								

LIMIT LOAD TEST SERIES

Speed 520 cpm		Lubricant Oil	
Angle ± 5°		Load 20,000 lb	
Test	Slope b	Intercept a	B-10 Life 10^6 cycles
L1	3.62	22.9	1.844
L2	1.74	27.1	1.388
L3	1.00	-5.3	0.147
L4	1.34	7.2	0.643
L5	4.95	-4.9	1.392

SUPPLEMENTARY TEST SERIES

Speed (520) (260) cpm	Lubricant Oil	
	Angle ± 5°	
Test	EE'	FF'
Load lb	20,000	14,000
Slope b	2.48	8.25
Intercept a	13.4	0.26
B-10 Life 10^6 cycles	0.894	1.938



The established practice exists of assigning to a particular design of bearing a load rating factor that is based upon such elements of the design as roller diameter, number of rollers, pitch diameter of roller set, effective length of roller contact, etc. (See American Standards Association B3.11-1959). For the bearing used in this test program this Basic Load Rating (BLR) is 9860 lb. This factor is to be used to estimate probable bearing life in conventional rotating service. However, the bearing of interest in the test results being reported herein operates under oscillating instead of rotating load. In this event the conventional practice is to derive a factor based on the "Aircraft Static Capacity" (ASC) of the bearing. This is calculated by using the expression ASC = 12,000 $D\ell$ (N-3) in which D is the roller diameter, ℓ is the effective length of the roller, and N is the number of rolling elements. In the bearing at issue D = 0.1568, ℓ = 0.8754, N = 36 for a value of the ASC of 54,300 lb. Now the bearing manufacturer recommends that, for oscillating service through a small angle, the Basic Oscillating Load Factor (BOLF) is determined by multiplying the ASC by 0.354. In the case at hand, the value of the BOLF is 19,200 lb.

All of the established practice discussed thus far is based on a B-10 life of one million cycles, i.e., that 90% of a group of apparently identical bearings will endure the rated load and complete or exceed a life of one million cycles before the first evidence of fatigue develops. This is conventionally expressed as:

$$L = \left(\frac{C}{P}\right)^3 \text{ million revolutions}$$

where L is the life in millions of revolutions

P is the constant stationary radial load

C is the factor discussed previously

i.e., BLR for rotating service

BOLF for oscillating service

If the life is expressed in hours (H) and the speed of rotation (or oscillation) (N, in rpm) is used, the following expression results:

$$HN60 = \left(\frac{C}{P}\right)^3 \quad \text{or} \quad C = P \sqrt[3]{HN60}$$

If the cubic relationship shown above is presumed to be valid, the quantity C is the all important factor for a designer to know in order to specify a bearing for an application with some assurance that 90% of such bearings will reach or exceed a predictable life.

The value of C discussed previously, BLR = 9860 lb, has been established from a calculation based on tests of many bearings under rotating load conditions. Few, if any, tests have been made previously to establish a similar factor of C for oscillating load conditions. The calculation of 19,200 lb shown above is the best available estimate of the factor for the bearing tested in the program just completed. In addition to any other results that may appear, a comparison of the values of this factor determined experimentally under a variety of operating conditions in this program should enable a designer to use this type of bearing under oscillating load conditions with more assurance than he could reasonably have had previously.

For each series of tests shown in Table IV the value of the 10% life (in hours) was calculated and, along with the appropriate value of load and speed, used to calculate a value of C (See Appendix E). These are shown in Table V. Since a requirement of this program was that the load rating for a B-10 life of 15 million cycles be established, Table V also includes the factor C calculated on the basis of 15 million cycles. As an additional aid in the use and interpretation of these data for the bearing tested the nomograph in Figure 3 was prepared. For a desired condition of load and speed the value of $P \sqrt[3]{N}$ is calculated. Entering the nomograph with this value, an estimate of the B-10 life in hours can be found by a straight line through the appropriate value of the load rating factor. Lines indicating some of the test results of this program are shown on the chart.

In terms of the conventional load rating factor C, the results leave something to be desired. For commercial expediency the value of C for a particular bearing is treated as a constant over limited ranges of load and speed, i.e., not changing with changes in speed or load or any other outside

TABLE V

LOAD RATING FACTOR DATA

LIFE TEST SERIES

Speed cpm	Angle	Lubricant	20,000 lb Load Test	20,000 lb Load C B-10 (1 x 10^6)	20,000 lb Load C B-10(15 x 10^6)	14,000 lb Load Test	14,000 lb Load C B-10(1 x 10^6)	14,000 lb Load C B-10(15 x 10^6)	10,000 lb Load Test	10,000 lb Load C B-10(1 x 10^6)	10,000 lb Load C B-10(15 x 10^6)
520	± 9°	Oil	A	19845	8047	B	21604	8760	G	27519	11158
		Grease	a	23776	9641	b	29216	11847			
	± 5°	Oil	E	19040	7720	F	26180	10615	H	26670	10814
260	± 9°	Oil	A'	17800	7218	B'	19355	7848			
		Grease				b'	25095	10176			
	± 5°	Oil	E'	21550	8738	F'	22810	9168			
	± 1°	Oil	C'	24000	9732	D'	25900	10502			
130	± 5°	Oil	E"	23915	9697						

LIMIT LOAD TEST SERIES

Speed 520 cpm; Lubricant Oil; Angle ± 5°; Load 20,000 lb

Test	C B-10 (1 x 10^6)	C B-10(15 x 10^6)
L1	24468	9921
L2	22267	9029
L3	10548	4277
L4	17235	6988
L5	22298	9041

SUPPLEMENTARY TEST SERIES

Speed (520) (260) cpm; Lubricant Oil; Angle ± 5°

Test	EE'	FF'
Load, lb	20,000	14,000
C B-10(1 x 10^6)	19248	17457
C B-10 (15 x 10^6)	7805	7078

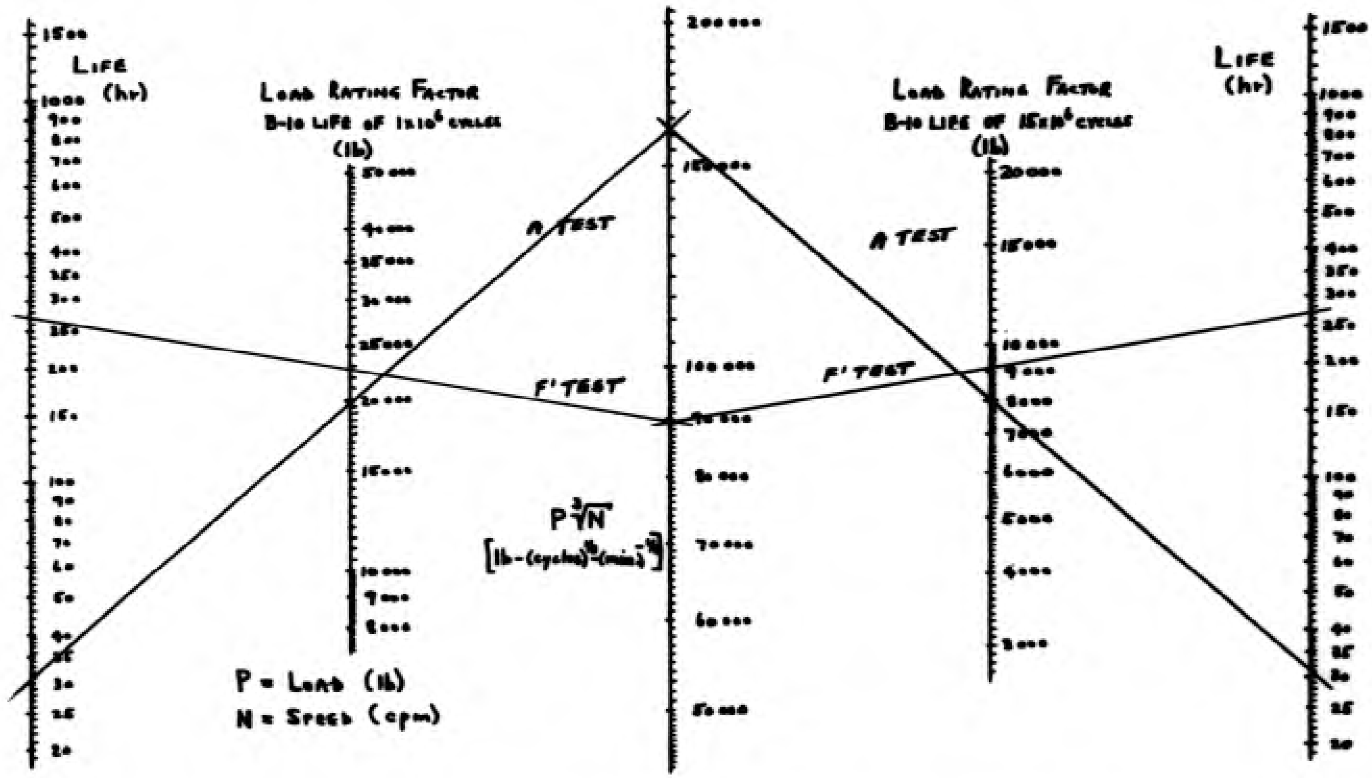


Fig. 3 - NOMOGRAPH OF LOAD RATING FACTOR

influence. However, the values of C for the conventional life tests (Table V) range from 17,800 to 29,216. Even if some separation is made, say on the basis of lubricant, the values range from 17,800 to 27,519 for oil and from 23,776 to 29,216 for grease. It is quite evident from this that C is not constant and that some degree of interaction exists between the several independent variables.

The results of the limit load tests can also be reviewed in terms of the load rating factor C. By comparing the results of these tests (Table V) with the pre-test conditions shown in Table II a potentially useful characteristic emerges. It appears that a higher value of load rating is produced by prestressing the bearing for a limited number of cycles at a higher load than the test load. As might be expected the gain appears to reach an optimum, i.e., too high a preload or too many cycles of preload can reduce the load rating to a value less than that without preloading. In addition to the load ratings, which are based on the 10% failure point, it is useful to consider the values of slope shown in Table IV and to calculate the median life (50% failure point). While more tests would be desirable, the data permit a tentative conclusion that prestressing the test bearing at 30,000 lb for between 5000 and 20,000 cycles willincrease the load rating.

The alternating speed tests (EE' and FF') shown in Table III were an attempt to simulate to a small degree the great variety of operating conditions to which a bearing is exposed in actual field service. Comparison of the load rating factor C in Table V, based on an average speed of 390 cpm, with their corresponding single speed tests (i.e., EE' to E and E', and FF' to F and F') shows no consistent trend. The value of C for the FF' series falls well outside the range bracketed by F and F'. The reason for this is not clear. A somewhat better result, at least more consistent, emerges upon comparing the slopes of these tests shown in Table IV. Here the slopes (failure rates) of both alternating speed tests fell between the extremes set by the single speed tests. This gives at least a small basis for confidence in the use of the values of C shown in Table V across a range of operating conditions.

Another method of presenting the results is to compare the respective values of the B-10 life for each test. Figure 4 is an attempt to show, as much as possible, the effects of load, speed, lubricant, amplitude of oscillation and prestress on one chart. Rather close scrutiny - and caution so that only similar conditions are compared - will enable one to observe the results of the effects mentioned above. This chart is simply a graphical presentation of the B-10 life results shown in Table IV.

In order to judge the effect of speed on the B-10 life, comparison is made of the tests within the groups A-A', E-E' - E", B-B', F-F', and b-b'. All of these except the E group indicate that the highest speed produces a B-10 life approximately 50% higher than that for the medium speed. The results of the E group indicate the reverse of this and offer the further confirmation of the lowest speed vs medium speed relationship. In searching for an explanation for this conflicting evidence, one observes that the four groups showing similar results are tests having the load or amplitude of oscillation or the lubricant different from the E group. Thus it would appear that an interaction between these independent variables has taken place and that a conclusion regarding speed is hardly justified on this evidence.

Groups A'-E'-C', A-E, B'-F'-D', and B-F are comparable in order to study the effect of the amplitude of oscillation. All of these except the A-E group indicate that the highest amplitude of oscillation produces the shortest life. Again, as with the speed effect, an interaction seems quite probable since the A-E group is the only one subjected to both high load and high speed.

The effect of the lubricant on the B-10 life can be studied by comparison of groups A-a, B-b, and B'-b'. Without exception the tests with grease produced appreciably longer life than the tests with oil. It is unfortunate, perhaps, that more tests with grease were not made under other combinations of load, speed and amplitude since the results obtained, though consistent, contradict those experienced in field service. Elaboration on the latter point will be made in the section entitled Interpretation of Results.

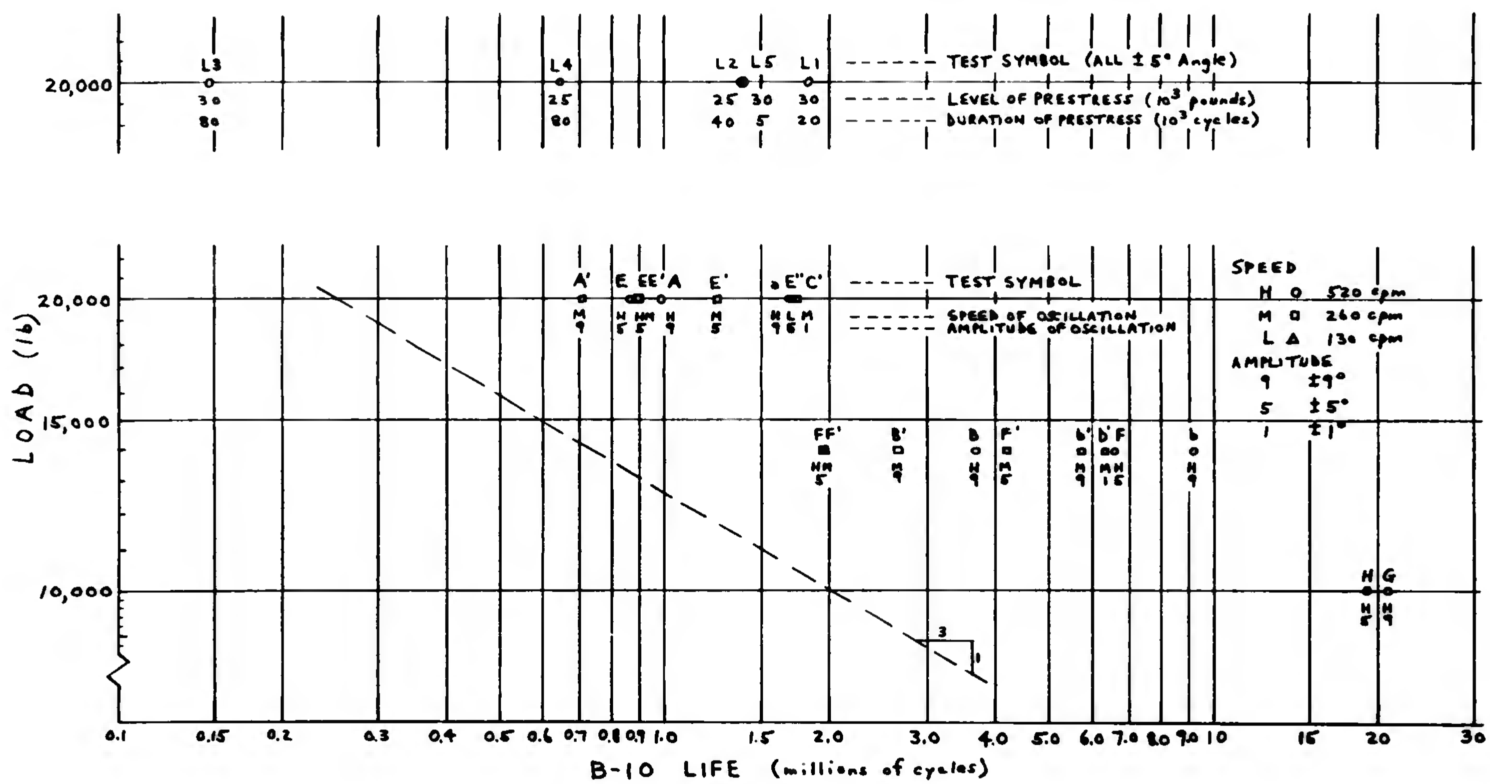


Fig. 4 - B-10 LIFE (Millions of cycles)

The load is perhaps the most important independent variable in that the convention of the life varying inversely as the cube of the load has had wide acceptance for many years for bearings under rotating rather than oscillating conditions. The groups A-B-G, E-F-H, E'-F', C'-D', and a-b are comparable in studying the effect of load on life. As a convenience in this comparison a line having a 1/3 slope is drawn on Figure 4. Lines connecting each pair of points in the groups just mentioned have slopes that indicate approximately the following exponents for the life-load relationship:

Points	A-B	B-G	E-F	F-H	E'-F'	C'-D'	a-b
Exponent	3.7	5.2	5.8	3.2	3.4	3.6	4.7

These values deviate quite a bit from an exponent of three. While the inconsistencies among these values leave much to be desired, at least one can state that they are all greater than three. Other than that general observation one is forced, once again, to conclude that interaction among the independent variables masks a more exact value for the life-load relationship.

The results of the limit load tests are plotted separately on Figure 4 to prevent confusion. One should bear in mind that all of these were run under conditions exactly the same as the E test. Thus the results should be compared with the B-10 life of 0.864×10^6 cycles resulting from the E test. It is apparent that increasing the number of cycles of prestress causes the B-10 life to pass through the optimum noted previously in the discussion of the load rating factor, i.e., increasing the number of cycles of prestress causes first an increase in life and then a decrease in life. The L1 test exhibited more than twice the life of the E test. It is possible that some other combination of prestress load and duration may do even better.

The results of the alternating speed tests (EE' and FF') were expected to fall between the results for the respective two single speed tests. Unfortunately, only the EE' test produced a B-10 life that fell between the value for the E test and the value for the E' test. The B-10 life for the FF' test fell well outside the range bracketed by the F and F' tests. Again, interaction among the independent variables seems to be the explanation.

An effort was made to explore further the condition of fretting corrosion, known to have frequently occurred in service operation. A number of bearings were cleansed in solvent to effectively remove all lubricant and were then assembled in the test machine. When operated without lubrication, it appeared that contact corrosion immediately set in and that the bearing rapidly lost its internal running clearance and approached the point of seizure. Upon removal these bearings showed an accumulation of oxide debris in the contact areas and pitting and attrition of the surface. No doubt had these bearings been continued in operation abrasive wear would have progressed in the sequence of fretting corrosion. In this instance the conditions were extreme, but the circumstances imply the dominant part that can be played by lubrication.

It seems probable that the earlier prevalence of fretting corrosion, experienced in the service operation of rotor bearings, was in a large measure induced by vagaries concerned with lubrication and contamination resulting from progressive accumulation of moisture. In some instances it is possible that a high level of ambient vibration in components of the assembly was conducive to that type of failure.

Importance was attached to a comparison of bearings removed from actual service with those subjected to machine test in this investigation. Bearings employed on the H-21 helicopter are essentially the same type and size as those used in the test program. A number of such bearings were obtained and subjected to examination.

A conspicuous difference affecting both failed and unfailed bearings removed from service as compared to bearings removed from the test machine, is the continuity of the load contact pattern impressed upon the raceways. This quite evidently is due to a progressive creeping displacement of the cage and roller assembly, during service, with respect to the raceways. On the other hand, both failed and unfailed bearings operated in the test machine consistently showed a load contact pattern in a fixed index relationship between raceways and rollers, and of an angular extent corresponding to the amplitude of oscillation at which it had been driven.

Failed bearings removed from service presented varying degrees of flaking breakout that appeared to duplicate in every way the condition obtained in machine-tested bearings. In some instances roller breakage had occurred, a condition that appears to result when the raceway flaking has progressed in continued service. Some such failures (roller breakage) were observed in the test program, usually in the case of heavy loads and the more advance degrees of flaking.

Less frequently, instances occur in service wherein extensive damage results from the effects of fretting corrosion, and the raceways acquire a typical "washboard" appearance.

The type of failure induced in these oscillating tests is to all appearances characteristic of failure in rolling-contact bearings subjected to rotation. Spalling or flaking breakout occurs at focal points in the contact of raceways and rollers and spreads progressively by breakdown of the adjacent boundary areas. This form of surface deterioration and damage appears to be a complex fatigue phenomenon and is generally accepted as such. The tests accomplished in the present investigation provide additional information as to various circumstances under which it will occur. However, no opportunity was afforded to explore the basic complex mechanism of surface fatigue.

None of the bearings operated in the test program gave discernible evidence of the occurrence of fretting corrosion in the contact areas of the raceways. However, all tests were conducted with adequate supply of lubricant, either oil or grease, and there was essentially no opportunity for contamination by moisture. The cyclic motion imparted to the bearing was essentially that of the driving speed and amplitude. There was no pronounced vibration imposed by surrounding components of the machine.

TECHNICAL DISCUSSION

Interpretation of Results

In a bearing rotating under radial load the entire load carrying area, save the unloaded side of the stationary race, is successively subjected to repetitive stress. The magnitude of the stress is dependent upon the position angle of the contacting roller, and the frequency of recurrence is dependent upon the kinematic relations of raceways and rollers. Thus, in the normal case of a rotating bearing the cumulative fatigue action is concerned with the bearing as a whole, and life performance is the integrated result of a complex pattern of stress and cyclic repetition.

In the circumstance of a bearing not rotating, but oscillating continuously about a fixed mean position, fatigue damage is concentrated in localized areas. Each bearing subjected to this condition presents within itself a composite of load tests, in steps commensurate with the load angle of successive roller contacts. Regarded as a stratified sampling of fatigue performance, this condition is fundamentally different from that obtained with the stress spectrum induced by rotation of the bearing. Moreover, in an oscillating bearing, the passage of a roller through its contact zone comprises the full spectrum of speed of loading within the half width of the zone. This is to be compared with the uniform speed of loading that characterizes the entire periphery of each raceway in a rotating bearing. These differences may explain why the constants for the life equation for oscillating bearings differ from those for rotating bearings.

Another consideration worth mentioning is that an oscillating bearing, where the outer race is not allowed to creep as it does in service, subjects only a small portion of each raceway to the maximum contact stress and subsequent fatigue damage. In service the outer race is allowed to creep and the resulting fatigue damage is likely to occur at any point over a much larger area. It would seem, therefore, that bearing lives in laboratory tests would tend to have considerable scatter since they use only a small portion of races that are not exactly homogeneous, e.g., directional properties in the metal. Such reasoning may explain the scatter of data for certain test conditions in which only a few tests were run.

An important disclosure in this investigation is that bearings subjected to relatively heavy loading, and oscillated rather than rotated, do not conform to the established pattern of life performance. It appears that two separate influences are present affecting the incidence of fatigue damage: first, the normal cumulative sequence of failure wherein the material structure is progressively impaired; second, a cumulative sequence of work hardening whereby the physical properties of the stressed material are raised. Each of these stress dependent phenomena are operating coincidentally, and together they determine the rate at which the deterioration develops and progresses.

Seldom is the failure of a ball or roller bearing catastrophic in nature, but usually it is a slowly progressive type of deterioration. This fact explains the widespread use of such bearings in airborne equipment and in other installations where failure constitutes a serious hazard. While continued operation of a bearing past the point of incipient failure is safe enough for a limited time, the operation obscures the early evidence of damage. This investigation afforded the opportunity to observe the progressive nature of fatigue damage, through the concentration of stress on contained and localized areas. In this investigation, the failures observed were invariably that of surface flaking affecting localized areas to a limited extent. The actual spalled or flaked areas are bounded by sharp edged and precipitous junctions with the unaffected surface. In some instances, the unstressed surface adjacent to the contact area shows a faint pattern of arborescent fissures or alternatively a symmetrical pattern of faint lines disposed obliquely to the boundary of the contact area. It is believed that these features are more or less characteristic of surface fatigue induced by contact stresses, but the relatively sharp line of demarcation between stressed and unstressed areas affords improved opportunity for their detection. Although some doubt exists as to the exact nature of the inception and propagation of fatigue damage and its relation to the elastic and plastic deformation incurred, there is no doubt whatever that the failures encountered in this test program are essentially a typical form of surface fatigue. As such it is to be expected that endurance as observed will relate directly to the amplitude and frequency of the imposed stress. Failure to comply with that relationship may be accepted as evidence of interaction among the variables involved, and in these tests the discrepancies observed are believed to originate in the degree of work hardening sustained with progress of the test. However, detailed metallurgical examination of the tested bearings is needed to establish the validity of this belief.

Considerable effort was expended in a search for an expression which more adequately describes the relationship among life, load, speed, angle of oscillation and lubricant, to no avail. The limitations of scope, funds, and time prohibited additional effort. While perhaps more can and should be done along these lines such effort will have to be the substance of future projects. The data are presented here in fairly complete fashion so that any reader can attempt such a task.

While the data shown in Table V do not fit into a neat package, still considerable use can be made of them by a designer. The extremes of load, speed, and angle of oscillation used for these tests to some extent bracket a range of each of these variables within which the operating conditions may be found. In this case the designer can benefit from some of the general trends in the data without necessarily waiting for the derivation of an exact relationship between the variables. For example, considering only the effect of changing the load, the value of C in all cases increased as the load decreased (Compare A to B, B to G, E to F, F to H, a to b, E' to F' and C' to D'). Considering the variation in lubricant alone, the value of C for grease was higher in all cases than that for oil. In making comparisons, the significance of the values of slope and intercept in Table IV should not be ignored. The slope is an especially good criterion since it extends over a broader range and a comparison of slopes is not dependent upon an almost accidental crossover point.

It is of interest to the designer how these results compare with the Basic Oscillating Load Factor (BOLF) mentioned previously. Based on a B-10 life of one million cycles the BOLF for this bearing was calculated as 19,200 lb. The results in Table V indicate that only two tests out of the sixteen tests in the Life Test Series produced values of C lower than 19,200. Thus, over the ranges of load, speed and amplitude used in these tests (and for two lubricants), the BOLF proves to be a conservative estimate.

The conservatism inherent in the BOLF can be expressed in a somewhat different manner. If an application calls for a B-10 life greater than one million cycles, an estimate is needed of the load that corresponds to the longer life. Based on the test results reported herein the conventional use of the cubic relationship between load and life to calculate the longer life would be conservative. The exponents tabulated in the previous section ranged from 3.2 to 5.8. **Obviously, this conservatism holds only** for loads equal to or less than the BOLF.

The tests using grease as a lubricant exhibited consistently longer B-10 life than the tests using oil. It should be noted that these results apply only to these two lubricants and not to greases or oils as general classes of lubricants. However, the differences in B-10 life were substantial (compare groups A-a, B-b, and B'-b') and well-founded (no less than 18 bearings tested under any one condition) and leave little doubt as to the relative performance of these two lubricants under controlled laboratory conditions. Any difference between these results and the results of these lubricants under field service conditions should be attributed to the lack of controlled conditions in the field. This applies particularly to the provisions for supplying adequate quantities of lubricant to the bearing and for insuring the absence of moisture and dirt contamination in the bearing housing.

If nothing more the results presented herein should strike a note of caution for the designer. It is apparent that the difference in life between lubricants under controlled conditions is already substantial and may be increased under the relatively uncontrolled conditions prevalent in the field.

Upon close scrutiny of Figure 4 the results regarding the effects of the speed and amplitude of oscillation appear to be reasonably consistent at first glance. However, the inconsistencies that remain are substantial differences in B-10 life and are well founded statistically; hence they cannot be cast aside. A number of explanations have evolved in the study of these results, but none has been capable of satisfying the diametrically opposed ordering of the B-10 lifes of the comparable groups of tests. This applies to both the effect of speed of oscillation and the effect of amplitude of oscillation.

A previous discussion of conventional design practice pointed out that a bearing is expected to have a B-10 life of a certain number of cycles regardless of the time (within reasonable limits) that is required to attain this number of cycles. Another practice, used by at least one manufacturer, is to ignore the amplitude of oscillation as long as it remains within one roller spacing (as is the case with all of the tests in this program). Taking these two effects together, then, common practice would say that the points on Figure 4 for each level of load should merge as one point. Ignoring tests FF' and b, the spread of points at each of the medium and high load levels encompasses a ratio of B-10 life of approximately 2.5 to 1. This approaches the difference in life that the cubic relationship yields for these two levels of load and, hence, offers a major obstacle to the use of these data with any sort of precision to a design application. Without further analysis (which may be as unproductive as the large amount done to date) or further testing, or both, not much more can be said about these effects. These tests have revealed, perhaps for the first time, characteristics of bearing performance that are important to efficient bearing design. It is unfortunate that they are also beyond logical explanation at this stage.

Reliability

Reliability is formally expressed as the probability that a given component will not fail before a specified time. Time between overhaul (TBO) in operational maintenance is based upon the experience gained in performance of critical components of an assembly. Operational efficiency, in terms of availability, is largely dependent upon an overhaul schedule based upon uniform levels of reliability. Thus, component parts, such as bearings, may be confidently returned to service when it is known that the characteristic pattern of reliability assures a continuing standard of performance throughout the ensuing period. Failures that occur and necessitate replacement in the interval between scheduled overhaul exact an undue penalty in operating efficiency. Unfortunately, the hazard in this respect counters the inducement to explore the potential increase of life, in service operation. Thus, frequently the practice of scrapping parts at every overhaul is established, and there is no opportunity to gain experience in extended operation.

The tests conducted in this investigation were unhampered by such limitations, and bearings were continued in operation until the first appearance of failure. The entire spectrum of operational conditions was simulated in the series of tests accomplished. In the aggregate these must reflect the characteristic trends that will result in actual service operation. The cumulative percentage of failure within a given interval of time is numerically equal to the complementary value of the reliability.

The conventional criterion used in evaluating bearing performance is the so called B-10 life. This is the life that will be attained by 90% of a group of bearings when subjected to essentially the same operating conditions in service. It serves as a convenient but arbitrary reference point, at which 10% of the group of bearings will be expected to show first visible flaking. Thus the B-10 life corresponds to a reliability of 90% and has found general use in industrial practice as a measure of bearing performance.

The more exacting requirements for airborne equipment, and the efficient scheduling of overhaul maintenance, are dependent upon a more extensive knowledge of the variation of bearing reliability with increasing time in service. Time between overhaul is a basic factor in maintaining operating efficiency and is established in terms of the principal components subject to deterioration in service. Ball and roller bearings are in this category, and their replacement at overhaul is usually a matter of major concern. Any indication of impairment or incipient failure is mandatory cause for rejection. However, the problem is posed by those bearings betraying no evidence of deterioration in service, in which case there is need to evaluate the probability that those bearings will accomplish the ensuing period of operation with no perceptible increase in the percentage rate of failure. Indication of failure encountered in the course of the operating period between overhauls necessitates the grounding of an operational unit and penalizes both efficiency and economy of operation. There is a need for factual information to evaluate the progressive degree of reliability of bearings through successive periods of operation between overhaul. Such informational data cannot be accumulated in service operation without accepting some degree of hazard, and the generally observed practice has been to replace all bearings at service overhaul.

The present test program has accumulated life data under controlled conditions of loading and affords a basis for empirical determination of reliability throughout the observed life span of the bearing. The conventional concept of reliability is the probability that the bearing will not fail before, that is, it will survive until, a given time. However, a useful concept in assessing the potential performance of a bearing is the "mean life before failure", characterizing the group. This mean life is a measure of future performance and applies to a group of bearings that are new, without prior service, or to a group that has already attained a given age before failure. Equal values of future "mean life before failure" imply the same level of reliability and any arbitrary disposal of such bearings is an unrewarding extravagance. For example, a typical set of test results are shown in Figure 5. The two lines marked H_F are the least square lines that represent the failed bearings over the regions below and above the 50% point. At 10% interval points along the curve, the area under the complete H_F curve above the level of that point is divided by the percentage life remaining and the values plotted as the H_E curve.

A cursory analysis of the failure pattern as developed in these tests suggests that virtually a constant level of reliability is maintained up to the median life. This means that unfailed bearings removed at a lower interval of life, actually have an average life expectancy equivalent to that of a new and unused bearing. These same circumstances were found to apply in prewar experience with aircraft engine main bearings, and in wartime operation unfailed bearings found at overhaul were repeatedly returned to service without unfavorable experience in the ensuing period.

More extensive analysis of the life performance of the bearing may well belie the prevalent practice of bearing replacement, in an effort to insure a higher level of performance. This may prove to be as ineffective as it is obviously uneconomical in operation.

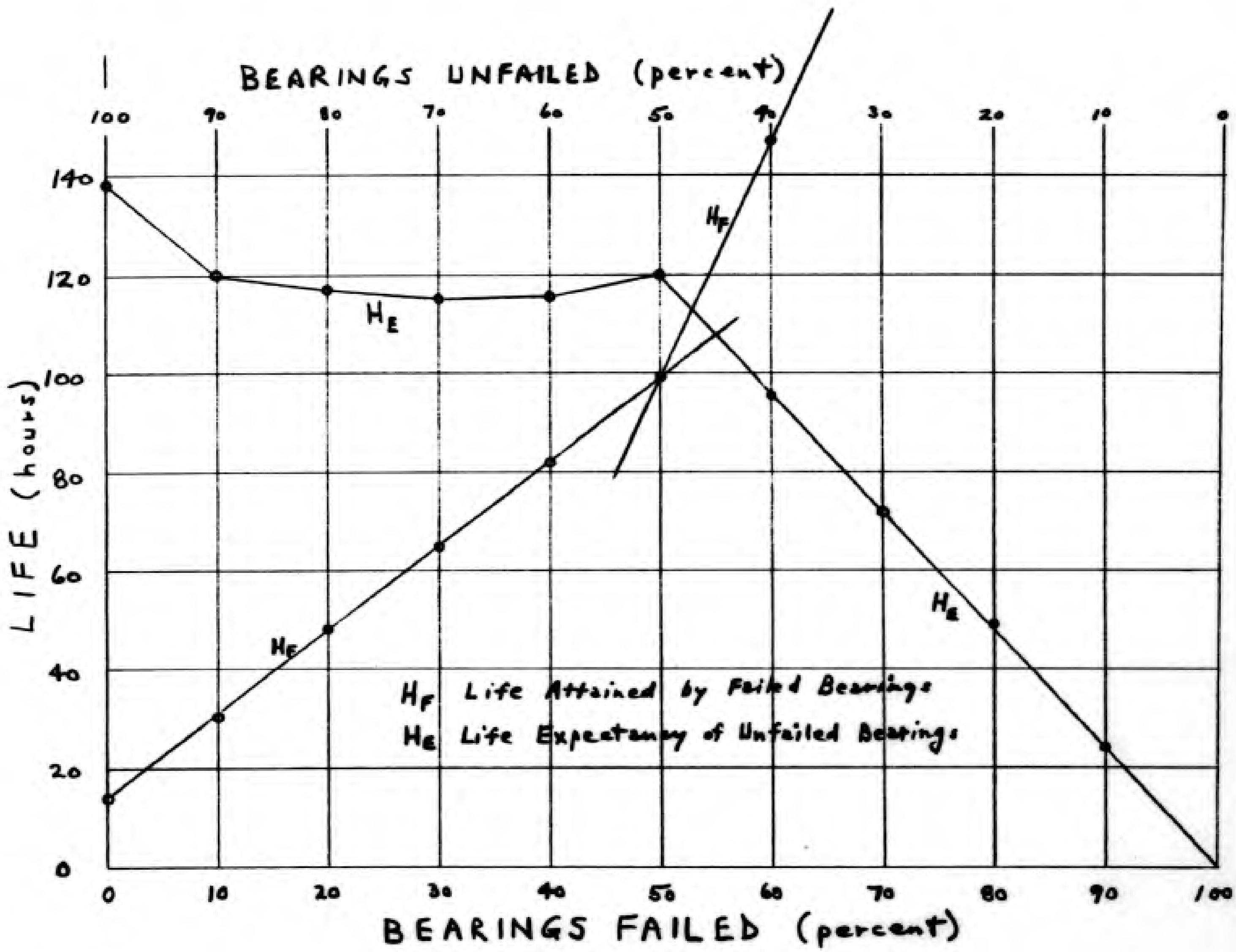


Fig. 5 - EXAMPLE OF RELIABILITY CONCEPT

CONCLUSIONS

1. The preponderance of tests show that the Basic Oscillating Load Factor (conventionally derived from the bearing geometry) is a conservative estimate of the load capacity rating of the test bearing used in this program.
2. Life prediction for values of load lower than the rating in terms of the cube of the applied load and the total cycles of oscillation, is also conservative. In the commonly used relationship of life and load, life = constant/$(\text{load})^n$, the value of n is greater than 3 for the test bearing used in this program.
3. The grease lubricant (MIL-G-25537) exhibited better results (larger load rating factor or larger B-10 life) than the oil lubricant (MIL-L-25336) tested in copious supply and free from moisture and dirt contamination. (Note that this is contrary to service experience).
4. These tests revealed an effect on the B-10 life of both the speed and the amplitude of oscillation, contrary to conventional design practice. However, no orderly relationship to describe either effect was found that could encompass all of the test data.
5. Flaking observed in the life tests conformed to the typical pattern of surface fatigue in rolling contact.
6. Bearings operated under these test conditions show an absence of fretting corrosion and wear.
7. Limit load tests gave an indication of the potential increase of life performance as the result of prestressing the bearing under appropriate conditions.

Other pertinent observations from the test results are noted below:

1. There was an unexplained predominance of outer race fatigue failure, affecting the outboard ends of the roller contact.
2. Flaking and breakage of rollers was of secondary importance.
3. There was no incidence of cage (retainer) failure throughout the test.
4. No evidence was observed of end thrust induced in the bearings.

RECOMMENDATIONS FOR EXTENDED INVESTIGATION

The basic mechanism of surface fatigue as involved in rolling contact is but partially understood. The some 700 bearings operated under controlled conditions in this investigation have been preserved for possible future reference.

Two principal categories of tested bearings are available:

(1) those lubricated with oil,

(2) those lubricated with grease.

A small group tested without lubrication is also available.

Both failed and non-failed bearings with known lives, and representative of the entire range of test conditions, are at hand.

These bearings comprise a veritable storehouse of information, to be elicited by planned and systematic metallurgical examination. Each bearing is identified with a completely known and recorded prior history.

1. Such parts are available for investigation of the process of work hardening and its relation to delay in the inception of fatigue.

2. Such parts are available for investigation of cumulative compressive stress and its relation to surface pitting, prior to occurrence of flaking break out.

3. Such parts are available for investigation of surface oxide film and its relation to the inception of surface break down.

4. It is known that certain surface treatment, as for example, the use of dodecyl alcohol, has proven effective in delaying the onset of surface fatigue damage. It is also known that surface roughness is intimately concerned in formation and persistence of lubricant films, and that these may have a forceful influence on the inception of fatigue. Such promising areas of investigation can be pursued with this operational test facility, permitting a full scale test of bearings.

Not the least important asset of the test installation employed in this investigation is the extensive use to which it has been put in a wide array of tests, serving to define a reference level of performance.

Thus a solid background is provided for comparison in continued experimental investigation, and testing as follows:

1. Determination of rate at which fatigue damage progresses following its inception.
2. Tests under scheduled program of loading, related to spectrum of conditions encountered in flight, serving to confirm basis for averaging of load cycles in prediction of service life.
3. Tests of the bearing under conventional conditions of load and rotation (by suitable modification of the test machine) in order to relate basic oscillating capacity to basic dynamic rating as previously established by the American Standards Association. This to serve as means for relating basic capacity of a bearing to its geometry and size, and so provide a basis for generalizing results obtained in this test.
4. Comparative tests of alternate bearing materials with the advantage of a large test facility in which performance of the existing standard bearing (SAE 52100 steel) has been well established.
5. Comparative tests of modified practice in bearing design, and race finishing procedures, observed in manufacture. The test facility provides means for obtaining prompt and conclusive results in comparison with the presently established standard.
6. The test machine construction can be modified at moderate cost to provide for full rotation in the lower speed range. By this means it would be possible to provide for prestressing of the bearing, with complete treatment of the bearing components, and thus afford a practical means for exploiting the capacity gain that was demonstrated in this investigation.

TEST PROGRAM

Bearing Life Investigation

The principal factors influencing life performance of the bearing were known to be:

1. Applied Radial Load
2. Frequency of Oscillation in cycles per minute
3. Angle of Oscillation
4. Lubrication
   (a) Oil
   (b) Grease

Accordingly a program was prepared, scheduling appropriate combination and range of these variables, to comprise a sequence of life tests under controlled conditions.

Applied Radial Load

Life tests were conducted under constant load, at three levels, consistent with design practice in the use of the bearing. These were:

(a) 10,000 lb.
(b) 14,000 lb.
(c) 20,000 lb.

Speed of Oscillation

Life tests were run at constant speed, at three levels, determined by a range appropriate to bracket contemplated values of rotor rpm. These were:

(a) 520 cycles per minute
(b) 260 cycles per minute
(c) 130 cycles per minute

Amplitude of Oscillation

The maximum angle of oscillation used in the life tests $\pm$ 9°, the minimum angle was $\pm$ 1°. The maximum angle is just short of the critical value of $\pm$ 9.5°, beyond which contact areas of adjacent rollers will overlap. An intermediate angle of $\pm$ 5° was employed in both the life tests and the limit load tests, and this was selected as representative of conditions encountered in cruise and level flight. Angle of Oscillation:

(a) $\pm$ 1°
(b) $\pm$ 5°
(c) $\pm$ 9°

Lubrication

It was not the purpose of these tests to qualify acceptable grades of lubricant, but rather to compare results obtained with:

(a) Oil lubrication MIL-L-25336
(b) Grease lubrication MIL-G-25537

The wide field of ball and roller bearing technology has benefited by extensive experimental investigation and practical development in the normal areas of application. A reliable basis has evolved for the life-load rating of bearings as used in conventional design, and such bearings can be applied with reasonable assurance of expected life performance.

In other circumstances, where a bearing does not rotate under the applied load, it is recognized that the life performance must be predicted on a substantially different basis. The concept of "aircraft static capacity" has been used to advantage, but there is a conspicuous scarcity of factual information serving to relate design to standards of performance.

Assuming that a minimum of three different values are assigned to each of the principal variables (load, speed and amplitude) it is evident that a total of 27 test runs would be required with each lubricant, and for each bearing type, in order to completely explore their combined effect on bearing life. Further, if it is assumed that a minimum of 25 bearings must be processed in each test run, to assure reliability, it is at once apparent that the whole program will be of stupendous proportion.

Fortunately, the wide experience gained in comparable tests performed on bearings subjected to rotation, gives credence to the though that certain basic relations may be found to apply. A formal statistical study of the problem was made in which a factorial design was prepared for the purpose of minimizing the very large amount of testing that would otherwise be required[2]. Although competently prepared, this analysis could not be expected to correctly anticipate results that would be disclosed with progress of the test. Contrary to the assumption made in the referenced factorial plan, i.e., that there be no interaction between the principal variables, there was found evidence of some interaction affecting both load and speed.

Because of the overriding importance of the caged needle roller bearing type in actual application, the decision was made early in the program to concentrate the initial effort on this type. Later, further investigation of the potential use of tapered roller bearings and spherical roller bearings resulted in decision to eliminate those types from the program. Accordingly, all effort and work was directed to investigation of the life performance of the caged needle roller bearing. After a portion of the test program had been completed, it became apparent that tests at 10,000 lb load, even those at high speed, would take an extremely long time to complete. The condition of ± 1° amplitude also gave evidence of producing tests of extremely long duration. Accordingly, a new approach was taken to the test planning such as to make the remaining time and funds most productive. Only enough tests were made under conditions of ± 1° amplitude and grease lubricant to establish a trend and most of the tests at 10,000 lb load were omitted. This permitted greater emphasis to be placed on some of the other test conditions of the Life Test Series as well as the Limit Load Test Series and the Supplementary Test Series.

Limit Load Investigation

Load and speed conditions imposed upon the hinge bearings vary with the normal range of power and maneuver capability of the helicopter in flight. However, transient conditions arise in which extreme values of loading may occur and persist for relatively short periods of operation. There is interest and concern in evaluating the effect of such overload upon the subsequent life performance of the bearing.

For this purpose, part of the test program was devoted to investigation of the so called "limit load" performance of the bearing. An effort was made to ascertain what limiting value of overload could be applied, without appreciable impairment of the normal life in service.

The procedure adopted was to assemble a machine with four new bearings, load these to the predetermined level, and operate at a speed of 520 cycles per minute, for a given period of time. The test load was then reduced to the base value of 20,000 lb. and the bearings subjected to conventional life test. Under these circumstances a fixed index relationship between rollers and raceways was maintained under both the preliminary and life test operation. Areas intermediate between the actual contact surfaces traversed, remained unstressed.

Two values of overload were selected, consistent with the design capacity of the bearing and related to the base load of 20,000 lb. employed in previous life tests of the bearing. Four values of the number of cycles of prestress were tested, corresponding approximately to 1/2%, 2%, 4% and 8% of the B10 life as observed in testing at 20,000 lb. load and 520 cycles per minute.

TEST EQUIPMENT

Test Machine

In considering various features of an appropriate design it seemed opportune to utilize the principles of the hydrostatic bearing. Such bearings have ability to sustain heavy loads under essentially static conditions, and are characterized by very low values of frictional torque. These properties provide means for transmitting load to an oscillating shaft, and for supporting load reactions with negligible friction.

Ability to test concurrently more than one bearing in the same machine is critically dependent upon the means adopted for distribution and equalization of the applied load among the several bearings. Study of this problem resulted in the decision to employ mechanical leverage, in a symmetrical configuration, serving to equalize the load imposed on each of four test bearings (Figure 6). Reduced assembly drawings of the machine may be found in Appendix A.

Special care was exercised in the design and fabrication of the machine parts in which the test bearings were installed. Inner races were a shrink fit on bearing seats, which were inserted in the ends of the hollow shaft. The bearing seats were ground precisely concentric and parallel with the main journal surface of the shaft. These inserts were made replaceable to accommodate other size bearings.

Outer raceways of the test bearings are an expansion fit in their housings. The bore is located accurately with respect to the outside diameter of the housing, and all parts were manufactured to precisely interchangeable dimensions and tolerances. Any test cell housing can be applied to any machine, and the fit in the supporting hydrostatic shoe will be correct. The housings were stress relieved prior to finish grinding; the outside diameter was case hardened and the surface treated to create a black phosphate coating.

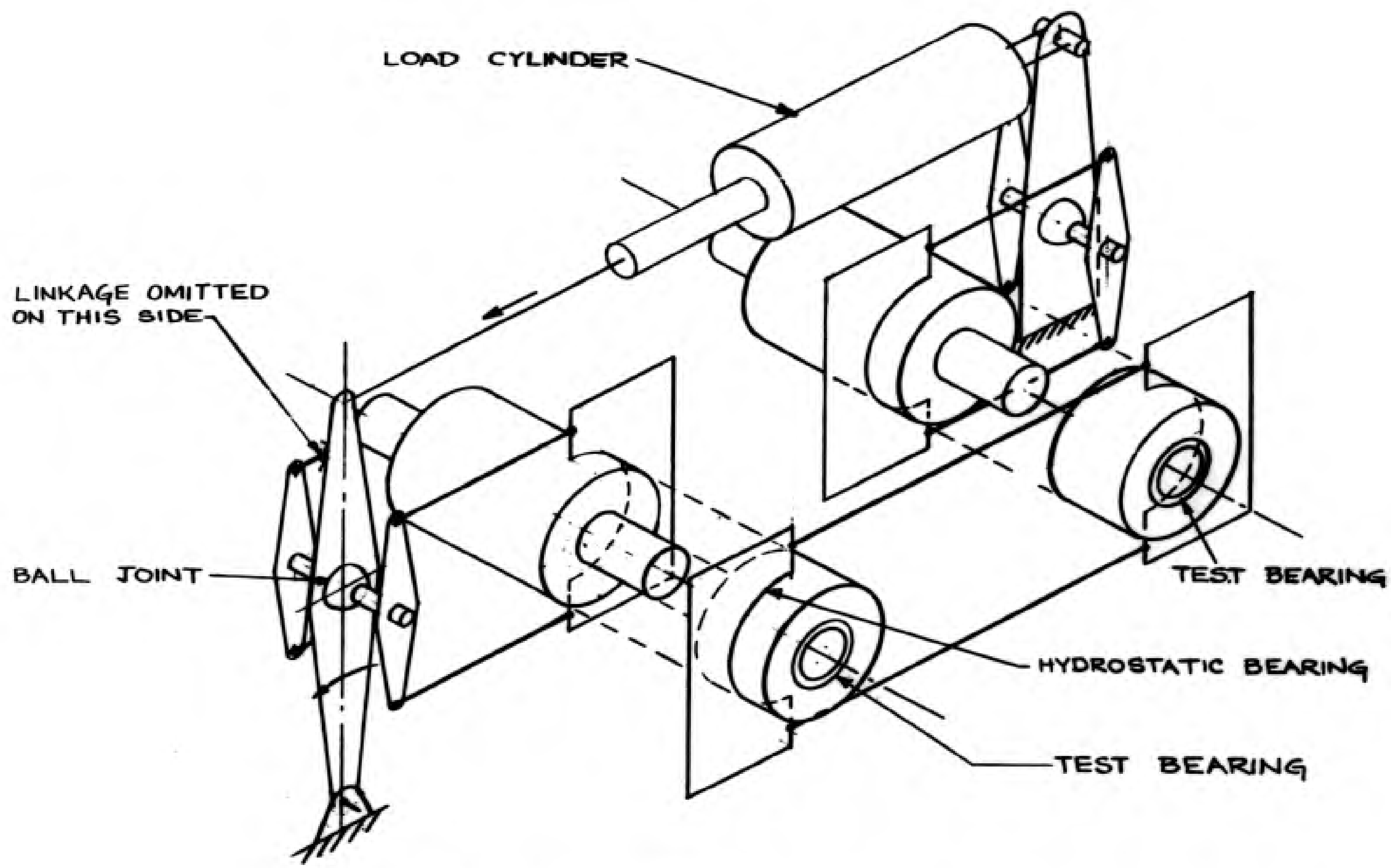


Fig. 6 - SCHEMATIC OF BEARING TEST MACHINE

Similarly, the welded main journal assemblies were stress relieved, and the journal surface ground to interchangeable dimensions and tolerances. This surface is also treated to creat a black phosphate coating.

The hydraulic oil necessary for operation of the hydrostatic bearings is supplied by a central system to the control panel of each machine. At this point a pressure-reducing valve is provided to establish the supply pressure to the machine, and a second reducing valve to establish the pressure in the loading cylinder and the applied test load.

As designed and constructed, the machine provides for a range of operating conditions intended to duplicate the spectrum of loading encountered in service on a hinge bearing of comparable size.

Loading System

The bearings are tested under radial load and are free from imposed axial constraint. An hydraulic cylinder mounted above, and in the longitudinal center plane of the machine, transmits the test load through an equalizer linkage to each of the test bearing locations.

Inner races of the test bearings are mounted with a shrink fit on the outboard ends of the two main journal shafts of the machine. Each of these journals is supported by two adjacent hydrostatic bearings, located inboard and serving to transmit the applied load to the shafts. The housings which carry the test bearings mounted on the stub ends of the shafts, are likewise supported in hydrostatic bearings, that are joined by parallel tension links, serving to transmit the applied load from the one main shaft to the other and equalizing the load reactions.

The configuration of the load linkage with duly-apportioned equalizer leverage and the provision of spherical aligning pivot reactions assure an effective equalization of the load among the four test bearing locations.

Oscillating Drive

The test bearings are driven in cyclic oscillation by a reciprocating wrist-pin motion imparted to each of the main journal shafts of the machine. This motion is derived from a rotating crankshaft, located midway between the main shafts, and driving through conventional crank-and-connecting-rod linkage. The two crank throws, each driving one of the two main journals, are phased 90° apart and serve to balance the torque reactions of the inertial forces of the oscillating masses.

The driving motor is floor mounted beneath the base plate of the machine, and is connected by a timing belt reduction to a jack shaft, and through a second timing belt to the crankshaft of the machine.

Applied load is regulated by a pressure-reducing valve in circuit with the main hydraulic loading cylinder and is indicated by a precision-type pressure gauge. The load can be accurately controlled in the range 2,000 pounds to 50,000 pounds on the test bearings. The machine components are designed for maximum loading with acceptable values of stress and deflection.

Amplitude or angle of oscillation is provided for by design of the driving crankshaft (Figure 7). Three alternate shafts are furnished for each machine having respectively throws designed for ± 1°, ± 5°, and ± 9° amplitude of oscillation. The crankshaft is supported on two ball bearings, and the housing is designed to permit easy interchange of shafts as required by the test program. The two quartered crank throws of each shaft are suitably counterbalanced to allow smooth operation at the designed speeds.

Speed variation for the different test conditions is varied in three steps of 2:1 ratio: High speed is 520 cycles per minute, intermediate speed 260 cycles per minute, and low speed 130 cycles per minute. Speed changes from 520 to 260 cycles is effected by shifting the timing belt; low speed of 130 cycles requires changing of the motor and jack shaft pulleys.

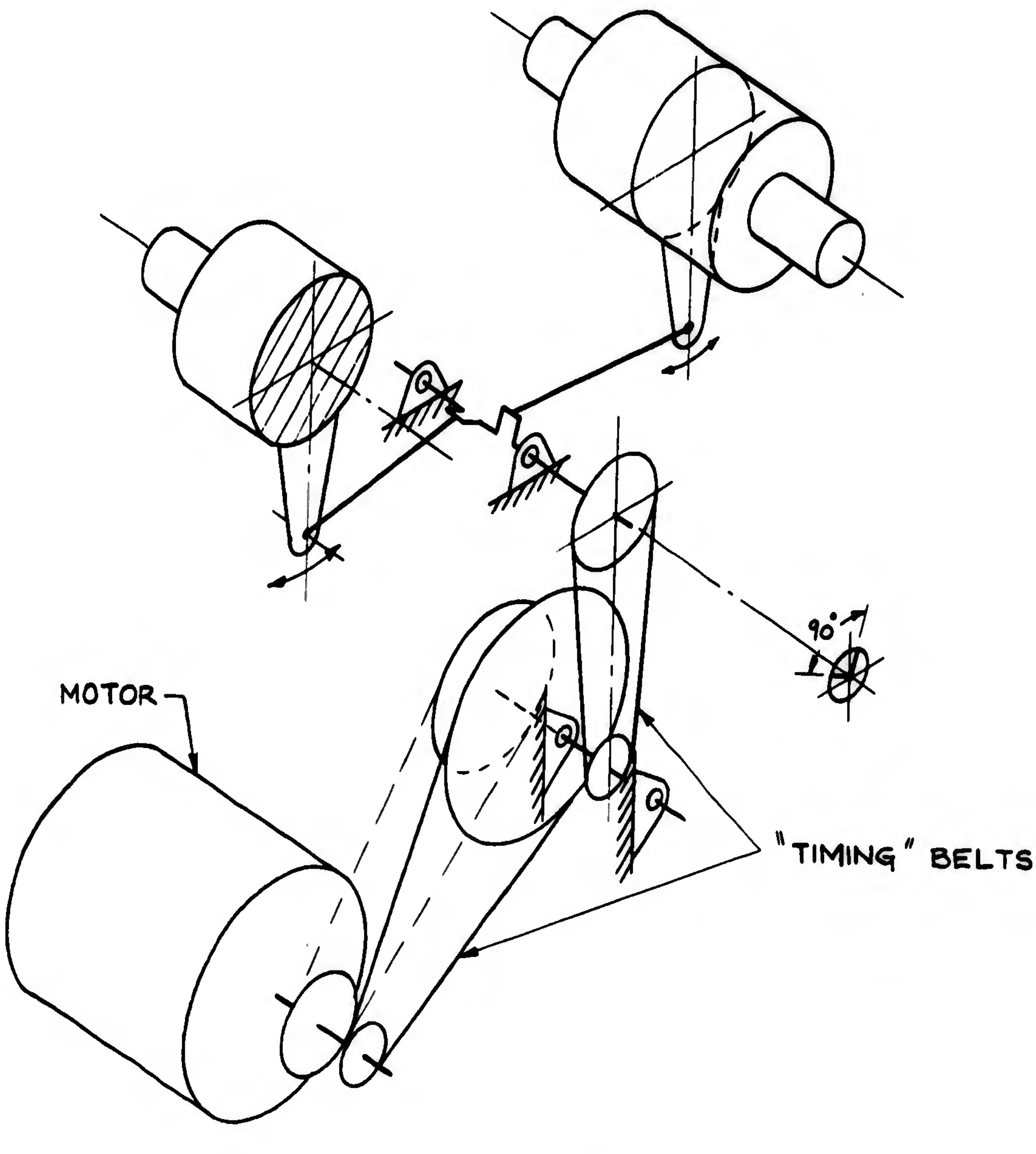


Fig. 7 - OSCILLATOR MECHANISM DRIVE

Lubrication is applied to the individual test bearings, and normally there is no occasion for its replenishment throughout duration of the test. Oil is provided to a level that essentially submerges the bearing, and with grease lubrication the bearing is completely packed with the lubricant. A spring-loaded leather seal engages the shaft to prevent leakage of the lubricant and to prevent influx of the hydraulic fluid from the adjacent hydrostatic bearings. The outboard face of the test bearing housing is sealed by a Plexiglass cover plate, and this allows visual inspection during progress of the test.

Continued performance of the machine was conspicuously satisfactory, and there was no untoward interruption of the test due to malfunction of any of the machines. It is of interest to note that the eight hydrostatic bearings, interposed in the loading system of the machine, gave uniform and manifestly-satisfactory service.

Although an appropriate allocation of spare parts had been made at the time of building the machines, there has as yet been no occasion for their use.

Instrumentation

For the purpose of definition, the life of the bearing is said to be terminated by the first evidence of failure in the form of fatigue damage affecting the load-carrying surfaces of the bearing, upon inspection with the unaided eye. This normally occurs as spalling or flaking of the surface in highly-localized areas from which small metallic spangles are sloughed, leaving sharp-edged pitting of the surface. Such damage is known to be early evidence in the sequence of progressive failure occurring under repeated heavy contact stress. It is also known that, prior to flaking of the surface, considerable fatigue action has taken place below the surface.

Some technicalities are concerned in implementing the definition of "first failure" (really, first visible evidence of damage) and not the least of these relates to the means available for its detection. Normally the functional performance of the bearing remains unimpaired for an extended period subsequent to inception of the failure. Invariably the damage progresses as a result of repeated stress and impact, and eventually large areas of the surface will be affected, and the condition will be reflected in rough running of the bearing. Because of the gradually progressive type of failure some difficulty is encountered in its early identification, and this condition applies both in service operation and in laboratory test. It is important to note that the degree of failure serving to terminate the test life of the bearing is not an absolute criterion, but is subject to chance variation in its discovery.

The plan for the conduct of these tests envisioned a means for monitoring the frictional torque of the test bearings. It was believed that this would promptly reflect the occurrence of first fatigue damage in the bearing, and serve to signal failure on a reliably uniform basis. An elaborate system of instrumentation, employing high gain bridge circuits and strain gauges, was developed for this purpose (See Appendix B). However, extensive experience with its use applied to the test installation proved it to be unreliable as a means of signaling first failure. The basic reasons for this appeared to be:

(a) An insensitivity of the bearing frictional torque to the occurrence and progress of fatigue damage.

(b) The relatively small torque signal compared with the extraneous signals resulting from the complex dynamics of the machine.

(c) The spurious displacements of the test bearing centers where the load was removed and reapplied upon replacement of test bearings.

Because of these difficulties and the inordinate amount of time and effort that would be required for their solution, the instrumentation system was operated as ancillary in the conduct of the test.

Persistent effort was made to correct the failure of the instrumentation system to yield significant results in monitoring frictional torque of the test bearing and signaling first failure. Although a continued record was maintained throughout duration of individual tests, there was conspicuous lack of correlation between meter indication and the inception and progress of fatigue failure. Reflection on this situation suggests that the minuscule proportion of flaking breakout from the surface, is imperceptible in its effect on the integrated value of bearing friction as a whole. Aside from the variable performance of the system in terms of its dynamic response, it became abundantly evident that frictional torque of the bearing is, at best, an insensitive indicator of the type of bearing failure with which we are concerned.

In lieu of the instrumental monitoring of the progress of the test it was necessary to devise alternate means for timely detection of failure when it occurred. For this purpose the test operation of the eight machines (32 test bearings) was pursued under the systematic supervision of a qualified technician. The use of a mechanic's stethescope, applied to individual test housings, permitted an objective estimate to be made of changes in operating level of smoothness. Appearance of fine metallic chips or spangles, visible through the housing cover plate, and progressive darkening of the oil, gave conclusive evidence of failure.

Reliability of the method employed in discovery of failure and termination of the test run, can be inferred from recorded data: first the numerical classification of the extent to which failure had advanced;

and second, the relative proportion of bearings that were removed and found to show no evidence of failure. In the latter case it should be noted that a substantial number of such removals was incident to a change in test phase of the machine or to the arbitrary termination of a test phase at approximately 1,000 or 2,000 hours. It should also be noted that the time error in estimating first failure is not random in a statistical sense and cannot be negative in anticipation of failure before it occurs. This would influence the values of life, corresponding to no failure, in the linear distribution of life versus percentage failure.

Although a high degree of precision is lacking in the definition of "first failure", it is quite apparent that the means employed for detecting failure in the test machine, were far more sensitive than those generally available in service operation. The latter quite often are limited to the observation of excessive vibration in service operation and periodically scheduled visual inspection. In this respect the results obtained in the test program may be considered conservative in terms of operating field experience.

Test Installation

The hydraulic system providing oil flow to the hydrostatic bearings of the machines, was designed as a central system suitable for operation of all machines at their designed capacity (Figure 8). This required a total of about 40 gallons per minute at a pressure of about 3,500 psi. Two pumping units were installed, each with a 20 gpm capacity and with their 100 gallon reservoirs interconnected in the system.

One pump was a constant volume type driven by a 30 hp motor, the other was a pressure-compensated variable-volume type driven by a 20 hp motor. Oil was delivered to two main headers, and the individual machines could alternatively be connected to either one. Pressure reducing valves installed at the control panel of each machine regulated the pressure supplied to the hydrostatic bearings and in turn to the loading cylinder of the machine.

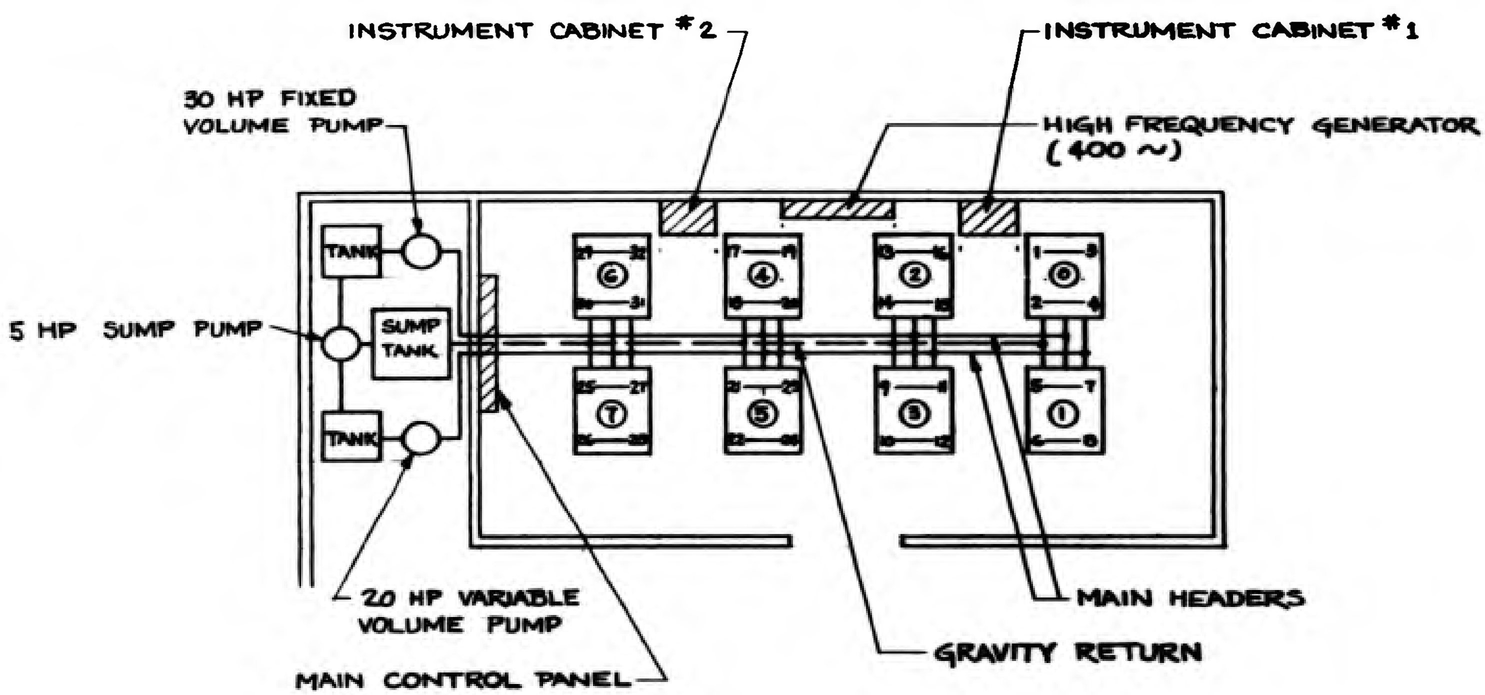


Fig. 8 - SCHEMATIC OF INSTALLATION

Oil flow from the hydrostatic bearings was returned by a gravity drain line, to an underground sump tank located in the pump room. The tank was fitted with a float-controlled magnetrol switch, serving to operate the start and stop control of a 5 hp electric motor, driving a 15 gpm fixed volume pump. This pump took its suction from the sump tank and delivered its output to the filter and heat exchanger of each of the two main pumping units, connected in parallel. The sump pump operated on a cycle, determined by adjustment of the floats in the tank, and provided a flexible and satisfactory control.

A pressure-sensitive control was connected to the terminal end of each main pressure header, and this was set to operate a complete electrical disconnect in the event of a preassigned loss of pressure.

The system was designed to provide complete flexibility and economy in operation, for the entire range of loading contemplated in the test program. One or more machines could be assigned to any phase of test, irrespective of assignment of the others. Operation of the system proved to be entirely satisfactory, and only minor difficulties were encountered.

Photographs of the test room, the instrument panel, and the hydraulic pumps may be found in Appendix C.

TEST PROCEDURE

The general arrangement and capabilities of the eight test machines and the methods of detecting bearing failures were discussed in the previous sections Test Installation and Instrumentation respectively. This section will include a number of details of the procedures associated with the complete cycle of test operations.

Prior to testing, the bearings were stored in their original packages in the air-conditioned room housing the test machines. Fitting of the bearings in the test machine conformed to the practice in actual aircraft usage. An interference fit was employed for both the inner and outer recess as follows:

Shaft	2.7602 2.7597	Bearing	2.7595 2.7588
Housing	3.6240 3.6232	Bearing	3.6250 3.6245

The slushing compound applied to the bearing for preservation in packaging was removed by washing the bearing in Varsol. Outer raceways and roller assemblies were chilled in dry ice and mounted in the test bearing housings in a bench assembly operation. Inner raceways were heated in oil on a bench adjacent to the test machines then mounted on the shaft in the test machine. The assembly of the side cover to the bearing housing permitted the addition of a new supply of oil or grease and the bearing was ready for test.

During some tests leakage of oil from the bearing housings required the addition of make-up oil. With few exceptions, the oil level in the bearing housing never went below the point of one-third of the bearing outside diameter down from the top.

Another minor difficulty was caused by the leakage of oil from the hydrostatic bearings into the test bearing housing. For the most part this in-leakage did not significantly dilute the test oil. When the in-leakage occurred at a time when the test lubricant was grease it probably had no effect on the test since the contaminant oil fell to the bottom of the test housing, essentially entirely out of the zone of loaded rollers.

The temperature of the bearing and lubricant during test was nearly constant. The temperature of the oil exiting from the hydrostatic bearing which surrounds the loaded half of the test bearing was controlled by regulating the pressure and flow to the hydrostatic bearing. This temperature ranged from 108° to 118°F and the oil in the test bearing housing is presumed to have been at essentially this same temperature.

During the progress of each test run, when failure was detected or presumed, the machine was stopped and that individual bearing was removed. The removal of the outer raceway and roller assembly was accomplished by pressing out of the bearing housing. The inner raceway was removed by a puller. Replacement was made with a preassembled housing unit and the test resumed with a delay normally of less than one hour. A careful record was maintained of the hours accumulated on each bearing. Thanks to adequate monitoring devices, and the provision of individual timing counters, it was found feasible to operate the test on a full-time basis and thus substantially conserve time in its accomplishment.

Conventionally accepted in the technology of ball and roller bearings, the concept of "first failure" is a matter of somewhat arbitrary degree. Contact fatigue is progressive in character. At inception it is essentially microscopic in extent, lies beneath the surface and precludes the possibility of visual detection. In this program, visual examination of bearings removed from test, and disassembled, revealed consistent and characteristic patterns of appearance. For the purpose of defining life performance, the single criterion of fatigue failure made evident by surface flaking breakout was employed. In the absence of this type of damage the bearing was accredited as a nonfailure, although other evidence of non-critical deterioration may be apparent.

The bearings removed following test were inspected then tagged and stored for future reference.

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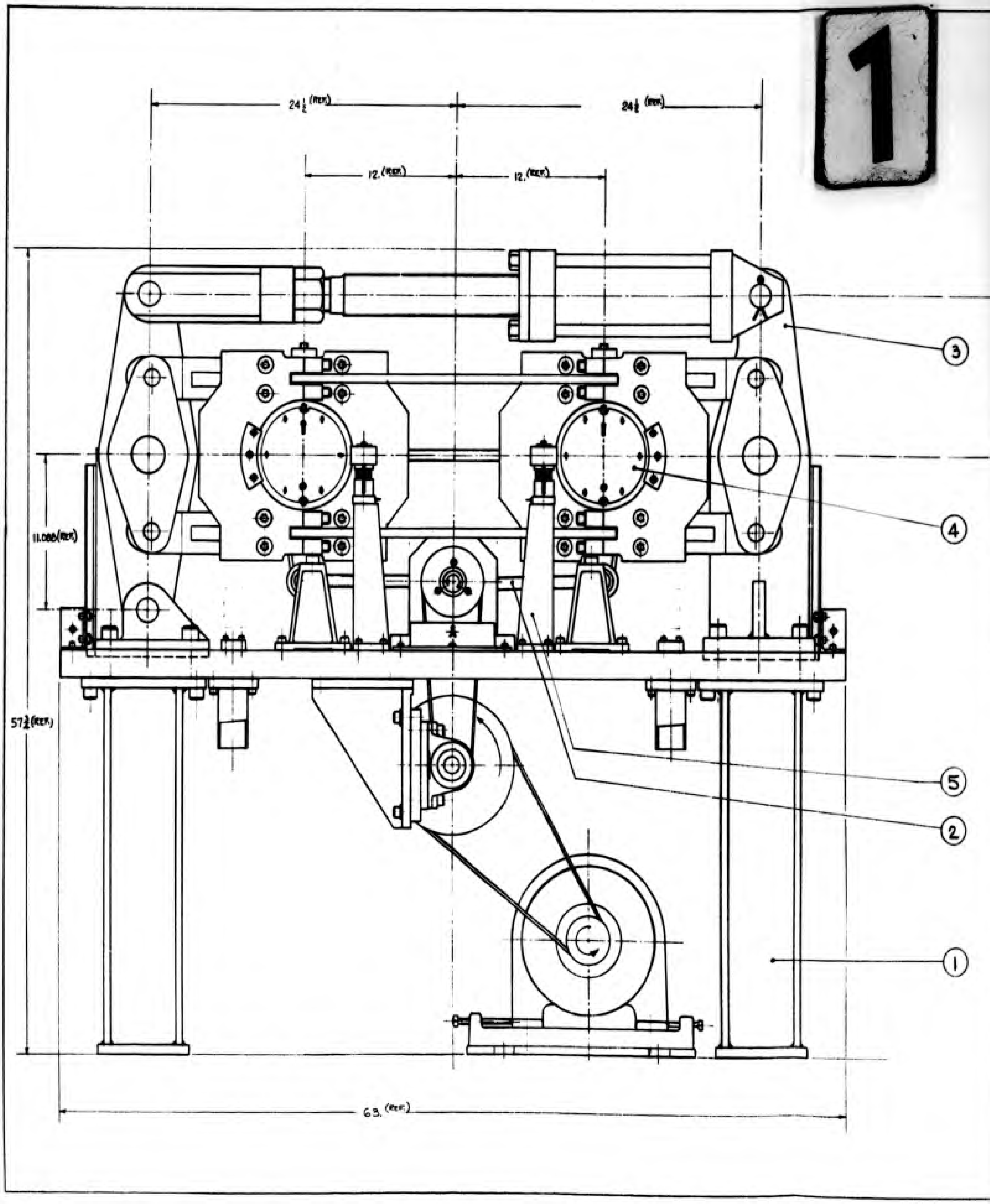
APPENDIX A

Test Machine Assembly Drawings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63. (REF.)
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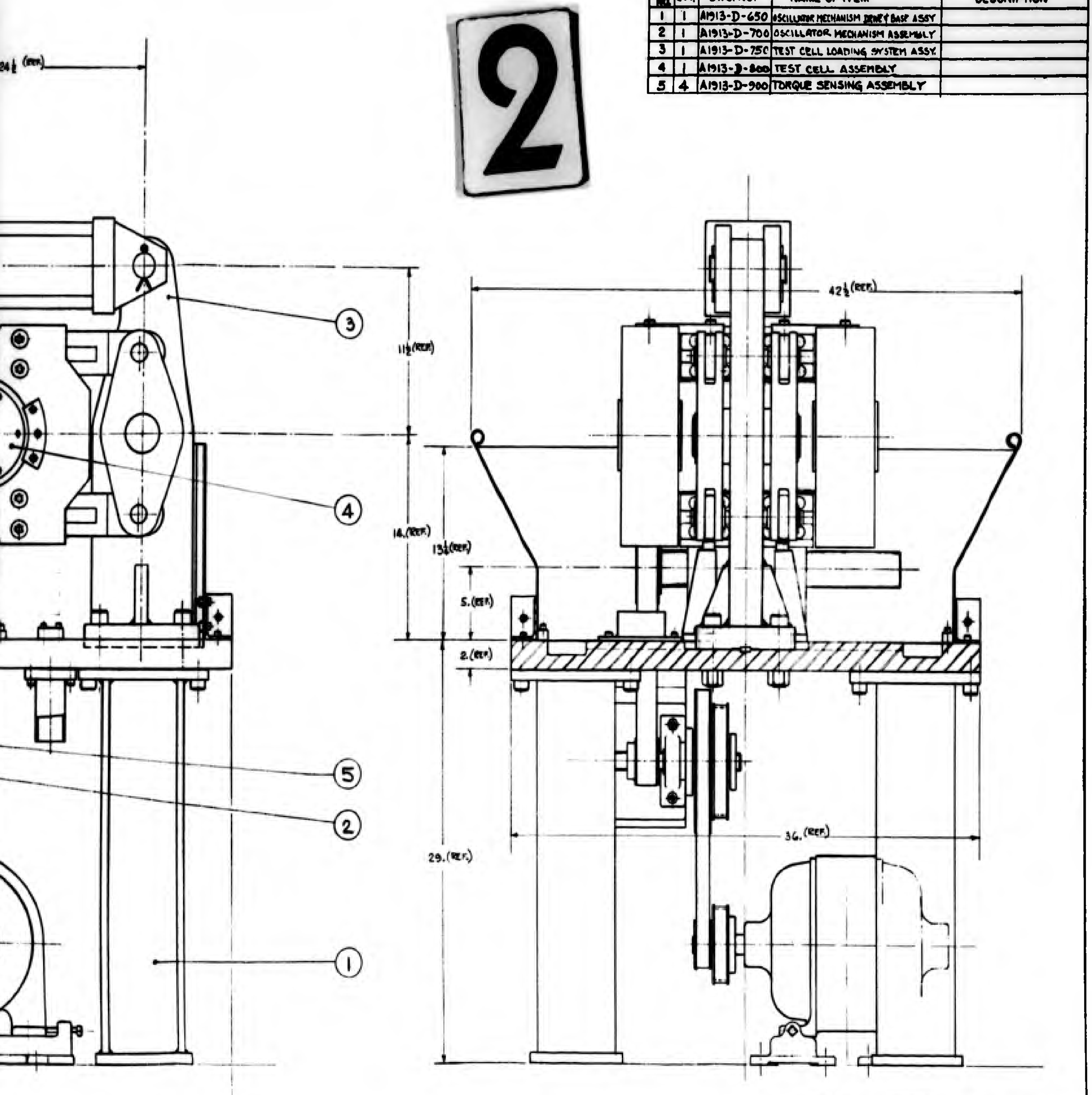
LIST OF PARTS				
ITEM NO.	QTY.	DWG. NO.	NAME OF ITEM	DESCRIPTION
1	1	A1913-D-650	OSCILLATOR MECHANISM DRIVE & BASE ASSY	
2	1	A1913-D-700	OSCILLATOR MECHANISM ASSEMBLY	
3	1	A1913-D-750	TEST CELL LOADING SYSTEM ASSY.	
4	1	A1913-D-800	TEST CELL ASSEMBLY	
5	4	A1913-D-900	TORQUE SENSING ASSEMBLY	



SCALE: 1/4
UNLESS OTHERWISE NOTED REMOVE BURRS & SHARP EDGES
NEXT ASS'Y.
LIST OF DWG'S.

BEARING TEST MACHINE GENERAL ASSEMBLY

DR. R.T. Smith | CH. | APPROVED: | DATE 6/28/61
DES. ENG. R.T. Smith | PROJ. ENG.

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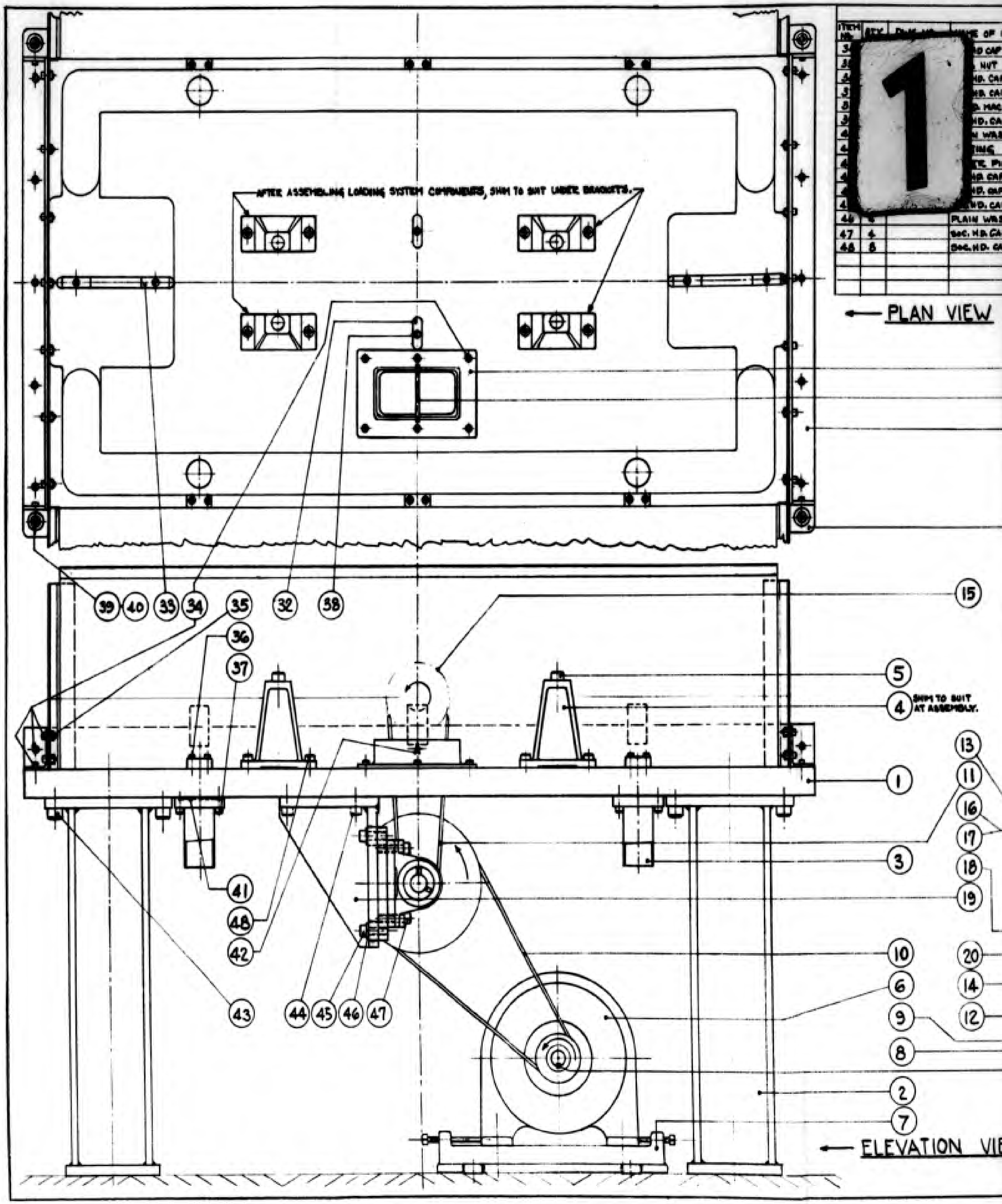
DWG. NO. A1913-D-600



AFTER ASSEMBLING LOADING SYSTEM COMPONENTS, SHIM TO SUIT UNDER BRACKETS.
PLAN VIEW
SHIM TO SUIT AT ASSEMBLY.
ELEVATION VI
39 40 33 34 35 32 38
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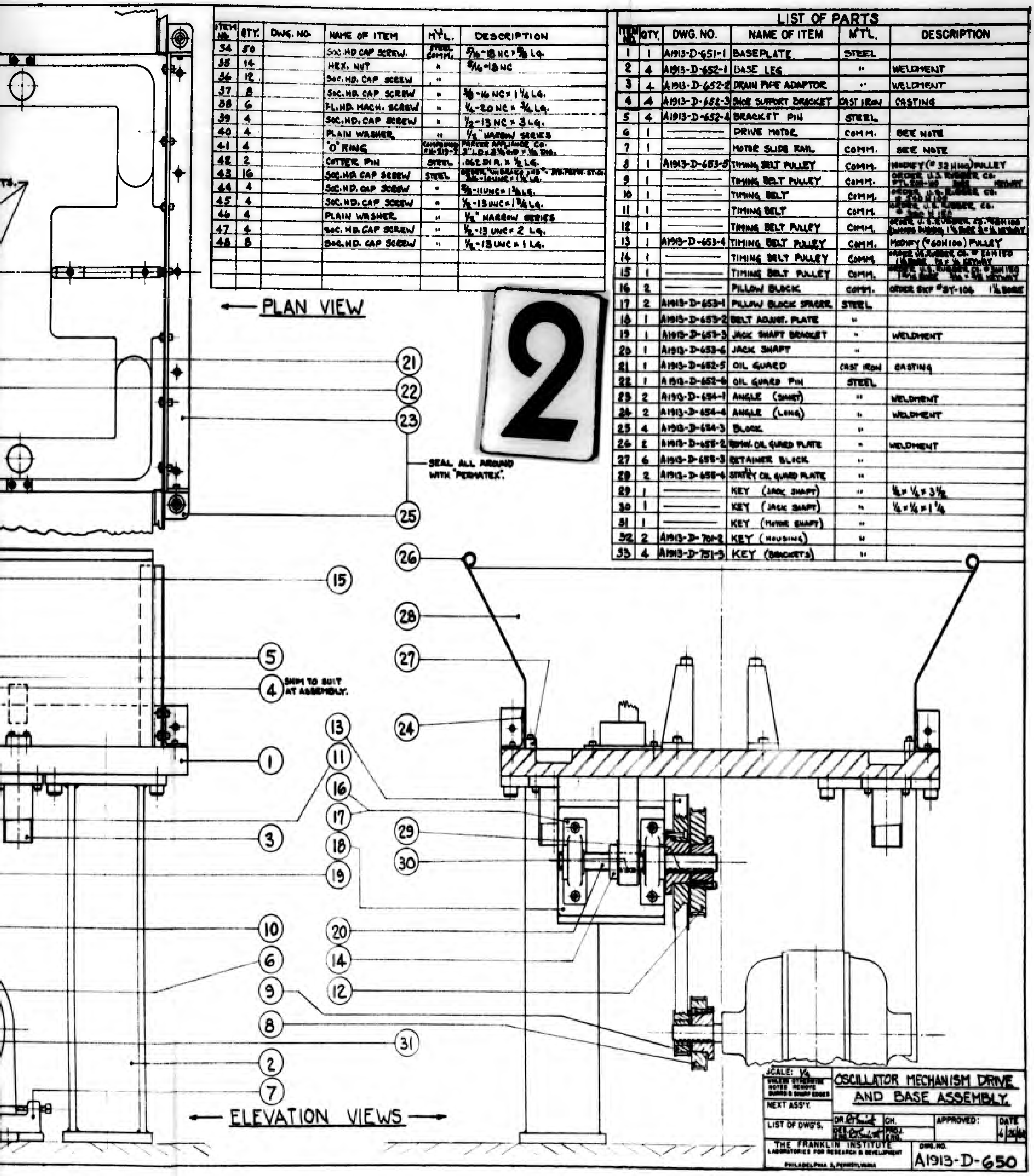
ITEM No.	QTY.	DWG. NO.	NAME OF ITEM	M'TL.	DESCRIPTION
34	50		SOC. HD. CAP SCREW.	STEEL COMM.	5/16-18 NC × 5/8 LG.
35	14		HEX. NUT	"	5/16-18 NC
36	12		SOC. HD. CAP SCREW	"	
37	8		SOC. HD. CAP SCREW	"	3/8-16 NC × 1 1/4 LG.
38	6		FL. HD. MACH. SCREW	"	1/4-20 NC × 3/4 LG.
39	4		SOC. HD. CAP SCREW	"	1/2-13 NC × 3 LG.
40	4		PLAIN WASHER	"	1/2" NARROW SERIES
41	4		"O" RING	COMPOUND #N-219-7	PARKER APPLIANCE CO. 2" I.D. × 2 1/8 O.D. × 1/16 DIA.
42	2		COTTER PIN	STEEL	.062 DIA. × 1/2 LG.
43	16		SOC. HD. CAP SCREW	STEEL	[illegible]
44	4		SOC. HD. CAP SCREW	"	5/8-11 UNC × 1 3/4 LG.
45	4		SOC. HD. CAP SCREW	"	1/2-13 UNC × 1 3/4 LG.
46	4		PLAIN WASHER	"	1/2" NARROW SERIES
47	4		SOC. HD. CAP SCREW	"	1/2-13 UNC × 2 LG.
48	8		SOC. HD. CAP SCREW	"	1/2-13 UNC × 1 LG.

LIST OF PARTS

ITEM NO.	QTY.	DWG. NO.	NAME OF ITEM	M'TL.	DESCRIPTION
1	1	A1913-D-651-1	BASEPLATE	STEEL	
2	4	A1913-D-652-1	BASE LEG	"	WELDMENT
3	4	A1913-D-652-2	DRAIN PIPE ADAPTOR	"	WELDMENT
4	4	A1913-D-652-3	SHOE SUPPORT BRACKET	CAST IRON	CASTING
5	4	A1913-D-652-4	BRACKET PIN	STEEL	
6	1	———	DRIVE MOTOR	COMM.	SEE NOTE
7	1	———	MOTOR SLIDE RAIL	COMM.	SEE NOTE
8	1	A1913-D-653-5	TIMING BELT PULLEY	COMM.	MODIFY (# 32 H100) PULLEY
9	1	———	TIMING BELT PULLEY	COMM.	ORDER U.S. RUBBER CO. [illegible]
10	1	———	TIMING BELT	COMM.	ORDER U.S. RUBBER CO. [illegible]
11	1	———	TIMING BELT	COMM.	ORDER U.S. RUBBER CO. [illegible]
12	1	———	TIMING BELT PULLEY	COMM.	ORDER U.S. RUBBER CO. [illegible]
13	1	A1913-D-653-4	TIMING BELT PULLEY	COMM.	MODIFY (#60H100) PULLEY
14	1	———	TIMING BELT PULLEY	COMM.	ORDER U.S. RUBBER CO. # 60H100 [illegible]
15	1	———	TIMING BELT PULLEY	COMM.	[illegible]
16	2	———	PILLOW BLOCK	COMM.	ORDER SKF #SY-104 1 1/4 BORE
17	2	A1913-D-653-1	PILLOW BLOCK SPACER	STEEL	
18	1	A1913-D-653-2	BELT ADJUST. PLATE	"	
19	1	A1913-D-653-3	JACK SHAFT BRACKET	"	WELDMENT
20	1	A1913-D-653-6	JACK SHAFT	"	
21	1	A1913-D-652-5	OIL GUARD	CAST IRON	CASTING
22	1	A1913-D-652-6	OIL GUARD PIN	STEEL	
23	2	A1913-D-654-1	ANGLE (SHORT)	"	WELDMENT
24	2	A1913-D-654-4	ANGLE (LONG)	"	WELDMENT
25	4	A1913-D-654-3	BLOCK	"	
26	2	A1913-D-655-2	[illegible] OIL GUARD PLATE	"	WELDMENT
27	6	A1913-D-655-3	RETAINER BLOCK	"	
28	2	A1913-D-655-4	STATIONARY OIL GUARD PLATE	"	
29	1	———	KEY (JACK SHAFT)	"	1/4 × 1/4 × 3 1/2
30	1	———	KEY (JACK SHAFT)	"	1/4 × 1/4 × 1 1/4
31	1	———	KEY (MOTOR SHAFT)	"	
32	2	A1913-D-701-2	KEY (HOUSING)	"	
33	4	A1913-D-751-3	KEY (BRACKETS)	"	



SCALE: 1/4

NEXT ASS'Y.

LIST OF DWG'S.

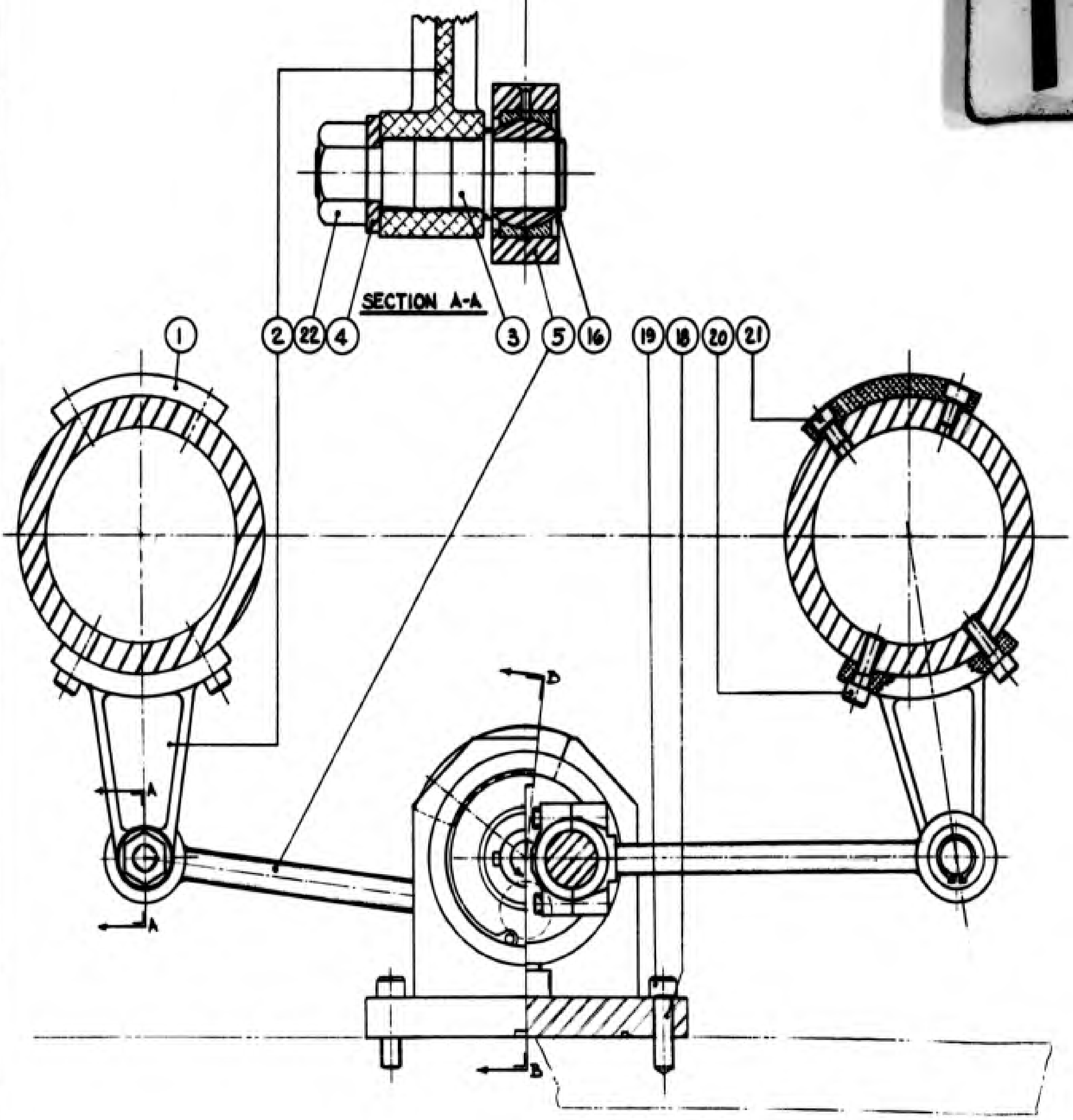
APPROVED:

DATE

OSCILLATOR MECHANISM DRIVE AND BASE ASSEMBLY.

THE FRANKLIN INSTITUTE
LABORATORIES FOR RESEARCH & DEVELOPMENT
PHILADELPHIA 3, PENNSYLVANIA

DWG. NO. A1913-D-650



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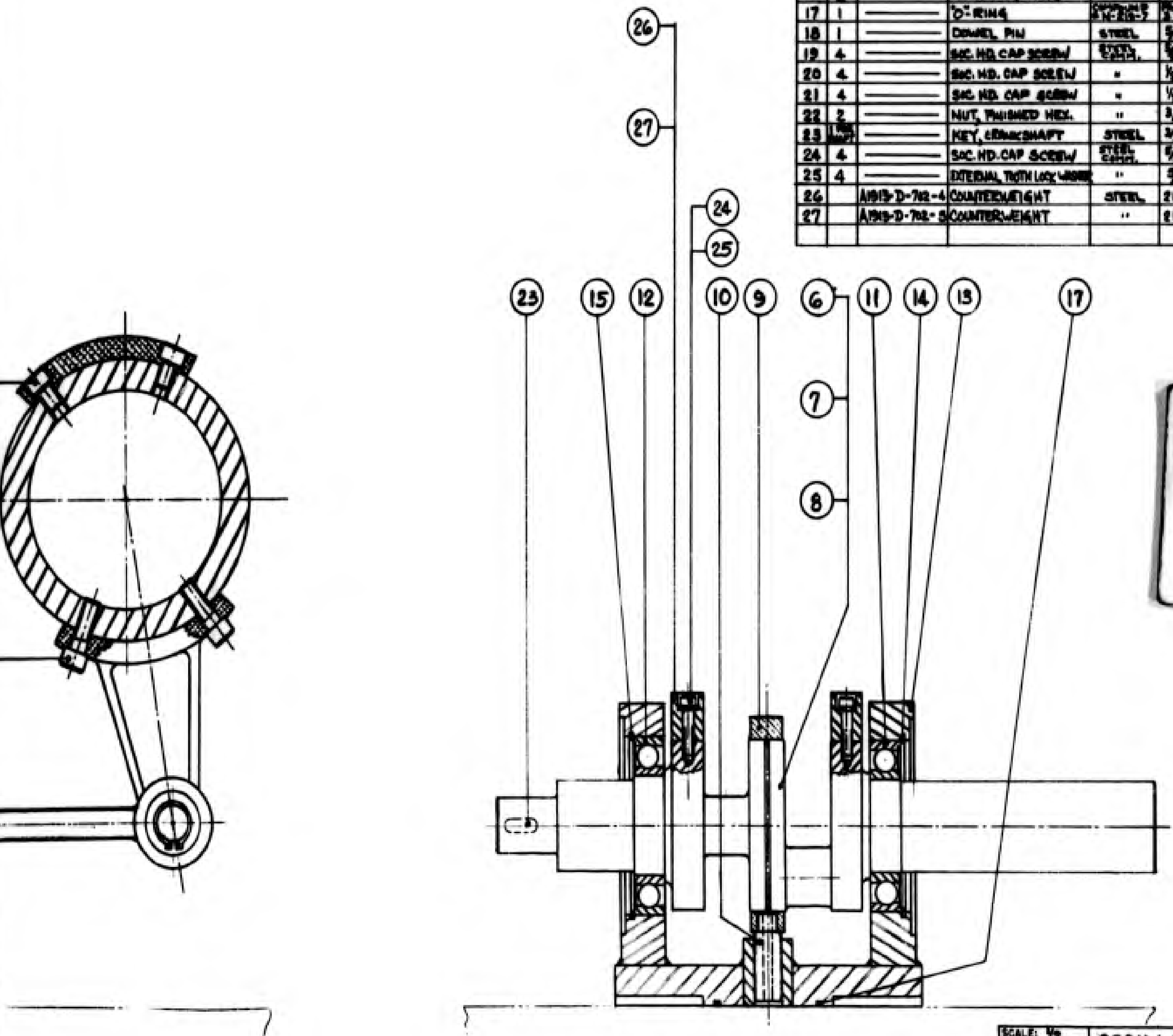
LIST OF PARTS					
ITEM NO.	QTY.	DWG. NO.	NAME OF ITEM	M'TL.	DESCRIPTION
1	2	A1913-D-705-3	SPACER	ALUMINUM	
2	2	A1913-D-705-1	OSCILLATING LEVER BRKT	"	CASTING
3	2	A1913-D-705-2	CRANK LEVER PIVOT PIN	STEEL	
4	2	A1913-D-705-4	PIVOT PIN WASHER	"	
5	2	A1913-D-704-1	CONNECTING ROD	"	WELDMENT & ASSEMBLY
6		A1913-D-703-1	CRANKSHAFT	"	.812 ECCENTRIC
7		A1913-D-703-2	CRANKSHAFT	"	.187 ECCENTRIC
8		A1913-D-702-1	CRANKSHAFT	"	1.437 ECCENTRIC
9	1	A1913-D-702-2	OIL SUPPLY RING	[illegible]	
10	1	A1913-D-702-3	RING STEM	STEEL	
11	1	————	BALL BEARING	COMM.	ORDER SKF #6213Z OR EQUIV.
12	1	————	BALL BEARING	"	ORDER SKF #6214 Z OR EQUIV.
13	1	A1913-D-701-1	CRANKSHAFT HOUSING	STEEL	WELDMENT
14	1	————	RETAINING RING	COMM.	ORDER TRUARC #5100-475
15	1	————	RETAINING RING	"	ORDER TRUARC #5100-500
16	2	————	RETAINING RING	"	ORDER TRUARC #5100-100
17	1	————	"O" RING	[illegible]	[illegible]
18	1	————	DOWEL PIN	STEEL	5/8 D. × 2 LG.
19	4	————	SOC. HD. CAP SCREW	STEEL COMM.	5/8-11NC × 2 LG.
20	4	————	SOC. HD. CAP SCREW	"	1/2-13NC × 1 1/4 LG.
21	4	————	SOC. HD. CAP SCREW	"	1/2-13NC × 3/4 LG.
22	2	————	NUT, FINISHED HEX.	"	3/4-10 NC AMERICAN STD.
23	[illegible]	————	KEY, CRANKSHAFT	STEEL	3/8 × 3/8 × 1 LG.
24	4	————	SOC. HD. CAP SCREW	STEEL COMM.	5/16-18NC × 1 1/4 LG.
25	4	————	EXTERNAL TOOTH LOCK WASHER	"	5/16
26		A1913-D-702-4	COUNTERWEIGHT	STEEL	2 REQ. FOR A1913-D-703-1
27		A1913-D-702-5	COUNTERWEIGHT	"	2 REQ. FOR A1913-D-702-1



SECTION B-B

SCALE: 1/2

UNLESS OTHERWISE NOTED REMOVE BURRS & SHARP EDGES

NEXT ASS'Y

LIST OF DWG'S

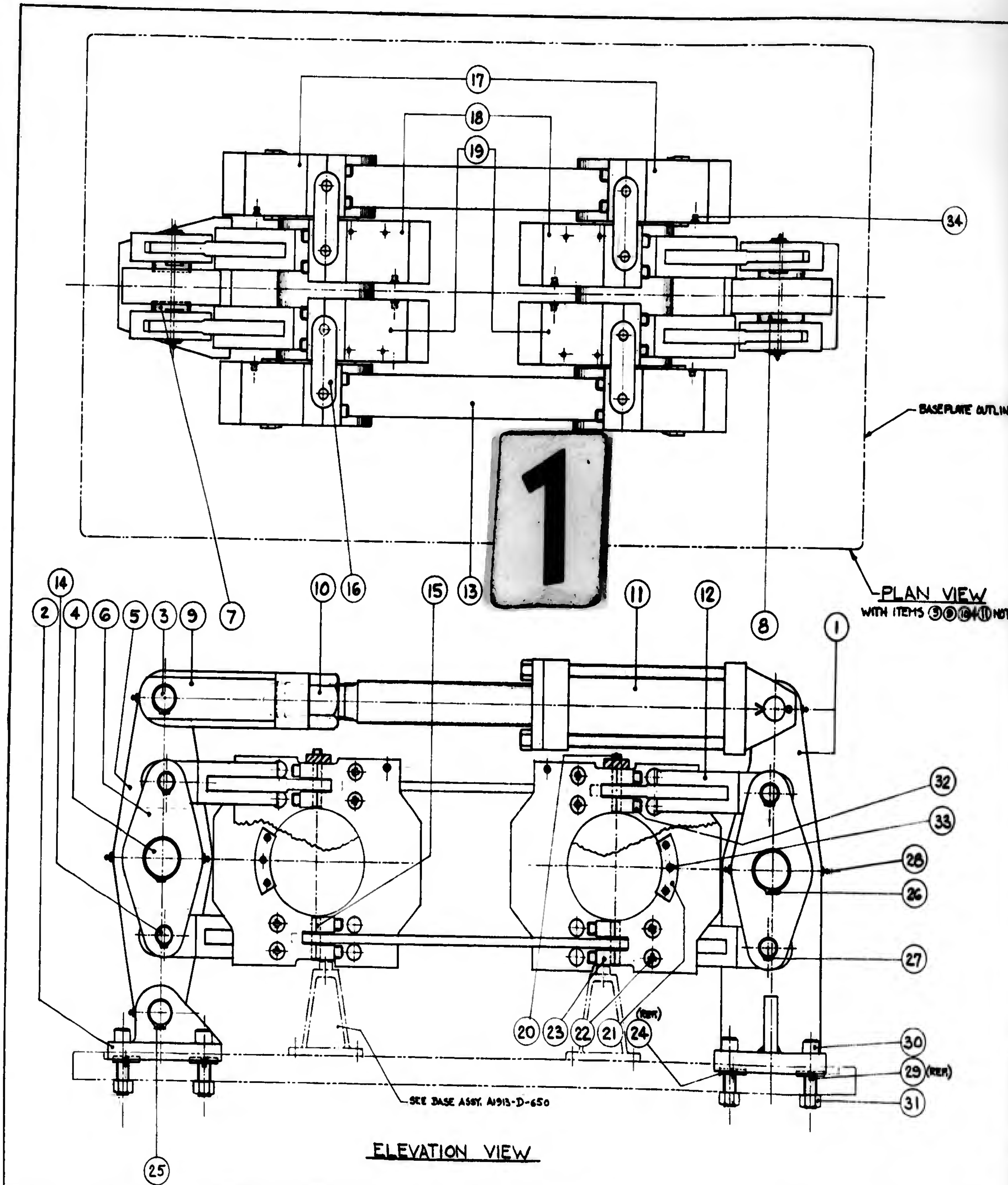
OSCILLATOR MECHANISM ASSEMBLY

DR. | CH. | APPROVED: | DATE

THE FRANKLIN INSTITUTE LABORATORIES FOR RESEARCH & DEVELOPMENT

PHILADELPHIA 3, PENNSYLVANIA

DWG. NO. A1913-D-700



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BASEPLATE OUTLIN
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SEE BASE ASSY. A1913-D-650
ELEVATION VIEW
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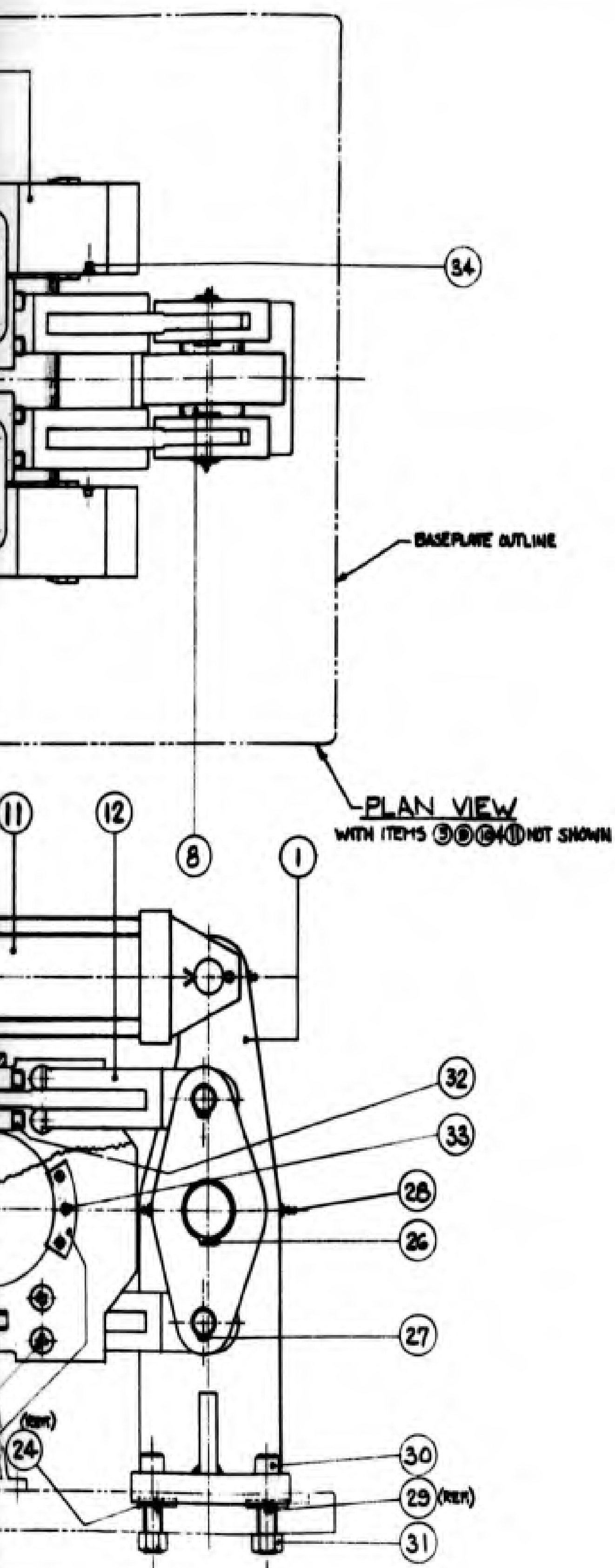
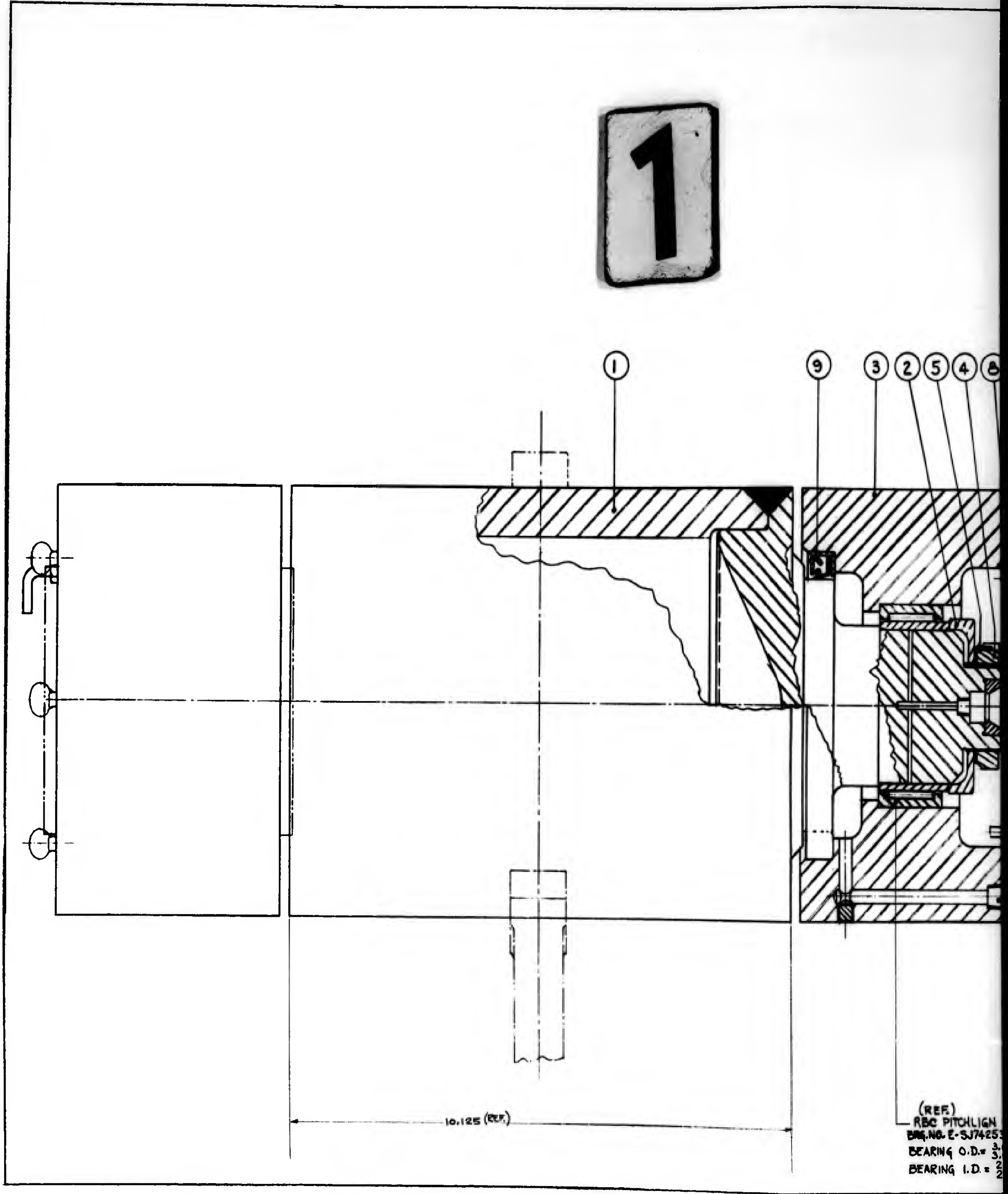


ITEM NO.	QTY.	DWG. NO.	NAME OF ITEM	M'T'L.	DESCRIPTION
1	1	A1913-D-751-1	SUPPORT	STEEL	WELDMENT & ASSY.
2	1	A1913-D-751-2	BEARING BLOCK	"	ASSY.
3	2	A1913-D-752-3	LEVER PIN (SHORT)	"	
4	2	A1913-D-753-4	LEVER PIN (LONG)	"	
5	1	A1913-D-752-2	LEVER (LONG)	"	ASSY.
6	4	A1913-D-754-1	LEVER (SHORT)	"	ASSY.
7	2	A1913-D-754-3	SPACER	"	
8	2	A1913-D-752-5	SPACER	"	
9	1	A1913-D-752-1	CYLINDER ROD CLEVIS	"	ASSY.
10	1	A1913-D-754-2	CYLINDER ROD NUT	"	
11	1	A1913-C-201	HYDRAULIC CYLINDER	"	ASSY.
12	8	A1913-D-753-1	CONNECTING LINK (SHORT)	"	WELDMENT & ASSY.
13	4	A1913-D-753-3	CONNECTING LINK (LONG)	"	ASSY.
14	8	A1913-D-752-4	LINK PIN (SHORT)	"	
15	8	A1913-D-753-5	LINK PIN (LONG)	"	
16	4	A1913-D-753-6	END LINK	"	
17	2 PAIR	A1913-D-852-1	HYDROSTATIC BRG. SHOE	"	
18	1 PAIR	A1913-D-852-2	HYDROSTATIC BRG. SHOE	"	
19	1 PAIR	A1913-D-852-3	HYDROSTATIC BRG. SHOE	"	
20	8	A1913-D-851-4	LINK PIN (LONG)	"	
21	8	A1913-D-851-3	RETAINING PLATE	BRONZE	
22	32	A1913-D-851-2	PIN NUT	STEEL	
23	32	A1913-D-851-1	CLAMP BLOCK	"	
24	—	A1913-D-751-3	KEY	——	
25	4	——	RETAINING RING	COMM.	TRUARC #5100-175
26	4	——	RETAINING RING	COMM.	TRUARC #5100-275
27	8	——	RETAINING RING	COMM.	TRUARC #5100-125
28	18	——	GREASE FITTING	COMM.	[illegible] ALEMITE #1610-B
29	—	——	FL. HD. MACH. SCREW	——	1/2-20NC × 3/4 LG. (REF.)
30	8	——	SOC. HD. CAP SCREW	COMM.	1-8NC × 4 LG. [illegible]
31	8	——	NUT, FINISHED HEX.	STEEL COMM.	1-8NC AMERICAN STD.
32	64	——	SOC. HD. CAP SCREW	ALLOY STEEL COMM.	5/8-11NC × 5 LG.
33	24	——	SOC. HD. CAP SCREW	ALLOY STEEL COMM.	1/4-20NC × 1/2 LG.
34	8	——	PIPE PLUG	STEEL COMM.	PARKER #4HP50N-S

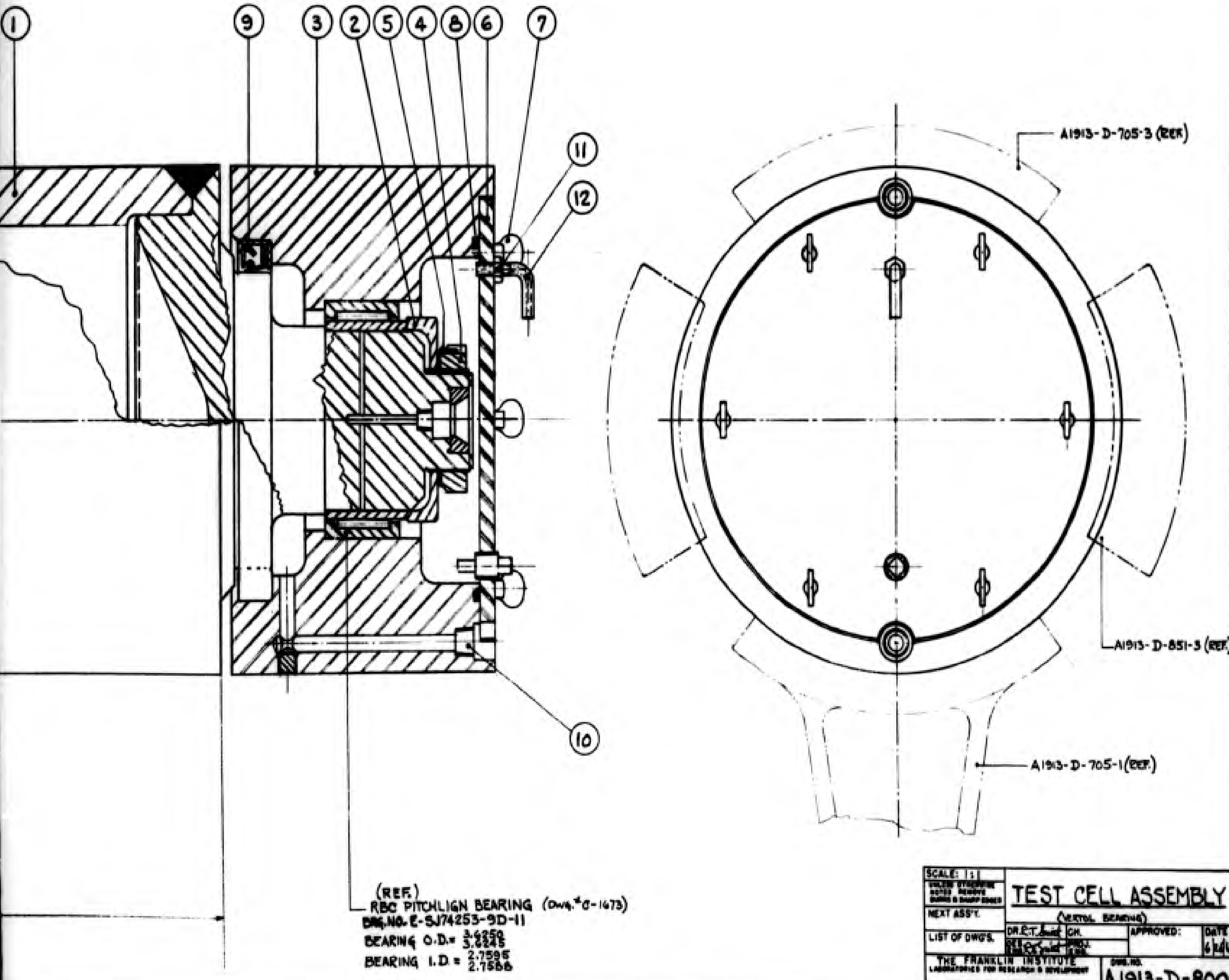
2

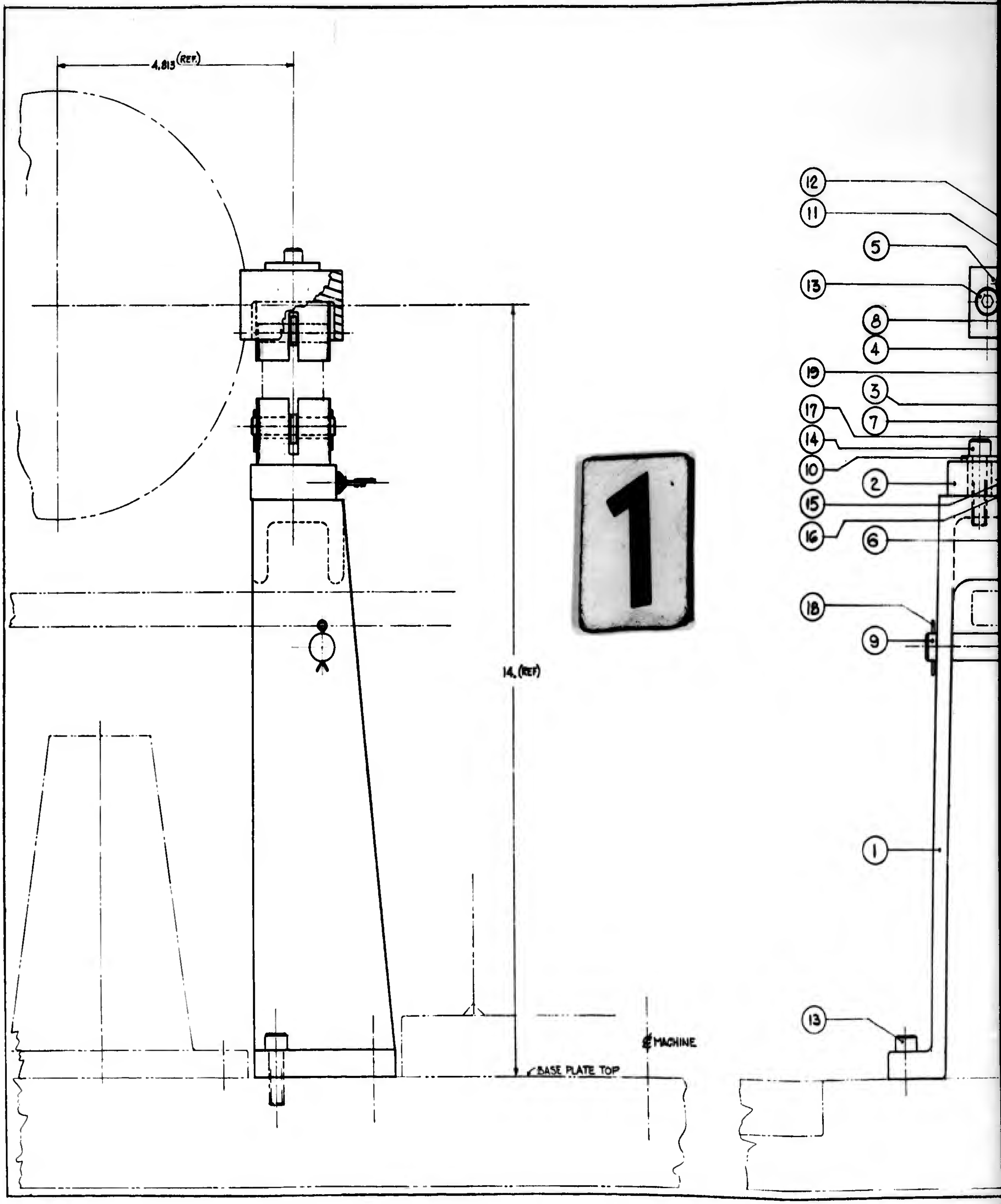
SCALE: 1/4 UNLESS OTHERWISE NOTED REMOVE BURRS & SHARP EDGES	TEST CELL LOADING SYSTEM ASSY.		
NEXT ASS'Y.			
LIST OF DWG'S.	DR. [illegible] DES. [illegible] CH. PROJ. ENG.	APPROVED:	DATE 6/26/61
THE FRANKLIN INSTITUTE LABORATORIES FOR RESEARCH & DEVELOPMENT PHILADELPHIA 3, PENNSYLVANIA	DWG. NO. A1913-D-750		


1
(1)
(9)
(3)
(2)
(5)
(4)
10.125 (REF.)
(REF.)
RBC PITCHLIGN
BRG. NO. E-SJ7425
BEARING O.D. =
BEARING I.D. =

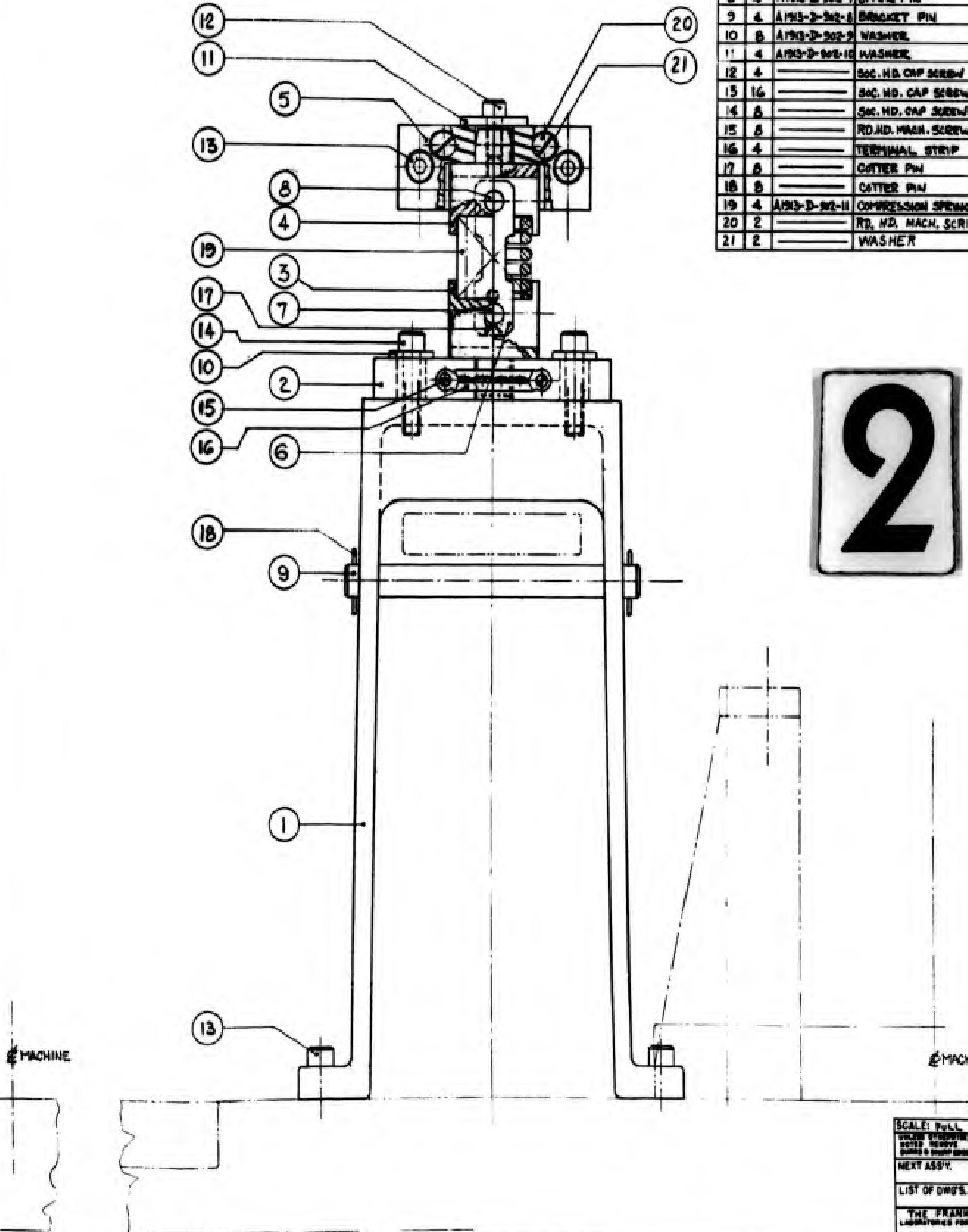
2

ITEM NO.	QTY.	DWG. NO.	NAME OF ITEM	M'T'L.	DESCRIPTION
LIST OF PARTS					
1	2	A1913-D-801-1	JOURNAL	STEEL	WELDMENT & ASSY.
2	4	A1913-D-801-2	RETAINER, N.BRG. RACE	"	
3	4	A1913-D-802-1	TEST CELL HOUSING	"	ASSY.
4	4	————	LOCK NUT	STEEL COMM.	SKF # N-08
5	4	————	LOCK WASHER	STEEL COMM.	SKF # W-08
6	4	A1913-D-801-4	JOURNAL COVER	PLASTIC	
7	24	————	THUMB SCREWS	STEEL COMM.	#10-32 NF × ½ LG.
8	4	————	'O' RING	COMPOUND # [illegible]	PARKER APPLIANCE CO. SIZE [illegible] [illegible]
9	4	————	OIL SEAL	COMM.	NATIONAL [illegible] [illegible]
10	8	————	PIPE PLUGS	STEEL COMM.	⅛-27 NPT SOC. HD.
11	4	————	HEX. LOCK NUT	STEEL COMM.	#10-24
12	4	A1913-D-801-5	BREATHER TUBE	STEEL	




4.813 (REF.)
14. (REF)
BASE PLATE TOP
℄ MACHINE
1
12
11
5
13
8
4
19
3
17
7
14
10
2
15
16
6
18
9
1
13

LIST OF PARTS					
ITEM NO.	QTY.	DWG. NO.	NAME OF ITEM	M'T'L.	DESCRIPTION
1	4	A1913-D-901-1	BRACKET	CAST IRON	CASTING
2	4	A1913-D-902-1	PLATE, BRACKET	STEEL	
3	4	A1913-D-902-2	SPRING RETAINER, LOWER	"	
4	4	A1913-D-902-3	SPRING RETAINER, UPPER	"	
5	4	A1913-D-902-4	PLATE, HOUSING	"	
6	4	A1913-D-902-5	STRAIN GAGE	SEE DETAIL	SOFT SOLDER
7	4	A1913-D-902-6	LOWER PIN	STEEL	
8	4	A1913-D-902-7	UPPER PIN	"	
9	4	A1913-D-902-8	BRACKET PIN	"	
10	8	A1913-D-902-9	WASHER	"	
11	4	A1913-D-902-10	WASHER	"	
12	4	———	SOC. HD. CAP SCREW	"	1/4-20 NC × 5/8 Lg.
13	16	———	SOC. HD. CAP SCREW	"	5/16-18 NC × 1 Lg.
14	8	———	SOC. HD. CAP SCREW	"	5/16-18 NC × 1 1/4 Lg.
15	8	———	RD. HD. MACH. SCREW	"	#4-40 NC × 1/4 Lg.
16	4	———	TERMINAL STRIP	COMM.	[illegible]
17	8	———	COTTER PIN	STEEL	3/64 D × 1/2 Lg.
18	8	———	COTTER PIN	"	1/16 D × 3/4 Lg.
19	4	A1913-D-902-11	COMPRESSION SPRING	"	
20	2	———	RD. HD. MACH. SCREW	"	1/4 - 20 NC × 1/2 Lg.
21	2	———	WASHER	"	



2

SCALE: FULL

NEXT ASS'Y.

LIST OF DWG'S.

TORQUE SENSING ASSEMBLY

DR. | CH. | APPROVED: | DATE

THE FRANKLIN INSTITUTE
LABORATORIES FOR RESEARCH & DEVELOPMENT
PHILADELPHIA 3, PENNSYLVANIA

DWG. NO. A1913-D-900

APPENDIX B

Instrumentation of Torque Indicating System

Appendix B

Instrumentation of Torque Indicating System

A. Introduction

For the purpose of monitoring the frictional torque in the test bearing during progress of the test, means were provided for obtaining an electrical signal from each test bearing and displaying this periodically in sequence on a suitable meter (See Figure B-1). It was thought that such an indication would provide sensitive indication of first failure in any of the bearing components. Also it appeared possible to readily obtain this from the torque restraint imposed upon the test bearing housing, because of its support on the fluid film of a hydrostatic bearing, in which the frictional drag is negligible.

Experience in operation of the test machine showed that the system was not a reliable means for the detection of bearing failure. This was primarily due to the fact that the comparatively low value of frictional torque in the needle roller bearing proved to be quite insensitive to the effects induced by surface discontinuity, in early stages of failure. Further test and study showed that disturbing effects resulted from variations in equilibrium of the loading system, and in dynamic response to the oscillating motion imposed upon the test bearing.

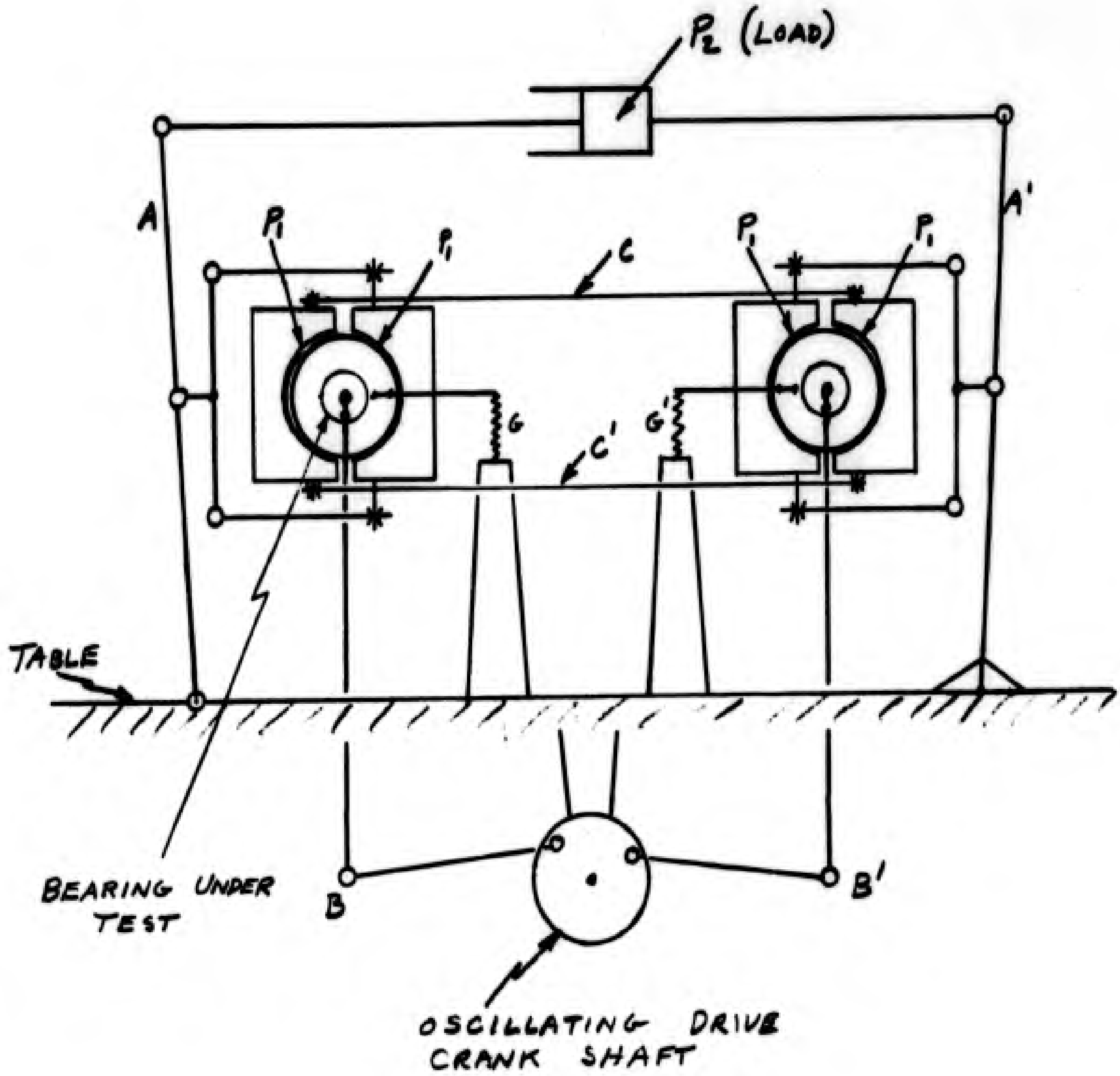


Fig. B-1 - MECHANICAL SCHEMATIC OF BEARING TEST MACHINE

B. Selection of Sensing Device

The object of the sensing element is to produce as large an electrical signal as possible for a given force that is to be measured. Since the mechanical signal to be measured is of a dynamic nature, the sensing element must not have a resonance within or close to the frequency spectrum of the driving frequency. An analysis was made of a strain gage type of sensing element mounted on a pre-stressed mounting strip. The natural frequency of the system was designed for 20 times the highest forcing frequency of the machine under test. The signal output for a minimum torque specified as 11.0 inch pounds must be in the order of 10^{-3} volts in order to have a good signal to noise ratio. These conditions required the use of a foil type of strain gage with a gage factor of 2 and a gage resistance of 1000 ohms. The voltage applied to the bridge circuit is adjustable at the 400 cps alternator used as the source of signal power. For the tests being conducted with this type of gage, the bridge voltage was set for 14 V rms. This level was found to be adequate with amplifier gain set to approximately .7 of maximum and individual channel gain control set at maximum.

C. Amplifier Design and Characteristics

The specifications for the amplifier are that it drive a 1 ma D.C. meter to full scale deflection for 1 mv rms of 400 cps signal at the input and have adjustable gain of 10:1. The gain accuracy must be held to 2%.

The amplifier designed for this system is shown in Figure B-2. The open loop gain of the voltage gain stages (3 stages) is approximately

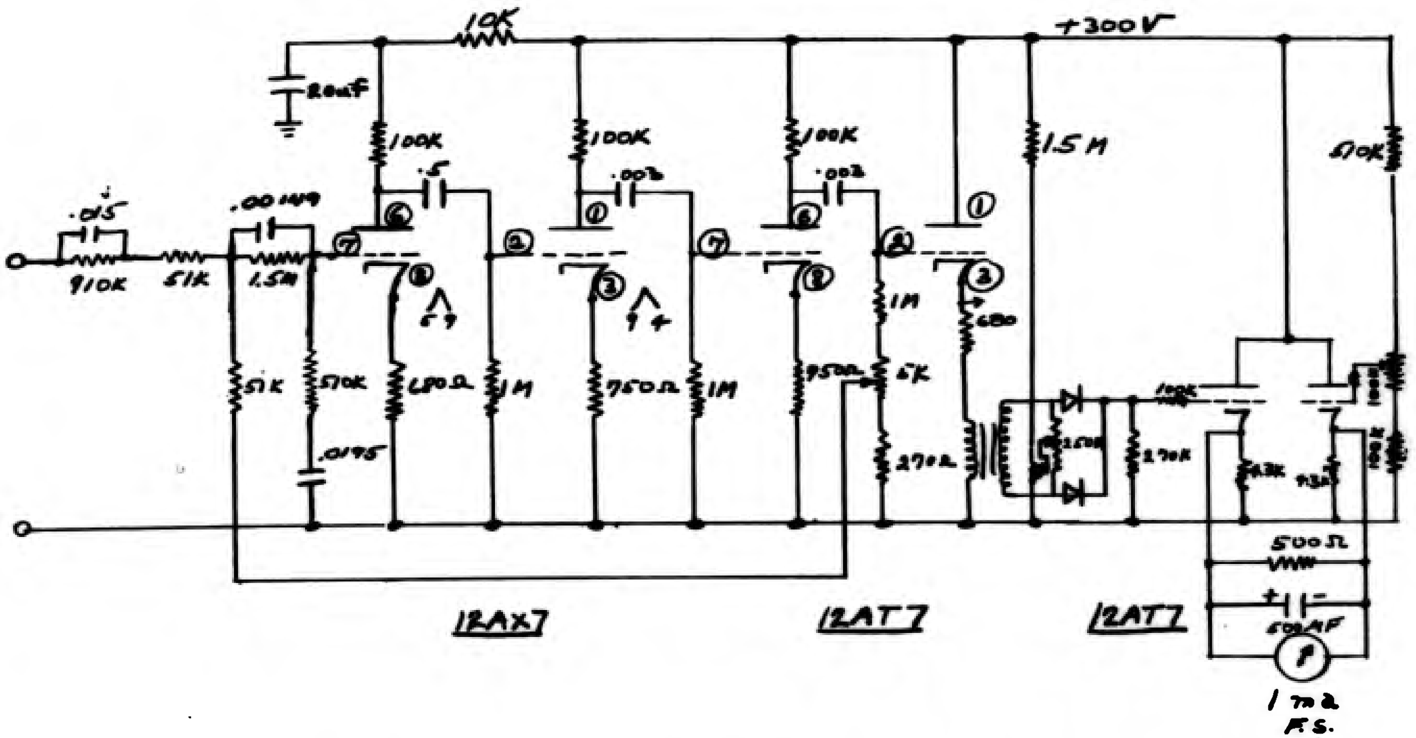


Fig. B-2 - SCHEMATIC OF AMPLIFIER FOR BEARING MONITOR

50,000 V/V or 94 db. The gain of the cathode follower stages is 1 ma F.S. on the read-out meter for 1.1 V rms input at the grid of the first cathode follower. Then in order to have full scale deflection for an input of 1 mv rms, the maximum gain of the amplifier must be $1.1/1 \times 10^{-3} = 1100$ V/V or 60.8 db which provides a gain accuracy of approximately $\frac{1}{94 - 60.8 \text{ db}} = \frac{1}{33.2 \text{ db}} = \frac{1}{46} = 2\%$. The minimum gain required of the amplifier is $1.1/10 \times 10^{-3} =$ 110 V/V or 40.8 db which provides a gain accuracy of approximately $\frac{1}{94 - 40.8 \text{ db}} =$ .67%. This gain accuracy must be at a frequency of 400 cps $\pm$ the oscillating frequency which in this application is less than 10 cps. For all gain settings, the amplifier has a bandwidth from 200 cps to 20 KC. The lower limit is determined by a network designed to have relatively low gain at 60 cps and still provide full gain at 400 cps as shown in Figure B-3.In this way, considerable 60 cps noise can be rejected at the input terminal of the amplifier. The other networks at the amplifier input are necessary to provide stability in the closed loop amplifier by providing the required phase shift and attenuation in the open loop characteristic of the amplifier.

The detector stage of the amplifier which consists of a transformer and full wave rectification to drive a double cathode follower meter stage, is provided with bias and balance adjustment for the diodes used to detect the 400 cps modulated wave. By adjusting these two controls, the nonlinear portion of the diode can be greatly reduced thereby producing a relatively linear output on the meter for increments of strain being measured by the transducer. The meter will indicate the average of the peak value of the demodulated carrier. This means then that the reading will vary for various

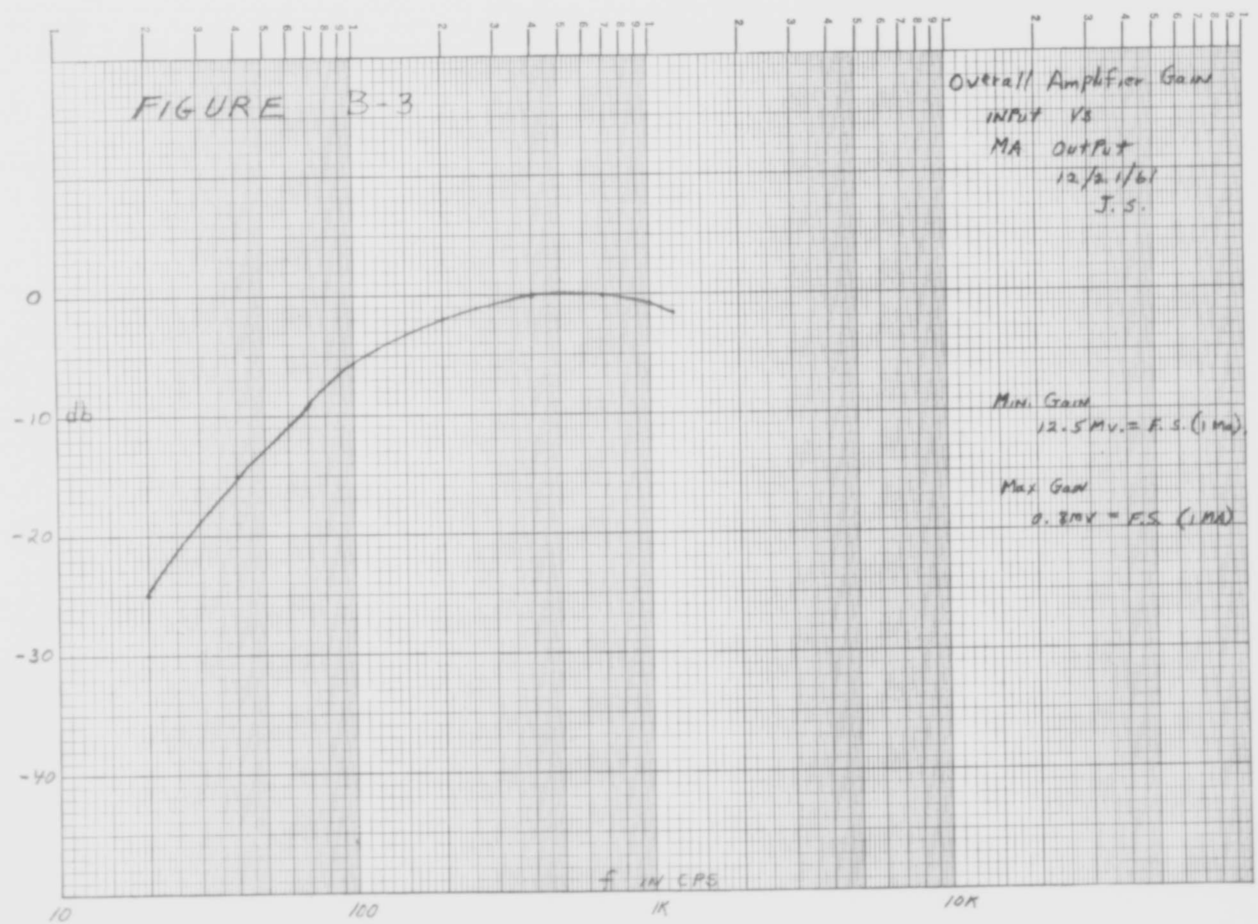
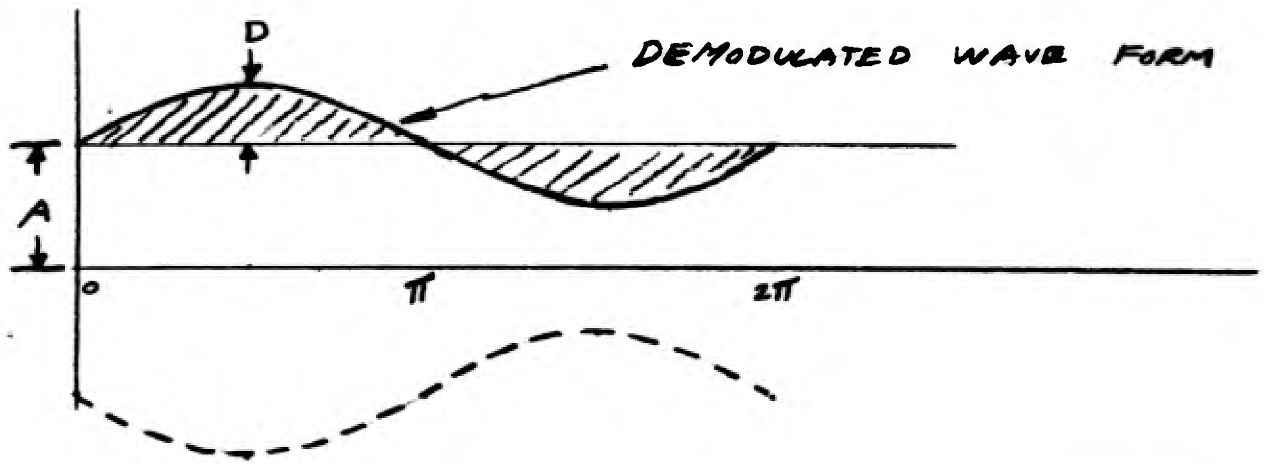


Fig. B-3 - OVERALL AMPLIFIER GAIN

wave forms whose peak value is the same. The average of a square wave forcing function input will be 1 and for a sine wave it will be .636. A triangular wave form will produce a reading of .5. This indicates the importance of knowing how the wave form is changing since the indicated force is a function of both the peak value of the forcing function and the wave form.

The initial balance of the system is also of considerable importance in reading the change of level of the forcing function. The analysis for the error incurred by initial unbalance in the transducer bridge circuit is shown in Figure B-4. If the electrical unbalance (A) at zero stress points indicated as 0, π, 2π, etc. is greater than the maximum stress signal (D) the average level will not vary as stress changes. Therefore, the most accurate setting for the system is for (A) to be zero when the stress is zero. If (A) has some finite value, there will be no change in meter reading until (D) becomes greater than (A). After (D) exceeds the value of (A) the meter will show a change proportional to the change in stress. The condition for maximum meter change for change in stress occurs when the value of (A) is zero. The electrical balance should be made at zero stress on the gage to obtain the most accurate indication of stress. The method of setting to a minimum meter reading while the machine is oscillating does reduce (A) to a value close to zero and will give an indication of change in stress to a limited accuracy. However, changes in low values of stress may be lost in this method of adjustment.



THE AVERAGE VALUE OF THE DEMODULATED WAVE FORM IS

$$\text{AVERAGE} = \frac{\int_0^{2\pi} A + D \sin x \, dx}{2\pi} = \int_0^{2\pi} \frac{A}{2\pi} + \int_0^{2\pi} \frac{D \sin x \, dx}{2\pi}$$

$$= \frac{Ax}{2\pi}\Big|_0^{2\pi} - \frac{D \cos x}{2\pi}\Big|_0^{2\pi} = \frac{A(2\pi)}{2\pi} - \frac{D \cos 2\pi}{2\pi} + \frac{D \cos 0}{2\pi}$$

$$= A - \frac{D}{2\pi} + \frac{D}{2\pi}$$

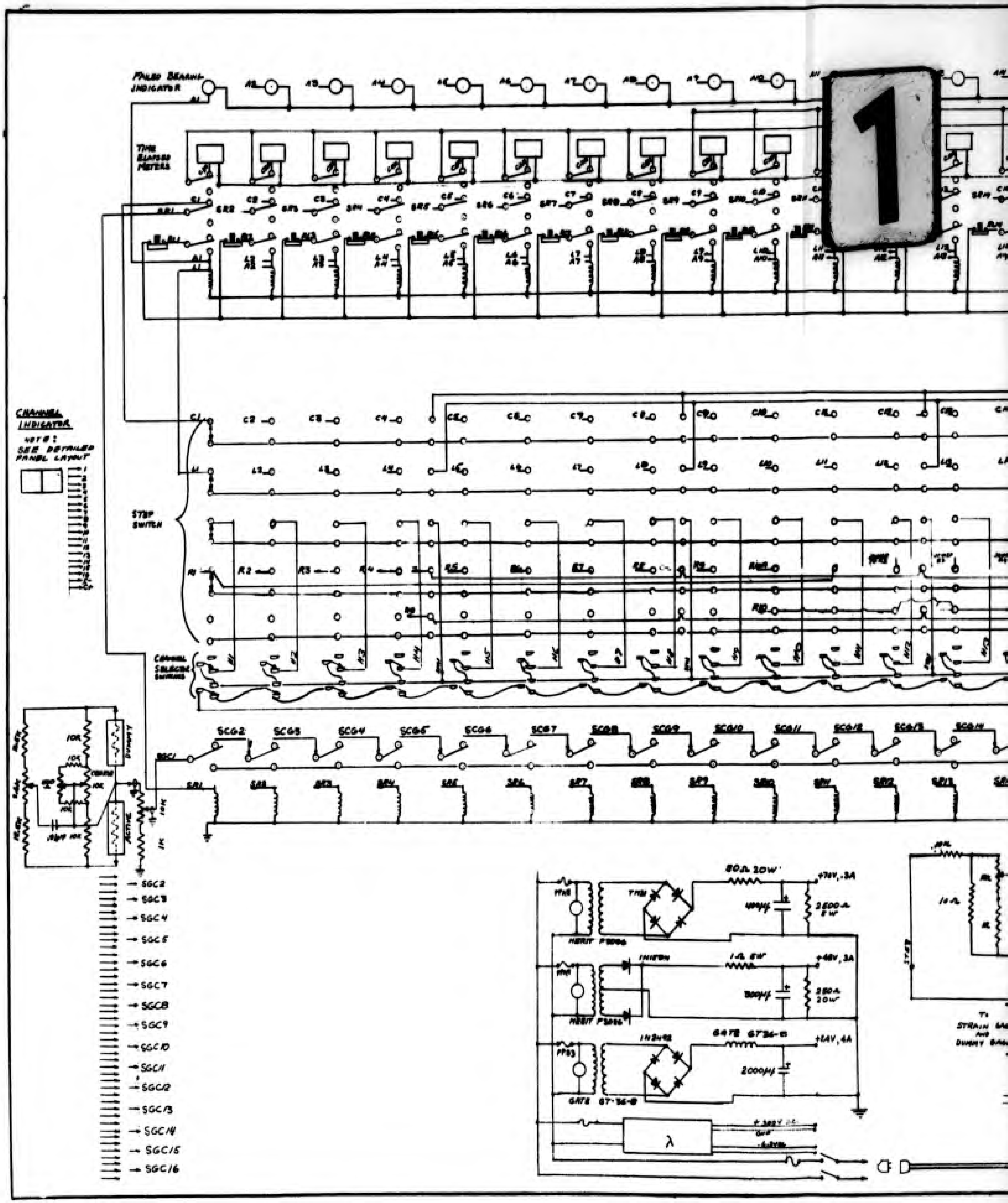
$$\text{AVERAGE} = A$$

Fig. B-4 - ANALYSIS OF BRIDGE BALANCE

D. Selection and Design of Multi-Channel Instrumentation

The transducer and amplifying system as discussed are adequate to measure the anticipated forcing functions of a bearing under test. The use of this type of system for monitoring many bearings presents a problem as to the most economical and reliable method of accomplishing this end result. An amplifier and meter for each channel is quite expensive and would require considerable space for the instrumentation. The phenomenon to be observed takes place over a period of many hours. A system that would measure and indicate the forcing function several times during each hour of operation would be sufficient to indicate the rate of change in the force level being monitored. The system selected for this instrumentation uses a force sensing load cell and a dummy gage for temperature compensation as one half of a bridge circuit for each bearing to be monitored. The other half of the bridge circuit is common to all of the channels. A single amplifier connected through switching circuitry amplifies the signal and indicates the level of force on a meter relay. A time elapsed meter records the length of time the bearing has been in operation and the elapsed time meter is disconnected by the meter relay when a preselected level has been reached. A red light associated with each time elapse meter indicates that the bearing under test has reached the selected level that is indicative of failure. The length of time for each reading is approximately 6 sec with a period of 1.5 sec required for switching. During a 1 hour period, each channel is monitored 20 times. This is for a 16 channel machine with 5 amplifier gain check positions, 1 synchronizing position and a homing period. The

complete circuit diagram of this system is shown in Figure B-5. Four power supplies are required to operate the relays, step switches and amplifiers. A 400/500 watt cps alternator is required to supply signal power to the strain gage bridges. The amplifier input is connected to the signal relay contacts through the input transfer relay. The signal relay is controlled by the step switch and connects the signal from either a bearing being tested or the reference signal used to check the amplifier gain. The step switch is controlled by a cam operated by a constant speed motor. The output circuit of the amplifier is used to drive a meter relay. An adjustable contact on the meter relay is used to energize a self-locking relay circuit. The meter relay in turn controls the time elapsed meters, the signal relays and the indicator lights to indicate bearing failure. The time elapsed meters are controlled by a cam and motor which operates 16 meters in groups of 8 at alternate intervals of 6 minutes. The meters read directly hours and tenths of an hour. The timer cam for the elapsed time meters is interlocked with the channel selector cam so that the switching pulse for the elapsed time meters cannot occur when a signal is being read by the instrumentation.
The sequence of the stepping operation is shown in Figure B-6. The power to operate the relays is connected through the channel selector switches. The 16 selector switches are wired in series. In this way, when a particular channel is selected, the selector control cam is disconnected from the circuit until the selector button is released. All of the signal relays are also disconnected at this time so that no signal appears at the input to the amplifier until the selector button is released. The step switch will


FAILED BEARING INDICATOR
TIME ELAPSED METERS
CHANNEL INDICATOR
NOTE: SEE DETAILED PANEL LAYOUT
STEP SWITCH
CHANNEL SELECTOR SWITCHES
SCG2
SCG3
SCG4
SCG5
SCG6
SCG7
DUMMY
ACTIVE
SGC2
SGC3
SGC4
SGC5
SGC6
SGC7
SGC8
SGC9
SGC10
SGC11
SGC12
SGC13
SGC14
SGC15
SGC16
λ

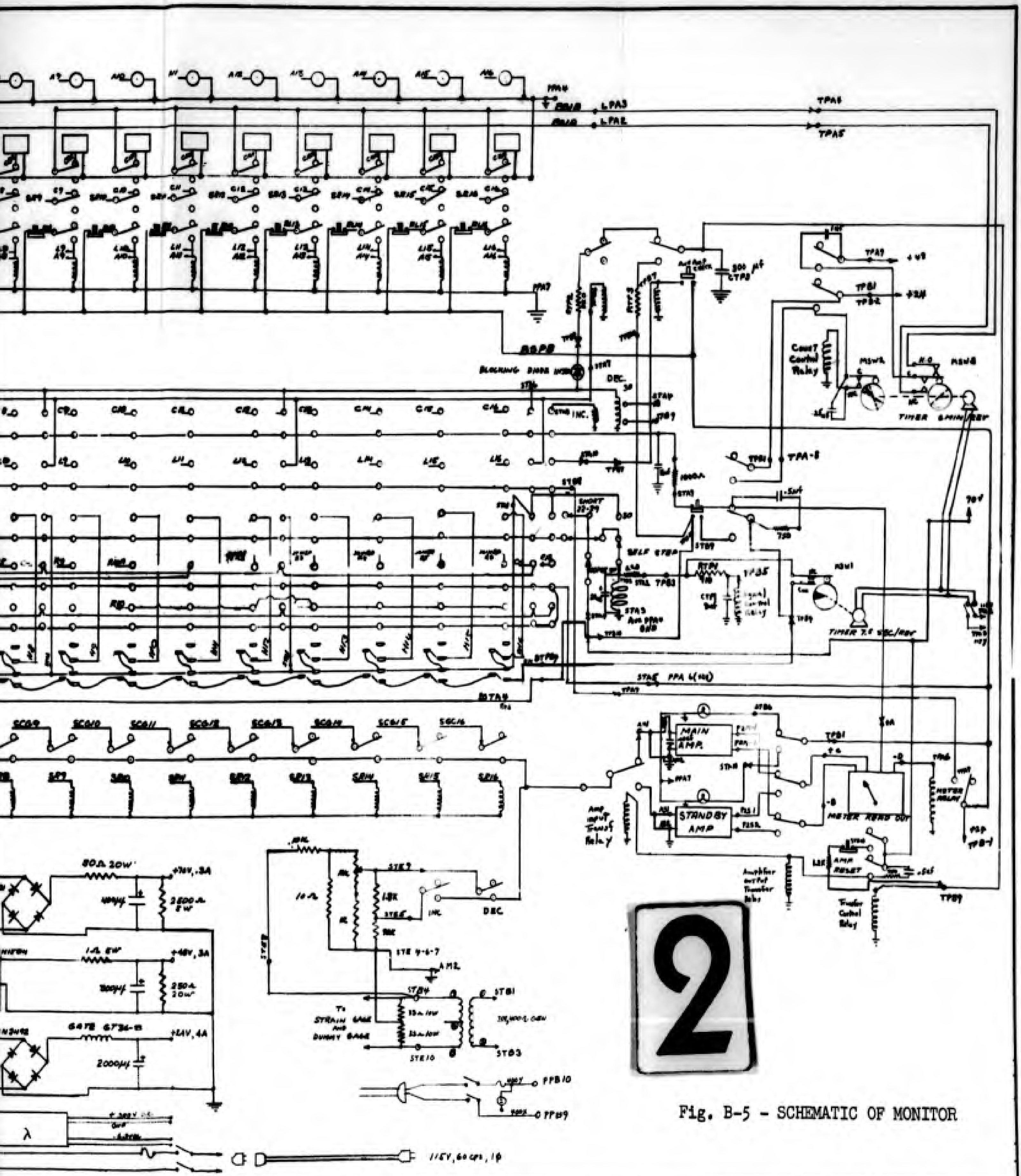


Fig. B-5 - SCHEMATIC OF MONITOR

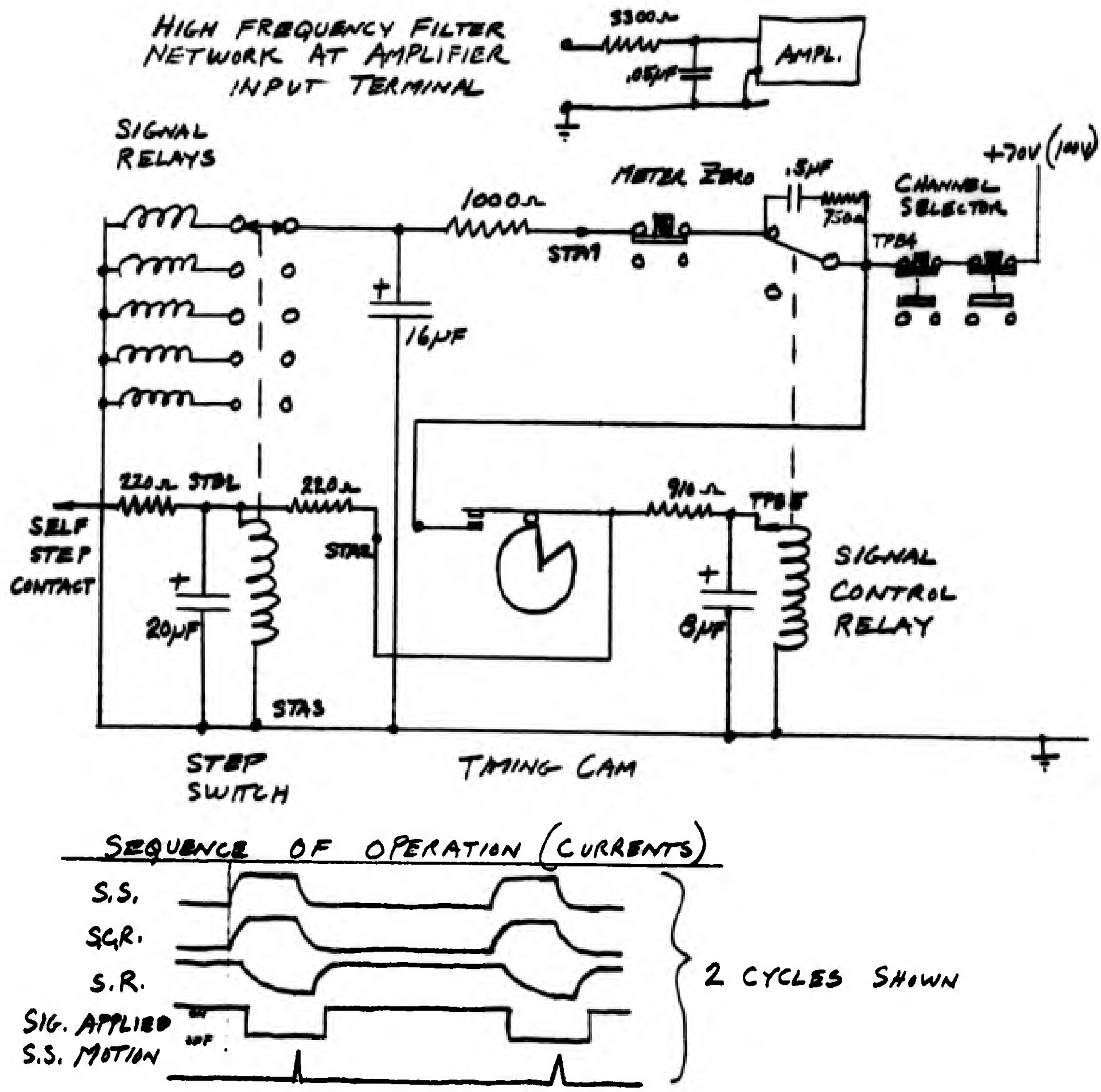


Fig. B-6 - CIRCUIT CHANGES REQUIRED TO ELIMINATE STEP PULSES IN AMPLIFIER INPUT self-step as long as the selector switch is depressed and will stop at the selected channel. The step switch can be jogged by momentarily depressing a selector switch other than the one associated with the channel at which the step switch is resting. When the step switch arrives at position 22, the self-stepping contact is energized and the step switch will continue stepping until the homing switch in the step switch is opened stopping the switch at position "0." This position was selected so that if the stepping control cam were in the closed position when the step switch arrived at "0," the wiper would stop at position 1 thereby not eliminating the reading at position 1. It also guarantees a full time interval for reading position 1. Returning now to the flow of power in the sequence of stepping, after the channel selector switches, the power is connected to the signal control relay arm and to the timing cam switch. After the signal control relay contact, it is connected to the selected signal relay through a meter zero push button switch and a time delay circuit. The timing of the system is shown at the bottom of the schematic. The step switch and signal control relay are both energized together through the timing cam. The delay circuits have been built in to both reduce switching transient voltages and delay the drop out time of the relay. After the signal control relay pulls in, the signal relay drops out removing the signal from the amplifier input terminals. When the timing cam opens the circuit again, the step switch steps the next position. The delay circuits hold the signal relay contact open until all of the switching has been completed. A high frequency filter circuit as shown at the top of the schematic is used at the amplifier input to reduce switching transients that are above 1 KC.

The bearing failure indicating system is controlled by the meter relay circuit. When the amplified signal level has reached a preselected level, the meter relay closes. This applies 24 V power to the time elapsed meter control relay of the channel being monitored. This relay has a self-locking circuit that is released by a push button switch. The relay when energized disconnects the signal relay, the time elapsed meter and applies power to the failed bearing indicator light for that channel. The meter relay is released when the signal control relay is energized for the next position of the step switch.

A self-checking system is built into the instrumentation to make certain that the amplifier gain and zero signal supply voltage does not drift more than 4% of the full scale reading. This check is made at position 5, 10, 15, 20 and 21. In the first four positions, the indicating arm of the meter relay must make contact with the contact arm that has been set for a given signal level. In the fifth position, the meter indicating arm must not touch the contact arm. The signal for this is taken from the source of supply for the strain gage bridges. The level can be adjusted over a wide range. Two relays are used to select the increase and decrease gain check signal levels to be applied to the amplifier input. The input to the amplifier being in the order of 100 K, the output from the 1.8 K and 90 K voltage divider will be 1 and .96 or a difference of 4%. The drift and gain of the amplifier must stay within this limit. If the amplifier is either higher or lower than this limit, a second amplifier whose gain is set to be similar to the first amplifier, is automatically connected into the circuit. In this way, readings are not lost and repair can be made without interruption of testing.

The setting of the amplifier gain is determined by the signal level of the bearing with the lowest forcing function output signal. Figure B-7 shows the wide range of adjustment available from the instrumentation. Each channel has a gain control by which the signal from the strain gage bridge can be adjusted in the ratio of 10 to 1. The amplifier, which is common to all channels, is also adjustable in a ratio of 10 to 1. The minimum signal to be measured is 11 inch lbs of torque which must produce 90% of the full scale reading on the meter. The maximum total gain of the system is then adjusted to the level of the smallest signal developed when the bearing is first installed. It is desirable that this signal should deflect the read-out meter to 10% of full scale. Once this level is set, all of the remaining channels are adjusted through their attenuator gain control potentiometers to give the same deflection. The signal level of the sampling circuit must be adjusted to produce a deflection of the read-out meter equivalent to the desired bearing failure level of the decrease gain check (C-).

$$OUTPUT\ (ma) = X\left(\frac{V_{rms}}{X}\right)\left(AMPLIFIER\ GAIN\ V/V\right)\left(ATTN.\right)\left(\frac{ma}{V_{rms}}\right)$$

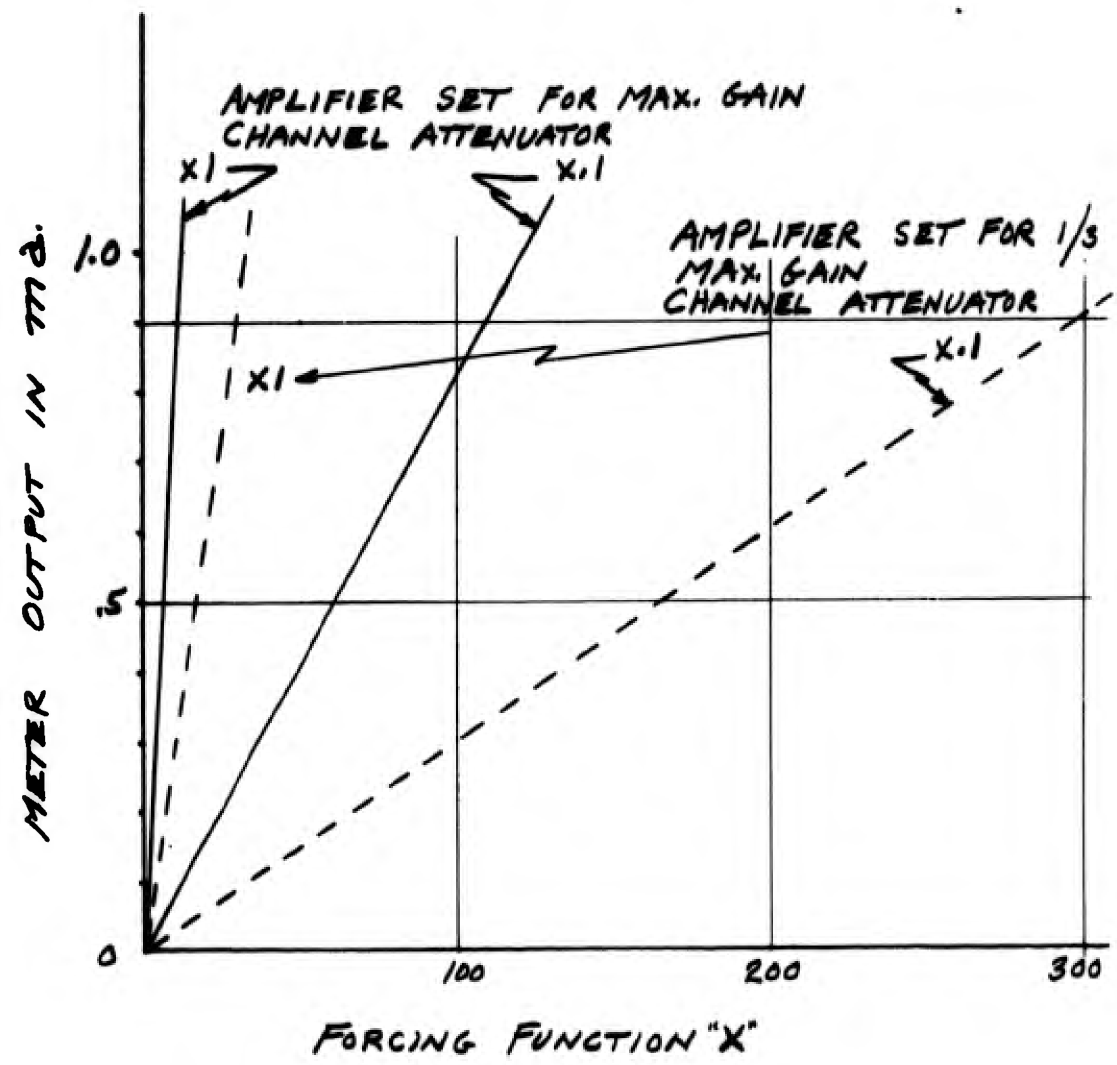


Fig. B-7 - RANGE OF GAIN CONTROL

APPENDIX C

Photographs of Installation

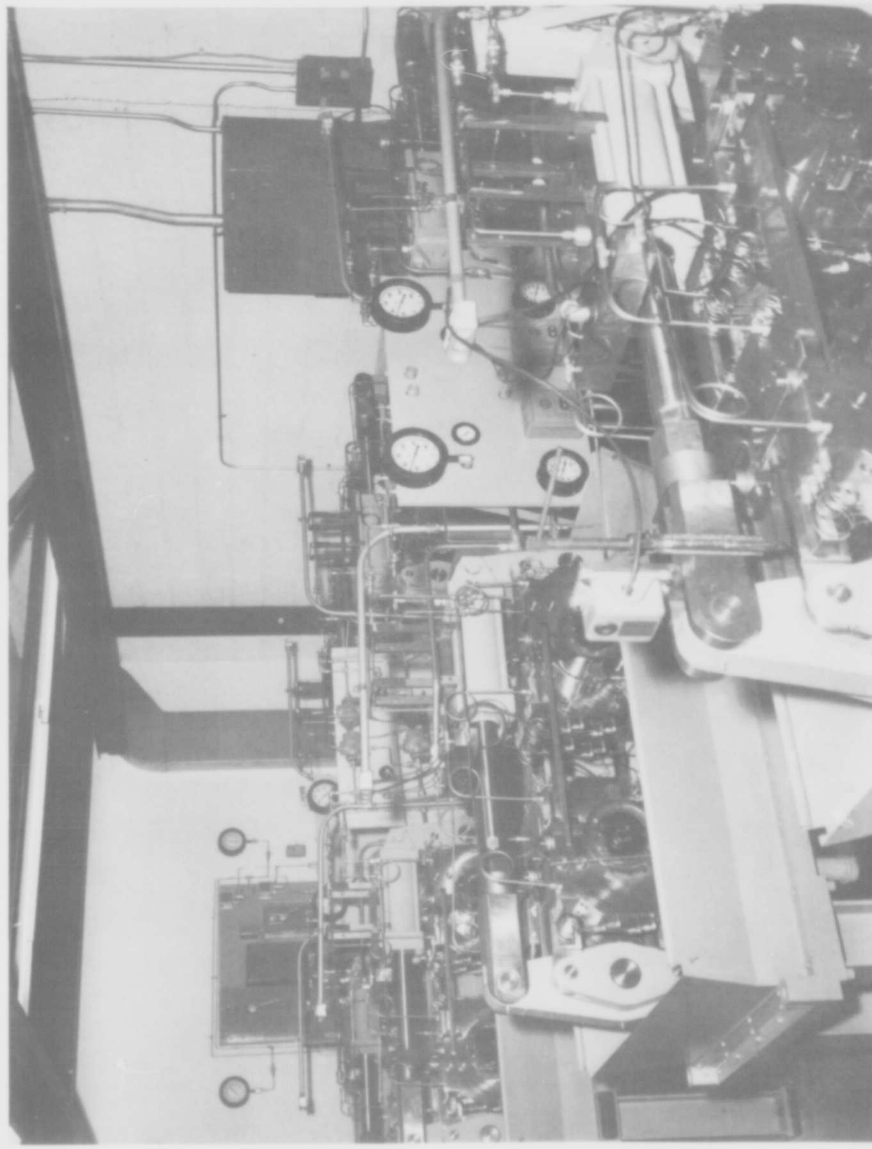


Fig. C-1 - OVERALL VIEW OF INSTALLATION

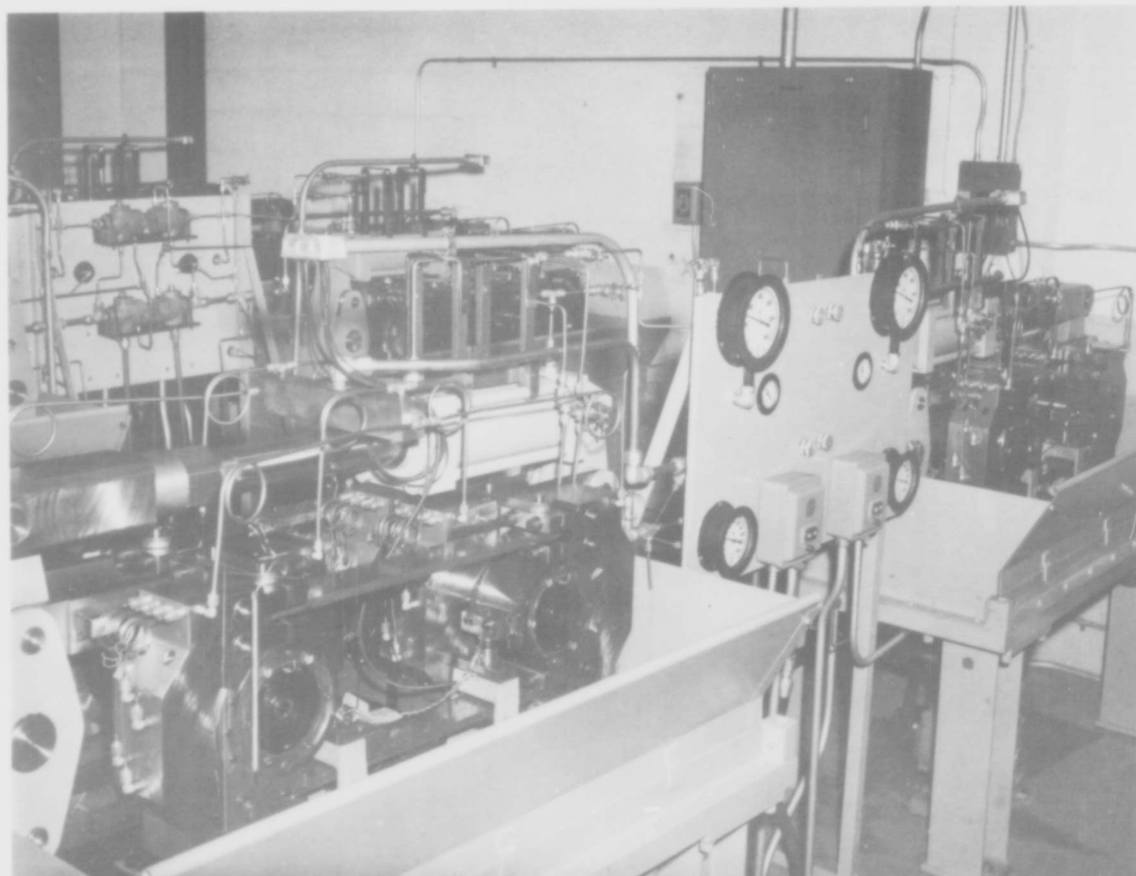


Fig. C-2 - VIEW OF ONE TEST MACHINE

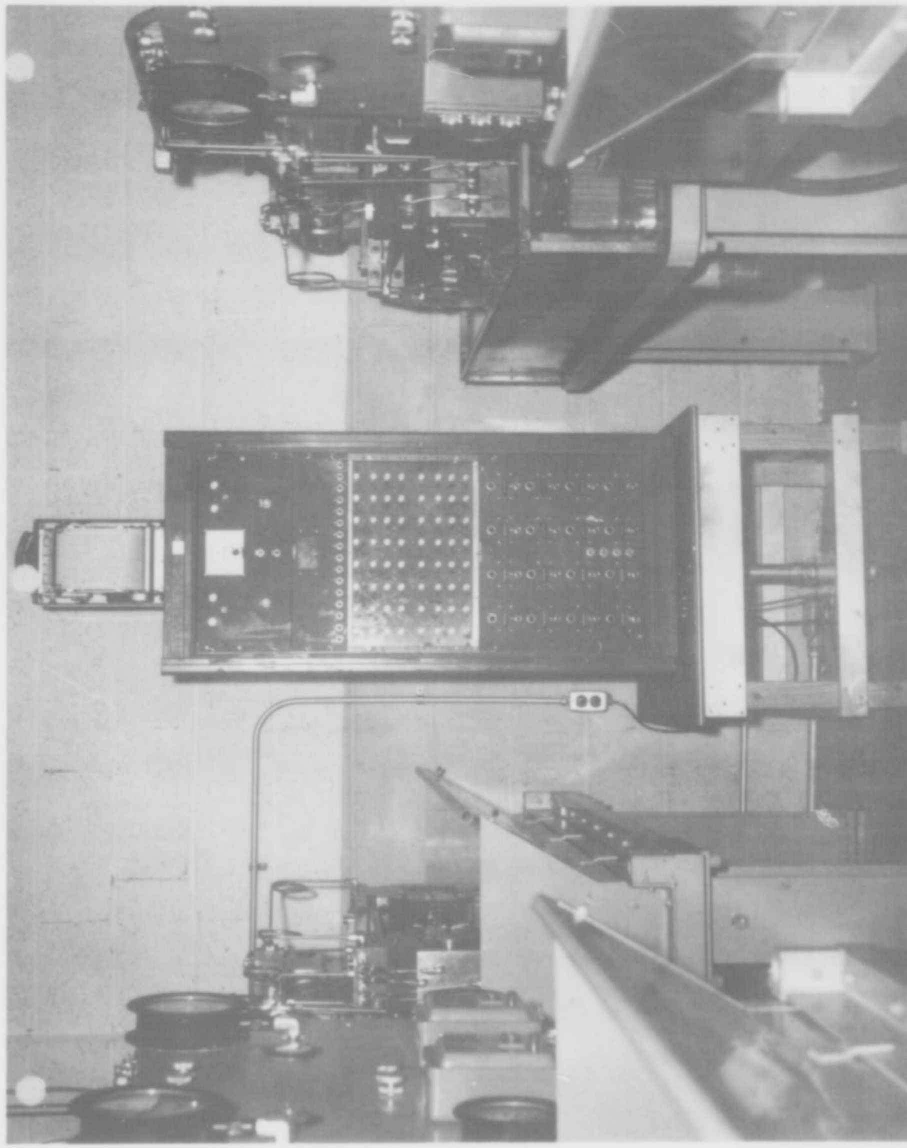


Fig. C-3 - VIEW OF INSTRUMENT CONSOLE

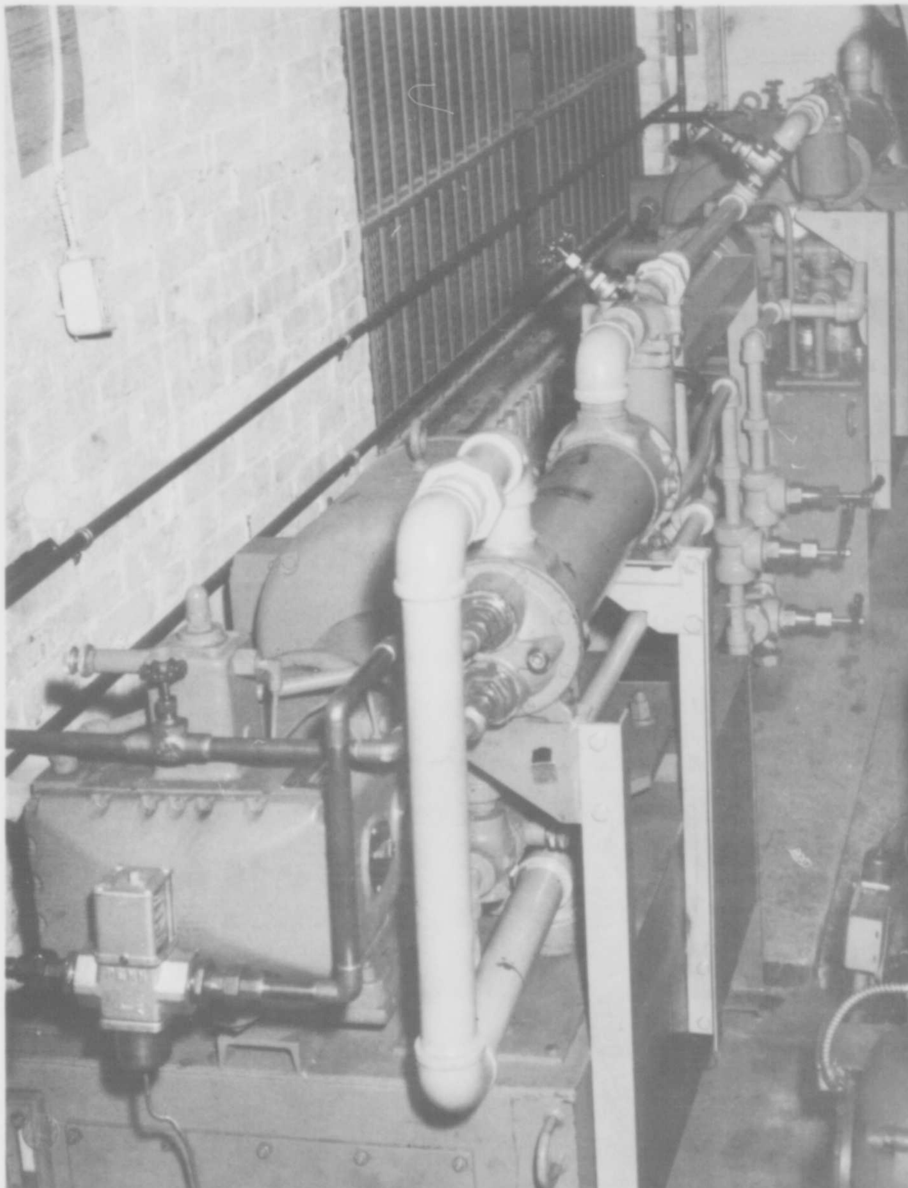


Fig. C-4 - VIEW OF HYDRAULIC PUMPS AND RESERVOIRS

APPENDIX D

Summary of Test Data

Table No. D-1

A TEST

Failure						No Failure		
			Failed Element					
Serial Number	Life in Hours	Order Number	Outer Race	Inner Race	Rollers	Serial Number	Life in Hours	Order Number
10	23	1	x					
55	26	2	x					
19	43	3	x		x			
53	44	4		x				
45	46	5	x					
47	46	6	x	x				
46	48	7		x				
12	50	8	x					
24	65	9	x		x			
						10'	67	1
31	70	10	x					
37	71	11	x	x	x			
33	75	12	x	x	x			
34	90	13	x					
70	93	14		x				
40	98	15	x		x			
39	104	16	x					
16	113	17	x		x			
17	126	18	x					
69	137	19	x		x			
71	137	20	x		x			
25	158	21	x					
28	178	22	x		x			
52	206	23	x					

Table No. D-2

B TEST

Failure						No Failure		
			Failed Element					
Serial Number	Life in Hours	Order Number	Outer Race	Inner Race	Rollers	Serial Number	Life in Hours	Order Number
73	47	1	x					
54	90	2	x	x				
14	93	3		x				
9	114	4	x					
83	127	5	x					
84	127	6	x			2'	134	1
						3'	134	2
						4'	134	3
						5'	134	4
41	138	7	x					
11	140	8	x	x				
23	144	9	x					
68	151	10	x					
						12'	155	5
38	160	11		x				
36	180	12		x				
57	187	13	x		x			
22	191	14	x	x				
29	204	15	x		x			
20	209	16	x	x				
63	236	17	x		x			
65	246	18	x	x				
51	247	19	x		x			
93	251	20		x				
42	252	21	x		x			
105	280	22		x				
18	287	23	x	x	x	16'	298	6
21	342	24		x	x			
32	353	25	x					
						21'	373	7
43	388	26	x		x			
						20'	423	8
13	428	27	x	x				
35	466	28		x				
77	468	29	x	x	x			
26	470	30	x		x			
						6'	478	9
15	478	31	x	x				
64	487	32		x				
30	516	33	x		x			
						19'	519	10
						13'	572	11
92	789	34			x			
58	887	35		x	x			
67	914	36	x					
						17'	930	12

Table No. D-3

C TEST

Failure						No Failure		
			Failed Element					
Serial Number	Life in Hours	Order Number	Outer Race	Inner Race	Rollers	Serial Number	Life in Hours	Order Number
27	470	1		x				
78	521	2	x	x				
79	991	3	x					
80	991	4	x					
						11'	991	1

Table No. D-4

D TEST

Failure						No Failure		
			Failed Element					
Serial Number	Life in Hours	Order Number	Outer Race	Inner Race	Rollers	Serial Number	Life in Hours	Order Number
44	817	1	x	x				
48	839	2	x	x				
49	839	3	x	x				
50	839	4	x					
56	972	5	x					
						7'	972	1
59	1039	6	x	x		8'	972	2
60	1039	7	x	x		9'	972	3
61	1039	8	x					
62	1039	9	x	x				

Table No. D-5

E TEST

Failure						No Failure		
			Failed Element					
Serial Number	Life in Hours	Order Number	Outer Race	Inner Race	Rollers	Serial Number	Life in Hours	Order Number
445	16	1	x			218'	6	1
114	18	2	x			216'	6	2
66	19	3	x			205'	10	3
451	20	4	x			222'	16	4
123	22	5	x			197'	17	5
86	22	6	x			200'	19	6
399	28	7	x	x		220'	26	7
167	31	8	x			221'	26	8
431	32	9	x			240'	42	9
427	40	10	x			229'	46	10
72	41	11	x			231'	46	11
74	45	12	x			33'	49	12
147	45	13	x		x	34'	49	13
104	48	14		x		155'	50	14
411	48	15	x		x	223'	68	15
299	50	16	x					
415	54	17	x					
82	58	18	x					
94	64	19	x					
400	65	20	x	x				
453	66	21	x					
452	66	22	x		x	193'	69	16
163	67	23	x	x		195'	69	17
174	69	24	x			217'	75	18
432	69	25	x		x	198'	77	19
444	73	26	x			199'	77	20
293	74	27	x					
154	76	28	x					
75	82	29	x					
180	88	30	x	x				
283	96	31	x					

Table No. D-5 (Cont.)

E TEST

Failure						No Failure		
			Failed Element					
Serial Number	Life in Hours	Order Number	Outer Race	Inner Race	Rollers	Serial Number	Life in Hours	Order Number
308	96	32	x			144'	84	21
113	98	33	x	x	x	215'	102	22
160	98	34		x		214'	102	23
465	99	35	x			212'	102	24
150	112	36	x	x	x	194'	117	25
81	117	37	x					
297	117	38	x		x			
164	119	39	x	x	x			
459	120	40	x	x	x			
173	120	41	x					
435	123	42	x	x		18'	124	26
90	128	43	x			148'	140	27
433	137	44	x	x	x	226'	213	28
142	141	45	x			168'	289	29
179	149	46	x			22'	441	30
140	150	47	x					
457	151	48	x					
134	159	49	x		x			
99	179	50	x	x				
119	191	51		x				
310	217	52	x					
158	464	53	x					

Table No. D-6

F TEST

Failure						No Failure		
			Failed Element					
Serial Number	Life in Hours	Order Number	Outer Race	Inner Race	Rollers	Serial Number	Life in Hours	Order Number
153	172	1		x				
187	190	2	x	x				
139	244	3	x					
95	378	4		x				
96	378	5	x		x			
97	443	6	x	x				
98	443	7	x	x	x			
127	479	8	x		x			
172	489	9	x					
186	507	10	x			35'	532	1
136	715	11	x		x			
185	749	12		x				
138	787	13	x	x	x			
						247'	803	2
						248'	803	3
						249'	803	4
						250'	803	5
151	868	14	x					

Table No. D-7

G TEST

Failure						No Failure		
			Failed Element					
Serial Number	Life in Hours	Order Number	Outer Race	Inner Race	Rollers	Serial Number	Life in Hours	Order Number
						122'	214	1
						152'	312	2
						127'	341	3
258	390	1		x				
						120'	431	4
260	526	2	x	x	x			
						154'	636	5
273	648	3	x	x	x			
262	665	4		x	x			
272	718	5		x				
						151'	745	6
						153'	745	7
						128'	771	8
266	794	6		x				
274	825	7	x		x			
						100'	854	9
						123'	862	10
						124'	862	11
271	903	8	x		x			
						119'	928	12
277	928	9	x					
						121'	928	13
						115'	997	14
269	1013	10	x	x	x			
268	1022	11	x		x			
						107'	1022	15
						108'	1022	16
						99'	1026	17
263	1026	12	x		x			
264	1026	13	x		x			
						106'	1052	18

Table No. D-7

G TEST (Cont'd)

Failure						No Failure		
			Failed Element					
Serial Number	Life in Hours	Order Number	Outer Race	Inner Race	Rollers	Serial Number	Life in Hours	Order Number
267	1052	14	x					
						126'	1059	19
						129'	1059	20
						125'	1089	21
						116'	1093	22
						101'	1115	23
						102'	1115	24
						103'	1115	25
						130'	1139	26

Table No. D-8

H TEST

Failure						No Failure		
			Failed Element					
Serial Number	Life in Hours	Order Number	Outer Race	Inner Race	Rollers	Serial Number	Life in Hours	Order Number
						92'	343	1
						138'	364	2
						145'	506	3
						147'	506.	4
292	506	1		x				
270	571	2	x		x			
259	577	3	x	x	x			
						137'	623	5
276	636	4	x	x	x			
						135'	709	6
265	872	5	x					
						114'	996	7
						110'	999	8
						111'	999	9
						112'	999	10
						113'	999	11
						136'	1000	12
						139'	1004	13
						140'	1004	14
						141'	1004	15
						142'	1004	16
						109'	1014	17
						117'	1049	18
275	1049	6	x					
						118'	1049	19
						146'	1059	20
						104'	1066	21
						105'	1066	22
						96'	1076	23
						97'	1076	24
						98'	1076	25

Table No. D-9

A' TEST

Failure						No Failure		
			Failed Element					
Serial Number	Life in Hours	Order Number	Outer Race	Inner Race	Rollers	Serial Number	Life in Hours	Order Number
						32'	24	1
115	33	1	x					
181	45	2	x					
76	46	3	x					
193	48	4	x					
189	53	5	x					
156	54	6	x					
148	57	7		x		14'	59	2
106	68	8	x					
169	68	9	x		x			
133	71	10	x		x			
170	74	11	x					
194	75	12	x					
108	91	13		x		27'	94	3
91	101	14	x	x	x	39'	101	4
85	105	15		x		15'	105	5
175	106	16	x					
182	107	17	x	x				
100	116	18	x					
125	119	19	x			40'	124	6
188	128	20	x	x				
177	152	21	x					
143	158	22	x					
157	180	23		x				
101	217	24	x					
102	217	25	x		x			
155	314	26	x					
141	327	27	x		x			
124	510	28		x				

Table No. D-10

B' TEST

Failure						No Failure		
			Failed Element					
Serial Number	Life in Hours	Order Number	Outer Race	Inner Race	Rollers	Serial Number	Life in Hours	Order Number
130	152	1		x	x			
162	154	2	x	x	x			
120	161	3		x				
190	162	4		x				
184	191	5		x	x			
176	206	6		x				
107	212	7	x		x	44'	220	1
126	231	8	x	x				
135	252	9		x	x			
149	272	10	x	x	x			
146	288	11	x					
122	307	12	x		x			
197	314	13		x				
117	317	14	x	x				
166	323	15	x	x	x			
131	325	16	x	x				
206	383	17	x		x			
118	388	18	x		x			
168	398	19	x		x			
159	445	20	x					
109	448	21	x	x	x			
110	448	22	x	x				
145	448	23	x	x	x			
171	455	24	x		x			
128	459	25	x		x			
129	459	26	x					
198	480	27	x					
116	539	28	x	x				
137	565	29	x		x	45'	571	2
161	611	30	x		x			
						41'	619	3
						46'	721	4
						47'	721	5
178	772	31	x		x			
200	866	32		x				
207	872	33	x					
						43'	915	6
						42'	958	7
						28'	977	8
						30'	1055	9
152	1121	34		x				

Table No. D-11

C' TEST

Failure						No Failure		
			Failed Element					
Serial Number	Life in Hours	Order Number	Outer Race	Inner Race	Rollers	Serial Number	Life in Hours	Order Number
						37'	27	1
87	89	1	x					
103	174	2	x					
165	218	3	x	x				
132	354	4	x	x		38'	382	2
191	382	5	x					
111	399	6		x				
121	568	7		x				
192	945	8	x	x				
						29'	1073	3
						26'	1098	4
						36'	1103	5

Table No. D-12

D' TEST

Failure						No Failure		
			Failed Element					
Serial Number	Life in Hours	Order Number	Outer Race	Inner Race	Rollers	Serial Number	Life in Hours	Order Number
						31'	343	1
88	353	1	x					
89	425	2		x				
183	601	3	x	x				
213	644	4		x		25'	617	2
112	740	5	x					
						51'	902	3
144	933	6	x	x		24'	1005	4
						50'	1245	5
						52'	1245	6
						23'	1380	7

Table No. D-13

E' TEST

Failure						No Failure		
			Failed Element					
Serial Number	Life in Hours	Order Number	Outer Race	Inner Race	Rollers	Serial Number	Life in Hours	Order Number
287	69	1	x					
						134'	77	1
280	77	2			x			
281	77	3	x					
						49'	93	2
196	138	4	x	x				
234	154	5	x					
242	166	6	x					
225	188	7	x					
227	207	8	x		x			
211	236	9	x					
						75'	239	3
231	269	10	x					
202	281	11	x		x			
243	301	12	x	x	x			
209	374	13	x		x			
210	374	14	x	x	x			
						76'	508	4

Table No. D-14

F' TEST

Failure						No Failure		
			Failed Element					
Serial Number	Life in Hours	Order Number	Outer Race	Inner Race	Rollers	Serial Number	Life in Hours	Order Number
195	162	1		x				
201	272	2	x	x	x			
219	595	3	x					
						56'	706'	1
						54'	867	2
						55'	867	3
235	1035	4	x					
						65'	1035	4
						66'	1035	5
						67'	1035	6
						93'	1112	7
261	1112	5	x					
						94'	1112	8
						95'	1112	9

Table No. D-15

TEST a

Failure						No Failure		
			Failed Element					
Serial Number	Life in Hours	Order Number	Outer Race	Inner Race	Rollers	Serial Number	Life in Hours	Order Number
239	44	1	x					
220	45	2	x					
232	52	3	x					
199	69	4	x					
205	109	5	x					
204	131	6	x		x			
203	133	7	x					
240	139	8	x		x			
236	140	9	x		x			
228	191	10	x					
224	193	11	x		x			
233	217	12	x	x	x			
						72'	248	1
229	250	13	x	x	x			
						53'	295	2
217	338	14	x		x			
222	471	15	x	x	x			
218	473	16	x					

Table No. D-16

b TEST

Failure						No Failure		
			Failed Element					
Serial Number	Life in Hours	Order Number	Outer Race	Inner Race	Rollers	Serial Number	Life in Hours	Order Number
208	171	1	x		x			
						48'	171	1
						63'	189	2
212	198	2	x		x			
237	355	3	x		x			
						70'	399	3
						71'	399	4
215	420	4	x	x	x			
						57'	511	5
						73'	546	6
214	616	5	x	x	x			
221	637	6	x	x	x			
223	646	7	x	x	x			
						88'	650	7
238	802	8	x	x	x			
						86'	814	8
						74'	835	9
						87'	857	10
230	869	9			x			
						68'	1031	11
						69'	1031	12
						64'	1034	13
248	1038	10	x					
226	1060	11	x		x			
						61'	1060	14
						62'	1060	15

Table No. D-17

c TEST

Failure						No Failure		
			Failed Element					
Serial Number	Life in Hours	Order Number	Outer Race	Inner Race	Rollers	Serial Number	Life in Hours	Order Number
						78'	355	1
241	590	1	x					
216	791	2	x	x				
						80'	945	2
						81'	945	3
						58'	1077	4
						59'	1077	5
						60'	1077	6
						79'	1231	7

Table No. D-18

d TEST

Failure						No Failure		
			Failed Element					
Serial Number	Life in Hours	Order Number	Outer Race	Inner Race	Rollers	Serial Number	Life in Hours	Order Number
						82'	1026	1
						83'	1026	2
						84'	1026	3
						85'	1026	4

Table No. D-19

e TEST

Failure						No Failure		
			Failed Element					
Serial Number	Life in Hours	Order Number	Outer Race	Inner Race	Rollers	Serial Number	Life in Hours	Order Number
250	305	1	x		x			
251	305	2	x		x			
252	305	3	x	x				
253	305	4	x					

Table No. D-20

a' TEST

Failure						No Failure		
			Failed Element					
Serial Number	Life in Hours	Order Number	Outer Race	Inner Race	Rollers	Serial Number	Life in Hours	Order Number
						77'	112	1
246	324	1	x					
244	357	2	x		x			
247	374	3	x		x			
245	396	4	x					

Table No. D-21

b' TEST

Failure						No Failure		
			Failed Element					
Serial Number	Life in Hours	Order Number	Outer Race	Inner Race	Rollers	Serial Number	Life in Hours	Order Number
						158'	65	1
						254'	125	2
						159'	159	3
430	285	1	x	x	x			
249	432	2	x		x	90'	432	4
						253'	478	5
						91'	484	6
						89'	485	7
340	520	3	x					
						181'	586	8
						211'	739	9
255	832	4		x				
256	832	5	x	x	x			
257	874	6	x					
254	877	7	x					
						251'	987	10
						252'	987	11
403	1027	8		x		196'	1027	12
						187'	1039	13
455	1109	9	x					
						236'	1147	14
370	1159	10	x		x			

Table No. D-22

d' TEST

Failure						No Failure		
			Failed Element					
Serial Number	Life in Hours	Order Number	Outer Race	Inner Race	Rollers	Serial Number	Life in Hours	Order Number
279	2182	1	x			131'	2182	1
						132'	2182	2
						133'	2182	3

Table No. D-23

L1 TEST

Failure						No Failure		
			Failed Element					
Serial Number	Life in Hours	Order Number	Outer Race	Inner Race	Rollers	Serial Number	Life in Hours	Order Number
285	17	1	x			176'	20	1
303	30	2	x		x	157'	23	2
301	43	3	x		x			
317	66	4	x					
318	66	5	x					
322	66	6	x	x				
291	72	7	x		x			
278	73	8	x		x			
290	84	9	x		x			
323	87	10	x					
325	113	11		x		156'	123	3
295	123	12	x					
296	123	13		x				
307	126	14	x	x				
328	130	15	x			173'	133	4
						174'	133	5
326	130	16	x	x		175'	133	6
300	134	17	x			161'	146	7
298	134	18	x			162'	146	8
282	161	19	x		x			
331	210	20	x					
330	210	21	x					
333	212	22	x	x		170'	212	9
309	225	23	x			171'	212	10
286	236	24	x			172'	212	11
314	241	25		x		163'	241	12
						164'	241	13
288	243	26		x				
289	243	27	x		x	143'	243	14
321	332	28	x			149'	310	15
320	332	29	x		x	150'	310	16
316	364	30	x			167'	364	17
						166'	364	18
						165'	364	19

Table No. D-24

L2 TEST

Failure						No Failure		
			Failed Element					
Serial Number	Life in Hours	Order Number	Outer Race	Inner Race	Rollers	Serial Number	Life in Hours	Order Number
353	13	1	x					
334	16	2	x					
339	54	3	x			179'	52	1
338	54	4			x	180'	52	2
343	63	5	x	x				
357	67	6	x					
358	67	7	x			177'	72	3
359	67	8	x			178'	72	4
337	72	9	x					
336	72	10		x	x			
335	84	11	x		x			
346	92	12	x					
345	92	13	x	x				
363	93	14	x					
368	100	15	x		x			
373	108	16	x		x			
341	112	17	x					
367	130	18	x		x			
356	171	19	x	x				
355	171	20	x		x			
388	228	21	x					
387	228	22	x		x			
390	228	23	x					
352	231	24	x		x			
350	231	25	x					
360	248	26	x	x	x	192'	270	5
381	270	27	x					
365	309	28		x	x	188'	450	6
364	309	29		x		189'	450	7
371	379	30	x			202'	454	8
						203'	454	9
414	454	31	x		x			

Table No. D-25

L3 TEST

Failure						No Failure		
			Failed Element					
Serial Number	Life in Hours	Order Number	Outer Race	Inner Race	Rollers	Serial Number	Life in Hours	Order Number
405	1.7	1	x					
410	2.1	2	x					
401	2.6	3	x					
404	20	4	x					
406	20	5	x					
407	20	6	x			201'	21	1
412	21	7	x					
413	21	8	x					
397	27	9	x		x			
396	27	10	x		x			
385	48	11	x					
386	48	12	x					
						209'	111	2
384	57	13	x					
395	90	14	x					
408	104	15	x		x			
409	104	16	x			206'	176	3
402	117	17	x		x	207'	176	4
440	302	18	x	x	x	224'	302	5
						225	302	6

Table No. D-26

L4 TEST

Failure						No Failure		
			Failed Element					
Serial Number	Life in Hours	Order Number	Outer Race	Inner Race	Rollers	Serial Number	Life in Hours	Order Number
416	20	1	x		x			
436	27	2	x					
437	27	3	x					
438	37	4	x					
424	48	5	x					
439	53	6	x					
425	68	7		x	x			
428	81	8	x			210'	81	1
429	81	9	x					
426	82	10	x			219'	164	2
						213'	183	3
434	222	11	x		x			
442	294	12			x			
443	294	13			x			

Table No. D-27

L5 TEST

Failure

Failed Element

Serial Number	Life in Hours	Order Number	Outer Race	Inner Race	Rollers
446	17	1	x	x	
449	63	2	x		
450	63	3	x		
441	70	4	x		x
447	106	5	x		
448	114	6	x		
462	152	7	x		x
454	165	8	x		
461	201	9	x		x
458	285	10	x		x
467	300	11	x		
466	300	12	x		x
463	304	13	x		x
464	304	14		x	

No Failure

Serial Number	Life in Hours	Order Number
228'	63	1
227'	86	2
230'	152	3
243'	195	4
242'	195	5
241'	195	6
232'	285	7
239'	300	8
238'	304	9
237'	304	10

Table No. D-28

E" TEST

Failure						No Failure		
			Failed Element					
Serial Number	Life in Hours	Order Number	Outer Race	Inner Race	Rollers	Serial Number	Life in Hours	Order Number
361	216	1	x					
372	404	2	x					
418	509	3	x		x			
419	596	4	x		x			
348	1267	5	x					
349	1267	6	x					
420	2178	7	x					
417	2178	8	x					

Table No. D-29

F" TEST

Failure						No Failure		
			Failed Element					
Serial Number	Life in Hours	Order Number	Outer Race	Inner Race	Rollers	Serial Number	Life in Hours	Order Number
421	1781	1		x		208'	1781	1
422	1781	2	x					
423	1781	3	x					

Table No. D-30

e" TEST

Failure						No Failure		
			Failed Element					
Serial Number	Life in Hours	Order Number	Outer Race	Inner Race	Rollers	Serial Number	Life in Hours	Order Number
469	1132	1		x		259'	1132	1
						260'	1132	2
						261'	1132	3

Table No. D-31

f" TEST

Failure						No Failure		
			Failed Element					
Serial Number	Life in Hours	Order Number	Outer Race	Inner Race	Rollers	Serial Number	Life in Hours	Order Number
						255'	1111	1
						256'	1111	2
						257'	1111	3
						258'	1111	4

Table No. D-32

EE' TEST

Failure						No Failure		
			Failed Element					
Serial Number	Life in Hours	Order Number	Outer Race	Inner Race	Rollers	Serial Number	Life in Hours	Order Number
284	33 33	1		x				
304	25 50	2	x					
306	53 91	3	x					
294	71 110	4	x	x				
305	115 165	5		x	x			
302	98 211	6	x					

Table No. D-33

FF' TEST

Failure						No Failure		
			Failed Element					
Serial Number	Life in Hours	Order Number	Outer Race	Inner Race	Rollers	Serial Number	Life in Hours	Order Number
398	8 69	1	x			160'	86 31	1
347	127 79	2	x		x	234'	82 82	2
468	121 121	3	x	x	x	244'	121 121	3
						245'	121 121	4
324	181 103	4	x	x	x	246'	121 121	5
377	140 146	5	x					
342	220 228	6		x	x			
374	251 297	7	x	x	x			
376	250 338	8	x					
362	400 396	9		x	x			
375	441 501	10	x	x	x			
456	478 481	11	x	x	x	235'	520 526	6
460	560 563	12	x			233'	560 563	7
						208'	890 891	8

Table No. D-31

TRANSCRIPT FROM TEST LOG

Bearing Failures

Serial Number of Failure	Date Removed From Test	Test Phase Designation	Machine No.	Brg. Location No.	Hours Life
	February 1962				
9	28	B	7	28	114
	March				
10	7	A	5	23	23
11	8	B	1	8	141
12	12	A	5	22	50
13	14	B	1	6	428
14	15	B	7	28	93
15	19	B	1	7	478
16	20	A	5	24	113
17	21	A	5	21	126
18	21	B	7	26	287
19	23	A	5	21	43
20	23	B	1	8	209
21	23	B	7	27	342
22	25	B	1	6	191
23	26	B	1	5	144
24	26	A	5	24	65
25	27	A	5	23	158
26	28	B	4	20	470
27	28	C	6	29	470
28	29	A	5	22	178
29	29	B	7	28	204
30	30	B	4	19	516
31	30	A	5	21	70
	April				
32	2	B	1	8	353
33	2	A	5	24	75
34	3	A	5	23	90
35	3	B	7	25	466
36	3	B	7	26	180
37	6	A	5	23	71
38	9	B	1	8	160
39	9	A	5	21	104
40	9	A	5	24	98
41	9	B	7	25	138

(Continued)

Table No. D-34

TRANSCRIPT FROM TEST LOG

Bearing Failures

Serial Number of Failure	Date Removed From Test	Test Phase Designation	Machine No.	Brg. Location No.	Hours Life
42	9	B	7	27	252
43	9	B	7	28	388
44	11	D	0	1	817
45	11	A	5	21	46
46	11	A	5	23	48
47	11	A	5	24	46
48	12	D	0	2	839
49	12	D	0	3	839
50	12	D	0	4	339
51	12	B	4	19	247
52	13	A	5	22	206
53	13	A	5	23	44
54	13	B	7	25	90
55	16	A	5	22	26
56	16	D	2	13	972
57	17	B	7	27	187
58	18	B	4	17	887
59	19	D	3	9	1039
60	19	D	3	10	1039
61	19	D	3	11	1039
62	19	D	3	12	1039
63	23	B	7	28	236
64	23	B	1	5	487
65	23	B	1	8	246
66	24	E	3	12	19
67	24	B	4	18	914
68	24	B	4	17	151
69	24	A	5	21	137
70	24	A	5	23	93
71	24	A	5	24	137
72	25	E	3	11	41
73	26	B	1	8	47
74	26	E	3	12	45
75	27	E	3	10	82
76	27	A'	5	23	46

(Continued)

Table No. D-34

TRANSCRIPT FROM TEST LOG

Bearing Failures

Serial Number of Failure	Date Removed From Test	Test Phase Designation	Machine No.	Brg. Location No.	Hours Life
77	27	B	7	26	468
78	30	C	6	29	521
79	30	C	6	31	991
80	30	C	6	32	991
	May				
81	1	E	3	9	117
82	2	E	3	10	58
83	2	B	4	18	127
84	2	B	4	19	127
85	2	A'	5	21	105
86	2	E	3	10	22
87	4	C'	6	31	89
88	4	D'	0	3	353
89	7	D'	0	2	425
90	7	E	3	12	128
91	7	A'	5	21	101
92	9	B	1	6	789
93	9	B	1	8	251
94	10	E	3	12	64
95	11	F	2	14	378
96	11	F	2	15	378
97	14	F	2	13	443
98	14	F	2	16	443
99	14	E	3	9	179
100	14	A'	5	21	116
101	14	A'	5	23	217
102	14	A'	5	24	217
103	14	C'	6	31	174
104	15	E	3	12	48
105	15	B	7	26	280
106	17	A'	5	24	68
107	18	B'	1	5	212
108	18	A'	5	21	91
109	21	B'	4	18	448
110	21	B'	4	19	448

(Continued)

Table No. D-34

TRANSCRIPT FROM TEST LOG

Bearing Failures

Serial Number of Failure	Date Removed From Test	Test Phase Designation	Machine No.	Brg. Location No.	Hours Life
111	21	C'	6	32	339
112	22	D'	0	4	740
113	22	E	3	12	98
114	23	E	3	12	18
115	23	A'	5	21	33
116	25	B'	4	20	539
117	25	B'	1	8	317
118	28	B'	1	6	388
119	28	E	3	9	191
120	28	B'	4	18	161
121	28	C'	6	30	568
122	28	B'	7	28	307
123	29	E	3	11	22
124	29	A'	5	22	510
125	29	A'	5	24	119
	June				
126	4	B'	1	8	231
127	4	F	2	16	479
128	4	B'	7	25	459
129	4	B'	7	27	459
130	4	B'	7	28	152
131	5	B'	4	19	325
132	5	C'	6	32	354
133	8	A'	5	24	71
134	11	E	3	12	159
135	13	B'	4	20	252
136	15	F	2	14	715
137	18	B'	1	5	565
138	18	F	2	15	787
139	18	F	2	16	244
140	18	E.	3	9	150

(Continued)

Table No. D-34

TRANSCRIPT FROM TEST LOG

Bearing Failures

Serial Number of Failure	Date Removed From Test	Test Phase Designation	Machine No.	Brg. Location No.	Hours Life
141	18	A'	5	23	327
142	19	E	3	11	141
143	19	A'	5	22	158
144	20	D'	0	2	933
145	21	B'	1	6	448
146	21	B'	1	8	288
147	21	E	3	9	45
148	21	A'	5	23	57
149	21	B'	7	28	272
150	22	E	3	12	112
151	25	F	2	13	868
152	25	B'	4	17	1121
153	26	F	2	15	172
154	26	E	3	11	77
155	26	A'	5	21	314
156	26	A'	5	23	54
157	26	A'	5	24	180
158	27	E	3	10	464
159	28	B'	7	27	445
160	29	E	3	9	98
161	29	B'	4	18	611
	July				
162	9	B'	1	7	154
163	9	E	3	11	67
164	10	E	3	12	119
165	10	C'	6	29	218
166	10	B'	7	28	323
167	11	E	3	11	31
168	11	B'	4	20	398
169	11	A'	5	22	68
170	12	A'	5	23	74
171	16	B'	1	6	455
172	16	F	2	16	489
173	16	E	3	10	120
174	16	E	3	12	69
175	16	A'	5	21	106
176	18	B'	4	18	206

(Continued)

Table No. D-34

TRANSCRIPT FROM TEST LOG

Bearing Failures

Serial Number of Failure	Date Removed From Test	Test Phase Designation	Machine No.	Brg. Location No.	Hours Life
141	18	A'	5	23	327
142	19	E	3	11	141
143	19	A'	5	22	158
144	20	D'	0	2	933
145	21	B'	1	6	448
146	21	B'	1	8	288
147	21	E	3	9	45
148	21	A'	5	23	57
149	21	B'	7	28	272
150	22	E	3	12	112
151	25	F	2	13	868
152	25	B'	4	17	1121
153	26	F	2	15	172
154	26	E	3	11	77
155	26	A'	5	21	314
156	26	A'	5	23	54
157	26	A'	5	24	180
158	27	E	3	10	464
159	28	B'	7	27	445
160	29	E	3	9	98
161	29	B'	4	18	611
	July				
162	9	B'	1	7	154
163	9	E	3	11	67
164	10	E	3	12	119
165	10	C'	6	29	218
166	10	B'	7	28	323
167	11	E	3	11	31
168	11	B'	4	20	398
169	11	A'	5	22	68
170	12	A'	5	23	74
171	16	B'	1	6	455
172	16	F	2	16	489
173	16	E	3	10	120
174	16	E	3	12	69
175	16	A'	5	21	106
176	18	B'	4	18	206

(Continued)

Table No. D-34

TRANSCRIPT FROM TEST LOG

Bearing Failures

Serial Number of Failure	Date Removed From Test	Test Phase Designation	Machine No.	Brg. Location No.	Hours Life
177	18	A'	5	24	152
178	18	B'	7	25	772
179	19	E	3	9	149
180	19	E	3	11	88
181	19	A'	5	21	45
182	19	A'	5	22	107
183	24	D'	0	4	601
184	24	B'	1	6	191
185	24	F	2	14	749
186	24	F	2	15	507
187	24	F	2	16	190
188	24	A'	5	23	128
189	25	A'	5	22	53
190	23	B'	7	25	162
191	26	C'	6	29	382
192	26	C'	6	32	945
193	27	A'	5	22	48
194	27	A'	5	23	75
195	31	F'	2	16	161
196	31	E'	3	12	138
197	31	B'	4	18	314
198	31	B'	4	20	480
	August				
199	2	a	5	23	69
200	2	B'	1	8	866
201	6	F'	2	14	272
202	6	E'	3	9	281
203	6	a	5	21	133
204	6	a	5	22	130
205	9	a	5	23	108
206	10	B'	7	25	383
207	10	B'	7	27	872
208	13	b	1	8	171

(Continued)

Table No. D-34

Bearing Failures

Serial Number of Failure	Date Removed From Test	Test Phase Designation	Machine No.	Brg. Location No.	Hours Life
209	14	E'	3	10	373
210	14	E'	3	11	373
211	14	E'	3	12	236
212	14	b	1	5	198
213	21	D'	0	4	644
214	27	b	1	7	420
215	27	b	4	20	615
216	29	c	6	30	790
	September				
217	4	a	5	21	338
218	4	a	5	24	473
219	5	F'	2	14	595
220	6	a	5	21	45
221	11	b	7	26	637
222	14	a	5	22	471
223	14	b	7	27	646
224	24	a	5	24	193
225	25	E'	2	13	187
226	25	b	4	17	1060
227	26	E'	2	14	207
228	26	a	5	21	191
229	27	a	5	23	250
230	28	b	7	25	869
	October				
231	1	E'	2	15	269
232	2	a	5	23	52
233	5	a	5	22	216
234	8	E'	2	13	154
235	9	F'	3	9	104
236	9	a	5	21	140
237	10	b	4	17	354
238	10	b	4	20	801
239	11	a	5	21	44
240	12	a	5	23	139

(Continued)

Table No. D-34

Bearing Failures

Serial Number of Failure	Date Removed From Test	Test Phase Designation	Machine No.	Brg. Location No.	Hours Life
241	15	c	6	29	589
242	22	E'	2	13	166
243	22	E'	2	14	301
244	23	a'	5	22	357
245	29	a'	5	21	396
246	29	a'	5	23	374
247	29	a'	5	24	374
	November				
248	2	b	7	27	1037
249	5	b'	1	5	432
250	7	e	2	13	305
251	7	e	2	14	305
252	7	e	2	15	305
253	7	e	2	16	305
254	16	b'	4	17	877
255	16	b'	4	18	832
256	16	b'	4	19	832
257	16	b'	4	20	874
258	26	G	1	7	390
259	26	H	6	30	577
260	26	G	7	27	526
261	27	F'	3	10	1112
	December				
262	10	G	5	23	665
263	17	G	7	25	1026
264	17	G	7	28	1026
265	18	H	2	15	872
266	26	G	4	20	794
267	31	G	5	24	1052
	January				
268	9	G	4	17	1022
269	14	G	1	7	1013
270	30	H	2	15	571
271	31	G	5	21	903

(Continued)

Table No. D-34

Bearing Failures

Serial Number of Failure	Date Removed From Test	Test Phase Designation	Machine No.	Brg. Location No.	Hours Life	
	February 1963					
272	1	G	1	6	718	
273	6	G	4	17	648	
274	6	G	5	24	825	
275	6	H	6	29	1049	
276	11	H	2	16	636	
277	11	G	7	26	928	
278	21	L_1	4	18	73	
279	22	d'	0	3	2182	
280	25	E'	0	1	77	
281	25	E'	0	3	77	
282	26	L_1	1	6	161	
283	26	E	4	18	96	
	March					
284	1	E-E'	0	3	33	33
285	1	L_1	2	13	17	
286	1	L_1	4	17	236	
287	4	E'	0	3	69	
288	4	L_1	1	7	243	
289	4	L_1	1	8	243	
290	4	L_1	2	15	84	
291	4	L_1	3	9	72	
292	4	H	6	31	506	
293	5	E	4	17	74	
294	6	E-E'	0	2	71	110
295	6	L_1	3	11	123	
296	6	L_1	3	12	123	
297	7	E	3	13	117	
298	7	L_1	2	14	134	
299	7	E	2	15	50	
300	7	L_1	2	16	134	
301	7	L_1	4	19	43	
302	11	E-E'	0	4	98	211
303	11	L_1	2	16	30	

Table No. D-34

Bearing Failures

Serial Number of Failure	Date Removed From Test	Test Phase Designation	Machine No.	Brg. Location No.	Hours Life	
304	12	E-E'	0	2	25	50
305	13	E-E'	0	1	115	165
306	13	E-E'	0	3	53	91
307	18	L_1	2	15	126	
308	18	E	2	16	96	
309	18	L_1	3	11	225	
310	18	E	4	17	217	
311	19	DR	2	15	20	
312	19	DR	2	16	20	
313	19	DR	3	11	16	
314	19	L_1	3	12	241	
315	20	DR	4	17	51	
316	22	L1	1	6	364	
317	22	L1	2	13	66	
318	22	L1	2	16	66	
319	22	DR	4	17	41	
320	22	L1	4	18	332	
321	22	L1	4	20	332	
322	25	L1	1	5	66	
323	26	L1	1	7	87	
324	27	FF'	0	1	181	103
325	27	L1	4	20	113	
326	28	L1	1	6	130	
327	28	DR	1	7	43	
328	28	L1	1	8	130	
329	28	DR	2	13	143	
330	28	L1	2	14	210	
331	28	L1	2	15	210	
332	28	DR	2	16	143	
333	28	L1	3	10	212	
334	29	L2	4	19	16	
335	April 1	L2	1	7	84	
336	1	L2	2	13	72	
337	1	L2	2	16	72	
338	1	L2	3	9	54	

Table No. D-34

Bearing Failures

Serial Number of Failure	Date Removed From Test	Test Phase Designation	Machine No.	Brg. Location No.	Hours Life	
339	1	L2	3	11	54	
340	1	b'	5	21	520	
341	2	L2	1	6	112	
342	3	FF'	0	3	220	228
343	3	L2	4	17	62	
344	5	DR	1	6	64	
345	5	L2	3	9	92	
346	5	L2	3	11	92	
347	8	FF'	0	1	127	77
348	9	E"	7	26	1267	
349	9	E"	7	27	1267	
350	10	L2	1	5	231	
351	10	DR	1	7	146	
352	10	L2	1	8	231	
353	11	L2	1	7	13	
354	15	DR	3	9	78	
355	15	L2	3	10	171	
356	15	L2	3	12	171	
357	17	L2	1	5	67	
358	17	L2	1	6	67	
359	17	L2	1	8	67	
360	18	L2	2	16	248	
361	18	E"	7	27	216	
362	19	FF'	0	4	400	396
363	19	L2	3	9	93	
364	22	L2	4	18	309	
365	22	L2	4	20	309	
366	23	DR	2	16	51	
367	23	L2	3	10	130	
368	24	L2	1	8	100	
369	25	DR	3	9	77	
370	25	b'	5	22	1159	
371	29	L2	2	13	379	
372	29	E"	7	26	404	
373	30	L2	4	19	108	

Table No. D-34

Bearing Failures

Serial Number of Failure	Date Removed From Test	Test Phase Designation	Machine No.	Brg. Location	Hours Life	
	May					
374	1	FF'	0	1	251	297
375	1	FF'	0	2	441	501
376	1	FF'	0	3	250	338
377	1	FF'	0	4	140	146
378	2	FC	0	3	5	
379	2	DR	2	13	66	
380	2	DR	3	10	140	
381	2	L2	3	12	270	
382	7	FC	0	1	35	
383	8	FC	0	1	2	
384	8	L3	2	13	57	
385	8	L3	3	9	48	
386	8	L3	3	12	48	
387	8	L2	4	17	228	
388	8	L2	4	18	228	
389	8	DR	4	19	121	
390	8	L2	4	20	228	
391	9	FC	0	1	1	
392	9	FC	0	2	38	
393	9	FC	0	3	33	
394	9	FC	0	4	38	
395	10	L3	2	14	90	
396	10	L3	4	17	27	
397	10	L3	4	18	27	
398	13	FF'	0	4	8	69
399	14	E	2	14	28	
400	15	E	2	13	65	
401	15	L3	3	9	2.6	
402	15	L3	3	10	117.	
403	15	b'	5	24	1027	

Table No. D-34

Bearing Failures

Serial Number of Failure	Date Removed From Test	Test Phase Designation	Machine No.	Brg. Location	Hours Life
404	16	L3	3	10	20
405	16	L3	3	10	1.7
406	16	L3	3	11	20
407	16	L3	3	12	20
408	16	L3	4	19	104
409	16	L3	4	20	104
410	16	L3	4	18	2.1
411	17	E	2	14	48
412	17	L3	3	9	21
413	17	L3	3	11	21
414	20	L2	1	5	454
415	20	E	2	13	54
416	21	L4	3	11	20
417	21	E"	7	25	2178
418	21	E"	7	26	509
419	21	E"	7	27	596
420	21	E"	7	28	2178
421	22	F"	6	30	1781
422	22	F"	6	31	1781
423	22	F"	6	32	1781
424	22	L4	3	9	48
425	23	L4	3	10	68
426	24	L4	2	13	82
427	24	E	3	9	40
428	27	L4	1	5	81
429	27	L4	1	7	81
430	27	b'	5	21	285
431	27	E	3	10	32
432	31	E	3	9	69
433	31	E	3	11	137

Table No. D-34

Bearing Failures

Serial Number of Failure	Date Removed From Test	Test Phase Designation	Machine No.	Brg. Location	Hours Life	
	June					
434	4	L4	2	16	222	
435	4	E	4	17	123	
436	5	L4	1	5	27	
437	5	L4	1	7	27	
438	6	L4	1	8	37	
439	7	L4	1	6	53	
440	7	L3	4	19	302	
441	10	L5	1	5	70	
442	10	L4	2	14	294	
443	10	L4	2	15	294	
444	10	E	2	16	73	
445	11	E	1	5	16	
446	11	L5	2	15	17	
447	12	L5	1	8	106	
448	12	L5	4	17	114	
449	13	L5	2	14	63	
450	13	L5	2	16	63	
451	13	E	4	17	20	
452	17	E	1	5	66	
453	17	E	1	6	66	
454	17	L5	4	19	165	
455	17	b'	5	23	1109	
456	19	FF'	0	3	478	481
457	25	E	4	17	151	
458	25	L5	4	18	285	
459	25	E	4	19	120	
460	26	FF'	0	2	560	563
461	July 1	L5	2	13	201	
462	8	L5	4	17	152	
463	10	L5	1	7	304	

Table No. D-34

Bearing Failures

Serial Number of Failure	Date Removed From Test	Test Phase Designation	Machine No.	Brg. Location	Hours Life	
464	10	L5	1	8	304	
465	10	E	2	13	99	
466	10	L5	2	14	300	
467	10	L5	2	16	300	
468	11	FF'	0	3	121	121
469	12	e"	7	25	1132	

Table No. D-35

TRANSCRIPT FROM TEST LOG

Bearings Removed Before Failure

Serial Number of Removal	Date Removed	Test Phase Designation	Machine No.	Brg. Location No.	Hours of Running
	March 1962				
2'	2	B	5	21	134
3'	2	B	5	22	134
4'	2	B	5	23	134
5'	2	B	5	24	134
6'	19	B	1	5	478
	April				
7'	17	D	2	14	972
8'	17	D	2	15	972
9'	17	D	2	16	972
10'	24	A	5	22	67
11'	30	C	6	30	991
	May				
12'	2	B	4	17	155
13'	2	B	4	20	572
14'	2	A'	5	23	59
15'	2	A'	5	24	105
16'	9	B	1	5	298
17'	9	B	1	7	930
18'	14	E	3	10	124
19'	15	B	7	25	519
20'	15	B	7	27	423
21'	15	B	7	28	373
22'	28	E	3	11	441
	June				
23'	20	D'	0	1	1380
24'	20	D'	0	3	1105
25'	20	D'	0	4	617
26'	25	C'	6	29	1098
27'	26	A'	5	22	94
28'	27	B'	1	7	977
	July				
29'	10	C'	6	31	1072
30'	10	B'	7	26	1054
31'	12	D'	0	2	343
32'	17	A'	5	21	24
33'	19	E	3	10	49

(Continued)

Table No. D-35

TRANSCRIPT FROM TEST LOG

Bearings Removed Before Failure

Serial Number of Removal	Date Removed	Test Phase Designation	Machine No.	Brg. Location No.	Hours of Running
29'	10	C'	6	31	1072
30'	10	B'	7	26	1054
31'	12	D'	0	2	343
32'	17	A'	5	21	24
33'	19	E	3	10	49
34'	19	E	3	12	49
35'	24	F	2	13	532
36'	26	C'	6	30	1103
37'	26	C'	6	30	27
38'	26	C'	6	31	382
39'	27	A'	5	21	101
40'	27	A'	5	24	124
41'	31	B'	4	17	619
42'	31	B'	4	19	958
	August				
43'	2	B'	1	5	915
44'	2	B'	1	6	220
45'	2	B'	1	7	571
46'	10	B'	7	26	721
47'	10	B'	7	28	721
49'	14	E'	3	9	93
50'	21	D'	0	1	1245
51'	21	D'	0	2	902
52'	21	D'	0	3	1245
	September				
54'	5	F'	2	13	867
55'	5	F'	2	15	867
56'	5	F'	2	16	705
57'	19	b	1	5	511
58'	20	c	6	29	1077
59'	20	c	6	31	1077
60'	20	c	6	32	1077
61'	25	b	4	18	1060
62'	25	b	4	19	1060
63'	26	b	7	26	189

(Continued)

Table No. D-35

TRANSCRIPT FROM TEST LOG

Bearings Removed Before Failure

Serial Number of Removal	Date Removed	Test Phase Designation	Machine No.	Brg. Location No.	Hours of Running
	October 1962				
64'	5	b	7	28	1034
65'	9	F'	3	10	1035
66'	9	F'	3	11	1035
67'	9	F'	3	12	1035
68'	10	b	1	6	1031
69'	10	b	1	8	1031
70'	12	b	4	18	399
71'	12	b	4	19	399
72'	12	a	5	24	248
73'	15	b	1	5	546
74'	15	b	1	7	835
75'	22	E'	2	15	239
76'	22	E'	2	16	508
77'	29	a'	5	22	112
78'	30	c	6	29	355
79'	30	c	6	30	1231
80'	30	c	6	31	944
81'	30	c	6	32	944
82'	31	d	0	1	1026
83'	31	d	0	2	1026
84'	31	d	0	3	1026
85'	31	d	0	4	1026
	November				
86'	2	b	7	25	814
87'	2	b	7	26	857
88'	2	b	7	28	650
89'	5	b'	1	6	484
90'	5	b'	1	7	432
91'	5	b'	1	8	484
92'	26	H	2	13	343
93'	27	F'	3	9	1112
94'	27	F'	3	11	1112
95'	27	F'	3	12	1112

(Continued)

Table No. D-35

TRANSCRIPT FROM TEST LOG

Bearings Removed Before Failure

Serial Number of Removal	Date Removed	Test Phase Designation	Machine No.	Brg. Location No.	Hours of Running
	December 1962				
96'	17	H	6	29	1076
97'	17	H	6	31	1076
98'	17	H	6	32	1076
99'	17	G	7	26	1026
100'	18	G	5	21	854
101'	31	G	1	5	1115
102'	31	G	1	6	1115
103'	31	G	1	8	1115
104'	31	H	2	14	1066
105'	31	H	2	16	1066
106'	31	G	5	22	1052
	January 1963				
107'	9	G	4	18	1022
108'	9	G	4	19	1022
109'	14	H	2	13	1015
110'	14	H	3	9	999
111'	14	H	3	10	999
112'	14	H	3	11	999
113'	14	H	3	12	999
114'	14	H	6	30	996
115'	14	G	7	27	997
116'	31	G	5	23	1093
	February				
117'	6	H	6	31	1049
118'	6	H	6	32	1049
119'	11	G	7	25	928
120'	11	G	7	27	431
121'	11	G	7	28	928
122'	15	G	4	17	214
123'	15	G	4	18	862
124'	15	G	4	19	862
125'	15	G	4	20	1089
126'	18	G	1	5	1059

(Continued)

Table No. D- 35

TRANSCRIPT FROM TEST LOG

Bearings Removed Before Failure

Serial Number of Removal	Date Removed	Test Phase Designation	Machine No.	Brg. Location No.	Hours of Running
127'	18	G	1	6	341
128'	18	G	1	7	771
129'	18	G	1	8	1059
130'	20	G	5	22	1139
131'	22	d'	0	1	2182
132'	22	d'	0	2	2182
133'	22	d'	0	4	2182
134'	25	E'	0	2	77
135'	27	H	2	13	709
136'	27	H	2	14	1000
137'	27	H	2	15	623
138'	27	H	2	16	364
139'	27	H	3	9	1004
140'	27	H	3	10	1004
141'	27	H	3	11	1004
142'	27	H	3	12	1004
	March				
143'	4	L_1	1	5	243
144'	4	E	1	6	84
145'	4	G	6	29	506
146'	4	G	6	30	1059
147'	4	G	6	32	506
148'	5	E	4	18	140
149'	5	L_1	4	19	310
150'	5	L_1	4	20	310
151'	5	G	5	21	745
152'	5	G	5	22	312
153'	5	G	5	23	745
154'	5	G	5	24	636
155'	6	E	3	9	50
156'	6	L_1	3	10	123
157'	6	L_1	4	17	23

Table No. D-35

TRANSCRIPT FROM TEST LOG

Bearings Removed Before Failure

Serial Number of Removal	Date Removed	Test Phase Designation	Machine No.	Brg. Location No.	Hours of Running
158'	8	b'	5	21	65
159'	12	b'	5	23	159
160'	19	F-F'	0	2	86 31
161'	19	L_1	2	13	146
162'	19	L_1	2	14	146
163'	19	L_1	3	9	241
164'	19	L_1	3	10	241
165'	22	L_1	1	5	364
166'	22	L_1	1	7	364
167'	22	L_1	1	8	364
168'	22	E	4	19	289
169'	28	DR	1	5	64
170'	28	L1	3	9	212
171'	28	L1	3	11	212
172'	28	L1	3	12	212
173'	28	L1	4	17	133
174'	28	L1	4	18	133
175'	28	L1	4	19	133
176'	28	L1	4	20	20
177' April	1	L2	2	14	72
178'	1	L2	2	15	72
179'	1	L2	3	10	54
180'	1	L2	3	12	54
181'	1	b'	5	24	586
182'	10	DR	1	6	55
183'	15	DR	3	11	78
184'	17	DR	1	7	54
185'	22	DR	4	17	246
186'	22	DR	4	19	293
187' May	1	b'	5	23	1039
188'	2	L2	2	14	450
189'	2	L2	2	15	450
190'	2	DR	2	16	141
191'	2	DR	3	9	99
192'	2	L2	3	11	270

Table No. D-35

TRANSCRIPT FROM TEST LOG

Bearings Removed Before Failure

Serial Number of Removal	Date Removed	Test Phase Designation	Machine No.	Brg. Location No.	Hours of Running
193'	15	E	3	9	69
194'	15	E	3	11	117
195'	15	E	3	12	69
196'	15	b'	5	21	1027
197'	16	E	3	9	17
198'	16	E	4	17	77
199'	16	E	4	18	77
200'	17	E	3	10	20
201'	17	L3	3	12	21
202'	20	L2	1	6	454
203'	20	L2	1	7	454
204'	20	DR	1	8	354
205'	20	E	2	14	10
206'	20	L3	2	15	176
207'	20	L3	2	16	176
208'	22	FF'	6	29	890 891
209'	24	L3	4	17	111
210'	27	L4	1	8	81
211'	27	b'	5	22	739
212' June	3	E	1	5	102
213'	3	L4	1	6	183
214'	3	E	1	7	102
215'	3	E	1	8	102
216'	3	E	3	9	6
217'	3	E	3	10	75
218'	3	E	3	11	6
219'	3	L4	3	12	164
220'	7	E	1	5	26
221'	7	E	1	7	26
222'	7	E	1	8	16

Table No. D-35

TRANSCRIPT FROM TEST LOG

Bearings Removed Before Failure

Serial Number of Removal	Date Removed	Test Phase Designation	Machine No.	Brg. Location No.	Hours of Running	
223'	7	E	4	17	68	
224'	7	L3	4	18	302	
225'	7	L3	4	19	302	
226'	10	E	2	13	213	
227'	11	L5	1	6	86	
228'	13	L5	2	13	63	
229'	13	E	2	15	46	
230'	17	L5	1	7	152	
231'	17	E	1	8	46	
232'	25	L5	4	20	285	
233'	26	FF'	0	1	560	563
234'	26	FF'	0	3	82	82[illegible]
235'	26	FF'	0	4	520	
236'	July 2	b'	5	24	1147	
237'	10	L5	1	5	304	
238'	10	L5	1	6	304	
239'	10	L5	2	15	300	
240'	10	E	4	17	43	
241'	10	L5	4	18	195	
242'	10	L5	4	19	195	
243'	10	L5	4	20	195	
244'	11	FF'	0	1	121	121
245'	11	FF'	0	2	121	121
246'	11	FF'	0	4	121	121
247'	11	F	3	9	803	
248'	11	F	3	10	803	
249'	11	F	3	11	803	
250'	11	F	3	12	803	
251'	12	b'	5	21	987	
252'	12	b'	5	22	987	

Table No. D-35

TRANSCRIPT FROM TEST LOG

Bearings Removed Before Failure

Serial Number of Removal	Date Removed	Test Phase Designation	Machine No.	Brg. Location No.	Hours of Running
253'	12	b'	5	23	478
254'	12	b'	5	24	125
255'	12	f"	6	29	1111
256'	12	f"	6	30	1111
257'	12	f"	6	31	1111
258'	12	f"	6	32	1111
259'	12	e"	7	26	1132
260'	12	e"	7	27	1132
261'	12	e"	7	28	1132

Appendix E

Sample Calculation

Using data from the F Test, Appendix Table D-6, the following tabulation is made

H	N_B	F	N_E	R	N_E/N_B	S_T	F_T
0	19					100.0	0
172	19	1	18		0.947	94.7	5.3
190	18	1	17		0.944	89.5	10.5
244	17	1	16		0.941	84.2	15.8
378	16	2	14		0.875	73.7	26.3
443	14	2	12		0.857	63.2	36.8
479	12	1	11		0.917	58.0	42.0
489	11	1	10		0.909	52.7	47.3
507	10	1	9		0.900	47.4	52.6
715	9	1	8	1	0.889	42.1	57.9
749	7	1	6		0.857	36.1	63.9
787	6	1	5		0.833	30.1	69.9
868	5	1	4	4	0.800	24.1	75.9

H — For an incremental time period ending at _____ hours.

N_B — This many bearings started this period.

F — This many bearings failed by the end of this period.

N_E — This many bearings were unfailed at the end of this period.

R — This many unfailed bearings were removed at the end of this period.

N_E/N_B — Ratio of number of survivors vs. number starting the period.

S_T — Proportion of total number that survived to the end of this period (Cumulative product of survivor ratio expressed in percent).

F_T — Proportion of total number that failed by the end of this period. (equal to $100\text{-}S_T$, expressed in percent).

We desire to calculate the best fitting straight line (see Fig. E-1) that represents the first portion of the data. This line can be expressed in the form $Y = a + b\ X$ ($H = a + b\ F_T$). The conventional least squares technique is used to determine the coefficients a and b as follows:

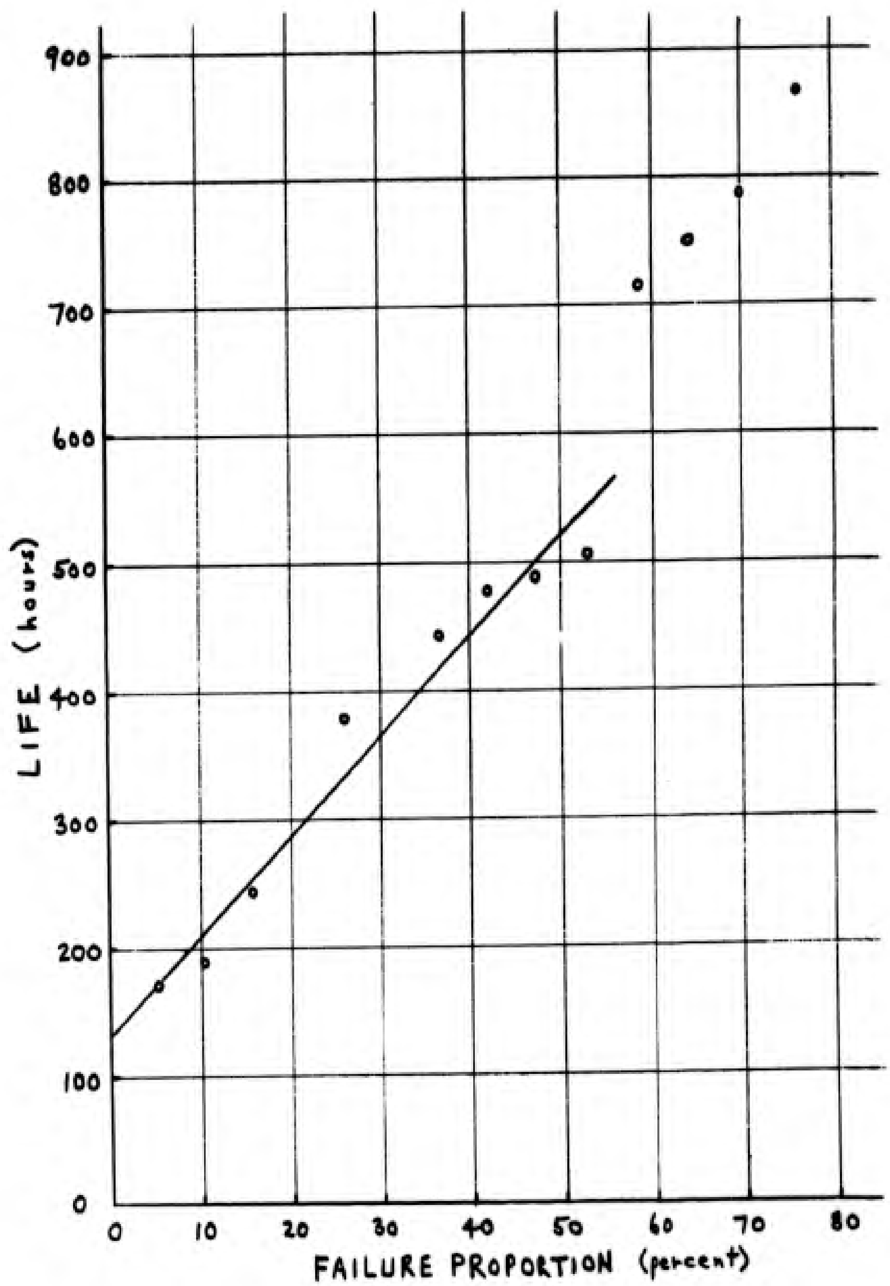


Fig. E-1 - PLOT OF F TEST DATA

X	Y	XY	X^2
5.3	172	911.6	28.09
10.5	190	1995.0	110.25
15.8	244	3855.2	249.64
26.3	378	9941.4	691.69
36.8	443	16302.4	1354.24
42.0	479	20118.0	1764.00
47.3	489	23129.7	2237.29
52.6	507	26668.2	2766.76
236.6	2902	102921.5	9201.96

$$Na + (\Sigma X)b = \Sigma Y$$
$$(\Sigma X)a + (\Sigma X^2)b = \Sigma XY$$

$$D = \begin{vmatrix} 8 & 236.6 \\ 236.6 & 9201.96 \end{vmatrix} = \begin{array}{r} 73,615.68 \\ \underline{55,979.56} \\ 17,636.12 \end{array}$$

$$a = \begin{vmatrix} 2902 & 236.6 \\ 102921.5 & 9201.96 \end{vmatrix} \div D$$

$$= \begin{array}{r} 26,704,087.92 \\ \underline{24,351,226.90} \\ 2,352,861.02 \end{array}$$

$$\frac{2,352,861.02}{17,636.12} = 133.4$$

$$b = \begin{vmatrix} 8 & 2902 \\ 236.6 & 102921.5 \end{vmatrix} \div D$$

$$= \begin{array}{r} 823,372.0 \\ \underline{686,613.2} \\ 136,758.8 \end{array}$$

$$\frac{136,758.8}{17,636.12} = 7.754$$

Thus $H = 7.754\ F_T + 133.4$ is the straight line functional relationship that best represents the data through the 50% failure point. The data beyond that point can be represented by another function (very often a straight line) using the same least squares technique.

Carrying on with the calculations concerning Test F, we use the following relationships:

$$H\ (\text{hours}) = F(\%)\ b + a$$

$$L\ (\text{Millions of cycles}) = \frac{60H\ N\ (\text{cpm})}{10^6}$$

$$C = P\ (\text{lb}) \sqrt[3]{L}$$

Then the B-10 Life and the B-10 Load Rating Factor for Test F (N = 520 cpm, P = 14,000 lb) are evaluated as follows:

$$H = (10)\ 7.75 + 133.4 = 210.9 \text{ hours}$$

$$L = \frac{210.9\ (60)\ 520}{10^6} = 6.58 \text{ million cycles}$$

$$C_1 = 14{,}000 \sqrt[3]{6.58} = 26{,}180$$

$$C_{15} = 14{,}000 \sqrt[3]{\frac{6.58}{15}} = 10{,}615$$

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