RE-ORDER NO. 65-733

October 1965

Tape Recorder - Reproducer Mechanism

Worst Case Analysis

Final Report

Prepared for

Jet Propulsion Laboratory
4800 Oak Grove Drive
Pasadena, California

Contract No. 950903

under Contract NASA 7-100

By

John Lueder, Project Manager

Co-Authors
Irving Karsh
Graham Barr
Robert Bruns

John R. Lueder
Project Manager

D F Burdorf
Director of Engineering

Serial No. ____

k

29 SO. PASADENA AVE., PASADENA, CALIFORNIA

TABLE OF CONTENTS

Section	Title	Page No.
1	STATEMENT OF PURPOSE	1
2	REFERENCE	1
3	SUMMARY	1-5
4	CONCLUSIONS and RECOMMENDATIONS	6-10
5	TECHNICAL DISCUSSION	11-40
6	CALCULATIONS and TABULATIONS	41-90
7	APPENDICES	
	1. Notes on Recorder Failure Problems	91-94
	2. Tape	95
	3. Start Time vs. Tape Speed	96-97
Bibliography		98-99

FINAL REPORT
ON
TAPE RECORDER-REPRODUCER MECHANISM
WORST--CASE ANALYSIS
JET PROPULSION LABORATORY

1.0 STATEMENT OF PROGRAM PURPOSE

Worst-case analysis as a useful tool in the design and functional specification preparation of a tape recorder-reproducer mechanism.

2.0 REFERENCES

Contract No. 950903

3.0 SUMMARY

This report covers the work done on Contract No. 950903, Worst-Case Analysis. This work was performed by personnel of Kinelogic Corporation.

The initial goals of the program were: (1) The development of a tool for analysis of a magnetic tape drive system which incorporates the use of an analog computer, and (2) a systematic study of the reliability of a tape recorder which has as its object the pin-pointing of those components which are more important to the reliability of this device in space missions. While the emphasis placed on each separate facet changed from time to time as the work progressed, these remained the major goals throughout the program. An operational recorder (Rocket Radar Recorder JPL Contract 951096) was used as the basis for this study program.

The work performed to attain these goals may be briefly described as follows:

k 29 SO. PASADENA AVE., PASADENA, CALIFORNIA

3.1 Mathematical Model and Flutter Dynamic Analysis

Early in the program major emphasis was placed on developing a mathematical model of the Iso-Elastic tape drive system which would be suitable for programming on an analog computer. By making several simplifying assumptions, models capable of simulation on the computer were obtained. While this line of analysis gave a greater insight into the mechanics of the Iso-Elastic drive, it had only theoretical results in terms of the dynamic behavior of the system. After some time had been spent on this approach it was decided that while a computerized tool for complete analysis of a tape drive system was a practical possibility, it required more effort than had been anticipated. As a consequence this phase of the program was de-emphasized in favor of the reliability study.

3.2 Test Failure Reports and Failure Mode Analysis

All the available data from in-house testing of the three units built were analyzed to determine the failures experienced and the corrective action taken. In the body of the report this data is tabulated and a discussion of all possible failure modes is presented.

3.3 Service Load Analysis Projected to Give Worst-Case Loads

The Iso-Elastic drive belt and the recording tape have varying degrees of tension during the transport cycle. The direction of these tension forces varies with the amount of tape transferred from one reel to the other. These loads have been analyzed to determine normal service conditions for the bearings and to allow the belt life test data to be applied to this recorder.

To permit evaluation of the mechanical components a set of Worst-Case Loads was generated. The governing factor here was taken to be the yield stress of the Mylar tape and belt material. It was assumed that the maximum possible loads would correspond to this condition.

3.4 Load Analysis Applied to Strength of Mechanical Elements

Using the Worst-Case Loads each of the mechanical elements is analyzed to determine the highest stress point. Where applicable, factors are used to modify the loading to account for environmental stresses. These stresses are then compared with standard handbook values of limiting yield stress for the materials in question. The relationship between the worst case stress and the endurance limit stress is also determined. The expected statistical scatter of these stress values has been estimated and used to demonstrate that all of the mechanical elements are designed with adequate safety margins.

3.5 Reliability, Life Analysis

In considering the reliability of the recorder system, it is possible to calculate the statistical reliability at any given lifetime of the individual components under normal operating conditions, and combine this with an assumption about the dependence of the system on each component to obtain a general figure of merit for a particular system.

This involves calculation of survival probabilities which would become extremely complicated if operating tolerances were included. With this in mind, the calculations are made assuming nominal values of size, load and speed for belts and bearings, under the assumption that these elements are the least reliable of the critical elements and are the controlling factors in recorder life -- after the correction of juvenile failures.

A second approach is to analyze individual components and their associated operating tolerances for a disabling build-up of tolerances. This includes manufacturing dimensional tolerances, differential thermal expansion, temperature variation in belt or bearing pre-load, and change of elastic modulus with temperature.

3.6 Mean Time Before Failure

In the RRR system we make the assumption that failure of any component will result in a system failure. Using this assumption and the reliability figures for individual components generated in the previous section, we compute the reliability of the entire system as the product of the individual reliabilities. The 50% reliability figure is generally used as a figure of merit. (Mean Time Before Failure).

3.7 Bearing Study

A general study of precision ball bearings was made with the aim of determining the criteria for reliable bearing performance in tape transports. This included manufacturing, testing, and installation procedures. Information was accumulated from reference works, personal communications and personal experience. This information is not an all-inclusive study of ball bearings but includes the information of a non-proprietary type available from and for the bearing industry and consumer.

3.8 The natural rotational and mechanical frequencies of the elements of the transport are calculated with the aim of discovering resonant frequencies in the environmental range and eliminates frequency coupled systems. This is valuable in evaluating system performance and preventing damage in a vibration environment.

4.0 CONCLUSIONS & RECOMMENDATIONS

The following conclusions and recommendations are made with the knowledge that the RRR recorder represents one design approach and one set of design compromises. The statements are made with the intent of being generally applicable to Iso-Elastic drive transports.

4.1 Mathematical Model and Flutter Dynamic Analysis

The work done on this study shows that it is feasible to construct a simple mathematical model of an Iso-Elastic tape drive and perform analog simulations with tape speed as the read-out parameter. For the sake of simplifying the simulation some assumptions are necessary. The effect of these assumptions must be evaluated by proceeding to the more complex analog or comparing the predictions with actual machine operation. The confirmed analog would be useful in design evaluation where the relative effect of configurations and environment disturbances could be determined.

4.2 Test Failure Reports

The initial tabulation and evaluation of the failure modes does pin-point the more important functional defects which may occur in the tape recorder mechanism as determined from engineering experience or confidence. Study of these areas would either result in increased confidence in their reliability or in strengthening the weak points.

Where the actual failure data from the testing period at Kinelogic was tabulated, information of a more empirical nature was obtained. This was so despite the sketchy nature of the records kept on this test operation. This data pointed out the necessity of an extensive test program to eliminate juvenile failure mechanisms and indicated the critical nature of the belts and bearings. The difficulty in analyzing the data indicates the need for proper documentation of test work. A report with the information in Table E would be a minimum and any other special test data would be added to this.

4.3 Load Analysis - No conclusions

4.4 Load Analysis of Mechanical Elements

Even the high values of applied load corresponding to yield stress in the belts and tape did not produce stresses approaching the accepted values for tape 303 stainless steel. The comparison with the Precision Elastic Limit (Ref. 19), which corresponds to a yield of 1 micro inch per inch as compared to 2000 micro inches for the normal 0.2% yield strength, also shows an acceptable safety margin.

The Precision Elastic Limit is a more meaningful criteria than the 0.2% yield strength where dimensional stability is important. It is recommended that it should be used as design standard and perhaps as a guide in material selection and choice of heat treatment, stress relieving and machining procedures for the more critical recorder components.

4.5. Reliability, Life Analysis

4.5.1. Statistical Reliability

As belts and bearings are critical to record-life and least reliable, careful consideration must be given to the load-speed or stress capacity of belts and bearings with respect to the desired operating lifetime or reliability. As in RRR it is possible to incorporate one or more weak elements which will nullify the reliability of all others.

Bearing capacity must be considered during design as some slight modification may allow significant improvement in capacity and result in drastic reliability improvement. It is also possible to obtain a significant increase in capacity by considering various manufacturers' bearings of the same nominal size. Some use different ball sizes, different widths, etc., with resulting different capacities.

Belt reliability is very sensitive to bending stress and several times less sensitive to installed tension stress. This dictates optimization of belt thickness between these two parameters, and consideration of pulley diameters and belt speeds. Choice of a material with a lower elastic modulus (DuPont "H Film") or using layered belts are alternatives which are also available but have little performance test background.

4.5.2 Tolerance Reliability

The dimensional tolerances of belts and bearings, already determined to be the critical recorder elements have a significant effect on recorder operation. This effect is accentuated by operating temperature and vibration.

Assembly using dimension coded and matched bearings, shafts, and housings should be considered to minimize the possibility of extreme fits and the degradation of these fits by temperature variations. Bearing mounts should be considered which do not change the preload with temperature. The use of flanged bearings or material more closely matching 440C thermal expansion are alternate methods.

Bearing preload procedure should be examined for a more reliable and more accurate method. If shims are to be used, they should be available in .0001" steps. An alternate method would be to use precision spacers with one spacer ground short to set the preload.

Belts should be fitted to each machine with a final tolerance in tension and stress determined by the acceptable limits of life at high tension and slip at the low tension. A method of determining the installed tension is desirable. Perhaps this could be done by measuring the torque to produce slip, or the resonant frequency of the installed belt.

The expected life of the belts as calculated for RRR seem short in view of the observed performance. A better estimate of performance could be made if additional belt tests were available to increase the predicted life at the 95% confidence level. Data on layered belt systems and "H Film" is desirable to determine their usefulness for long life.

4.6 Mean Time Before Failure

No satisfactory estimate of the "Mean Time Before Failure" was obtainable from the RRR test records. The short MTBF calculated on the basis of belt and bearing reliability is mainly dependent on the low reliability figures for belts which in turn are based on limited samples and thus are conservative. However, the recorder tests do indicate a life on the order of 100 hours.

While only a figure of merit, the analysis of the MTBF does provide a starting point for equipment evaluation and improvement.

4.7 **Bearing Study**

The bearing industry produces a product capable of satisfactory recorder performance with perfect handling, loads, lubrication and fit. Degradation of any of these results in premature failure. Solutions to the reliability problem are to perfect the operation environment or to make a less delicate bearing. Both approaches have merit and improvement in either direction will result in increased performance.

There are improvements available in bearing qualities and application but cost and time are the limiting factors. The most certain improvements can be made in the handling, fit and loading areas where cleanliness and accuracy can obviously be improved by technique and gaging improvements. With these improvements higher quality bearings would be commensurate, but the cost of high quality bearings requires their efficient use.

The specific facilities required should be determined by recommendations from bearing manufacturers and other consumers with like requirements. Section 5.7 considers specific areas in both manufacture and use of bearings; recommendations are included there.

4.8 **Natural Frequencies**

The fundamental frequencies of all free lengths of tape are above 3000 cps which is well beyond the rotational frequencies of the components and the vibration frequency band in most environment specifications.

5.0 TECHNICAL DISCUSSION

In the interest of clarity the substantiating calculations for the subsequent discussion are to be found under Section 6.0 Calculations and Tabulations. Adequate cross--references are given which permit the continuity of the discussion to be maintained.

5.1 Mathematical Model and Flutter Dynamic Analysis

Initially a simplified model of the RRR recorder was developed using the torque-voltage analogy. A paper analysis using Laplace Transform was made of the undamped case ($B_2 = 0$) which yielded two longitudinal frequency components which could cause flutter at 205 cps and 246 cps. The high-frequency component had the greater amplitude by a factor of 21. This analysis was made for the equal-reel case ($J_1 = J_2$). *

The system was programmed on the analog computer as shown. Results from this test substantially confirmed the paper analysis, a high-frequency component in the order of 250 cps being obtained. The lower frequency could not be obtained directly since the computer produced the combined (in tape recorder terminology "mixed") frequencies. By examining the envelope of the output, ($X_1 - X_2$), the frequency difference was obtained and this was then subtracted from the high-frequency component. The result was 213 cps which corresponds with the analytical result.

A further check was made using a program for the IBM 7094 which permits circuit analysis to be performed on the digital computer. The TAG Program used provided a check at

*See figure -page 41

$J_1 = J_2$ for the undamped case, being 246 cps for the dominant high-frequency mode and 206 cps for the low. Amplitudes also checked very well. This particular program on the 7094 takes a total time of 6 minutes, 2 minutes of which are utilized in execution.

The undamped case was extended to include a variable tape pack. In the analytical case, as indicated previously, frequencies of 390 cps and 165 cps were obtained for J_1 empty and J_2 full. However, making this run on the analog computer showed frequencies of about 600 cps and 140 cps. Re-checks were made of the analytical results but a re-run on the analog computer was not made. Although the analog computer checked results of the analytical calculation in the equal-reel condition, it did not do so in the case in which one reel was full and the other empty. It is believed at this point that the analytical results at these extremes are to be trusted.

With damping added to the system, the order of the characteristic equation doubles, thus making an analytical solution impractical. However, it was a simple matter to make a series of runs on the analog computer. These results revealed that the addition of the damping prescribed on Page 52 of the calculations essentially leaves the system relatively undamped and frequency is pulled only slightly from the undamped case. For example, results on both the analog and digital computer indicate that the equal-reel case in which $J_1 = J_2$ the two frequencies change from 246, 206 cps to 205, 203 with the addition of damping. It is interesting to note that from runs made on the analog computer with parameters set for 75°C., these frequencies go down further to 186, 186 cps with damping in. The results of the IBM 7094 TAG Program at JPL are shown in Figures 1, 2 and 3. Figure 1 is a graph trace of the digital data for a program involving the Laplace Transform operation from a paper analysis for the undamped, equal-reel case. The quantity on the ordinate is the longitudinal velocity which appears at the magnetic head and which could cause flutter components in the playback. This preliminary calculation was done to serve as a check for the TAG Program on the computer. One discrepancy was noted in this Laplace solution. The time scale does not conform with the algebraic results from the paper study and should be extended by a factor of about 1.03. These corrections are noted in the time trace in Figure 2. However, the plot is accurate enough to show the excellent correspondence obtained between paper analysis, digital computer and analog computer results of this circuit analysis.

Figure 2 shows the time plot of $\frac{\omega_1(t) - \omega_2(t)}{/T'/}$ where the symbols denote velocity across the head divided by the disturbing torque. This is for the undamped equal-reel case. It shows excellent correpondence with the analog computer results obtained previously under the same conditions. The analog computer results received from the program on page 55 of the calculations are shown in Figure 4. The top graph represents the velocity output at the magentic head for an applied step torque disturbance / T' / and so has identical coordinates with the digital results of Fig. 2. The correspondence between the analog and digital computer is very good. Both, in turn, agree with the paper analysis for the undamped case. Three other variables on the trace of Fig. 4 are shown for the recorder analysis. X_2, belt tape position from J_2, X_1 belt tape position from J_1 and X_1 or ω_1 longitudinal velocity variations from J. Some of these results have been checked with paper analysis and found to be in good agreement. The relative amplitudes of the two frequency modes of each variable can be determined by analyzing each waveform.

For the case in which damping is introduced into the tape recorder by considering the viscous friction produced at the magnetic head-tape junction, the paper analysis is too long. The results obtained from the digital computer are given in Figure 3. The important frequency component is about 200 cps and it is damped out in about 12 cycles. If the cogging frequency of the motor is anything greater than about 50 cps, it is readily seen that this mode of longitudinal vibration will be re-excited and sustained to produce unwanted flutter upon playback. It is interesting to note that these frequencies determined theoretically correspond very well with what has been experimentally observed on the RRR recorder both in experiments at Kinelogic and at JPL. The same physical system studied on the analog computer produced the results shown in Figure 4. The correspondence between the analog and digital computer is excellent. The top trace of Figure 5 corresponds very well with the IBM 7094 trace of Figure 3. The decrement is the same; even in fine detail where the lower frequency affects the waveform around the 7th cycle of the transient, there is good agreement. Thus, we conclude that we have a powerful tool for the worst-case analysis of any physical system for which a network model can be drawn, with good means of cross--checking the results.

Approximately 50 runs were made on the analog computer for various conditions, with and without damping, with varying tape packs on the two reels and at different temperatures. The resulting frequency modes were all obtained on charts such as those shown in Figures 4 and 5. The results are summarized in the graphs of Figures 6 and 7. Figure 6 shows the important mode, the high-frequency mode which in every case has a much greater amplitude than the low-frequency one shown in Figure 7. As the tape shifts from J_1 to J_2, the frequency increases. This is due to the unsymmetrical nature of the recorder. Since the effect of motor coupling into the system is not considered here, there is some question whether this frequency shift with tape pack would be so pronounced in the real system. Also, some of the measurementschange when the tape packs shift and this was not taken into consideration in the analyses. One important feature is the small amount of frequency "pulling" in going from the undamped to the damped condition. This indicates that for this particular mode of vibration, very little damping action is provided. This explains why the possibility of large flutter in the 200 cycle frequency ranges exists. To shift this high-frequency mode out of the playback range would require either some special form of damping the longitudinal mode, the addition of rather large amount of inertia or even a radical re-design of the support points for the Iso Belt and magnetic tape.

Another interesting point in this worst-case study is indicated by the effect of temperature change on the operating modes. The effect in both Figures 6 and 7 is to reduce the frequencies by a small amount. Comparisons can be made between the dashed and solid lines on both graph sets. The correspondence between the results of the TAG Program and the analog computer program is shown in both graphs; in both cases the agreement is good, particularly so on the high-frequency mode. It would have been interesting to make other checks, but the emphasis on this program was shifted to historical rather than theoretical effort and so further computer work was not attempted.

On page 57 of the calculations, a more complete circuit of the RRR recorder is given. It was thought that this model would be studied on the computer in a way similar to that described in the preceding paragraphs. However, certain problems were discovered in connection with this circuit, in particular with respect to the way in which the drive system is coupled into the tape motion. At the same time, the emphasis was shifted from the theoretical work to a careful logging of failures which had occurred on the RRR recorder under life tests. Consequently this part of the study was abandoned.

5.2 Test Failure Reports and Failure Analysis

There are a number of failure producing mechanisms which should be considered in the design of a high reliability tape transport. The principal failure modes are listed in Table 1. Table 2 lists the environmental factors. In Table 3 the major sub-assemblies of the RRR recorder are listed along with the failure modes associated with each one. Table 4 shows an attempt to weigh the relative importance of each of these failure modes together with its probability of occurrence. This was rather inconclusive as it was necessarily based largely on engineering judgment. It is interesting to note that a subsequent tabulation of actual failures of the RRR recorders did show the end of tape sensor, which heads the list in Table 4, was a frequent cause of failures. This tabulation of actual failures of the three RRR recorders built to date is given in Table 5. The failures were recorded during the testing period at Kinelogic. Included in the headings are: "Purpose" - purpose of test; "Conditions"- environment during the test; "Element" - part of machine that failed; "Symptoms" - mode of failure; "Cause" - reason for failure as best determined from the records; "Acc. Hours" - approximate running time between failures; "Acc. hours on failed part" - approximate time part was in service prior to failure; "Fix" - means chosen to remedy the cause of failure.

This list proved very useful in guiding the subsequent course of the Worst-Case Analysis. A significant number of the failures listed were of the juvenile or "break-in" type which can be expected with a device of this kind. Several bearing failures were observed and this resulted in an extensive review of the state-of-the-art of ball bearings and their handling requirements. Belt failures began to show the low expected machine life after juvenile failures were eliminated by continual testing.

5.3 Load Analysis

Using the standard design values for the belt and tape tensions, and ignoring the small unbalance produced by the motor torque the load arrangement shown in Figure 8 was prepared. These values were used for the computation of belt lives.

For each of the rotating components the worst combination of tape and belt tensions was used to compute the maximum radial load for the corresponding bearing pair. These loads are tabulated in Table 6. The calculations for bearing life used these loads combined with the appropriate axial preload.

In Table 7 the belt and tape tensions corresponding to yield stress level of 15,000 psi are listed. These values were computed for use in the strength analysis of the mechanical components. The worst case distribution of loads used for this analysis is shown in Figure 9.

The assumption here was that each of the rotating components would be in torsional equilibrium under the belt and tape loadings. Thus the tape packs were assumed to see a maximum load of 5.6 # or twice the belt tension at yield.

5.4 Load Analysis Applied to Strength of Mechanical Elements

The extremely high loads corresponding to yielding of the belts and tape were used to determine the highest possible stress points in the tape transport. Sample calculations for the worst cases are shown in Section 6.4. In each instance these worst case stresses were well below the yield stress of the material (303 stainless steel) used to fabricate these parts.

Since correct tape tracking calls for good dimensional stability, it was felt that the normal 0.2% offset yield strength might be an inadequate criteria of performance. In consequence, The Precision Elastic Limit for 303 stainless was estimated and compared with the worst case stress loads. The Precision Elastic Limit may be defined as the stress required to produce a permanent deformation (strain) of 1×10^{-6} inch per inch.

Since the critical stress section on the drive capstan assemblies is on the rotating shaft, it is subjected to a cyclic stress. The accepted value of the endurance limit stress was used to evaluate the effect of fatigue on its life.

For clarification, all of these stress values -- normal service stress, worst case stress, precision elastic limit, endurance limit and yield point are plotted on a bar chart. Figs. 10, 11, 12, 13. This demonstrates that safety margins are adequate.

In addition, the typical statistical distribution of yield strength for this type of material was sketched in on Figure 10, in order to show the confidence level of the yield strength.

5.5 Reliability, Life Analysis

The reliability study of the transport was divided into two major sections: Statistical Reliability, Tolerance -- Reliability. The first section consisted of applying the results of the load analysis (Section 5.3) to belt and bearing statistics, available through the Kinelogic Belt Study (JPL Contract #950899) and manufacturers' catalogs. The second section was aimed at investigating the effects of mechanical and environmental tolerances in belt and bearing assemblies.

5.5.1 Statistical Reliability

The belts and bearings used in the RRR were analyzed for life reliability using the manufacturers dynamic load ratings (Refs. 20, 21) and the normal load values developed in Section 5.3. These reliability figures are significant in that they indicate the lifetime available under ideal conditions. In system operation the life can be expected to be significantly shorter, depending on the control effected over design, installation, and operation environments; however, these figures set a statistical upper limit on the life expectancy. Improved performance can be expected from components having greater performance ratings.

For bearings the figures were obtained using a technique (Ref. 22) for evaluating the Weibull parameters of characteristic life and slope. The results are shown (Fig. 14) in graphic form for clarity. The calculations were made using the largest preload allowed by the tolerance $\pm$ .5 lb.

From this graph, one group of the bearings stands out as relatively short lived: another group is of intermediate life expectancy and a third has a relatively long life expectancy. This indicates that either special care must be taken with these critical bearings to assure their relative life in the system, or their design load must be changed, or they must be eliminated and a more compatible substitute found. In this particular case the indicted bearing is a small one .0937" x .1875" which in the double shielded version used has .025" balls associated with a low capacity. Since life is proportional to $\left(\frac{\text{capacity}}{\text{load}}\right)^3$ a small increase in capacity would drastically increase the life. Similarly, a decrease in the load by decreasing, or holding closer tolerances on, the preload would result in longer life. It may even be possible to substitute a more durable bearing in its stead.

For belts, the situation is an analogous one except that there is more freedom of choice in designing for longevity, but less control over the manufacturing variables. The graph (Fig. 15) indicates the reliability-life of the individual belts calculated using the results from a belt study performed at Kinelogic (JPL Contract #950899). It seems apparent that the reliable recorder lifetime with this system of belts is comparatively short. If reliable recorder life beyond these lifetimes is desired, it is apparent that belts will necessarily be the most critical factor.

This situation may not really be as critical as is indicated by the reliability figures, as life tests of the machine have indicated some significant difference between the calculated and actual lifetimes. The calculations are made on the basis that the prediction be the lower limit of reliability-life 95 times out of 100. If we accept the possibility of only 50 correct predictions out of 100 for reliability-life the expected lifetime can be increased by a factor of 3 or more. This still indicates that the belts are the most unreliable elements.

In the stress range where the belts normally operate, small changes (10%) in the installed stress result in small changes (10--20%) in the expected life, but small changes (10%) in the bending stress result in much larger (50%) changes in the expected life. These changes can be accomplished by reducing belt thickness, increasing pulley diameters, decreasing the elastic modulus of the belt through use of another material, or by using layered belts. The system can be optimized within the constraints of size, power transmitted, etc.

5.5.2 Tolerance Reliability

The dimensional tolerances of the belts and bearings already determined to be the critical recorder elements, have a significant effect on the recorder operation. This effect will be accentuated by operating temperature and vibration.

The shaft housing and bearing dimension tolerances were tabulated (Table 8) and the range of possible radial fits at room temperature was calculated to be -.00010" to +.00050" for shafts in bearings and bearings in housings. The fits at the temperature extremes $0^{\circ}F$ and $180^{\circ}F$ were then calculated for their worst case (loose fits becoming looser, tight fits becoming tighter). The shafts and housings are of 303 stainless steel while the bearing parts are 440 C stainless.

The results of these calculations show that with the materials and tolerances used there can be some excessively loose or tight fits resulting from random assembly and temperature variations. Normally the tight fits can be excluded since mounting procedures should be such that the parts will not be mated with interference fits. Tight fits will result in decreased radial play or load increase and in distortion of bearing races leading to speed variations and failures. The loose fits are generally considered undesirable and will be the cause of vibration damage and fretting corrosion.

In the axial direction there is a similar situation with shaft and housing loading or unloading the bearing due to differential expansion. The amount of pertinent dimension change depends on the type of mounting and size of the bearings. The differential change in axial dimension between the RRR mounts and bearings is equal to change in temperature x width of bearing pair x differential expansion rate. $\Delta T 2W \ (\alpha - \beta)$

The method of preloading the bearings requires duplexed pairs of precision shims. The duplex pairs are supplied with a nominal load $\pm 1/2$ lb. The bearings loaded by the shim technique have their inner rings deflected a fixed amount with shims usually in steps of .0002" thick. This means the preload is fixed within approximately $\pm .0001''$. If we consider the SR-2-5PP bearing used for the RRR capstans and use a deflection graph (Ref 18) we find a deflection of .00027" for a 2 lb load and a deflection of .00037" for a 4.3 lb load. This indicates that a tolerance of .0001" per bearing in the loading shim can result in a large preload tolerance. The effect of this load on life is shown in the graph (Fig. 16) where the 90% life is plotted against preload for the three types of bearings used excluding the motor bearings.

The temperature effects previously considered also have an effect on the preload condition of the bearing. To calculate the loading change due to a change in axial dimensions we use the deflection vs. load graphs mentioned above and consider the worst case, (high loads increasing, low loads decreasing).

To calculate the loading produced by changes in the radial dimensions is more difficult. To simplify, we find an average of 3 for $\frac{\text{change in axial play}}{\text{change in radial play}}$ in the unloaded condition, and assume a like relation for the loaded condition. We use .8 as the fraction of radial interference that is transmitted into the bearing to increase the load. The effects of these temperature variations on bearing load are shown in Table 9.

The belts are manufactured with a length tolerance of $\pm$.25% which will result in an installed stress tolerance of $\pm$ .0025 x 750,000 psi = 1625 psi in applications without an automatic tensioning device, where 750,000 psi is the room temperature elastic modulus of Mylar. This results in significant variability in installed tension. Lower tension results in increased life but decreases the non-slip load carrying ability; higher tension has the opposite effect. In addition, the width and thickness have tolerances of 15% and 4% respectively. This results in an overall stress tolerance of $\pm$ 68% in the motor capstan belt on RRR, for example. Normally, belts with an exceptionally low tension will not be used but the upper limits can be approached.

Temperature changes will affect belts both in their dimensions and elastic modulus. The thermal expansion coefficient of $1.5 \times 10^{-5} \frac{\text{inch}}{\text{inch } F^{o}}$ for Mylar is matched to that of Aluminum or Magnesium so that temperature changes will not affect the relative dimensions, but lower temperatures increase the elastic modulus, while higher temperatures decrease it. These changes in modulus could be used to predict longer expected life at lower modulus values but no tests have been made of temperature effects on belt fatigue.

5.6 Mean Time Before Failure

The term Mean Time Before Failure is usually considered the average time before failure for many units of the same design and is usually loaded with statistical uncertainties such as sample size, standard deviation, etc. However, this may be considered as a figure of merit and if additional information is provided it can convey considerable information and prove useful in comparing relative performance.

In this case, Mean Time Before Failure is calculated theoretically from fatigue life information on individual components, the number of units and test records being such that no satisfactory MTBF was generated by life tests. We combine the results of bearings and belt statistics from Section 5.5.1 with the assumption that failaure of any one of these elements will be a system failure. Statistically, this means that the survival probability of the system is the product of the individual elements' survival probability.

This assumption is warranted by the degrading effects noted when a bearing fails, increasing the system torque requirements and stalling the motor, or by the obvious consequence of a broken belt. We also assume that the belts and bearings are the limiting factor in recorder life to the extent that other survival probabilities are considered to be essentially one for times on the order of a thousand hours. The graph (Fig. 17) represents this reliability figure as a function of time and clearly shows the dependence on belts and bearings.

5.7 Reliable Bearings for Tape Recorders

The object of this study is to determine criteria for bearings which can be expected, with some degree of certainty, to operate for the desired recorder lifetime.

In general the problem is two fold. At this time, bearings intrinsically have a very uncertain lifetime: a group of identically prepared bearings under identical ideal conditions will exhibit an average failure time from 4 to 5 times as long as the failure time for the first 10%. The controls necessary to decrease this spread apparently are not yet understood or available.

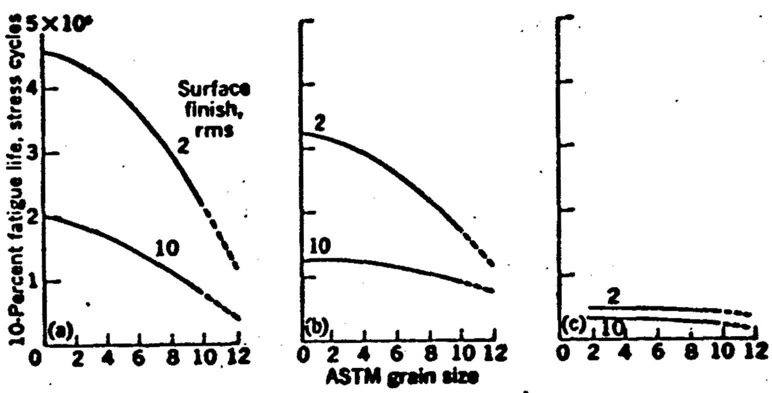
Secondly, normal applications of bearings are not ideal; lubrication is limited, fits of bearings are not perfect, they operate in an atmosphere which cannot be cleaned, and they must be handled several times before final operation. These and other problems militate against bearings attaining their intrinsic life. In an application where failure of any one of many bearings is catastrophic, this means low reliability at an appreciable fraction of the expected bearing life.

Solution to the first situation requires both design of the best bearing and proper speed-loading application of this bearing in the recorder. In general it seems that standard bearing techniques - i.e., steel balls and races, oil or grease lubricants, are best from the standpoint of reliability in the mild temperature environment of a recorder. Other techniques, dry lubrication, silver or gold plated balls, etc., are under test for extreme environments of vacuum and temperature but as yet seem even more unreliable than standard methods. (Ref. 1, 3, 10). The following comments have been gleaned from the listed references and from personal communication and experience:

1. Consumable electrode vacuum melt steel produces better bearing material than vacuum melt which in turn is better than the normal air melt. This can be specified to the manufacturer. (Ref. 1, 2).

2. Grain size and surface finish are critical to fatigue life. Large grain size and finest surface finish contribute to extended life. These factors may be controlled by the manufacturer. See Fig. 18 (Ref. 1, 9).

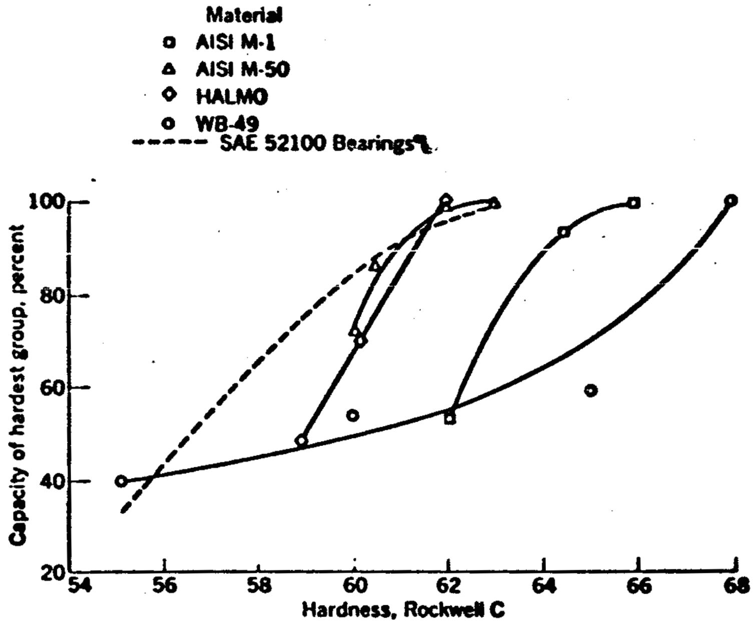
M-50 Steel



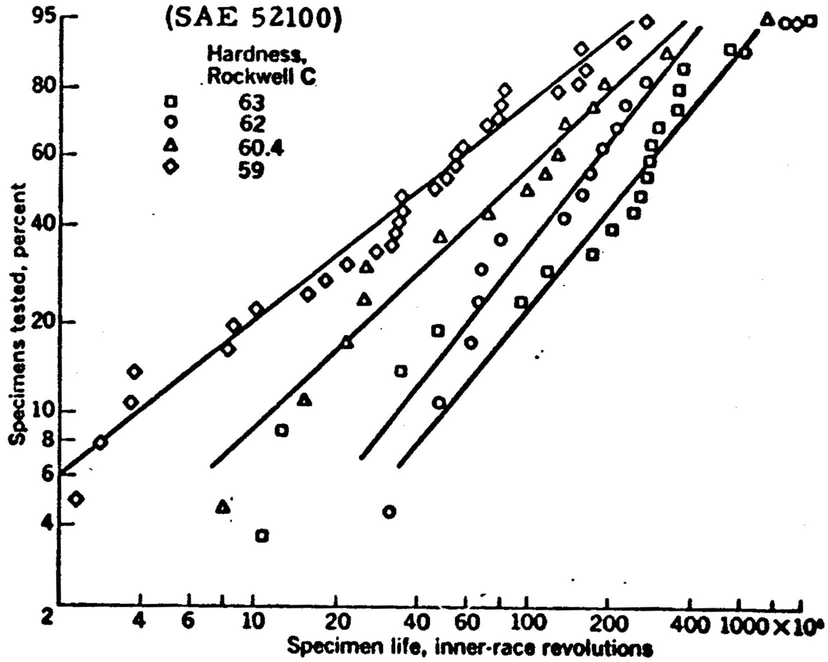
(a) Hardness, Rockwell C-62½. (b) Hardness, Rockwell C-58. (c) Hardness, Rockwell C-54.

Effect of grain size on 10 per cent fatigue life for various hardnesses and surface finishes
Fig. 18 Ref. 9

3. In any one material the fatigue life is very dependent on the actual hardness rating. The maximum of the hardness range for the desired steel should be specified. See Fig. 19 and 20 (Ref. 1, 7, & 8)



Relative load--carrying capacity of bearing steel balls tempered to different hardness levels. Fig.19. Ref. 8



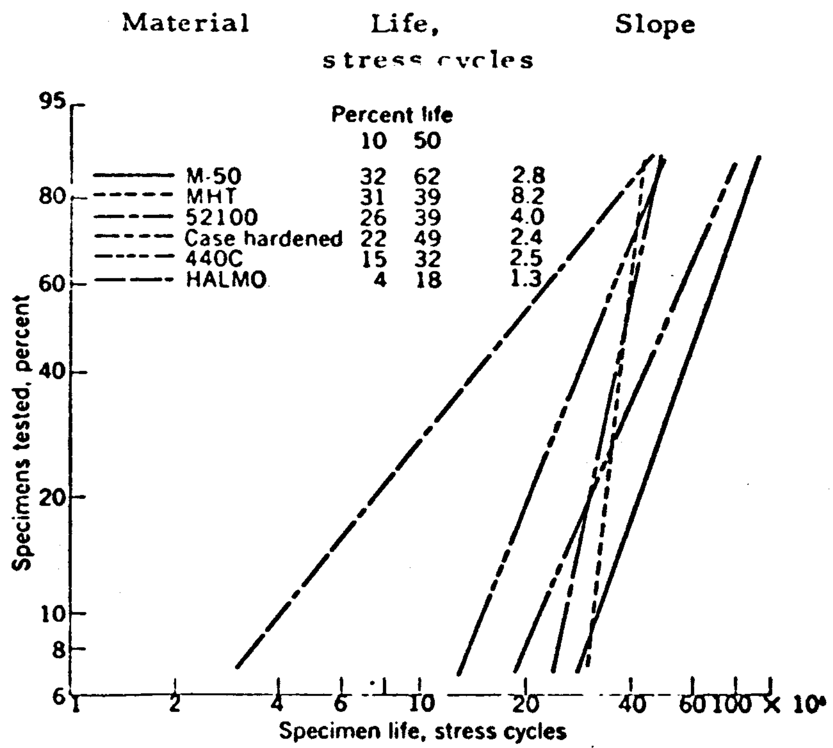
Ball bearing fatigue life at four hardness levels. Fig. 20 Ref. 7

4. The grain or fiber orientation to the wear surface significantly affects the fatigue life with respect to both balls and races. The orientation of the balls during operation with respect to their equator (a line determined by the ball forming operation) is a critical factor. A load path consistently over the poles results in early fatigue failure of the ball. This is uncontrolled in ball bearings. (Ref. 1).

5. Most fatigue failures are initiated at shallow subsurface imperfections. Significant improvements in life have been effected by magnetic particle inspection of bearing parts. This may be possible, but the small sizes involved makes this inspection difficult. Some sort of subsurface inspection is desirable but none is presently available for miniature size bearings. (Ref. 1, 2).

6.

While 440 C. stainless and 52100 chrome are the steels used predominantly in bearings, MHT (1% Aluminum in 52100) and AISI M-50 are considered to have better performance. See Fig.21. (Ref. 1, 2, 6)



Best results obtained with each of six materials tested.

Fig. 21 Ref. 6

7. Of 52100 and 440 C steels the choice depends on the environment the bearing encounters. 52100 is harder but more subject to degradation of its higher load capacity by corrosion. This corrosion readily occurs during handling or operation in a humid atmosphere. However, Reference 3 reports no performance difference in vacuum operation.

(Ref. 3, 4)

8. The choice of retainer material and type for this application must be made among steel ribbon or crown type which provide higher strength and phenolic type which provide increased lubrication due to oil impregnation. Results of tests are such that a clear choice cannot be made. Bearing manufacturers recommend impregnated phenolic retainers for long life with minimum lubrication. Ref. 3 states that phenolic retainers are definitely superior. Chipping of corners and flaking of phenolic retainers have been observed in bearings used in our tape recorders. Ref. 1 shows a Tiros installation using oil impregnated sintered nylon as a reservoir adjacent to the bearing.

9. Once the proper materials have been chosen it is necesary to consider the dimensional and operational specifications. Normally instrument bearings are supplied in ABEC 7 class which specifies the dimensional tolerances of the inner and outer rings. There exists a class ABEC 9 which holds tolerances even closer, but comparative performance figures

are not available from the industry. Several manufacturers are willing to supply bearings with tolerances to 20 microinches. The question of whether this affords a real or imaginary increase in reliable performance remains, as no figures are available. Tolerances are important but the law of diminishing returns may be operating. Since these dimensions control run out, noise and torque variations, lower levels in these properties can be gained by closer tolerances. (Ref. 4)

10. The shape of the bearing races is an important factor in bearing performance, but relatively little information was found concerning the effect of this parameter. Within a specific type bearing no choice is offered in standard catalogues but special applications could result in special design by a manufacturer. (Ref. 9, 11)

11. The balls normally used in the bearings have dimensional variations of the order of .000010" but tolerances as low as .000003" are available and are provided in the aforementioned 20 microinch bearings. Since the largest in a set of balls will be highly loaded close matching will minimize the overload. (Ref. 11)

12. The number and nominal size of the balls are important in that larger balls will sustain larger loads while a decrease in the number of balls increases the reliability of the set of balls. The

limiting factors to be considered are ball-load capacity, ball space capacity, ring deformation, ball reliability. Differences exist among manufacturers as to ball size and number and associated capacity for the same size bearing. (Ref. 4, 17, 18)

13. One element not included in the tolerance specifications is the retainer which generally is a more geometrically intricate part than ball or races. An attempt should be made to exert some control over the tolerances of these elements as they are vital to bearing performance.

14. Bearings can be provided with the high points of eccentricity marked on the inner and outer rings. This will enable these points to be aligned so as to oppose the high points of the mating shaft or housing. (Ref. 17)

In considering these tolerances versus reliability, it would seem that the more perfect geometry would provide the most reliable operation. The question of cost-effectiveness is unsettled.

15. Once a bearing is assembled, other operational parameters are to be considered: radial play, preload torque, torque variation. Most of these are functions of the previous parameters and they are controlled within some specific limits. It is possible to cull from a standard run of bearings those meeting a tighter set of limits.

16. The specification for radial play should be determined by the load characteristics. Axial loads require larger radial play to increase the contact angle, thus decreasing the ball loading. Reducing radial play causes radial loads to be spread over a greater number of balls but ball loading is increased for thrust loads. For all preloaded bearings radial play greater than .0005" is recommended. Ranges of radial play as small as .0002" can be specified within the .0001" to .0012" range generally available. Excessive radial play can cause the balls to override the edge of the race under load. High speed operation at a large contact angle results in gyroscopic forces on the balls, producing additional friction loading. (Ref. 5)

In angular contact bearings the contact angle itself may be specified as close as $\pm .5^{\circ}$ (Ref. 4)

17. The force required to rotate the inner race is usually measured statically and at 2 RPM. This force depends on many factors, geometry, load, lubrication, etc. Any run of bearings all the same will provide a distribution of starting, average running, and peak running torques. The variation in running torque as the bearing rotates (hash width) will also differ between individual bearings. The most desirable bearing would have the smallest difference between average and peak torque and the least hash width. The actual limits to values of these properties may be specified but the ability of the manufacturer to control them requires sorting by test to obtain the best. The range from peak or starting to average torque may be

2 to 1. (Ref. 4). Figures concerning yield of a particular % hash width or % peak torque are not available from the manufacturers. The average torque will depend on surface finish, load, lubricant, contact angle, number of balls, etc.

18. Preload values are set within customer specified limits and materially affect the life and operating torque of a duplex pair. Several pairs of duplexed bearings have been received in a qualitative unsatisfactory condition. It is advisable that torque measurements be made on duplex pairs as a pair in the specified preload condition.

19. Lubrication requirements are set by operating temperatures, environment, torque requirements bearing material. Reliable lifetimes require a clean, non-reactive low water content oil or grease. In general, oil viscosity is very important in increasing life except for silicones which produce essentially viscosity independent lives. The synthetic silicone oils have consistently performed very well under a wide variety of conditions. Mineral oils show a greater dependance on viscosity and thus on temperature but have outperformed the silicones in some applications. The present situation is such that no one lubricant is an outstanding choise. However, test data is available on a few lubricants which have demonstrated long life capability in 20-100 mg quantities (over 10,000 hours in R-3 bearings). Apiezon K. Versilube F-50, Diphenyl-bis-n-dodecylsilane, OS-124 polyphenyl ether. (Ref. 1, 3, 12, 13, 14, 15.)

20. Once the bearings are lubricated it remains to keep the lubricant in and contamination out. This is accomplished through the use of shields, although greases are effective by themselves. Shields are available in contacting or noncontacting types. Contact type (usually called "seals") are usually made of some filled teflon or other low friction material fixed to the outer ring and rubbing on the land of the inner race. This type results typically in 5 to 20 times the unsealed running torque but are the most effective of the standard methods. This material will be an additional source of wear particles itself, but should be considered where external conditions are dirty. Non-contacting shields are fixed to the outer ring and either extend below the inner land into an undercut or extend to within about .003" of the land. The former provides a labyrinth type of seal and is more effective than the butt type of seal. (See Fig.22)

Here maximum effectiveness is indicated by the lowest "effectiveness number" as measured in the MPB test system.

(Ref. 4, 5, 16, 17)

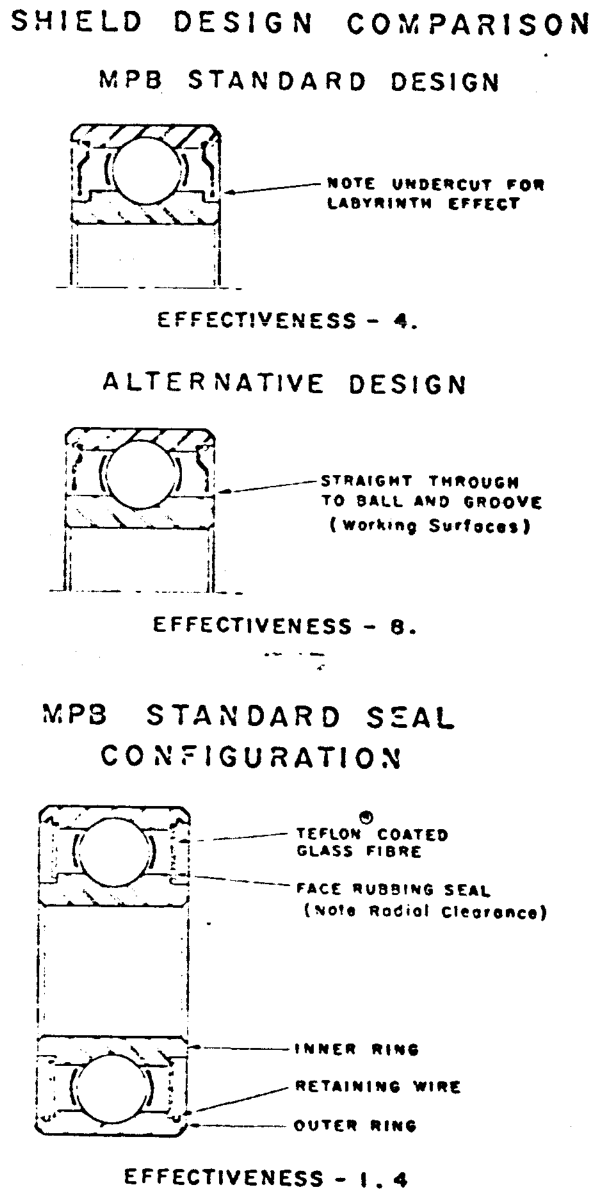


Figure 22 Ref. 16

21. Once the bearings have been received by the consumer the second situation requires attention. Not bearing experts, the consumers must use the engineering services and installation recommendations available from the manufacturers. The rigid cleanliness requirements set up by them may even require a change in the philosophy of the assembler. A paperwork system to confirm conformation to specifications is a necessity. A list of handling and installation recommendations is included as a separate list compiled from manufacturers' catalogues.

BEARING HANDLING AND INSTALLATION

1. All bearing handling should be done by a qualified person in a qualified clean area, and be kept to a minimum.

2. Bearing vials should be stored in an upright position.

3. Bearings should be demagnetized before installation.

4. Remove bearings from package only in the clean area and transport open bearings in covered glass petri dishes.

5. Handle bearings only with tweezers, finger cots, or special handling tools.

6. Shafts and housings must be carefully checked for variation from tolerance limits.

7. Bearings should not be placed on shafts or in housings until all finishing operations have been completed and shafts and housings have been thoroughly cleaned.

8. Bearings should be matched as close as possible to shafts and housings for line-line to .0002" loose fits, unless another fit is specifically required.

9. Bearings should be installed with special handling tools, with mounting pressure always applied at the ring to be fitted, and with the bearing at a right angle to the axis of fit.

10. Preloading should be done with shims of dimensions parallel within .0001" without burrs, and in thickness steps of .0001".

11. With bearings having the high points of ring eccentricity marked, the high points should be aligned opposite the indicated high point of the bearing seat - i.e., high points of inner rings 180° from high point of shaft, high points of outer rings 180° from high point of housing.

5.8 Natural Frequencies

There are two types of natural individual element frequencies considered in this section as opposed to the system frequencies considered in the Mathematical Model section. We consider the rotational frequencies of shafts and bearing parts which are the generators of mechanical vibrations and the fundamental frequencies of the free spans of tape and belt which are the absorbers of the mechanical vibrations.

Rotating systems are composed of elements having different rotational frequencies; shaft; individual balls; retainer. A damaged area on a race will be contacted by balls several times during one revolution of either race. A ball will also make several revolutions for each revolution of a race. Retainer hangup will occur at its rate of revolution. All these rotating elements of a bearing can and do contribute their particular frequency to the mechanical "noise" spectrum of a rotating part.

Lengths of belt or tape clamped at the ends are resonant at some fundamental frequency and, provided with energy of the frequency, they will absorb it without damping causing damage or speed fluctuations.

The calculations are made using the 30 ips tape speed for the rotational frequencies and the free span fundamental frequencies are calculated for both the transverse (in the thickness direction) and longitudinal (in the length direction) modes. The combination of these individual parts was attempted in the Mathematical Model to give a more complete analysis. This mechanical noise; friction noise and electrical disturbances in the motor are sources of transport flutter, except for environment. Adjusting the free spans of tape to eliminate a fundamental frequency near that of a generator, or the environmental frequency is a design necessity. The particular frequencies of the RRR are recorded in Table 10.

Longitudinal vibration frequency $= \frac{1}{2L}\sqrt{\frac{E}{P}}$

Transverse vibration frequency $= \frac{1}{2L}\sqrt{T/_{PA}}$

L = Length

E = Elastic modulus

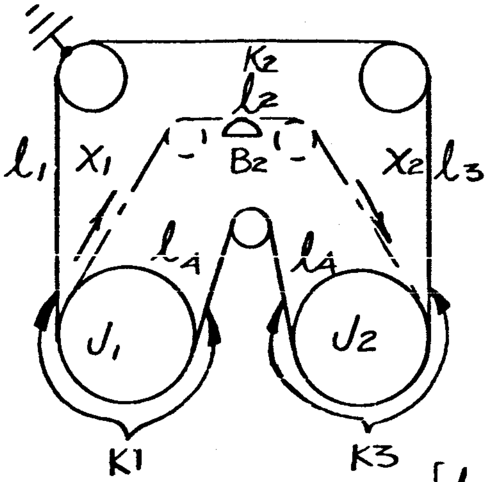
T = Tension

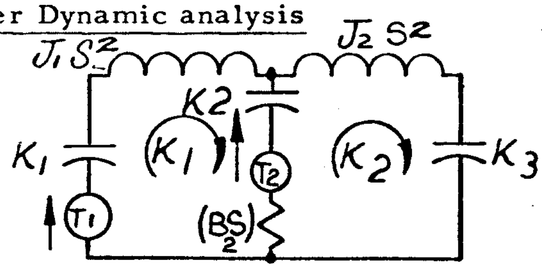
A = Cross-section area

P = Mat. density

6.0 CALCULATIONS

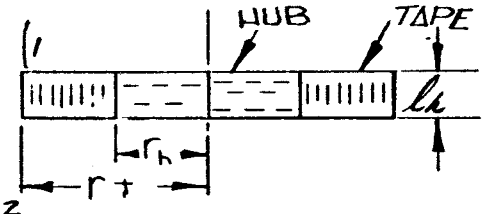
6.1 Mathematical Model and Flutter Dynamic analysis





J's and K's are functions of tape position and B_2 is nonlinear.

S = Laplace transform variable.



$$K_1 = 16E_IA_I r_{T1}^2\left[\frac{l_1+l_4}{l_1\,l_4}\right]\frac{oz\text{-}in}{rad}$$

$$K_2 = 4E_TA_T\frac{(r_{T1}+r_{T2})^2}{l_2}\ \frac{oz\text{-}in}{rad} \qquad (2$$

$$K_3 = 16E_IA_I r_{T2}^2\left[\frac{l_3+l_4}{l_3\,l_4}\right]\frac{oz\text{-}in}{rad} \qquad (3$$

$$J_1 = J_h + J_{T1} = \frac{\pi r_h^4 l_h \delta_h}{2g} + r_T l_{T1}(r_{T1}^2 + r_h^2) \qquad (4$$

$$J_2 = J_h + J_{T2} = \frac{\pi r_h^4 l_h \delta_h}{2g} + r_T l_{T2}(r_{T2}^2 + r_h^2) \qquad (5$$

$$B = \frac{T}{\Omega}\ \text{(Assume Linear for first Analysis)} \qquad (6$$

E = Modulus of Elasticity Isobelt in lbs/in.^{2}

A = Area of Isobelt in sq. in.

A_T = Area of Magnetic Tape in sq. in.

E_T = Modulus of Elasticity Magnetic Tape in lbs. in.^{2}

Analysis RRR Drive, continued

r_{T1} = Tape radius reel 1 in inches

ℓ_1 = Effective Isobelt length left side reel 1 in inches.

ℓ_4 = Effective Isotbelt length right side reel 1 in inches.

ℓ_4 = Effective Isobelt length left side reel 2 in inches.

ℓ_3 = Effective Isobelt length right side reel 2 in inches.

ℓ_2 = Total length of tape reel (1) to reel (2)

r_{T2} = Tape radius reel (2) in inches

r_h = Hub radius in inches

ℓ_h = Hub width in inches

δ_h = Hub density in oz/in.^{2}

γ_T = Tape weight in oz/ft.

ℓ_{T1} = Length of tape on reel 1 in inches

ℓ_{T2} = Length of tape on reel 2 in inches

g = Gravity constant = 386 in/sec.^{2}

The matrices are:

$$\tilde{T} = \begin{bmatrix} T_1 - T_2 \\ T_2 \end{bmatrix} ; \quad \tilde{X} = \begin{bmatrix} \chi_1 \\ \chi_2 \end{bmatrix} ; \quad \tilde{Z} = \begin{bmatrix} J_1 S^2 + B_2 S + K_1 + K_2, & -B_2 S - K_2 \\ -B_2 S - K_2, & J_2 S^2 + BS + K_2 + K_3 \end{bmatrix} \quad (7$$

The matrix equation is (8

$$\tilde{X} = \tilde{Z}^{-1} \cdot \tilde{T}$$

$$\begin{bmatrix} \chi_1 \\ \chi_2 \end{bmatrix} = \frac{1}{\Delta(S)} \begin{bmatrix} M_{11}(S) & M_{21}(S) \\ M_{12}(S) & M_{22}(S) \end{bmatrix} \cdot \begin{bmatrix} T_1 & -T_2 \\ & T_2 \end{bmatrix} \qquad (9$$

$$\chi_1(S) = \frac{M_{11}(S)\left[T_1(S) - T_2(S)\right] + M_{21}(S)\,T_2(S)}{\Delta(S)} \qquad (10$$

$$\chi_2(S) = \frac{M_{12}(S)\left[T_1(S) - T_2(S)\right] + M_{22}(S)\,T_2(S)}{\Delta(S)} \qquad (11$$

$$\chi_1(S) - \chi_2(S) = \frac{(M_{11} - M_{12})\,T_1 - (M_{11} - M_{21} - M_{12} + M_{22})\,T_2}{\Delta} \qquad (12$$

For simultaneously applied disturbances $T_1(S) = \frac{|T_1|}{S}$ and $T_2(S) = \frac{|T_2|}{S}$

$$\Omega_1(S) - \Omega_2(S) = \frac{(M_{11} - M_{12})\,|T_1| - (M_{11} - M_{21} - M_{12} + M_{22})}{\Delta}\,|T_2| \qquad (13$$

where $\Omega(S) = S\chi(S)$ are the velocity vectors.

$$M_{11}(S) = J_2 S^2 + B_2 S + K_2 + K_3 \qquad (14$$

$$M_{12}(S) = B_2 S + K_2 = M_{21}(S) \qquad (15$$

$$M_{22}(S) = J_1 S^2 + B_2 S + K_1 + K_2 \qquad (16$$

$$\Delta(S) = J_1 J_2 S^4 + (J_1 + J_2) B_2 S^3 + \left[J_1(K_2 + K_3) + J_2 (K_1 + K_2)\right] S^2 + B_2(K_1 + K_3) S + (K_1 K_2 + K_2 K_3 + K_1 K_3) \qquad (17$$

Assume a system with the following constants:

$$E_I = 7.5 \times 10^5 \text{ lbs/in.}^2$$

$$A_I = (2.50 \times 10^{-1})(10^{-3}) = 2.5 \times 10^{-4} \text{ in.}^2$$

$$\ell_1 = 6 \text{ in.}$$

$$\ell_4 = 1 \text{ in.}$$

$$r_{T1} = 1.313 \text{ in.} = r_{T2}$$

$$E_T = 7.5 \times 10^5 \text{ lbs/in.}^2$$

$$A_T = 2.5 \times 10^{-4} \text{ in.}^2$$

$$\ell_2 = 4 \text{ in.}$$

$$\ell = 8 \text{ in.}$$

$$r_h = 0.75 \text{ in.}$$

$$\ell_h = 0.25 \text{ in.}$$

$$\delta_h = 0.101 \text{ lbs/in.}^2 = 1.617 \text{ oz. in.}^3$$

$$\gamma_T = 4.05 \times 10^{-3} \text{ oz/ft.}$$

$$\ell_{T1} = 250 \text{ ft.} = \ell_{T2}$$

$$g = 386 \text{ in/sec}^2$$

$$B_2 = O$$

$$T_2 = O$$

From Equation (1)

$$K_I = 16\,(7.5 \times 10^5)(2.5 \times 10^{-4})(1.313)^2 \left[\frac{6+1}{6 \times 1}\right]$$

$$= 6040 \text{ oz. in/rad.}$$

$$K_2 = \frac{(4)\ (7.5 \times 10^5)(2.5 \times 10^{-4})(2.625)^2}{4}$$

$$= 1,293 \text{ oz. in/rad.}$$

$$K_3 = (16)(7.5 \times 10^5)(2.5 \times 10^{-4})(1.313)^2 \left[\frac{8+1}{8-1}\right]$$

$$= 5,820 \text{ oz. in/rad.}$$

$$J_1 = J_2 = \frac{(\pi)(0.75)^4(0.25)(1.617) + (4.05 \times 10^{-3})(250)(1.313^2 + 0.75^2}{(2)\ (386)}$$

$$= J = 3.55 \times 10^{-3} \text{ oz. in. sec.}^2$$

Subst. Eq's. (14) - (17) into (13),

$$\Omega_1(S) - \Omega_2(S) = \frac{(JS^2 + K_3)\ |T_1|}{J^2S^4 + J\,(K_1 + 2K_2 + K_3)S^2 + (K_1K_2 + K_2K_3 + K_1K_3)}$$

$$JS_1^2,\ JS_2^2 = \frac{-A \pm B}{2}$$

$$\frac{J\left[\Omega_1(S) - \Omega_2(S)\right]}{|T_1|} = \frac{S^2 + a_o}{(S^2 + \beta^2)(S^2 + \lambda^2)}$$

where $a_o = \frac{K_3}{J} = \frac{5,820}{3.55 \times 10^{-3}} = 1.64 \times 10^6$

$$\beta^2 = \frac{A - B}{2J}$$

$$\lambda^2 = \frac{A + B}{2J}$$

and

$$A = K_1 + 2K_2 + K_3$$

$$B = \sqrt{K_1^2 + 4K_2^2 + K_3^2 - 2K_1 K_3}$$

For the numbers given,

$$A = 1.445 \times 10^4$$

$$B = 2,600$$

$$A - B = 11,850$$

$$A + B = 17,050$$

$$\beta^2 = \frac{11.850 \times 10^3}{7.10 \times 10^{-3}} = 1.67 \times 10^6$$

$$\lambda^2 = \frac{17.05 \times 10^3}{7.10 \times 10^{-3}} = 2.40 \times 10^6$$

from which

$$\beta = 1.29 \times 10^3$$

$$\lambda = 1.55 \times 10^3$$

Letting $|T_1'| = \dfrac{|T_1|}{J}$ and writing the inverse transform of (18

$$\frac{W_1(t) - W_2(t)}{|T_1|} = \frac{a_0 - \beta^2}{\beta(\lambda^2 - \beta^2)} \sin \beta t + \frac{a_0 - \lambda^2}{\lambda(\beta^2 - \lambda^2)} \sin \lambda t$$

$$= -3.18 \times 10^{-5} \sin 1.29 \times 10^3 t + 6.71 \times 10^{-4} \sin 1.55 \times 10^3 t$$

Rewriting (13 and including J_1 and J_2.

$$\Omega_1(S) - \Omega_2(S) = \frac{(J_2 S^2 + K_3)\ |T_{,1}|}{J_1 J_2 S^4 + [J_1(K_2+K_3) + J_2(K_1+K_2)]S^2 (K_1 K_2 + K_2 K_3 + K_1 K_3)}$$

$$\frac{J_1[\Omega\ (S) - \Omega\ (S)]}{|T_1|} = \frac{S^2 + a_o}{(S^2 + \beta^2)(S^2 + \lambda^2)}$$

where $$a_o = \frac{K_3}{J_2}$$

$$\beta^2 = A - B$$

$$\lambda^2 = A + B$$

and $$A = \frac{1}{2}\left[\frac{K_2 + K_3}{J_2} + \frac{K_1 + K_2}{J_1}\right]$$

$$B = \frac{1}{2}\sqrt{\frac{J_1^2(K_2^2 + 2K_2K_3 + K_3^2) - 2J_1J_2(K_1K_2 + K_2K_3 + K_1K_3 - K_2^2) \ldots\ldots}{J_1^2 J_2^2}}$$

$$\ldots\ldots + J_2^2(K_1^2 + 2K_1K_2 + K_2^2)$$

Now consider the case in which reel J_1 is empty and J_2 is full.

The numbers on Page 44 are the same except

$$r_{T1} = 0.75 \text{ in.}$$

$$r_{T2} = 1.700 \text{ in.}$$

$$l_{T2} = 500 \text{ ft.}$$

$$l_{T1} = 0 \text{ ft.}$$

$$K_1 = (16)(7.5 \times 10^5)(2.5 \times 10^{-4})(0.75)^2 \left[\frac{6+1}{6 \times 1}\right] = 1969 \text{ oz. in/rad.}$$

$$K_2 = \frac{4\,(75 \times 10^5)(2.5 \times 10^{-4})(2.45)^2}{4} = 1126 \text{ oz. in/rad.}$$

$$K_3 = (16)(7.5 \times 10^5)(2.5 \times 10^{-4})(1.700)^2 \left[\frac{8+1}{8 \times 1}\right] = 9760 \text{ oz. in rad.}$$

$$J_1 = \frac{\pi\,(0.75)^4\,(0.25)(1.617)}{(2)(386)} = 5.21 \times 10^{-4} \text{ oz. in sec.}^2$$

$$J_2 = \frac{\pi\,(0.75)^4\,(0.25)(1.617) + (4.05 \times 10^{-3})(0.500 \times 10^3)(1.7^2 + 1.75^2)}{(2)\;(386)}$$

$$= 9.58 \times 10^{-3} \text{ oz. in sec.}^2$$

$$A = 1/2\left(\frac{1126 + 9760}{9.58 \times 10^{-3}} \qquad \frac{1969 + 1126}{5.21 \times 10^{-4}}\right) = 3.54 \times 10^6$$

$$B = 1/2\sqrt{\frac{(5.21\text{x}10^{-4})^2(1126^2 + 2\text{x}1126\text{x}9760 + 9760^2) - 2(5.21\text{x}10^{-4})(\ldots\ldots}{(5.21\text{x}10^{-4})^2 \quad (9.58\text{x}10^{-3})^2}}$$

$$\ldots\ldots(9.58\text{x}10^{-3})(1969\text{x}1126 + 1126\text{x}9760 + 1969\text{x}9760 - 1126^2) + \ldots\ldots$$

$$\ldots\ldots + (9.58\text{x}10^{-3})^2\,(1969^2 + 2\text{x}1969\text{x}1126 + 1126^2)$$

$$= 1/2\sqrt{24 \times 10^{12}} = 2.45 \times 10^6$$

$$\beta^2 = 3.53 \times 10^6 - 2.45 \times 10^6 = 1.09 \times 10^6$$

$$\lambda^2 = 3.53 \times 10^6 + 2.45 \times 10^6 = 5.99 \times 10^6$$

$$\alpha_0 = \frac{9,760}{9.58 \times 10^{-3}} = 1.02 \times -10^{6}$$

$$\beta = \sqrt{1.09 \times 10^{6}} = 1.04 \times 10^{3}$$

$$\lambda = \sqrt{5.99 \times 10^{6}} = 2.45 \times 10^{3}$$

Substituting these into (i9 ,

$$\frac{W_1(t) - W_2(t)}{|T_1|} = \left[\frac{(1.02 \times 10^{6}) - (1.09 \times 10^{6})}{(1.04 \times 10^{3})(5.99 \times 10^{6} - 1.09 \times 10^{6})} \right] \sin 1.04 \times 10^{3}\, t$$

$$\left[+ \frac{(1.02 \times 10^{6}) - (5.99 \times 10^{6})}{(2.45 \times 10^{3})(1.09 \times 10^{6} - 5.99 \times 10^{6})} \right] \sin 2.45 \times 10^{3}\, t$$

$$= -1.36 \times 10^{-5} \sin 2\pi (165)\, t + 4.14 \times 10^{-4} \sin 2\pi (390) t$$

So there are two components present at the head which could cause flutter, one at 165 cps, the other at 390 cps. The amplitude of the latter exceeds the former by a factor of 30:1.

Compare this with the equal-reel condition, the results of which are given at the bottom of Page 46 . In this case, the two frequencies are 205 cps. and 246 cps, the latter exceeding the former by 21:1

The numbers given on Page   refer to room temperature.

If the temperature is raised to 100^0 C, the E_1 and E_T decrease to $E_1 = E_T \quad 5.0 \times 10^5$. Therefore, all the K's are reduced to 2/3 their room temperature values. For the empty-reel, full-reel case,

$$K_1 = \frac{2}{3} \times 1969 = 1313 \text{ oz. in/rad.}$$

$$K_2 = \frac{2}{3} \times 1126 = 751 \text{ oz. in/rad.}$$

$$K_3 = \frac{2}{3} \times 9760 = 6507 \text{ oz. in/rad.}$$

The J's remain:

$$J_1 = 5.21 \times 10^{-4} \text{ oz. in - sec}^2$$

$$J_2 = 9.58 \times 10^{-3} \text{ oz. in - sec}^2$$

From Pages 47 and 48

$$\alpha_0 = \frac{K_3}{J_2} = \frac{2}{3} \times 1.02 \times 10^6 = 6.80 \times 10^5$$

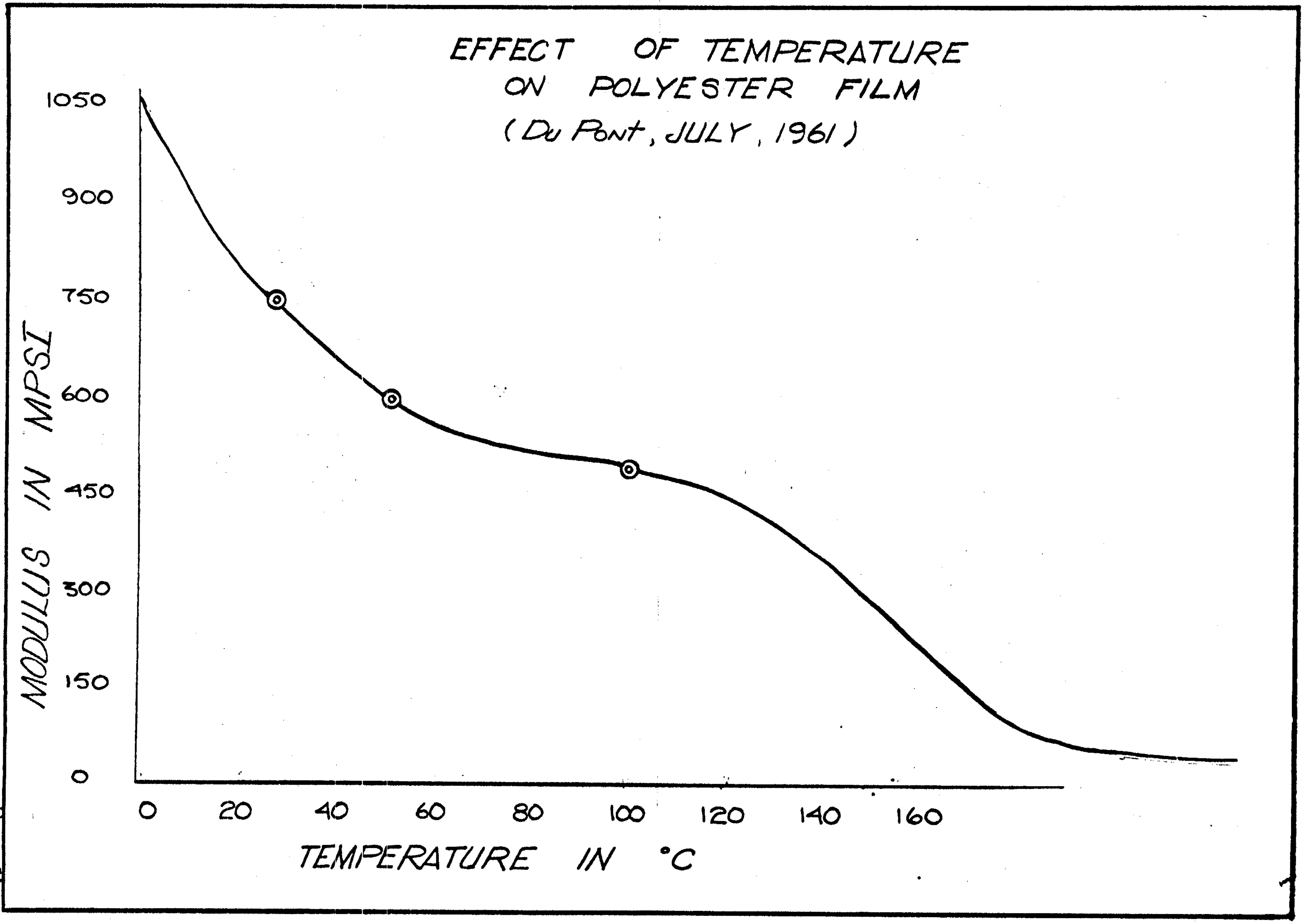
$$A = \frac{1}{2}\left(\left[\frac{K_2 + K_3}{J_2} + \frac{K_1 + K_2}{J_1}\right]\frac{2}{3} \times 3.54 \times 10^6 = 2.38 \times 10^6\right.$$

$$B = \frac{1}{2J_1 J_2}\left[J_1^2 (K_2 + K_3)^2 - 2J_1 J_2\left[(K_2 + K_3)(K_1 + K_2) - 2K_2^2\right]\right.$$

$$\left.\ldots + J_2^2 (K_1 + K_2)^2\right]^{\frac{1}{2}} = \frac{2}{3} \times 2.45 \times 10^6 = 1.63 \times 10^6$$

$$\beta^2 = A - B = .75 \times 10^6 \qquad \beta = .87 \times 10^3$$

$$\gamma^2 = A - B = 4.01 \times 10^6 \qquad \gamma = 2.00 \times 10^3$$


EFFECT OF TEMPERATURE
ON POLYESTER FILM
(Du Pont, JULY, 1961)
MODULUS IN MPSI
1050
900
750
600
450
300
150
0
0
20
40
60
80
100
120
140
160
TEMPERATURE IN °C

Substituting these results into (19

$$\frac{W_1(t) - W_2(t)}{|T_1|} = \frac{a_0 \beta^2 \sin\beta t}{\beta(\lambda^2-\beta^2)} + \frac{a_0 \lambda^2}{\lambda(\beta^2-\lambda^2)} \sin\lambda t$$

$$= \sqrt{2/3}\,(-1.37 \times 10^{-5}) \sin \sqrt{2/3}\; 2\pi(165)t$$

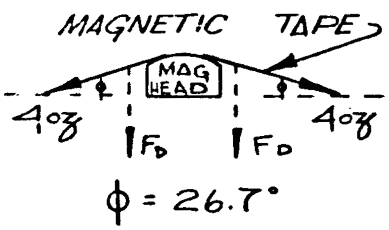
$$+ \sqrt{2/3}\,(4.14 \times 10^{-4}) \sin \sqrt{2/3}\; 2\pi(390)t$$

$$\frac{W_1(t) - W_2(t)}{|T_1'|} = -1.12 \times 10^{-5} \sin 2\pi(135)t + 3.38 \times 10^{-4} \sin 2\pi(318)t$$

The determination of Coulomb friction, T_2, and viscous friction, B_2.

The downward components are

$$F_o = 4 \sin 26.7^o \text{ oz.}$$



The static friction, F_s, is taken as $\frac{1}{3}$. ∴ The frictional force,

$$F_2 = 2F_s F_o = (2)\left(\frac{1}{3}\right)(4 \sin 26.7^o)$$

$$= 1.196 \text{ oz.}$$

This, converted to the appropriate radius, is T_2.

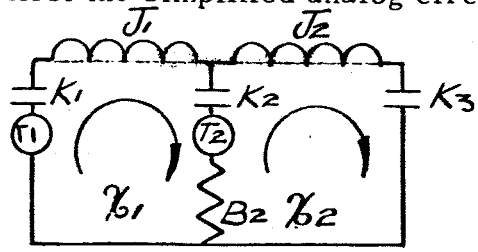
To get B_2, at 3 3/4 ips, the RRR recorder comes to a stop in 0.125 sec. Assuming this takes place in 3 time constants $\frac{J^1}{B}$,

and $J = 2 \times 3.55 \times 10^{-3}$

$$B_2 = \frac{3 J^1}{} = \frac{(3)(2)(3.55 \times 10^{-3})}{0.125} = 0.1703 \ \frac{\text{oz. in}}{\text{rad/sec}}$$

ANALOG COMPUTER SOLUTION

Consider first the simplified analog circuit from P .



$J \approx L$

$K \approx 1/C$

$B \approx R$

$T \approx V$

$\chi \approx q$

Writing the differential equations for this circuit:

$$T_1 = K_1 \chi_1 + J_1 \frac{d^2 \chi'}{dt^2} + K_2 \chi_1 - (\chi_1 - \chi_2) + T_2 + B_2 \left(\frac{d\chi_1}{dT} - \frac{d\chi_2}{dt} \right)$$

$$T_2 = K_2(\chi_2 - \chi_1) + J_2 \ddot{\chi}_2 + K_3 \chi_2 + B_2 (\dot{\chi}_2 - \dot{\chi}_1)$$

Rearranging:

$$\ddot{\chi}_1 = -\frac{1}{J_1} \left\{ (T_2 - T_1) + \chi_1 (K_1 + K_2) - \chi_2 K_2 + B_2 (\dot{\chi}_1 - \dot{\chi}_2) \right\}$$

$$\ddot{\chi}_2 = \frac{1}{J_2} \left\{ -T_2 + \chi_2 (K_2 + K_3) - \chi_1 K_2 + B_2 (\dot{\chi}_2 - \dot{\chi}_1 \right\}$$

The value of the coeffs. from Pg. 44, 45:

$K_1 = 6.04 \times 10^3$; $K_2 = 1.29 \times 10^3$; $K_3 = 5.82 \times 10^3$ in. oz/rad.

for 1/2 tape run $J_1 = J_2 = 3.55 \times 10^{-3}$ oz. in sec^2

from Pg. 52 B_2 = .17 oz in/rad/sec

The J values depend on the amount of tape on each pack, and vary from 10^{-2} to 5×10^{-4}.

The K's also depend on tape position but vary only slightly from full-empty to empty-full and are presently assumed constant. Substituting all but J_1, J_2 in the equations:

$$\ddot{\chi}_1 = -\frac{1}{J}\{(T_2 - T_1) + 7.33\times10^{3}\chi_1 - 1.29\times10^{3}\chi_2 + .17(\dot{\chi}_1 - \dot{\chi}_2)\}$$

$$\ddot{\chi}_2 = -\frac{1}{J_2}\{-T_2 + 7.11\times10^{3}\chi_2 - 1.29\times10^{3}\chi_1 + .17(\dot{\chi}_2 - \dot{\chi}_1)\}$$

Since the paper recorder cannot operate at $>$ 10 cps, we scale the time by a factor of 100, so that: machine time τ = 100 t.

To prevent any large values of the variables from peaking the amplifiers in the computer we scale χ by $\frac{1}{100}$, so that:

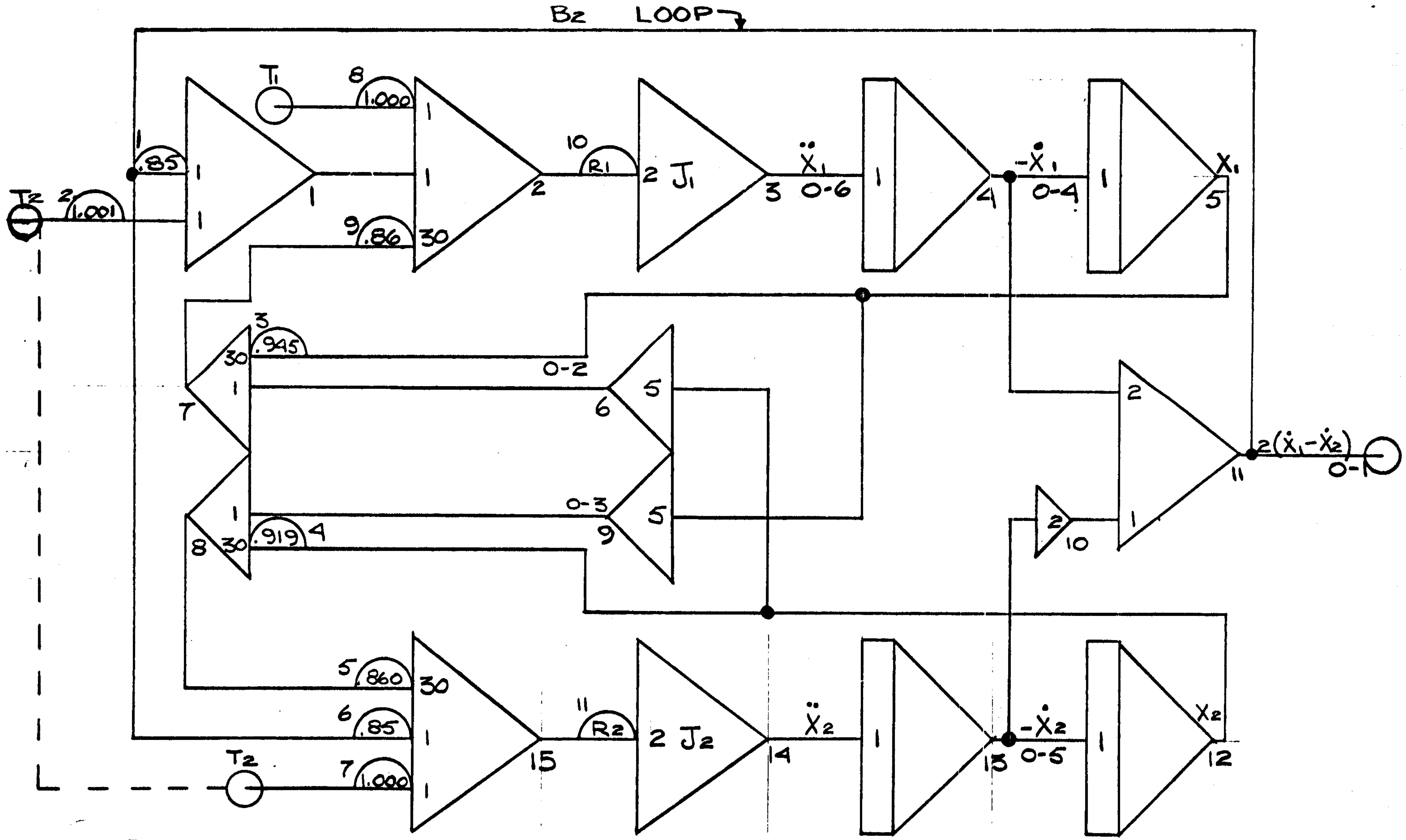
machine $X = \frac{1}{100}\chi$

Now $\ddot{X}_1 = -\frac{10^{-3}}{J_1}\{10^{-3}(T2 - T_1) + 733X_1 - 129X_2 + 1.7(\dot{X}_1 - \dot{X}_2)\}$

$$\ddot{X}_2 = -\frac{10^{-3}}{J_2}\{-10^{-3}T_2 + 711X_2 - 129X_1 + 1.7(\dot{X}_2 - \dot{X}_1)\}$$

is the system to be run through the computer as shown by the program on the following page. Tape position (as it influences J) may be changed by adjusting the values of pot. s #10 and 11 according to a set schedule. B_2 may be included in any amount by adjusting pot. s #1 and 6. Changes in tape modulus may be accommodated by adjusting pot. s #5 and 9.

ANALOG SOLUTION

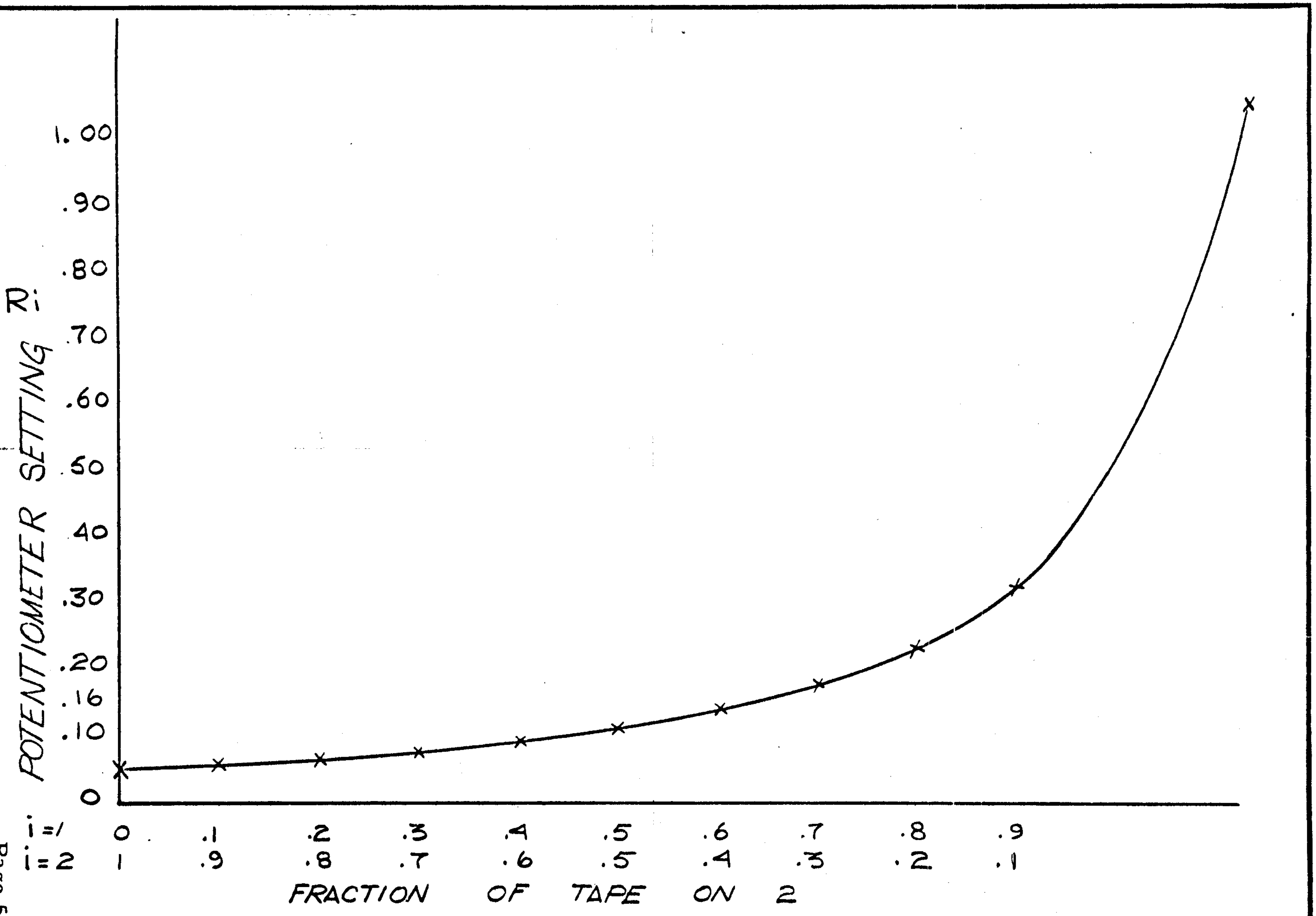


$$\ddot{X}_1 = -\frac{10^{-3}}{J_1}\left[10^{-3}(T_2 - T_1) + 733X_1 - 129X_2 + 1.7(\dot{X}_1 - \dot{X}_2)\right]$$

$$\ddot{X}_2 = \frac{10^{-3}}{J_2}\left[-10^{-5}T_2 + 711X_2 - 129X_1 + 1.7(\dot{X}_2 - \dot{X}_1)\right]$$

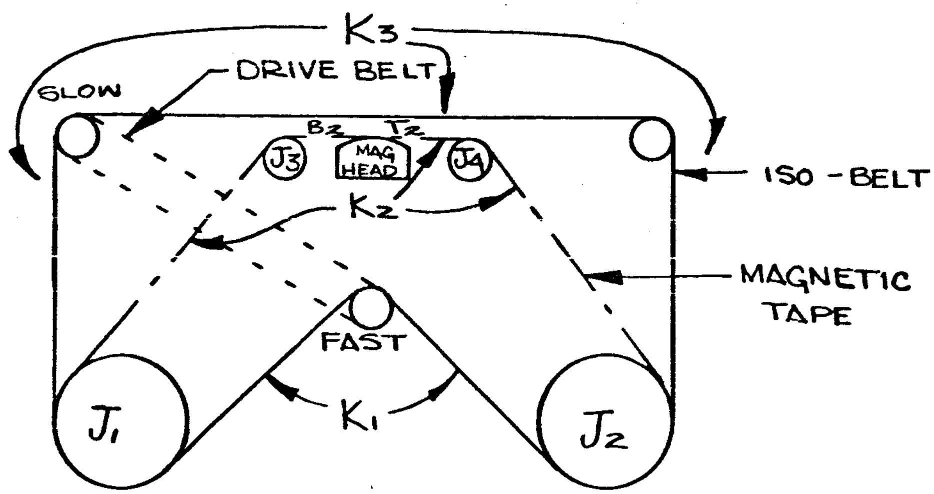
$$2(\dot{X}_1 - \dot{X}_2) = 2\times10^{-4}(\dot{Y}_1\ \dot{Y}_2)$$

MACHINE REAL

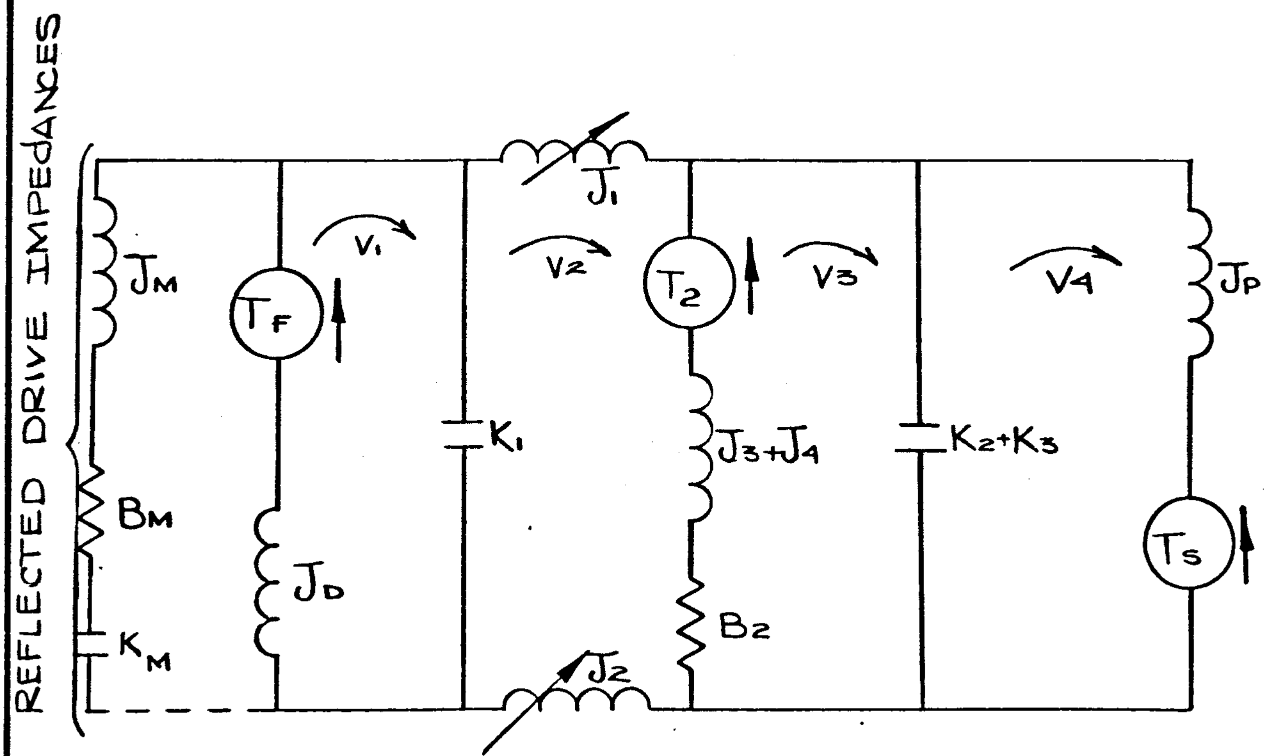


POTENTIOMETER SETTING R_i
1.00
.90
.80
.70
.60
.50
.40
.30
.20
.16
.10
0
i=1 0 .1 .2 .3 .4 .5 .6 .7 .8 .9
i=2 1 .9 .8 .7 .6 .5 .4 .3 .2 .1
FRACTION OF TAPE ON 2

From fundamental considerations, a more accurate model of the RRR recorder is shown below



The electrical equivalent is



29 SO. PASADENA AVE., PASADENA, CALIFORNIA

FIG. 1. LAPLACE TRANSFORM SOLUTION FOR CASE 1

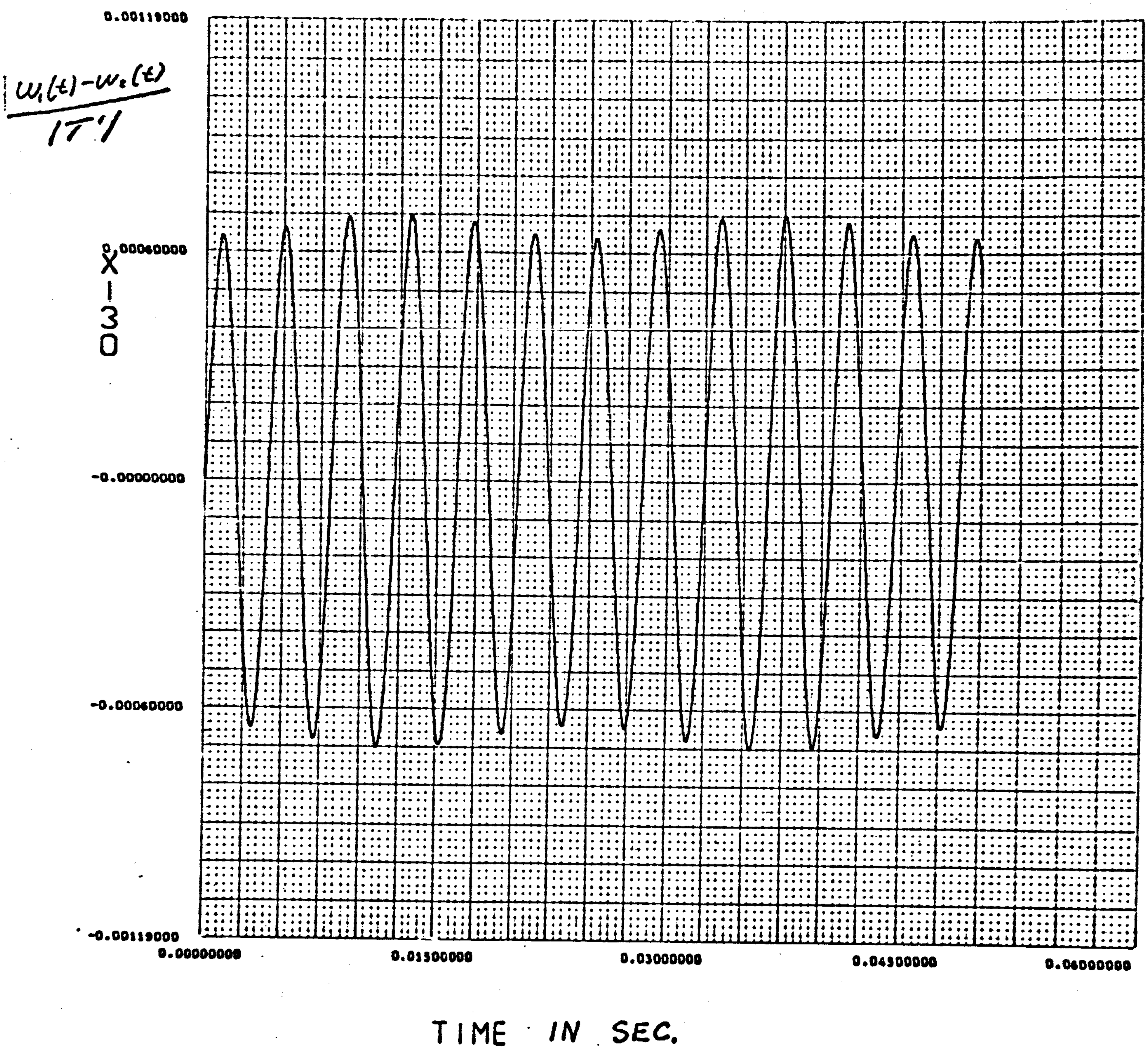


FIG. 2. 7094 TAG PROGRAM Results (Digital computer solution for CASE I)

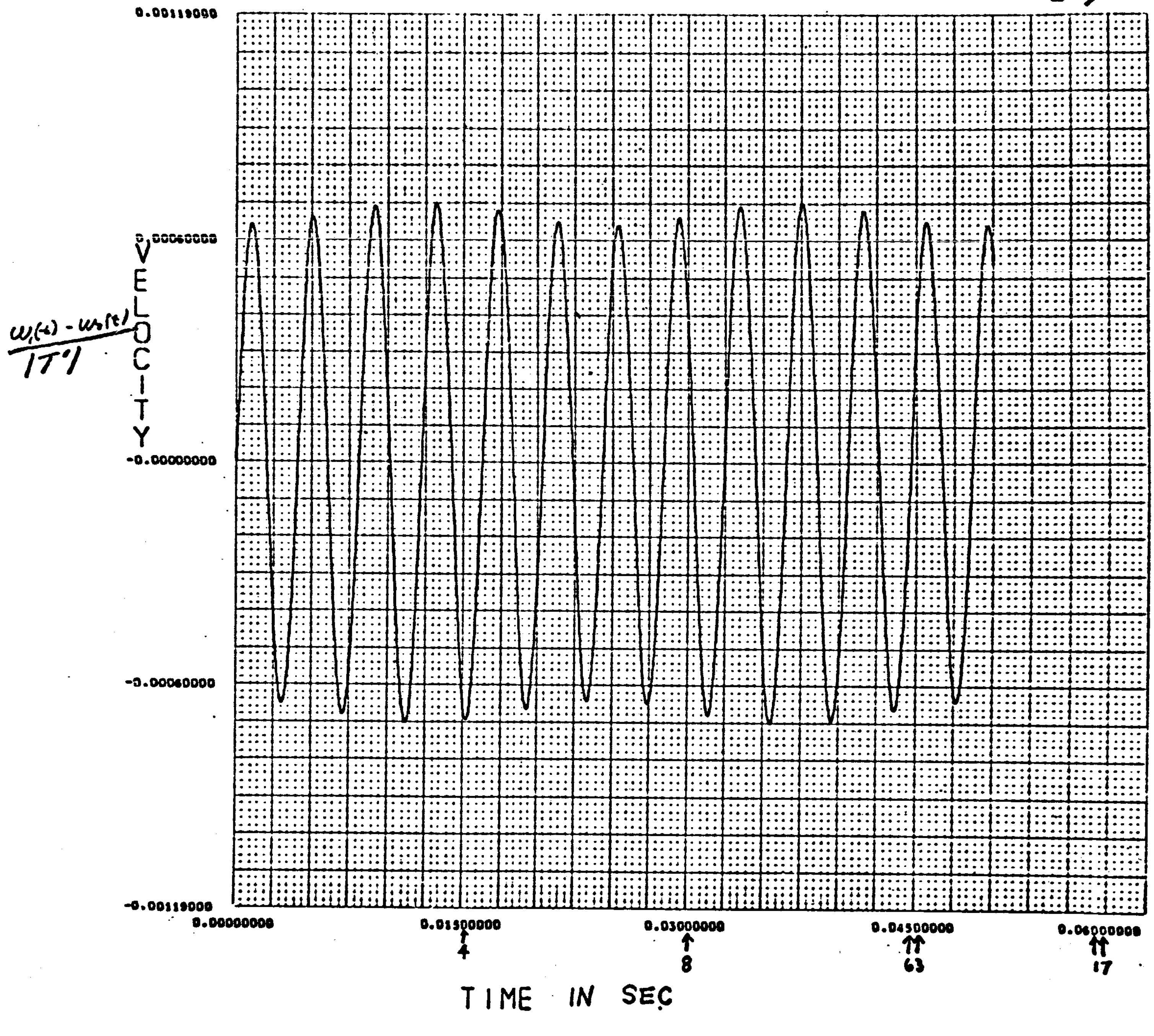
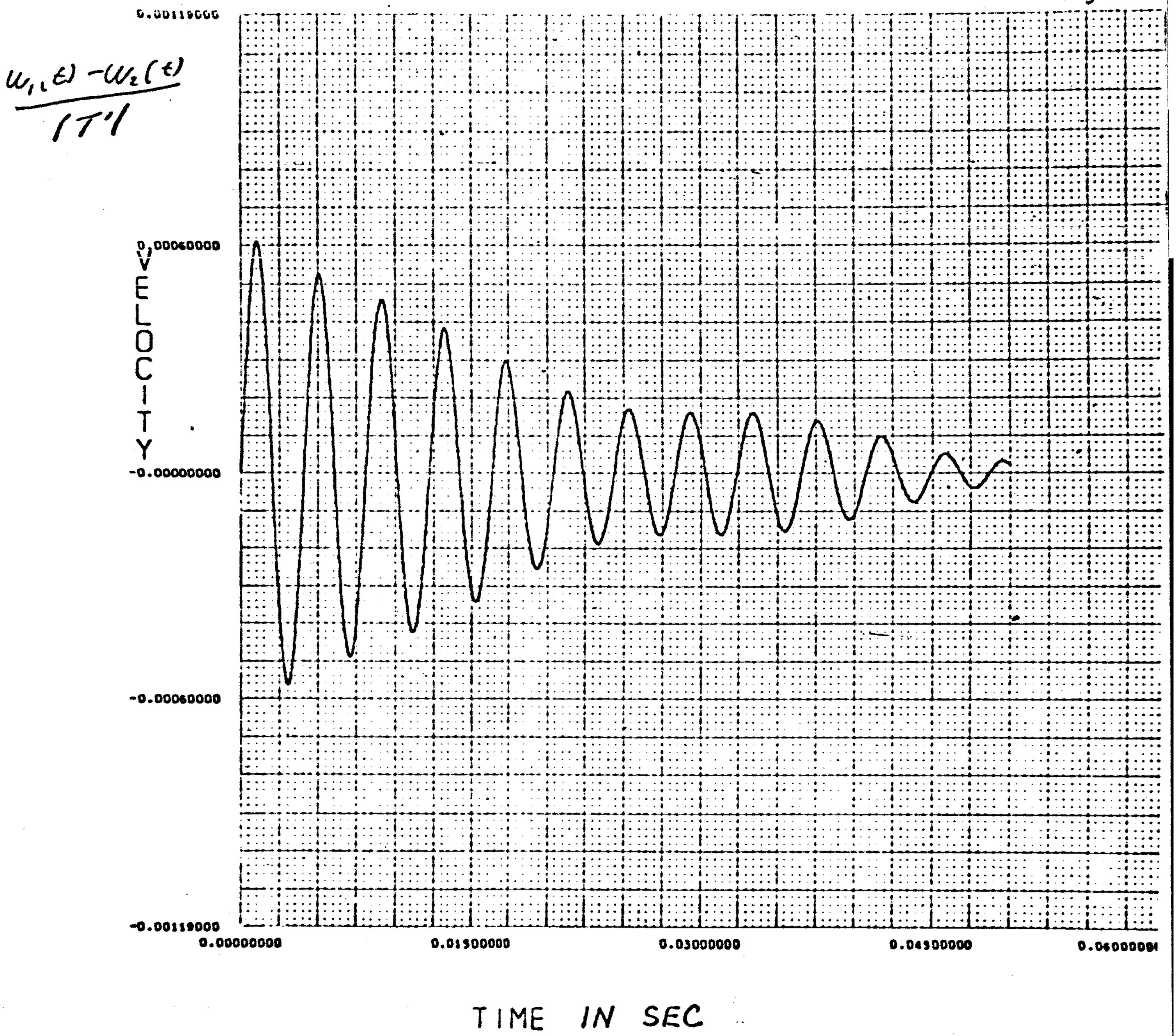


FIG. 3. 7044 TAG PROGRAM RESULTS (Digital computer solution for CASE 4)



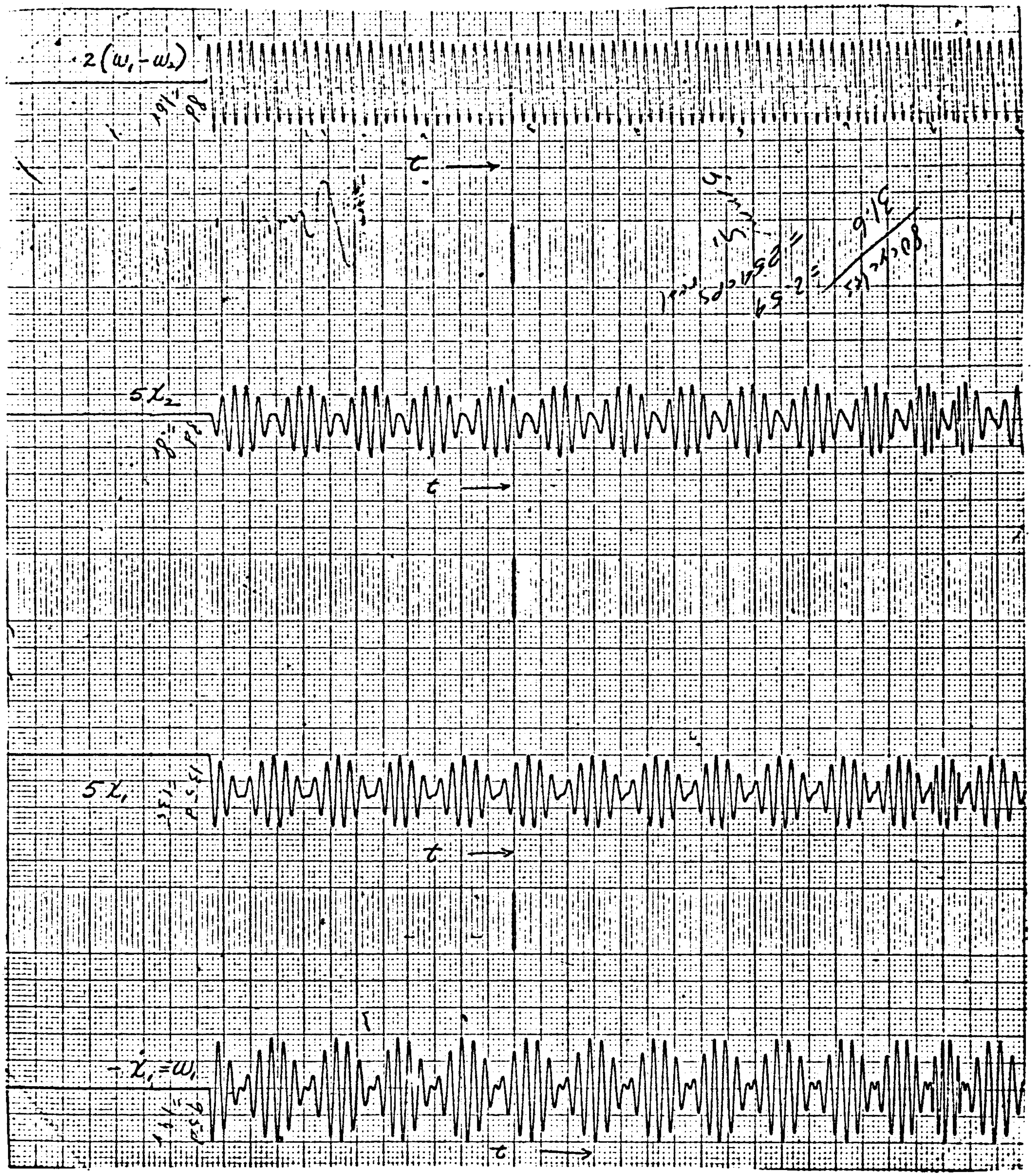


FIG. 4. Analog computer results for undamped, equal-reel case.

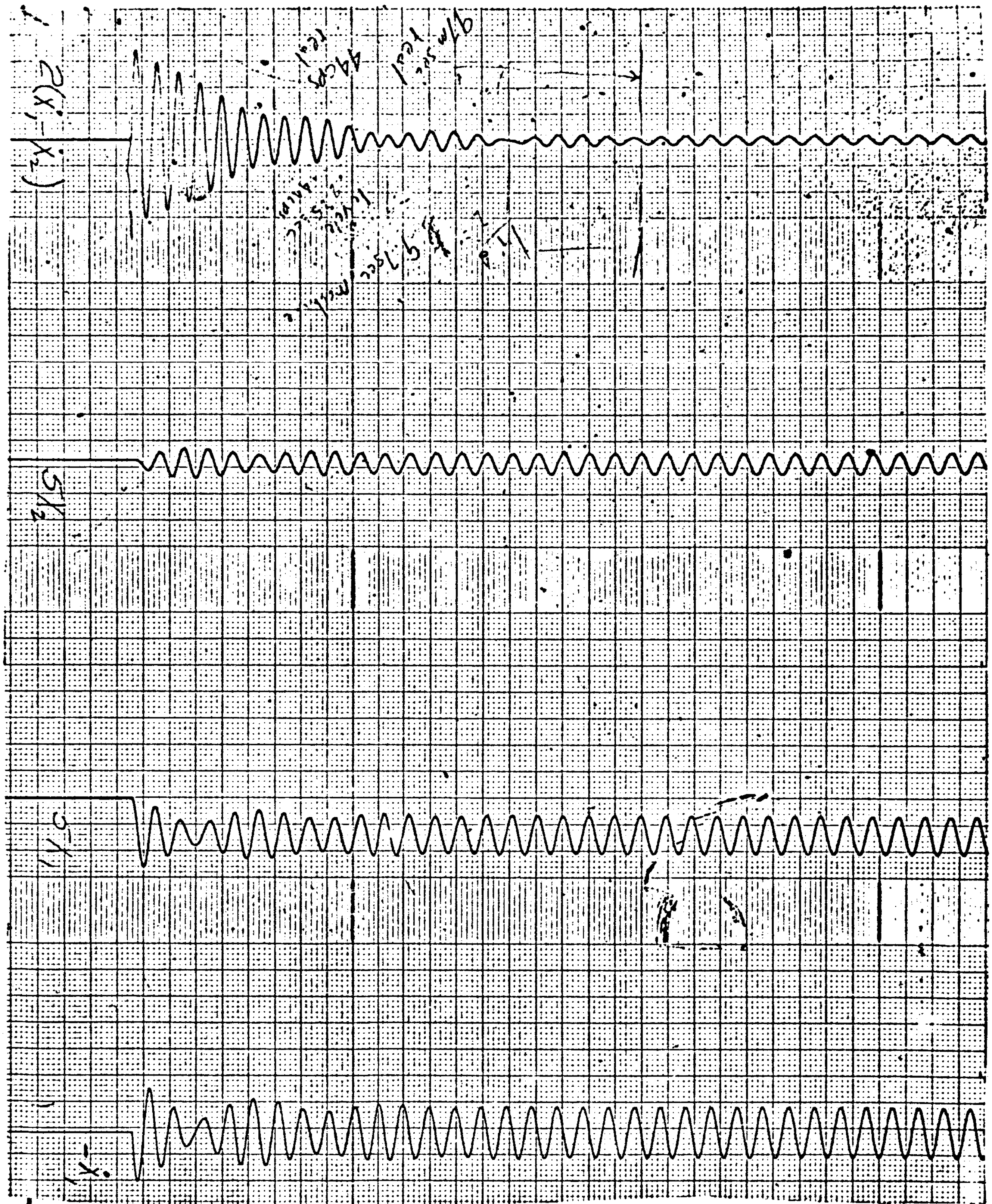


FIG. 5. Analog computer results for damped, equal-reel case.

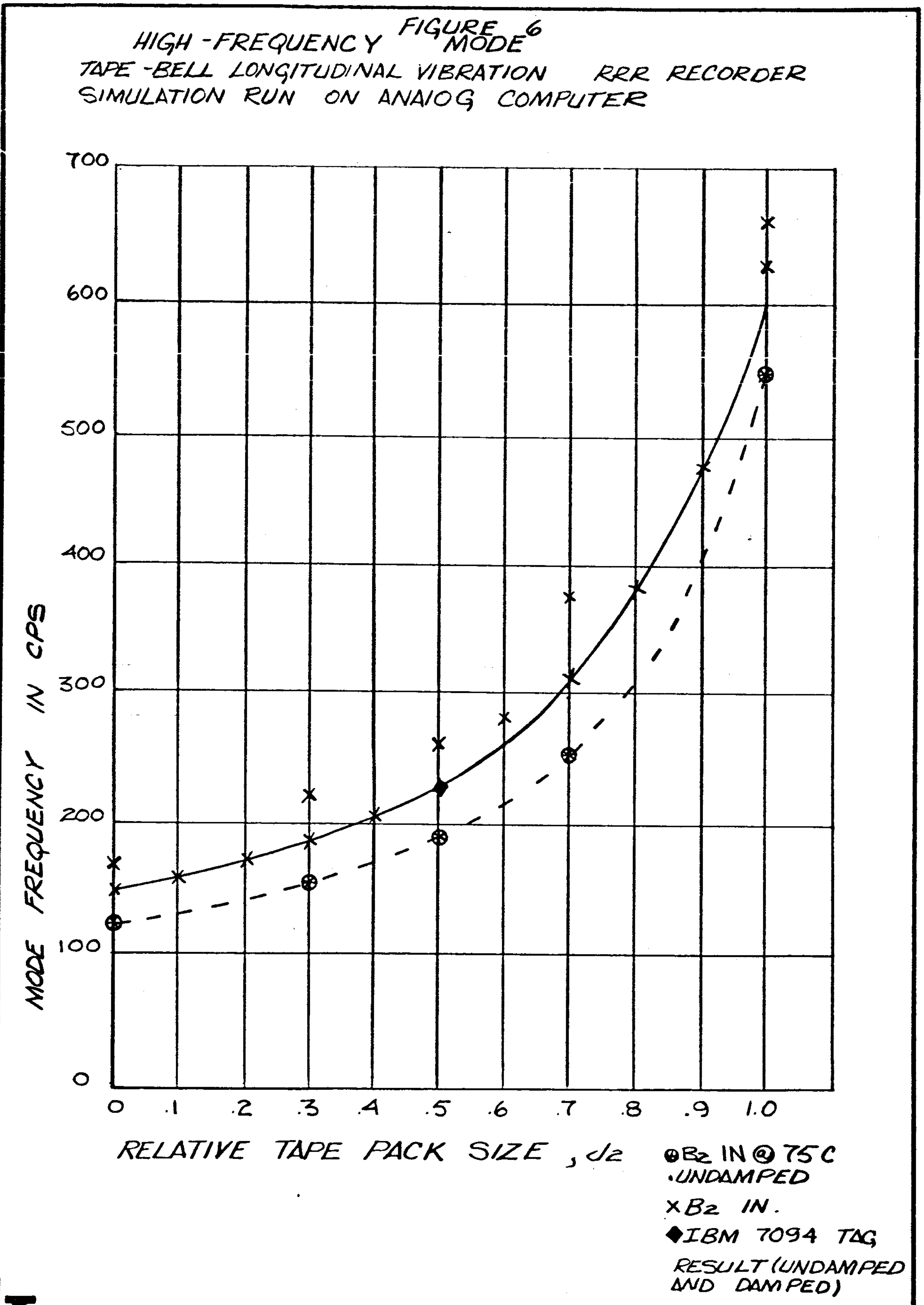


FIGURE 6
HIGH-FREQUENCY MODE
TAPE-BELL LONGITUDINAL VIBRATION RRR RECORDER
SIMULATION RUN ON ANAIOG COMPUTER
700
600
500
400
300
200
100
0
MODE FREQUENCY IN CPS
0
.1
.2
.3
.4
.5
.6
.7
.8
.9
1.0
RELATIVE TAPE PACK SIZE, α_2
⊗ B_2 IN @ 75C
· UNDAMPED
× B_2 IN.
◆ IBM 7094 TAG
RESULT (UNDAMPED AND DAMPED)

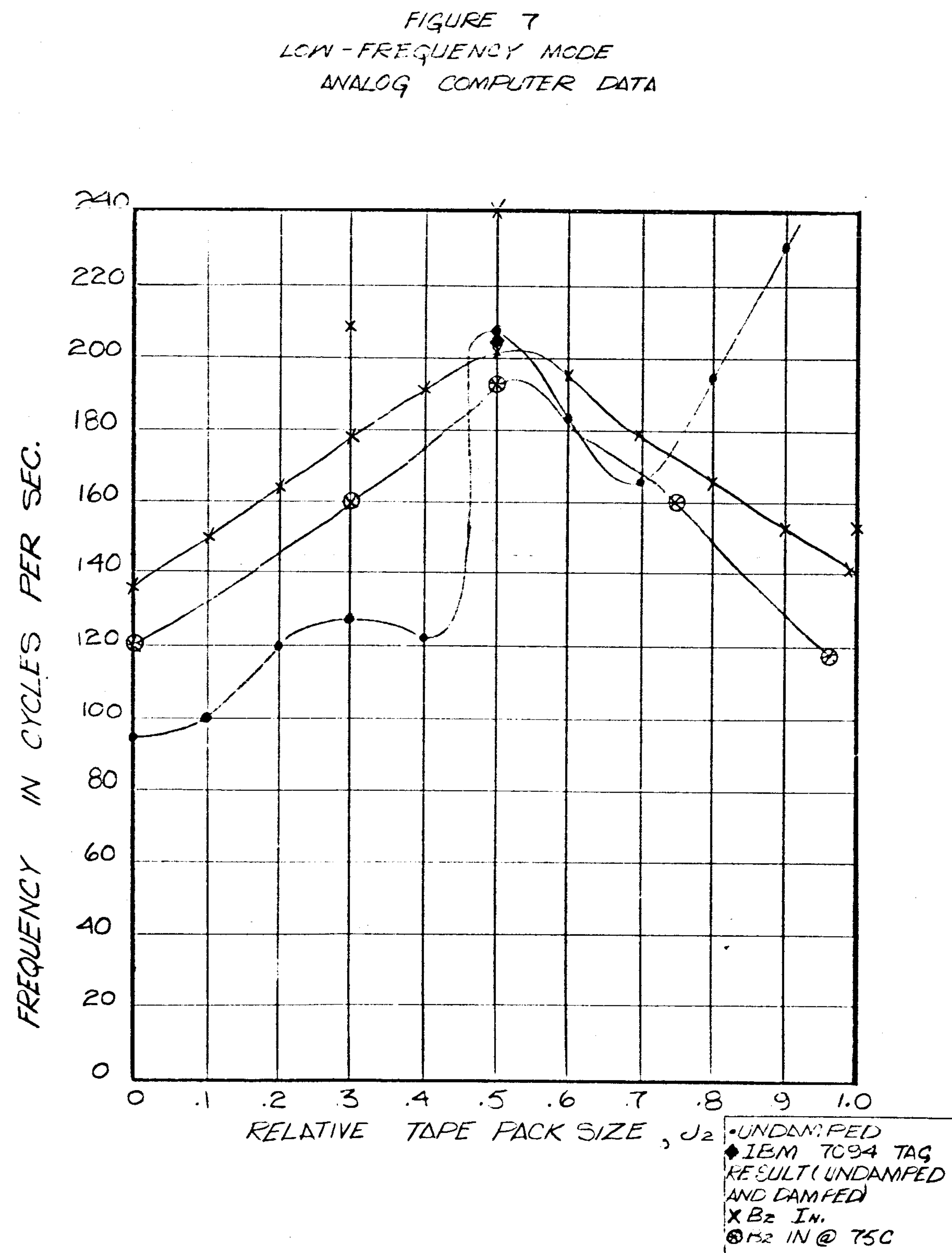


FIGURE 7
LOW - FREQUENCY MODE
ANALOG COMPUTER DATA
FREQUENCY IN CYCLES PER SEC.
240
220
200
180
160
140
120
100
80
60
40
20
0
0
.1
.2
.3
.4
.5
.6
.7
.8
.9
1.0
RELATIVE TAPE PACK SIZE, J_2
• UNDAMPED
◆ IBM 7094 TAG RESULT (UNDAMPED AND DAMPED)
X B_Z IN.
⊗ B_Z IN @ 75C

6.2 TEST FAILURE REPORTS AND FAILURE MODE ANALYSIS:

TABLE 1

TYPE OF FAILURE

A. Loosening, lightening, or slipping

B. Deforming (permanent)

C. Material transfer or wear

D. Breaking

E. Chemical breakdown or combination

F. Oscillating

G. Loss of signal

H. Dynamic (intermittent or temporary)

I. Adhesion, or loss of adhesion

TABLE 2

ENVIRONMENTAL FACTORS

I. Temperature

II. Vibration

III. Shock

IV. Acceleration or free fall

V. Sealed Environment

VI. Atmosphere

VII. Radiation

VIII. Electrical fields or charges

IX. Magnetic Fields

X. Particle Impact

TABLE 3

SUBASSEMBLIES AND FAILURE MODES OF TAPE RECORDER

A REEL ASSEMBLY

- α - Is not aligned
- β - Turns erratically
- γ - Does not turn
- δ - Broken off

B HEAD ASSEMBLY

- α - Guide Roller misaligned
- β - Guide Roller turns erratically
- γ - Guide Roller does not turn
- δ - Some tracks do not record or playback
- ϵ - Excessive skew
- ζ - No output signal
- η - Head tilted
- θ - Assembly tilted
- κ - Guide Roller broken off
- λ - False signals

C TAPE GUIDE ROLLER ASSEMBLY (EOT)

- α - Misaligned
- β - Turns erratically
- γ - Does not turn
- δ - Broken off

D END-OF-TAPE SENSOR

- α - False Signals
- β - No Signals
- γ - Misaligned (tape drags)

E FAST CAPSTAN ASSEMBLY (11 Items)

- α - Is not aligned
- β - Turns erratically
- γ - Does not turn

F SLOW CAPSTAN ASSEMBLY

- α - Misaligned
- β - Turns erratically

29 SO. PASADENA AVE., PASADENA, CALIFORNIA

TABLE 3 Continued

F SLOW CAPSTAN ASSEMBLY (Continued)

γ- Does not turn
δ- Incorrect speed differential

G. TAPE MOTION SENSOR ROLLER ASSEMBLY

α- Misaligned
β- Turns erratically
γ- Does not turn
δ- Broken off

H ISO-BELT TENSION ARM ASSEMBLY

α- Misaligned
β- Turns erratically (roller)
γ- Swings erratically
δ- Does not turn (roller)
ϵ- Does not swing
ζ- Incorrect tension
η- Broken off

J ISO-BELT GUIDE ROLLER ASSEMBLY

α- Misaligned
β- Turns erratically
γ- Does not turn
δ- Broken off

K MOTOR

α- Misaligned
β- Turns erratically
γ- Does not turn
δ- Incorrect speed (out of sync)
ϵ- Overheats

L DRIVE BELT

α- Broken
β- Mistracks
γ- Does not drive (excessive drag)
δ- Drives erratically

M CAPSTAN INTERCONNECTING BELT
α- Broken
β- Mistracks
γ- Does not drive
δ- Drives erratically

TABLE 3 Continued

N ISO-ELASTIC BELT

- α- Broken
- β- Mistracks
- γ- Does not drive (excessive drag)
- δ- Drives erratically

O HOUSING AND COVER ASSEMBLY

- α- Press lost
- β- Warped permanently
- γ- Warped dynamically
- δ- Vibrates (resonates)

P MAGNETIC TAPE

- α- No signal on tape
- β- Tape mistracks
- γ- Falls off reel hub
- δ- Oxide spalls or blocks
- ε- Layer-to-layer adhesion
- ζ- Tape breaks
- η- Incorrect tape speed and false signals

TABLE 4
RESULTS OF FAILURE MODE SCORING, RRR RECORDER

DESIGNATION	DESCRIPTION Component, Failure Mode	SCORING Imp.	SCORING Occ.	SCORING Tot.	RANKING Imp.	RANKING Occ.	RANKING Tot.
D-β	End-of-Tape sensor, no signals	9	24	33	2a	3a	1
E-β	Fast capstan ass'y, turns erratically	15	21	36	7a	1a	2a
P-α	Magnetic Tape, no signal	8	28	36	1a	7a	2b
L-α	Drive belt, broken	8	29	37	1b	8a	3a
E-γ	Fast capstan ass'y, does not turn	8	29	37	1c	8b	3b
A-α	Reel ass'y, not aligned	17	21	38	9a	1b	4a
P-β	Magnetic tape, mistracks	14	24	38	6a	3b	4b
F-α	Slow capstan ass'y., does not turn	8	30	38	1d	9a	4c
B-$\mathcal{E}$	Head ass'y., no output signal	8	30	38	1e	9b	4d
K-γ	Motor, does not turn	8	30	38	1f	9c	4e
D-α	End-of-tape sensor, false signals	16	23	39	8a	2a	5a
F-δ	Slow capstan ass'y., incorrect speed diff.	13	26	39	5a	5a	5b
L-δ	Drive belt, doesn't drive	8	31	39	1g	10a	5c
A-γ	Reel ass'y., doesn't turn	8	31	39	1h	10b	5d
A-β	Reel ass'y., turns erratically	16	24	40	8b	3c	6a
F-β	Slow capstan ass'y., turns erratically	17	23	40	9b	2b	6b
P-γ	Magnetic tape, falls off reel	10	30	40	3a	10c	6c
K-δ	Motor, out of synchronism	16	25	41	8c	4a	7a
H-δ	Iso-belt tension arm ass'y., roller doesn't turn	10	31	41	3b	11a	7b
N-δ	Iso-belt, drives erratically	18	24	42	10a	3d	8a
P-δ	Magnetic tape, oxide spalls or blocks	17	25	42	9c	4b	8b
B-δ	Head ass'y., incomplete recording &/or playback	12	30	42	4a	10d	8c

Table 5 RRR FAILURES

MACHINE NO.	DATE 1964	PURPOSE	CONDITIONS	ELEMENT	
1.	5/14	Tape Curing -stick test	Vacuum, 75° C.	Tape Guides	2
	5/18	Start-Stop Mode	Recorder Pressurized - room temperature	Guide Flanges Head Ass'y.	T m
		Outgassing check			
	5/25		Vacuum, 78° C.	Tape	Ta to
	7/14		Vacuum, 80° C.	Iso Belt	Is of

NO RECORD OF FAILURES BEYOND HERE

MACHINE NO.	DATE	PURPOSE	CONDITIONS	ELEMENT	
2.	10/1	Machine Check Out	(room conds)	Motor	No
	10/7			E.O.T.S.	Ta
	10/12			E.O.T.S.	Sl at

#1

SYMPTOMS	CAUSE	Accumulated Hours Before Failure Machine	Element
Guides Frozen	Bearing Failure	28	28
ape mistracking achine jammed	Perpendicularity of guides off; improper guide heights.	4	32
ape sticking heads	Bad tape	21	53
o Belt slips f guides	Not recorded	24	77
isy Bearings	Too little lubrication	Not Recorded	Not Rec.
e jammed	Erratic reverse signal near end of tape; last turns slipped on hub.	2	2
ggish reversals end of tape.	Idler before sensor rough; oxide deposited on sensor windows.	16	16

#2

nt

FIX	REMARKS
Replace Bearings	Cause of Bearing Failure Not Determined
Align and adjust guides.	Heads also aligned; tension arm adjusted.
Replace tape with new 951	Heads may have exuded some epoxy to cause this.
Not recorded	Temperature went to 90°C. up to 21 hours without heads
Replace and properly lube bearings	This operation is the first noted.
Filter capacitor changed from .005 μf $\rightarrow$ .001 μf	Real cause and fix may be same as below.
Clean windows, repolish idler.	Unit shipped now to JPL

H3

MACHINE NO.	DATE	PURPOSE	CONDITIONS	ELEMENT
	1964			
3	7/2	Tape Curing Outgassing	Vacuum, 78° C.	Tape
	Break			
	7/16	same	Vacuum, 80° C. continuous run	Iso Belt Idler
	7/20	same	same	Iso Belt Idler
	7/21	same	same	E.O.T.S.
	7/29	same	same	E.O.T.S.
	8/7	Machine Life	75° C.	Motor
	8/10	same	same	Tape Heads pack on hubs
	8/14	Outgass check	Vacuum, 70° C.	Motor
	8/17	same	Vacuum, 75° C.	Motor
	8/17	same	same	Motor
	RECORDER CHECKED AND ALIGNED HERE. NEW ISO BELT INSTALLED HERE			
	9/10	Machine check clock track record	Room conds	Tape

SYMPTONS	CAUSE	Accumulated Hours Before Failure Machine	Eleme
Tape jammed	Not recorded		
Belt Left Guide and Capstan	Same as below	22	22
Belt Life Guide and Capstan	Loose Belt idler	3	3
Run off end of tape	EOTS failed to actuate relay (reason below)	7	32
Run off end of tape	EOTS failed to actuate relay. Window too short	40	40
Unit stalled	Increased friction load overloaded motor	36	108
Freeloop of tape jams machine	Tape absorbs substance from heads excessive pressure in pack during heating causes blocking which upsets tape reeling.	16	124
Erratic starting	Increased friction load overloaded motor	3	19
Stalled	Overloaded motor	14	14
stalled	Overloaded motor	11	11
Tape slack and mistracking	Attributed to low tape tension	~12	164

#2

Fix	REMARKS
nt	
Not recorded	Two week break in record following this
Same as below	Proper cause not discovered here
Loctite roller in place	This is problem not corrected 7/16
None attempted	Same problem below
Window lengthened to 2 feet	This is same as 7/16 problem
Increase motor voltage→15 V	Is this motor too marginal at low volts
New tape installed	Perhaps the heads should be cured.
Increase motor voltage→12 V	Same problem again
Increase motor voltage→13 V	and again
Increase voltage to start, then return to 13 V	and again
Increase differential .22→.31	Machine has many hours at this tension; why is it too low

#3

MACHINE NO.	DATE	PURPOSE	CONDITIONS	ELEMENT	S
3 continued	9/14	Machine check	Room conds	Tape	Tape as t
	9/25	Life Test	Room conds	Motor	Mach
	10/17	Life Test	Room conds	E.O.T.S.	Run of t
	10/20	Life Test	Room conds	E.O.T.S.	Run of t
	10/21	Life Test	Room conds	Motor	Mach
	10/25	Life Test	Room conds	Drive Belt	Belt
	BREAK IN TEST RECORDS HERE. UNIT RECONDITIONED; SOME BEARINGS REPLAC				
	11/24	Life Test	Room conds	Iso Belt	Belt
	12/5	Life Test	Room conds	Iso Belt	Belt
	12/8	Life Test	Room conds	Motor	Mach
	12/14	Life Test	Room conds	Iso Belt	Belt

#1

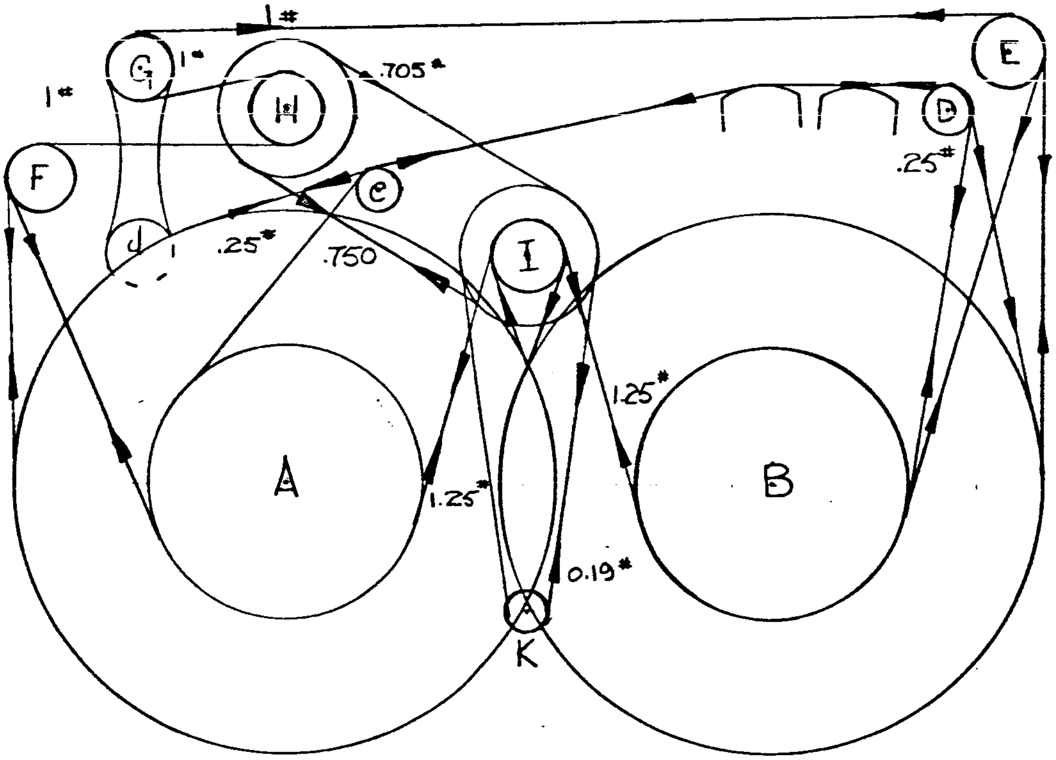
YMPTOMS	CAUSE	Accumulated Hours Before Failure Machine	Accumulated Hours Before Failure Element
mistracking efore	Damaged tape ?	2	42
ine stalled	Motor bearings frozen; improper lubrication	13	179
off end ape	Sensor transistor failure ?	93	272
off end pe	Noise signal to control rack causing re-start in same direction	3	See remark
ine stalled	Oxide buildup on motor shaft caused drive belt to jam	22	297
Broken	Fatigue, previous damage during failure	90	387
ED; NEW DRIVE BELT,	NEW CAPSTANS		
Broken	Fatigue	19	234
Broken	Fatigue	56	56
ine stalled	Overloaded Motor	26	309
Broken	Fatigue	78	104

#2

Fix	REMARKS
Replace tape	Tape did not appear damaged on inspection.
Replace and properly lube bearings	This happened on No. 2 also. Perhaps previous low voltage failures were due to motor bearing deterioration.
Replace transistor	Next cause may be the real problem
Decoupling diodes added to external circuit	This may be the solution to above failure. This is not a machine failure.
Clean shaft and belt	Motor drive belt upper edge scuffed during this operation
Replace belt	This is the damaged belt that broke
Replace Belt	Old Belt used as Replacement
Replace Belt	This is the old belt which broke
Increase voltage 25 V → 27 V	Cause of overload unknown
Replace Belt	Should we look for cause of short life

6.3 LOAD ANALYSIS

Figure 8 Normal Service Loads in Belts and Tape for RRR



6.3 (Continued)

TABLE 6

MAXIMUM SERVICE LOADS ON BEARINGS

Bearing Pair	Maximum Service Load (pounds) Upper	Lower
A	3.13	0.63
B	3.13	0.63
C	.06	.06
D	.187	.187
E	.85	.85
F	.85	.85
G	1.0	1.0
H	2.34	1.74
I	4.25	1.75
J	7.0	1.5
K	.76	.14

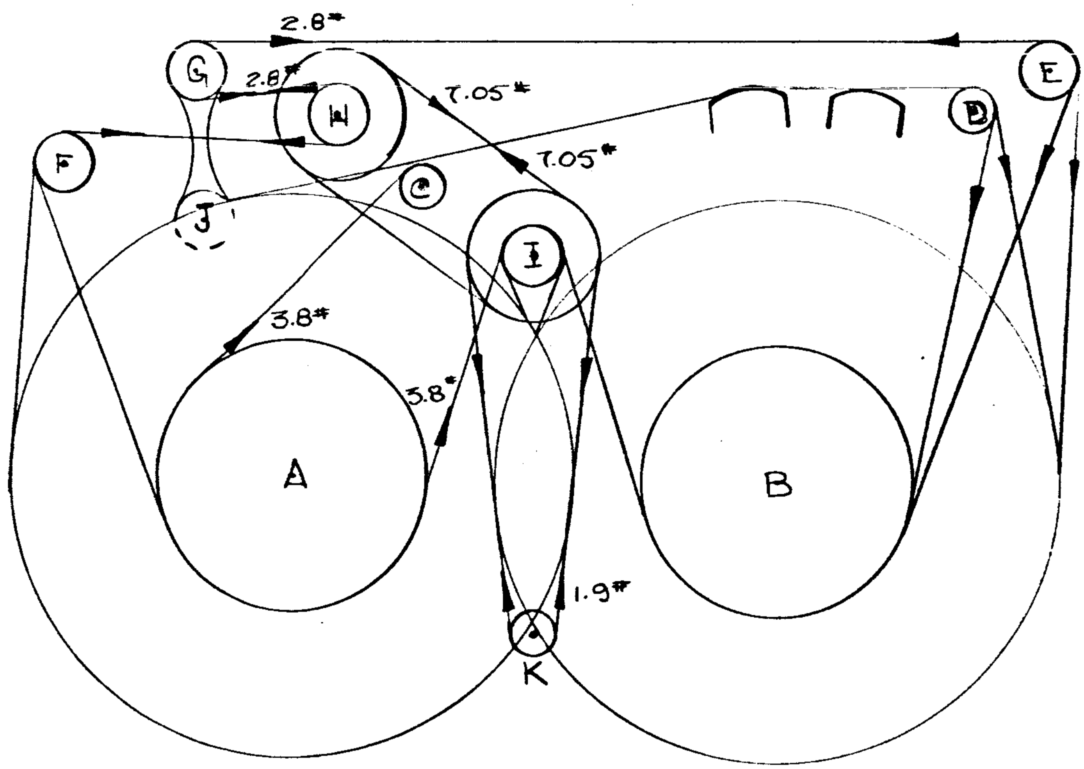
TABLE 7

BELT AND TAPE TENSIONS AT YIELD POINT

Component	Width	Thickness	Yield Load (lbs)
Iso-Elastic Drive Belt	0.187 in.	0.001 in.	2.8
Capstan Interconnecting Belt	0.094 in.	0.005 in.	7.05
Transmission Belt	0.125 in.	0.001 in.	1.9
Tape	0.25 in.	0.001 in.	3.8

6. 3 (Continued)

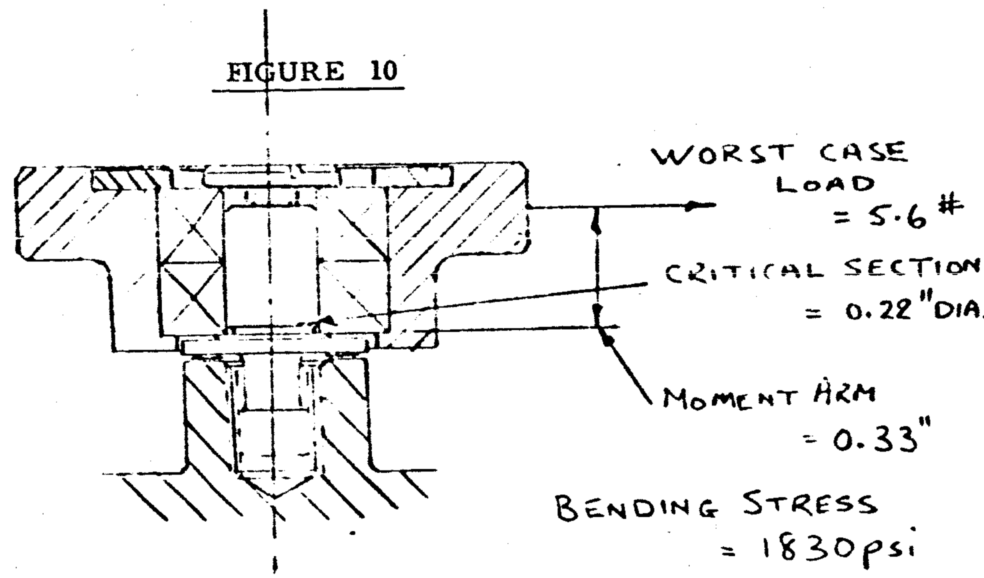
Figure 9 Worst Case Loads for Belts and Tape on RRR



6.4 LOAD ANALYSIS - REEL HUB ASSEMBLY.

LOCATION A + B TYP.

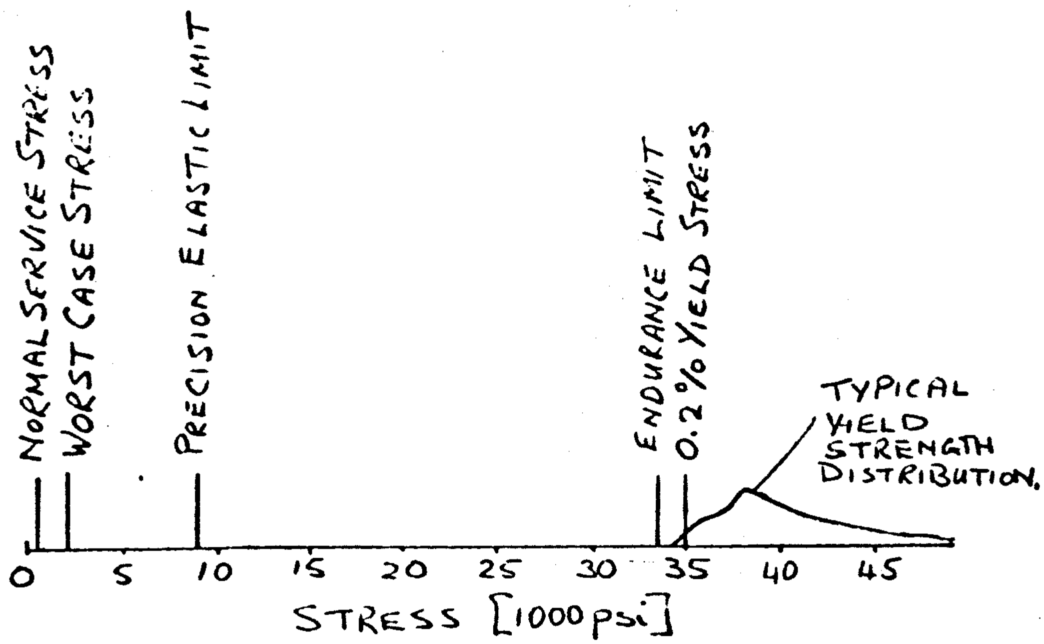
FIGURE 10



BENDING STRESS = 1830 psi

SHEAR STRESS = 147 psi.

TENSILE STRESS = 40 psi

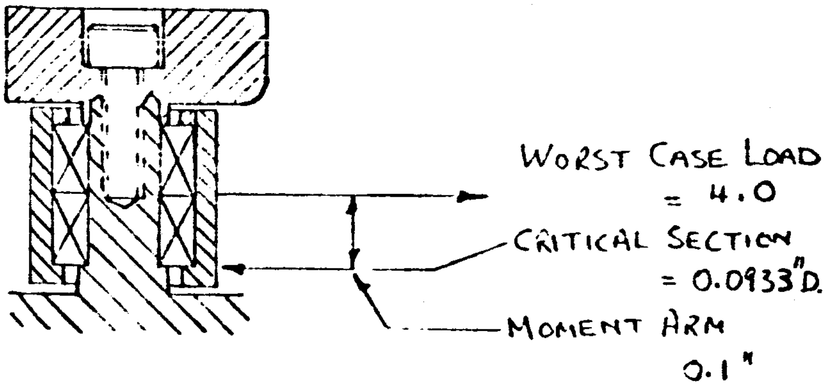


SCALE 2/1

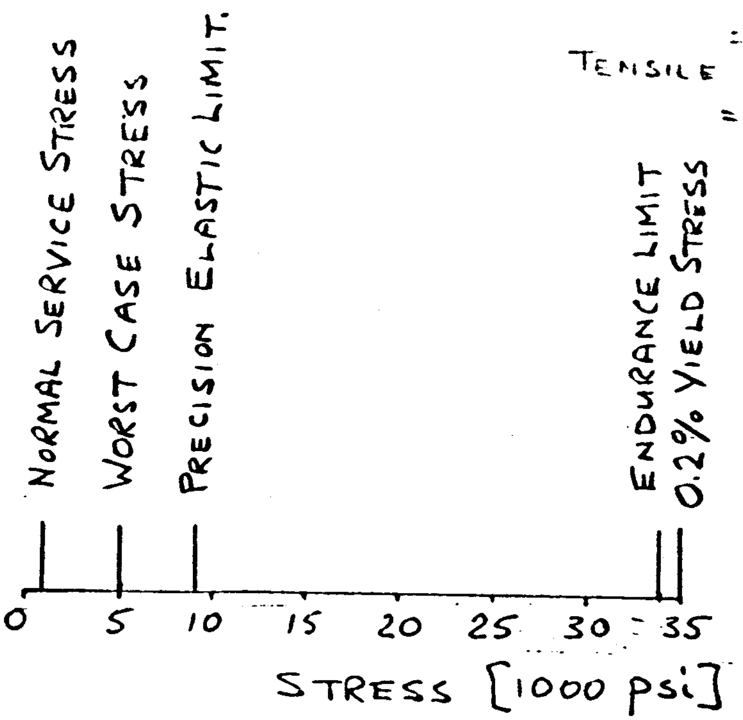
LOAD ANALYSIS - GUIDE ROLLER ASSY.

LOCATION C & D TYP.

FIGURE 11



BENDING STRESS
= 5,000 psi
SHEAR STRESS
= 585 psi.
TENSILE STRESS
= 157 psi



SCALE 4/1

FIGURE 12 LOAD ANALYSIS - BELT IDLER ASSY.

LOCATION E, F + G TYP.

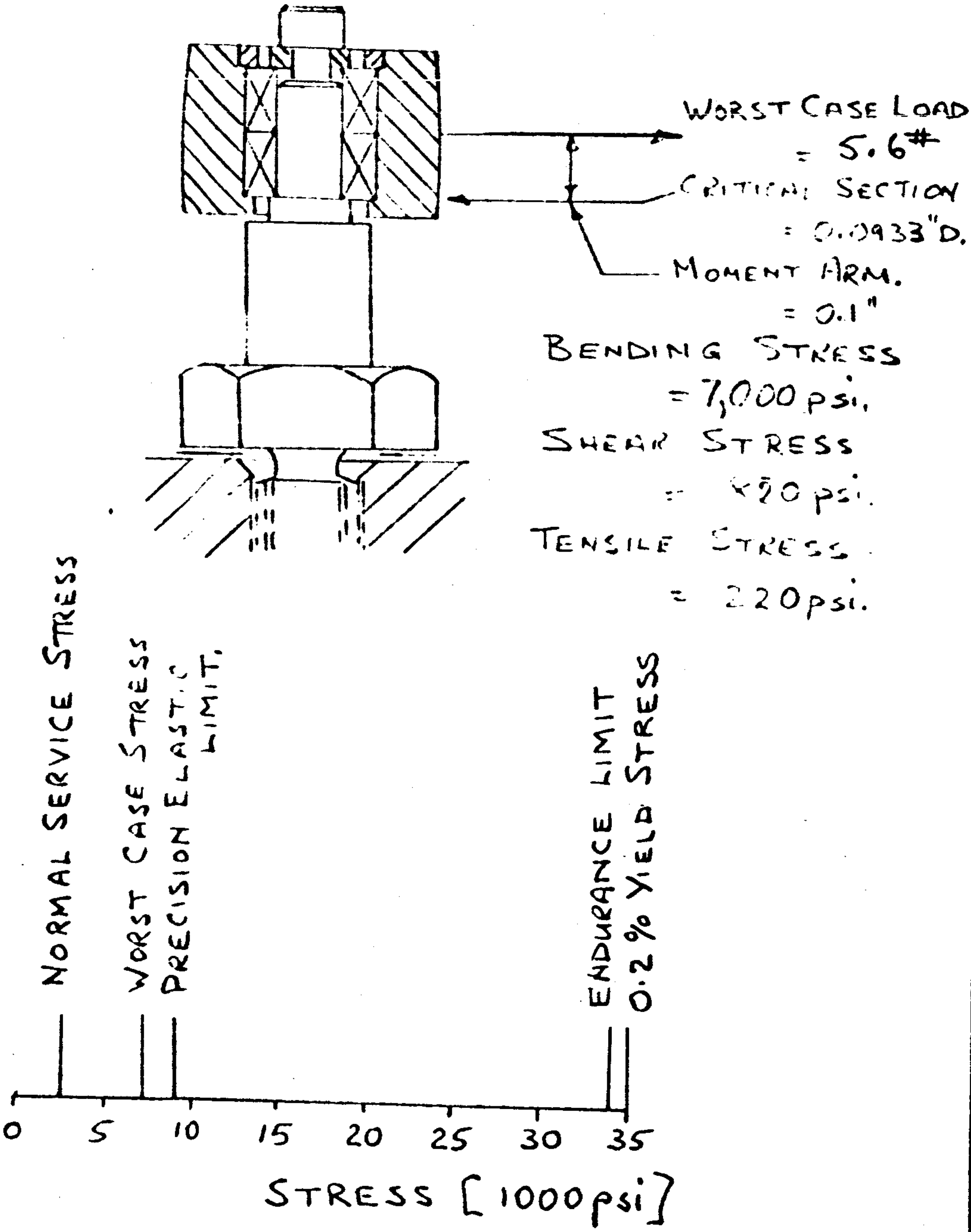
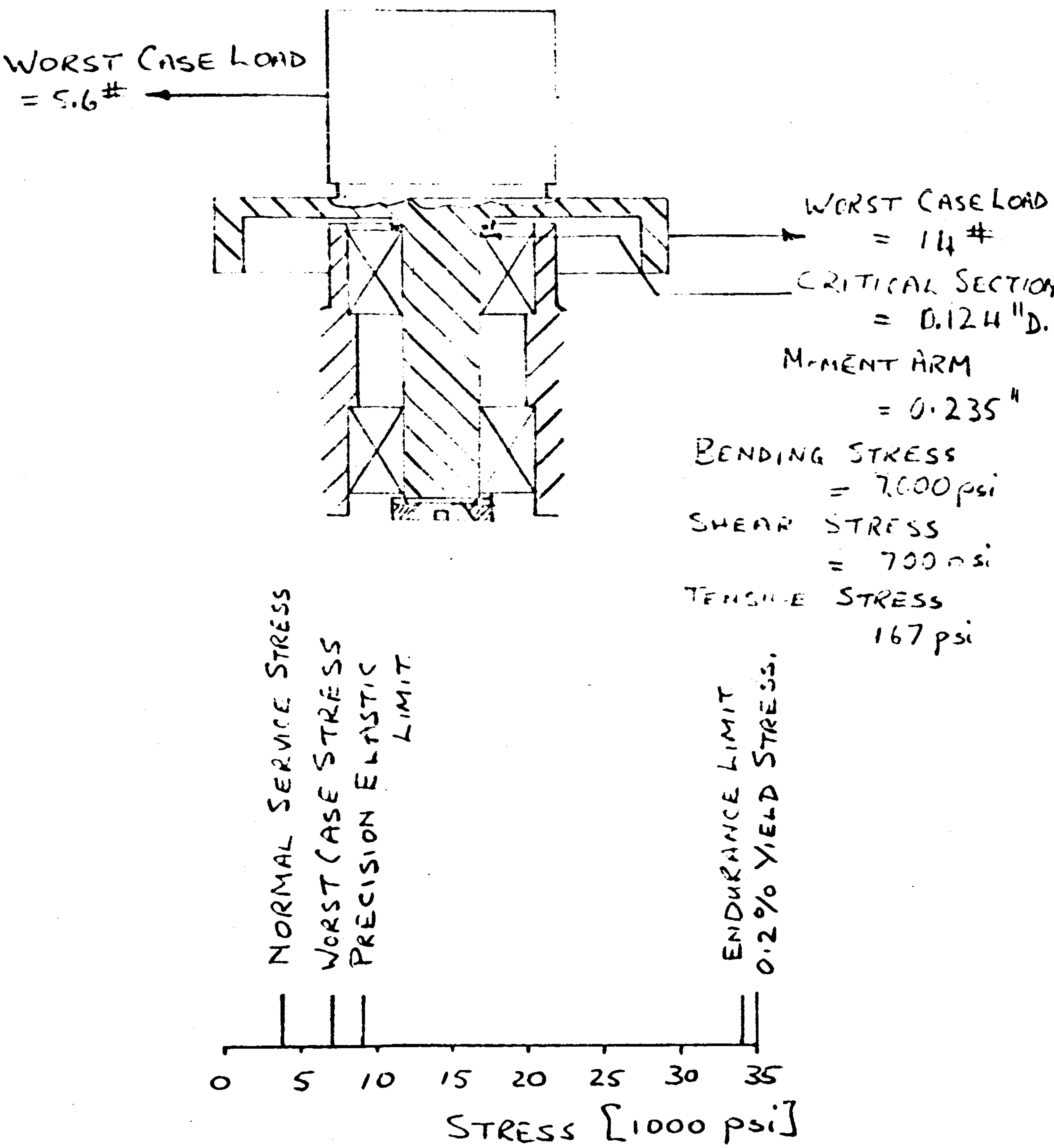


FIGURE 13 LOAD ANALYSIS DRIVE CAPSTAN ASSY.

LOCATION H & I TYP.



BEARING SURVIVAL PROBABILITY

B_{10} Life = $\left(\frac{C}{P}\right)^3$ Rev. $\qquad B_{50}$ Life $\cong$ 4.08 B_{10},

since we will be using high reliability bearings. (Ref.22) We presume a 1.5# preload which would be the upper limit of 1 $\pm$ .5 lbs preload.

Dynamic load rating of duplexed pair = 1.625 C_{dyn} of individual bearings.

New Hampshire Ball Bearings

			# Balls	Pairs
Reel Hubs	SR F PP-1			2
C_{dyn} Factor		134	8 (3/32")	
Belt Idlers & Guide Rollers	SR 133 PP-1			5
C_{dyn} Factor		12	8 (.025")	
Fast, Slow Capstans	SR 2 PP K58			2
C_{dyn} Factor		51	6 (1/16")	
Tension Arm	4215 (SR 166)			1
C_{dyn} Factor		64	8 (1/16")	

Microtech Bearings

Motor bearings	MR 618, 518			
C_{dyn} Factor		60	6 (1/10")	1

Ref. 22 Ball Bearing Survival, E. Shube
Machine Design, July 1962

RRR BEARINGS

LOCATION	Bearing Type	Duplex Preload	I-inner O-outer ring rotate	#Preload Thrust	#Radial Load	#Dyn. Capacity	RPM Av.@ 30 ips	B_{10} in hrs.
A. Lt. reel hub	R4-PP	P	0	1.5 +.5	3.13	134	252	3×10^5
			0	1.5 +.5	.63	134	252	6×10^5
B. Rt. reel hub	"	P	0	1.5 +.5	3.13	134	252	3×10^5
			0	1.5 +.5	.63	134	252	6×10^5
C. Lt. tape guide	R133-PP	D	0	1.5	.12	19.5	2290	800
D. Rt. tape guide	"	D	0	1.5	.375	19.5	2290	509
E. Rt. belt idler	"	D	0	1.5	1.7	19.5	1530	485
F. Lt. belt idler	"	D	0	1.5	1.7	19.5	1530	485
G. Tension arm Idler	"	D	0	1.5	2.0	19.5	1530	419
H. Slow Capstan	R2-5PP	P	I	2.0	2.34	51	1530	5750
		P	I	2.0	1.74	51	1530	6800
I. Fast Capstan		P	I	2.0	4.25	51	1530	3580
		P	I	2.0	1.75	51	1530	6800
J. Tension arm	R 166	P	0	2.0	7	64	1/3	4.7×10^7
		P	0	2.0	1.5	64	1/3	1.9×10^8
K. Motor	MR618	P	I	1.0	.76	60	8000	1.34×10^4
	MR518	P	I	1.0	.14	60	8000	1.89×10^4

Brg.	HOURS 100	200	300	400	500	600	700	800	900
			PROBABILITY OF SURVIVAL						
A^1	.999998	.999994	.999990	----85	----80	----75	----69	----63	----56
A_2	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0
B^1	----98	----94	----90	----85	----80	----75	----69	----63	----56
B_2	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0
C	.994	.984	.974	.960	.945	.930	.916	.900	.885
D	.987	.970	.950	.930	.905	.880	.850	.825	.800
E	.986	.968	.945	.921	.895	.867	.844	.814	.784
F	.986	.968	.945	.921	.895	.867	.844	.814	.784
G	.984	.963	.935	.905	.875	.845	.810	.778	.744
H^1	.9995	.9988	.9980	.9970	.9960	.9949	.9937	.9924	.9911
H_2	.9996	.9991	.9984	.9976	.9968	.9959	.9950	.9940	.9930
I^1	.9991	.9978	.9962	.9944	.9924	.9904	.9882	.9858	.9834
I_2	.9996	.9991	.9984	.9976	.9968	.9959	.9950	.9940	.9930
J^1	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0
J_2	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0
K^1	.99985	.99962	.99934	.99903	.99870	.99834	.99794	.99756	.99712
K_2	.99991	.99976	.99959	.99939	.99919	.99899	.99872	.99848	.99821
π	.93	.86	.77	.68	.59	.51	.44	.37	.31

Where π is the product of the numbers in the column.

RRR Bearing Reliability Fig 14

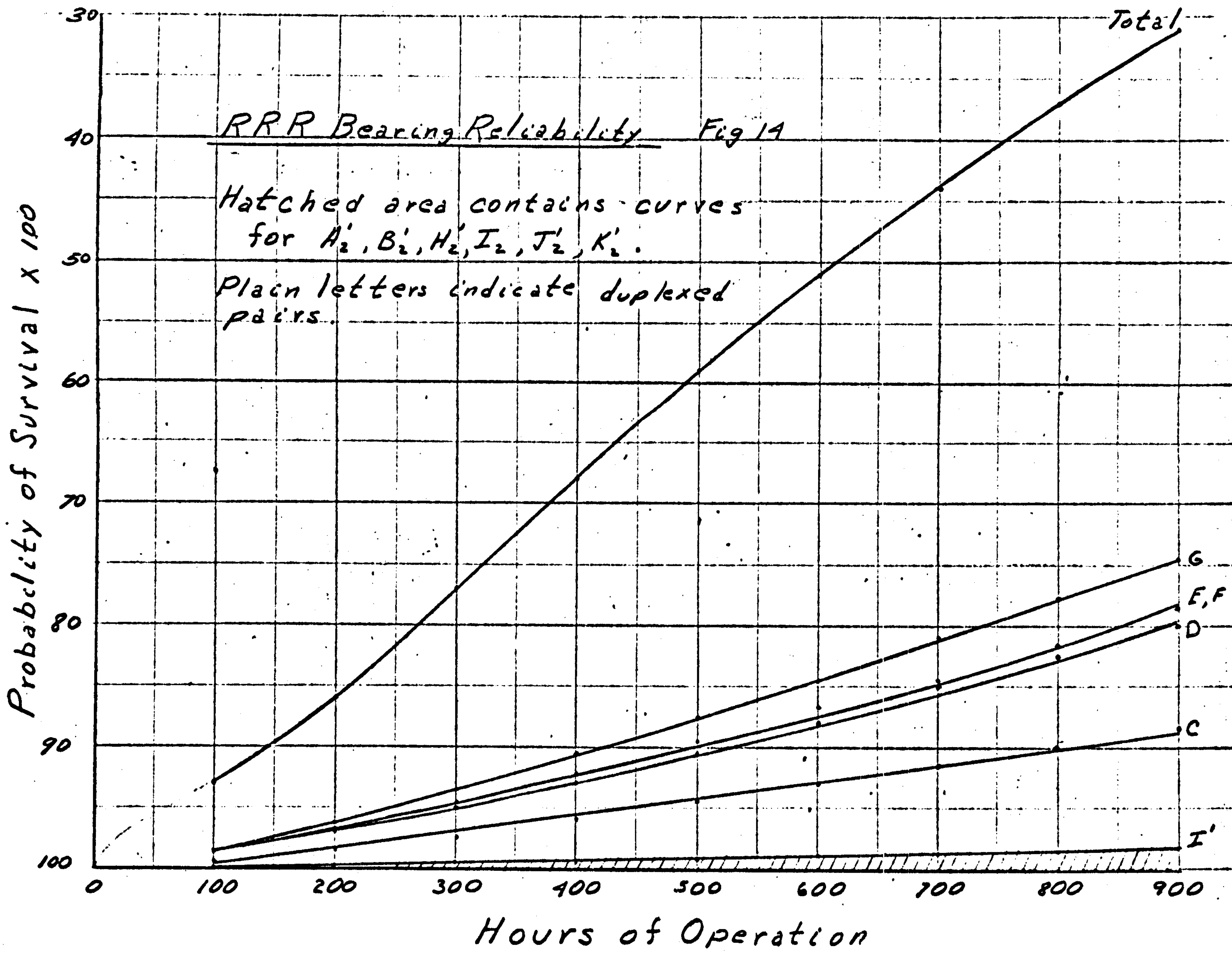


Figure 15

BELT RELIABILITY

Probability of Survival vs. Operating Time

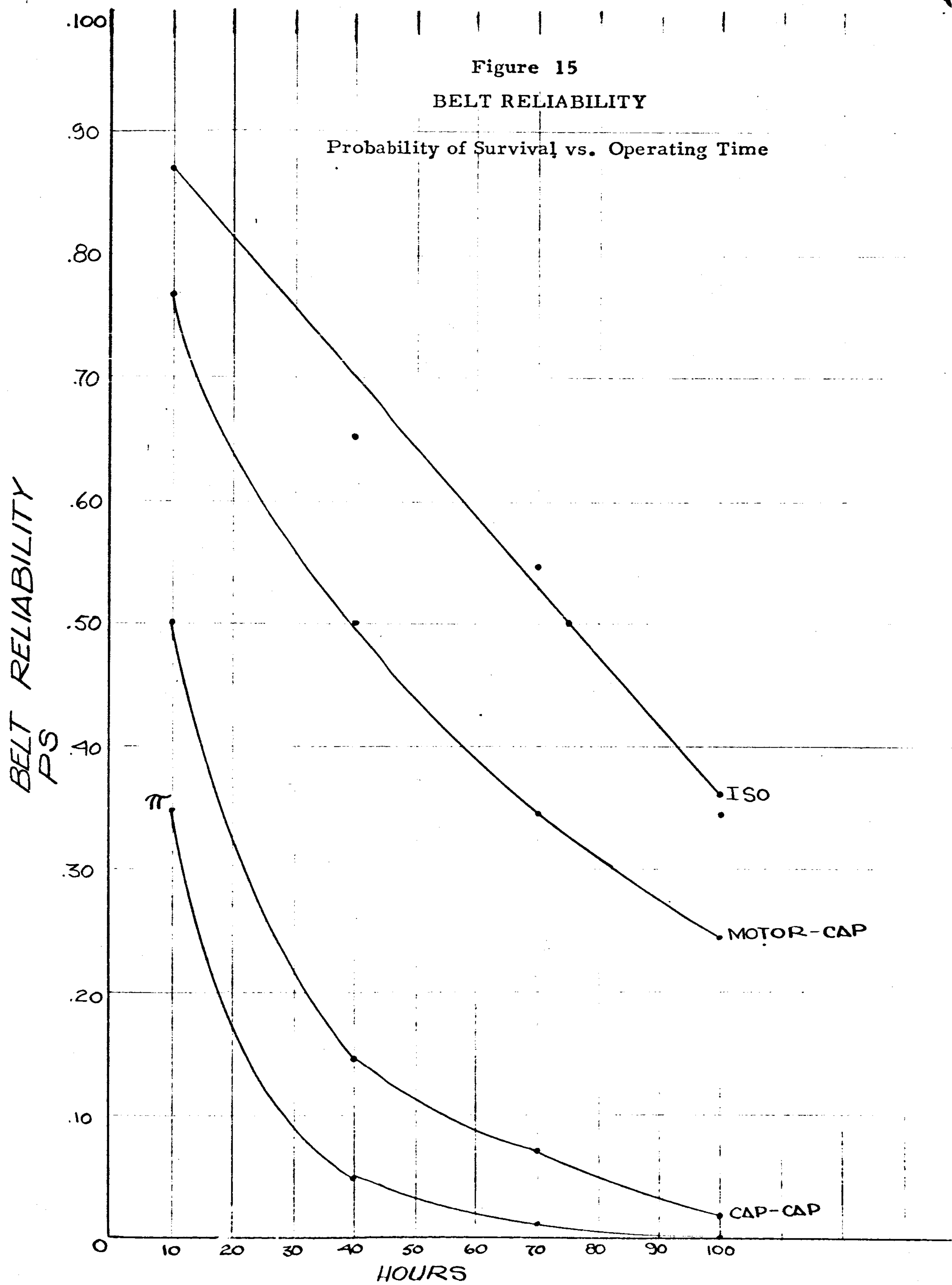


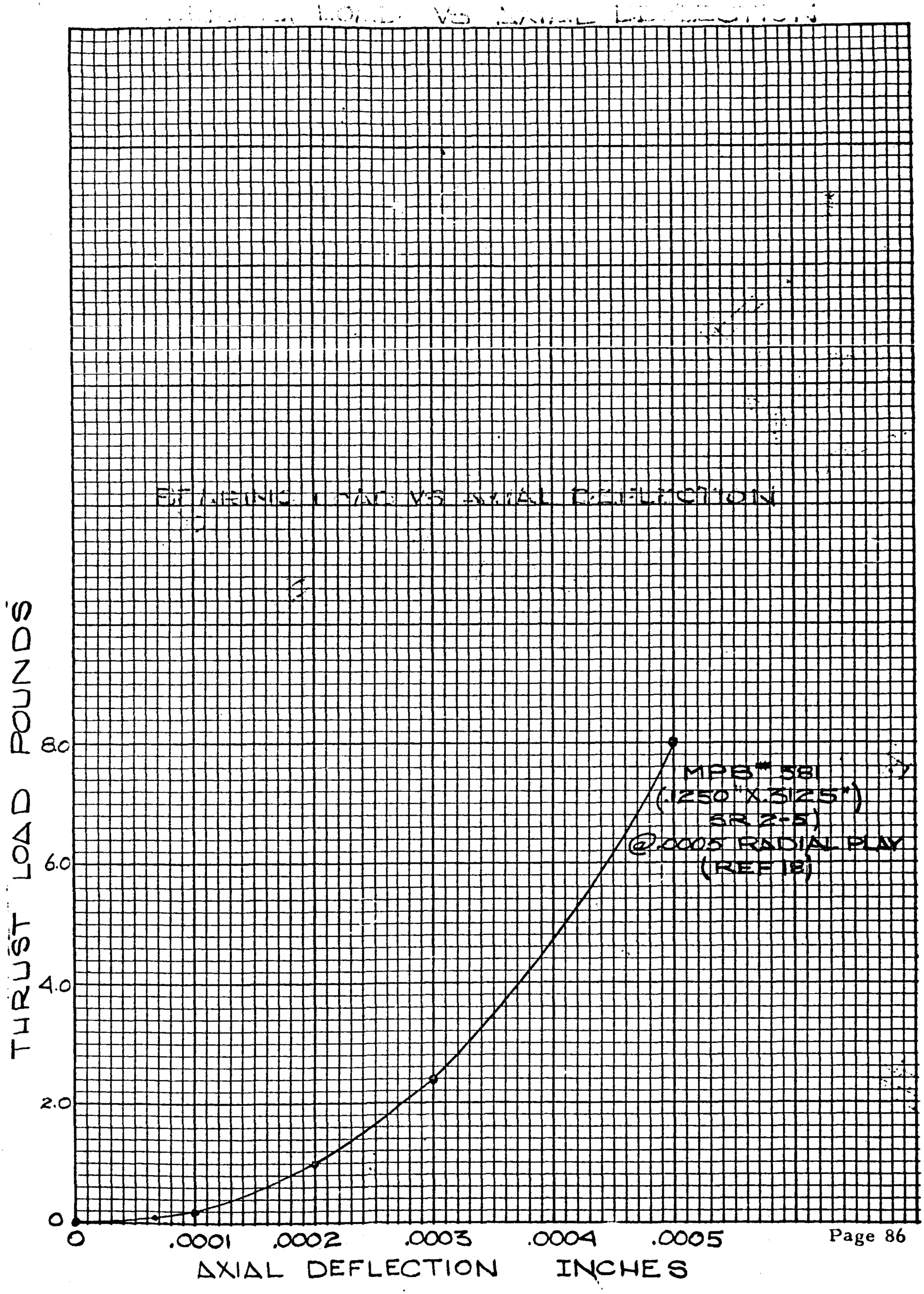
TABLE 8

BEARINGS - WORST CASE TOLERANCE VS. ENVIRONMENT

Bearing +.00005 -.00025	Pair Width	Shaft +.00005 -.00025	Housing +.00025 -.00005	Nominal Fit Min	Nominal Fit Max.	Differential Expansion -	Differential Expansion +	Operational Fit Min	Operational Fit Max.
						0°F	180°F	@ 180°F	@ 0°F
.0937"		.0935"		-.00010"	.00050"	-.00001"	.00005"	-.00015"	.00051"
.1250		.1248		"	"	-.00001	.00006	-.00016	.00051
.1875		.1873		"	"	-.00002	.00009	-.00019	.00052
.2500		.2498		"	"	-.00002	.00012	-.00022	.00052
								@ 0°F	@ 180°F
.1875"		.	.1875"	-00010"	.00050"	-.00002"	.00009"	-.00012"	.00059"
.3125			.3125	"	"	-.00003	.00016	-.00013	.00066
.3750			.3750	"	"	-.00004	.00019	-.00014	.00069
.6250			.6250	"	"	-.00006	.00031	-.00016	.00081

TABLE 9

Pair Width	Nom. Preload # Min.	Nom. Preload # Max.	Axial Dimens. Change		Oper. Preload # Min.	Oper. Preload # Max.
.1874"	.5	1.5	.00000"	.00009"	.15	1.5
.2812	1.0	2.0	.00003	.00014	.4	2.3
.2500	1.0	2.0	.00000	.00012	.22	2.0
.3920	.5	1.5	.00000	.00019	.00	1.5
					@180° F.	@ 0° F.



THRUST LOAD POUNDS
8.0
6.0
4.0
2.0
0
0
.0001
.0002
.0003
.0004
.0005
AXIAL DEFLECTION INCHES
MPB # 581
(.1250" X .3125")
SR 2-5)
@.0005 RADIAL PLAY
(REF 18)

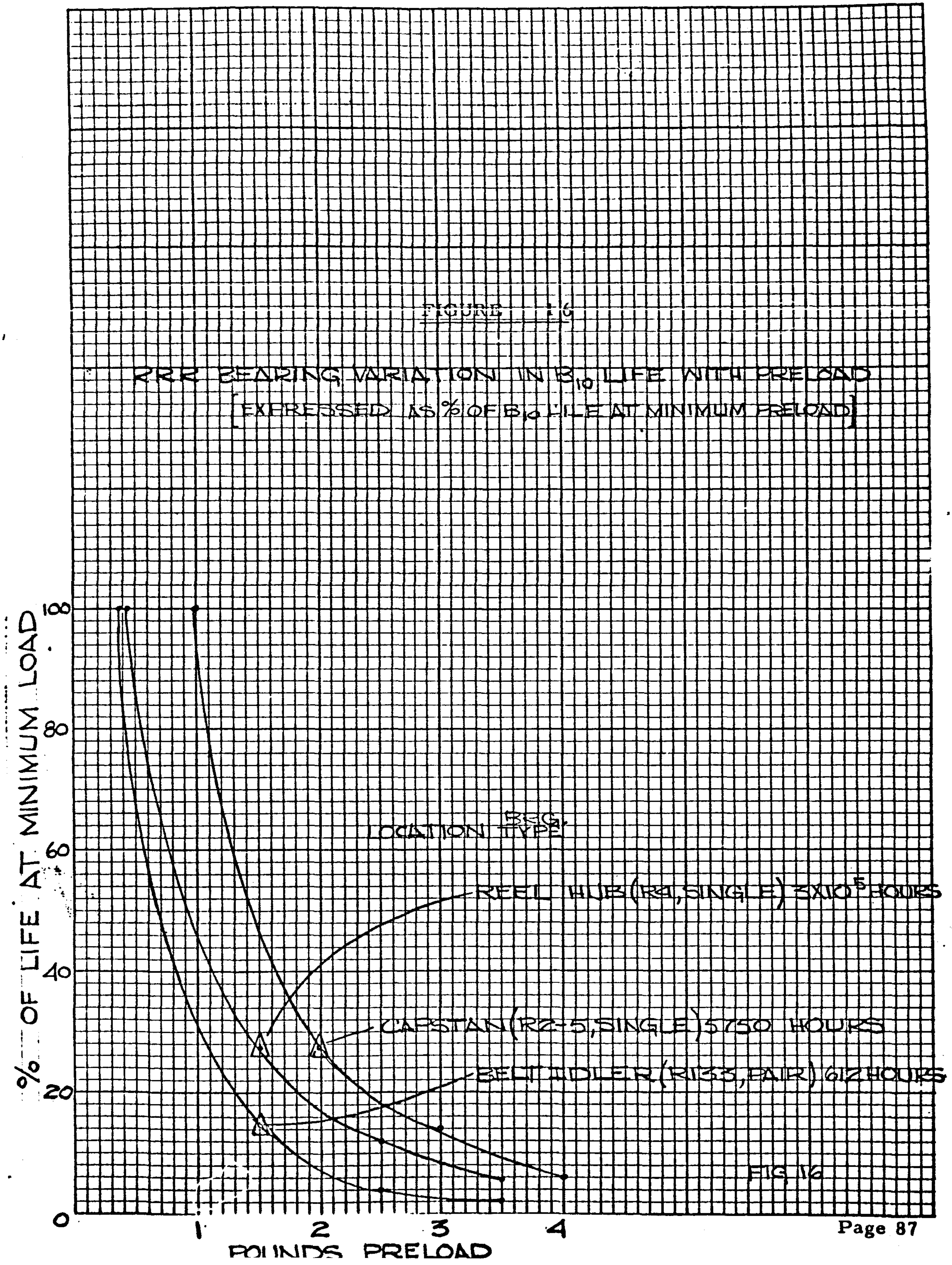
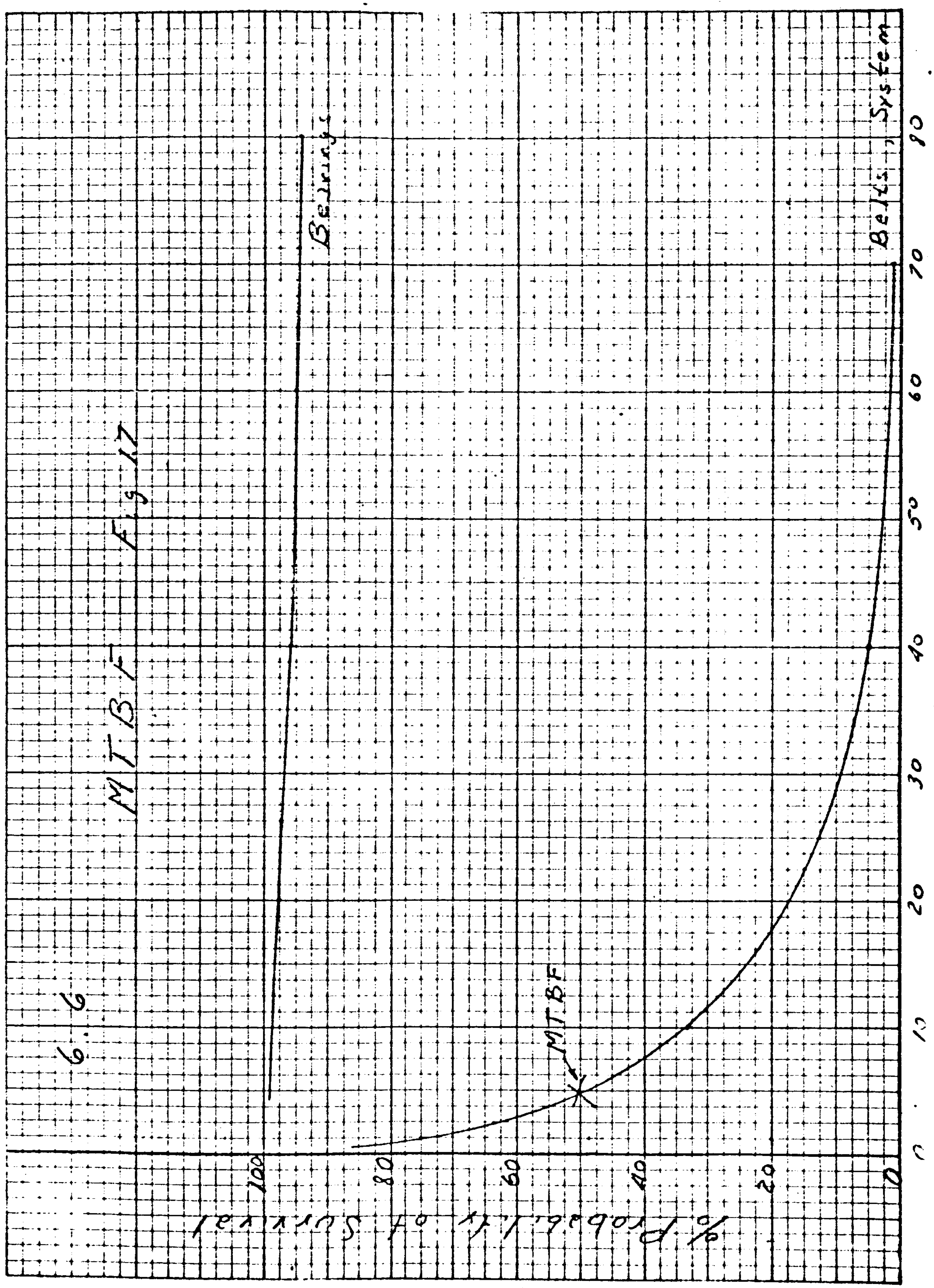


FIGURE 16
RRR BEARING VARIATION IN B_{10} LIFE WITH PRELOAD
[EXPRESSED AS % OF B_{10} LIFE AT MINIMUM PRELOAD]
LOCATION BRG. TYPE
REEL HUB (R4, SINGLE) 3×10^5 HOURS
CAPSTAN (R2-5, SINGLE) 5750 HOURS
BELT IDLER (R133, PAIR) 612 HOURS
FIG 16
% OF LIFE AT MINIMUM LOAD
100
80
60
40
20
0
1
2
3
4
POUNDS PRELOAD


6.6
MTBF Fig 17
Bearings
Belts, System
MTBF
% Probability of Survival
0
20
40
60
80
100
0
10
20
30
40
50
60
70
90

6.8 TABLE 10-A ROTATION FREQUENCIES AT 30 IPS

Element	D Inches	d Inches	n	Element Freq. RPS	Ball Freq. cps,	Retainer Freq. cps	Balls on Fixed Spot		Balls o Rotating S
Capstan	.179	.0625	6	25.5	36.8	10.9	65.5		87.6
Idler	.118	.025	8	38.2	90.1	17.3	139		167
Motor	.174	.0625	6	133.3	185	62.3	374		426
Hub	.375	.0938	8	6.9-2.9	14-5.8	3.1-1.3	25 - 10		30 - 13

D = land diameter

d = Ball Diameter

n= number of balls

	Rotation Frequency cps
Iso-Belt	1.3
Motor Belt	9.1
Capstan Belt	12

TABLE 10-B
TRANSVERSE BELT, TAPE VIBRATIONS

Location	Tension	LENGTH Start	LENGTH Mid	LENGTH End		FREQUENCY Start	FREQUENCY Mid (x 10^3 cps)	FREQUENCY End
TAPE								
L. pack - guide	3 oz	1.0"	1.3"	1.6"		6.0	4.6	3.7
L. guide - head	3 oz.	2.0"	2.0"	2.0"		3.0	---	---
Head - head	3.5 oz	.5"	.5"	.5"		13.0	---	---
Head - R. guide	4 oz	.3"	.3"	.3"		23.1	---	---
R. guide - pack	4 oz.	2.0"	2.0"	2.2"		3.5	3.5	3.1
BELTS								
Motor-Capstan	5 oz.	2.25"	---	---		10.8	---	---
Capstan - capstan	34 oz.	1.3"	---	---		26.6	---	---
L. pack - Idler	10 oz.	1.7"	2.0"	2.2"		21.6	17.0	15.4
L. Idler-Slow Cap.	10 oz.	1.4"	---	---		24.2	---	---
Slow cap. - Tens. Arm	10 oz.	.8"	---	---		42.4	---	---
Tens. Arm -R. Idler	10 oz.	4.65"	---	---		7.3	---	---
R. Idler - pack	10 oz.	2.5"	2.3"	2.2"		13.6	14.7	15.4
R. pack - fast cap.	10 oz.	1.5"	1.4"	.45"		22.6	25.1	75.4
Fast capstan-L. pack	10 oz.	.45"	1.4"	1.5"		75.4	25.1	22.6

Appendix 1

January 15, 1965

NOTES ON RECORDER FAILURE PROBLEMS

by John Lueder

It seems apparent that most recorders prior to test have misalignments and seemingly minor manufacturing errors which will cause an operational failure long before the design lifetime is attained.

The following are recommendations based on 1" tape transport experience. Valves for 1/4" machines should not exceed these and if it is possible closer tolerances should be held, about half of these values. All items must be checked for manufacturing errors and must be within these tolerances.

Alignment is most critical in the tape handling system of guides and capstans. Accuracy of the order of $\pm$ .0005" or less must be maintained for tape guide heights above the deck plate and guides should be within .0005" of each other in height. Width between guide flanges must be greater than tape width by up to .001".

Reel hubs must be centered vertically within $\pm$ .002" of the tape guides' centers.

Operation out of the vertical results in the tape mistracking and being damaged by guides. 1" tape is more susceptible than 1/4" due to its lack of stiffness and the high friction load to be overcome by edge guiding. Belt elements are less critical but will adversely affect a marginal tape system.

Tape guides and capstans must be vertical during operation, but the tape and belt tensions exert a bending moment, therefore these elements must be aligned out of the vertical in order that normal operating tensions result in vertical operation of these elements. Since the amount of bearing

In bearings where preloading is required .0005" should be specified as a minimum radial play to prevent high ball loading which would result from the small contact angle associated with low radial play. Shims, if used, should be available in steps of .0001" and parallel within .0001". They should be applied against the stationary ring where possible.

Wherever possible, bearings should be used with the inner ring rotating to reduce the thrust loading of the balls. Where highly loaded duplex pairs are required non-separable angular contact bearings should be considered for their higher load capacity.

Vacuum processed steel is producing higher reliability in bearings and should be specified as the material to be used for the bearings. Consumable electrode vacuum melted steel is even better, but may not be as available.

In guiding the tape, slip must occur between the tape and guide roller. The amount of force to move the tape depends on the coefficient of friction between tape and roller, and must be exerted by the guide flange on the tape edge. This action will damage the tape edge. If the friction is great enough and the misalighment sufficient the tape will not be returned to the proper position, and further tape damage leading to failure will occur. Reduction of the friction results in a marked increase in the misalignment which can be corrected. (Teflon tape was wrapped around rollers.) A suitable method of decreasing this friction should be investigated. Sharp edges in the tape path cause severe damage during loading and unloading and if tape is momentarily mistracking, these should be avoided.

Motor torque values should exceed the minimum required values by a significant margin, determined perhaps by the available power and space.

For design consideration, the machine failures have shown that belts and bearing lives are the limiting factors. The Iso-belt, by reason of its many direction reversals and high tension, has a comparatively short expected life, (of the order of hundreds of hours). An attempt should be made to minimize the number and severity of directinn reversals perhaps by eliminating the necessity of a tension arm or locating the capstan to eliminate an idler. Thickness and width of the belt, and the pulley diameters should all be considered toward lengthening the belt life.

If a tension arm is necessary, an attempt should be made to minimize the amount of travel and concomitant belt tension change as this necessitates higher stress levels to maintain proper driving at the minimum tension.

In considering bearings it is noted that the life expectancy of $>90\%$ of a group (L_{10}) is 1/5 of the life expectancy of $>50\%$, (L_{50}). This indicates a wide spread in actual life for each individual bearing and that some have intrinsically a longer life. If these could be culled from a group of bearings much higher reliability could be obtained. It is noted that ball failures occur primarily because of subsurface imperfections.

A doubling of L_{10} and L_{50} has been accomplished by one manufacturer in a lot using balls which have been magnetically inspected for imperfections. If manufacturers can provide this type of selection it may significantly improve the reliability of operation.

In view of the dust produced from the tape, bearings with double labyrinth type shields should be considered; and where possible in excessively dirty areas seals should be considered.

radial play affects this alignment, verticality in operation must be the criterion; but as a guideline, a tilt opposing the action by the tape or belt, of the order of .0003" in 1" results in satisfactory tracking. 1/4" transports having much smaller moments, due to lower tensions and shorter moment arms should be aligned to verticality $\pm$ .0002".

Misalignment of heads to their baseplate is common and will result in poor signal and tape mistracking. This must be checked in place and should be of the order $\pm$ .0001" tilt top to bottom

Belt oscillations due to misalignments cause significant flutter; therefore, pulleys mounted on shafts must be concentric and perpendicular on the order of .0005". If the pulleys are not cut in place in the shaft they may be Loctited in place. It has proven difficult to obtain the proper perpendicularity with a pinning operation.

To avoid excessive bearing damage it is desirable to balance all high inertia rotating components in assembly. This also reduces tape flutter.

Handling of belts and bearings must be of a standard to minimize the possibility of damage during installation and removal. Belts to be force fit over pulleys should be installed using a piece of Mylar sheet as a shoe horn and must be sized to fit with the proper tension. Iso belts must be carefully fitted and clear of all roller edges before tensioning.

Bearings must be carefully installed and preloaded accurately. No one should remove or replace a bearing except in the clean area. Bearing handling must be done only by a qualified person.

Machines should be run under cover and away from contaminating substances (cigarette smoke, dust, water, etc.).

Appendix 2

TAPE

The ability of the magnetic tape to survive many operational cycles is an additional factor in the long term reliability of a recorder. As with bearings and belts, there is an ultimate and a practical limit. In tests run at the National Bureau of Standards, 1/2" tape with 7 tracks at 30 ips survived for over 10^6 cycles. In normal use, tape is replaced after 50-100 cycles. The wide difference is attributed to the condition of cleanliness and mechanical perfection in a laboratory situation. The tape manufacturers have produced an oxide coat which minimizes wear off and whose weat particles produce minimal clumping and further wear. Most operational failures are promoted by external dirt entering the tape system and causing erosion of the coat, displacement at head, local stress in a pack, etc. It is apparent that in a sealed recorder the available dirt can and should be at a minimum. Internal sources of dirt should be isolated if possible.

Ref: Memorex Monographs #3

Appendix 3

START TIME VS. SPEED

Considering stop time as a measure of the rate of energy loss to the system analagous to a force-mass deceleration F = Ma & Fxd = work or $\frac{dE}{dt}$ = power loss.

Most losses are velocity dependent: as speed goes up belt hysteresis loss k per cycle is a constant, but cycles per second increase linearly. As bearing speed goes up the same effect occurs. The head friction will have 2 components -- one steady and one velocity dependent -- so

$$\frac{dE}{dt} = k_{belts} V + k_{bearings} V + k_{head} V + i_{head}$$

Let's presume $\frac{dE}{dt} \propto V$ neglecting i_{head}

The system has a total inertia I composed of bodies at various speeds

$$I = I_1 + \frac{w_2^2}{w_1^2} I_2 + \frac{w_3^2}{w_1^2} I_3 + -----$$

where the effective system speed is taken as w_1.

For a given speed w_1 we have a source of energy (the motor) which produces torque at various speeds (a characteristic curve can be assumed).

If, for hysteresis-synchronous motors we assume a constant torque-speed relation for starting, we have power function $\frac{dE}{dt}$ = Tf (f is in cycles/sec.) (T is constant torque) but let's use f as the speed relative to which the inertias are are measured.

$$f = 2\pi w_1 \therefore dE = T\, 2\pi w_1\, dt$$

To find start time we need to evaluate the proportionality constant in the energy loss system. $\frac{dE}{dt} = K^1 V = Kw\ (t)$

Since we know total E contained in the system $E = w_1^2 I$ and the time to run run down by measurement (t_{stop}) for a particular w_1.

energy removed $dE = \left(Kw + i\right) dt$

energy available $E = w^2 I$

$$\frac{dE}{dt} = 2wI \frac{dw}{dt}$$

$$\therefore \quad K = 2I \frac{dw}{dt} \qquad \frac{dw}{dt} = \frac{K}{2I}$$

$$\int_{w_1}^{o} dw = \frac{K}{2I} \int_{o}^{t_{stop}} dt \qquad -w_1 = t_{stop} \frac{K}{2I}$$

$$K = \frac{w_1 2}{t_{stop}} \quad \text{constant for all } w$$

Start time requires the difference between the energy available and the energy loss be used to increase w.

$$E_{in} - E_{loss} = Iw^2 \quad \Rightarrow \quad \frac{dE_{in}}{dt} - \frac{dE_{loss}}{dt} = 2Iw \frac{dw}{dt}$$

substituting: $2\pi wT - Kw = 2Iw \frac{dw}{dt}$ (for $w \neq o$)

(neglecting what happens at $t = o$; $w = o$)

$$\frac{dw}{dt} = \frac{2\pi T - K}{2I}$$

$$\int_{o + s}^{w_i} dw = \frac{2\pi T - K}{2I} \int_{o + e}^{t_i\, start} dt \qquad \text{for small } s \;\&\; e$$

$$\therefore \quad w_i = \frac{2\pi T - K}{2I} t_i\, start = \frac{\pi T - w_1 I / t_1 stop}{I} t_i\, start$$

$$\text{or } t_i\, start = \frac{I}{\pi T - \frac{w_1 I}{t_1 stop}} w_i$$

REFERENCES

1. ADVANCED BEARING TECHNOLOGY, E. E. Bisson and W. T. Anderson, NASA 1964

2. SKF RESEARCH REPORT, Dec. 1961 Report AL61T027

3. LUBRICATION UNDER SPACE VACUUM CONDITIONS F. J. Clauss, Lockheed M & S Co. 1962

4. BARDEN SALES CATALOGUE, Barden Corporation 1962

5. DESIGN & PURCHASING MANUAL, New Hampshire Ball Bearings, Inc. 1959

6. ROLLING CONTACT FATIGUE EVALUATION OF BEARING MATERIALS AND LUBRICANTS. E. G. Jackson. Trans. ASLE, Vol. 2, No. 1, 1959

7. EFFECT OF BEARING TEMPERATURE ON CAPACITIES OF BEARINGS OF VARIOUS MATERIALS. Arthur S. Irwin. Paper presented at ASME Lubrication Symposium, New York. March 14-15, 1960.

8. ROLLING CONTACT PHENOMENA. J. B. Bidwell. Elsevier Pub. Co. 1962. PP 317-345.

9. EFFECT OF HARDNESS, SURFACE FINISH, AND GRAIN SIZE ON ROLLING CONTACT FATIGUE LIFE OF M-50 BEARING STEEL. R. A. Baughman. Jour. Basic Eng. (ASME Trans.) Ser. D Vol. 82, No. 2, June 1960, pp 287 - 294.

10. BEARINGS FOR VACUUM OPERATION, Phase I, II. H. E. Evans, T. W. Flatley, Goddard Space Flight Center

11. MPB ENGINEERING NEWS Vol. 7 #1. C. H. Hannan

12. PROBLEMS OF LUBRICATION IN SPACE. M. M. Freundlich, C. H. Hannan, Lubrication Engineering, Feb. 1961.

13. LUBRICANT MANUAL FOR INSTRUMENT BALL BEARINGS. R. F. Irwin, MPB Reprint #5801. 1959

14. NEW WAY TO EVALUATE BEARING LUBRICATION. T. A. Harris Product Engineering, April 1965.

REFERENCES

(Continued)

15. LUBRICATION IN SPACE ENVIRONMENTS. R. L. Adamczak, R. J. Benzing, H. Schwenker

16. MPB ENGINEERING NEWS. Vol. 8 #2. C. H. Hannan

17. NEW DEPARTURE GENERAL CATALOGUE 1961

18. MPB SALES CATALOGUE #6

19. PROCESSING AND TESTING GUIDANCE SYSTEM MATERIALS FOR MAXIMUM DIMENSIONAL STABILITY, C. G. Jennings, A. D. Shankman, Dr. L. E. Colteryahn.

20. NEW HAMPSHIRE BALL BEARINGS SALES CATALOG

21. MICROTECH BEARINGS SALES CATALOG

22. BALL BEARING SURVIVAL, E. Shube, Machine Design, July 19, 1962.