CHAPTER 3.4
ELECTROMECHANICAL AND MICROELECTROMECHANICAL SYSTEMS (MEMS) RELIABILITY

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INTRODUCTION

Reliability of electromechanical and microelectromechanical systems (MEMS), like other important design parameters, is established by the design and manufacturing process. Actual reliability (time-to-failure) rarely exceeds desired reliability primarily because of wearout of these systems. The reliability design and analyses techniques discussed in this chapter will aid the designer in using reliability tools to provide a more reliable electromechanical component or system. One must focus attention on design weaknesses so that they may be corrected, protected against, or accepted after consideration. These techniques also provide a means of ensuring that a design will meet specified time-to-failure (reliability to life) requirements.

It is generally true that one can demonstrate elecromechanical reliability by test with a sufficient degree of statistical confidence prior to and during production. This is because of the short life that these electromechanical systems have. Their wearout time or cycles is relatively short and the mean-time-between-failures (MTBFs) is measured in hours rather than hundreds of years as is the case with electronic parts. These actual MTBFs can be demonstrated during accelerated life testing.

There are two fundamental types of failures. One results from higher stresses than the part can withstand. The second is caused by the part literally wearing out. Both types of failures are covered in this chapter. However, the second is the most intriguing since wearout comes in many different forms. Wearout can be a result of fatigue, like a spring failing after being subjected to the same load for many years. It can be the result of abrasion, like tire wear on a car. Wear and wearout also are caused by friction, thermal cycling, and so forth. Generally all mechanical parts have a finite life. It can be very long like the pyramids of Egypt or very short like some MEMS motors. We will attempt to quantify the wearout time for all mechanical parts based on stresses and wear in this chapter. In this way, one can predict the MTBF and determine when to perform maintenance.

The classic bathtub curve shown in Chap. 3.1 also applies to mechanical failures. Stage 1 (infant mortality) is very short. It is therefore not necessary to test mechanical components for extended lengths of time to screen out infant mortality failures. The majority of a mechanical component's life is spent in Stages 2 and 3, which show a gradual increase in failure rate during Stage 2 and an accelerated failure rate during Stage 3. This is because mechanical components have continuous wear and fatigue throughout their operating life. An excellent method of selecting an appropriate service time is to determine when (number of operating hours) the failure rate curve begins to accelerate the sharpest, sometimes called the knee of the hazard rate curve. As illustrated in Fig. 3.4.1, higher operating stresses result in shorter life of the part.

Examples of electromechanical devices include motors, sensors, switches, connectors, and so forth. Samples of the MEMS devices and additional applications are presented in Chap. 8.5, "Microsensors and

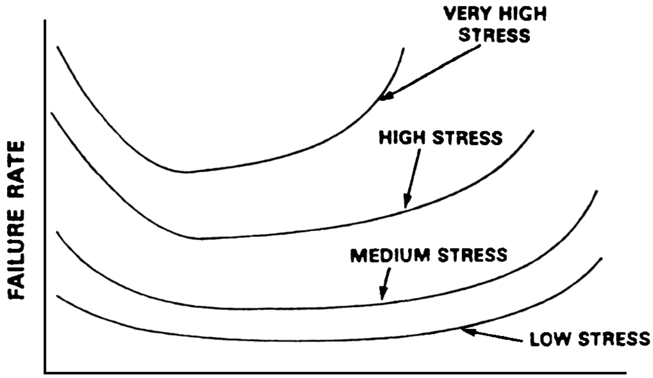

FIGURE 3.4.1 Effect of stress levels on mechanical failure rates.

Actuators." In the design of these systems, and operational requirements must be translated into design requirements, which are then verified by analysis (to provide an assessment of the design) and possibly by testing. The results can be compared with the operational requirements.

MECHANICAL STRESS ANALYSIS

Mechanical stress analysis is an important design procedure by which the maximum actual stresses induced on a material in its application are identified, so that adequate safety margins can be designed in.

Life-cycle phase. Stress analysis is normally conducted during full-scale development as the detailed design definition progresses.

Purpose. Stress analysis provides identification of the stresses, and confirmation of compliance with safety margins.

Objectives. The primary objective of stress analysis conducted on electromechanical systems/equipment is to identify any life-limiting stresses. This is performed by comparing stress-strength characteristics of parts and materials with internal and/or external loadings so that when the design is exposed to stress levels experienced in service, its life meets the requirements for that equipment. Stress analysis requirements should be invoked as a design technique to reveal system/equipment design deficiencies.

Material Stress-Strength Comparison

A material's strength or capability of handling a given stress varies from lot to lot and manufacturer to manufacturer. This variation for all materials of the same type can be represented by a statistical distribution of material strength. Similarly, the stress applied to a material changes from one point in time to another, with instantaneous changes occurring in temperature, mechanical stresses, transients, vibration, shock, and other deleterious environments.

At a random point in time the environmental effects can combine, reaching stress levels beyond the material's strength, resulting in failure of the material.

This strength-stress relationship can be described graphically by two overlapping statistical probability density distributions as shown in Fig. 3.4.2. For materials rated at the mean distribution used in an environment with an average stress equal to the mean of the stress distribution, the probability of failure is equal to the product of the areas of the two distributions where overlap occurs (i.e., the probability of the stress being greater than the minimum strength times the probability of the strength being lower than the maximum stress). This probability of failure is represented by the solid dark area of Fig. 3.4.2.

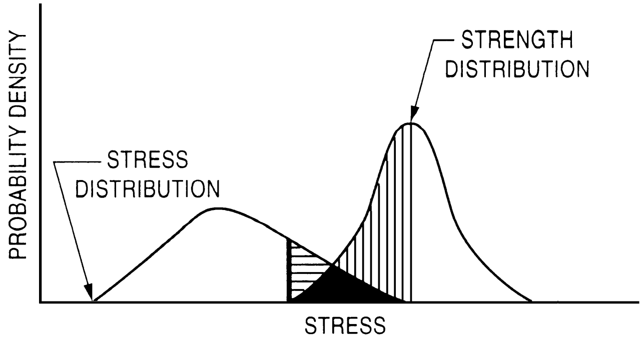

FIGURE 3.4.2 Two overlapping statistical probability-density distributions.

To reduce the probability of failure the potential stress levels can be reduced to a point where there is a very small probability of the stress exceeding the material's strength (stress distribution is shifted to the left), or the material's strength can be increased so that the probability of the combined stresses reaching or exceeding this strength is very small (strength distribution is shifted to the right). In most cases, the strength of the material cannot be increased and the only approach is to decrease the stress on the part. Alternatively, sometimes a different or stronger material can be used that can withstand the higher stresses. Either technique reduces material internal temperatures, decreasing the rate of chemical time-temperature reaction.

Procedure for Mechanical Stress Analysis

This chapter explains the fundamental mechanical stress analysis principles and shows the primary methods and formulas necessary for calculating mechanical stresses, safety factors, and margins of safety associated with electromechanical designs. These calculations are used in the mechanical stress analysis in two ways: (1) They ensure that the structural part has sufficient strength to operate in its service environment and under the loads specified, and (2) to a lesser extent, they ensure that there is no appreciable excess of material or over-design resulting in higher costs and burdening the program with excessive weight, mass, volume, and so forth.

Mechanical stress analysis is a detailed and complex engineering discipline. However, for our purposes simplifying assumptions will be made which provide equations for evaluating the approximate values of the stress, loads, margins of safety, and safety factors.

Several important factors must be considered before performing a mechanical analysis:

1. Important parameters are the loads, the material properties, and the actual calculated stresses.
2. Factors to be considered include the size, material joining techniques, welding, bonding, loads, material properties, moisture, thermal environment, and thermal limitations of the material.
3. Environmental factors such as corrosion and temperature and their resultant stresses must be considered.
4. Actual stress values must be calculated under steady-state loading conditions.

Symbols and Abbreviations

Symbols and abbreviations used in this chapter are defined as per US MIL-HDBK-5E. These symbols are abbreviations will be used throughout this section. with the exception that special statistical symbols are presented as necessary. To go further than this information, the reader should be familiar with the symbols and abbreviations used in mechanical structural analysis.

STANDARD DEFINITIONS, NOMENCLATURE, AND FUNDAMENTAL EQUATIONS

All electromechanical systems, while in operation, are subjected to external loads. These supplied loads must be transmitted through components and between components to the supports, which provide the necessary reactions to maintain equilibrium. In the process, forces and moments are developed in each component of the system to resist the externally applied loads; providing equilibrium for the components. The internal forces and moments at any point within the system are called *stress resultants*.

Normal Force. If a uniform bar is subjected to a longitudinal force P, the force must be transmitted to its base. Under this loading the bar will increase in length and produce internal stress resultants called *normal forces*, denoted by P. Normal forces are associated with electromechanical components that lengthen under tensile loads and shorten under applied compression loads.

Shear Force. If a uniform square plate is subjected to a uniformly distributed force f on three edges of the plate, then these forces must be transmitted to the base edge. This form of deformation is called *shear deformation* and is associated with an internal stress resultant called *shear force*, denoted by S.

Bending Moment. If a uniform beam is subjected to a couple C, the beam is no longer straight, but has become bent. Deformations that bend electromechanical components are associated with internal stress resultants called *bending moments*, denoted by M.

Torque. If a uniform circular shaft is subjected to a couple, the shaft must twist as the couple is transmitted to the base support. Deformations that twist electromechanical components are associated with internal stress resultants called *torque*, denoted by the vector C.

Fundamental Stress Equations

Internal stress resultants represent the total force or moment acting at any point within a electromechanical system. In reality these stress resultants are distributed over surfaces within the components, and their magnitudes per unit area are called *stress*. Stress is defined by force per unit area. Therefore, bending moments and torques must be described by statically equivalent force systems made up of distributed normal or shear forces in order to evaluate their effect in terms of stress.

Normal and Shear Stress. Stresses acting perpendicular or normal to a surface are called *normal stresses*, while stresses acting in the plane of the surface are called *shear stresses*. Internal stress resultants do not necessarily act normal or in the plane of the surface in question. For convenience, in mechanical stress analysis, internal force resultants are always represented in component form so that stresses are either normal stresses or shear stresses.

Consider an incremental internal force resultant ΔR acting on an incremental surface area ΔA. The vector sum of ΔP (normal force) and ΔS (shear force) may be used to replace ΔR. The normal force per unit area and the shear force per unit area are then called normal stress f_n and shear stress f_s, respectively. Therefore, average values are defined by

$$f_{n(\text{avg})} = \frac{\Delta P}{\Delta A} \qquad f_{s(\text{avg})} = \frac{\Delta S}{\Delta A} \tag{1}$$

Notice that the correct stress involves more than magnitude and area, but also direction, sense, and location.

Fundamental Strain Equations

All electromechanical components deform under load. They may change size, shape, or both.

Normal Strain. Normal strain e_n is defined as a change in length per unit of original length. The uniform bar increased in length by δ when subjected to the normal force *P.* Therefore, average values for normal strain are defined by

$$e_{s(\text{avg})} = \frac{\delta}{L} - \frac{L_{\text{final}} - L_{\text{initial}}}{L_{\text{initial}}} \tag{2}$$

Shear Strain. Shear strain e_s is defined as change in angle between any two originally perpendicular lengths, measured in radians. A uniform square plate changes in shape when subjected to the shear force *S.* The square plate becomes a trapezoid under load and forms an interior angle α of less than 90°. Therefore, average values for shear strain are defined in radians by

$$e_{s(\text{avg})} = \frac{\pi}{2} - \alpha \tag{3}$$

When the deformations are small and the angle change is also small, average shear strain can be defined in terms of the shear deformation δ_s,

$$e_{s(\text{avg})} \geq \tan\,(e_{s(\text{avg})}) = \frac{\delta_s}{L} \tag{4}$$

COMMONLY USED FORMULAS

Formulas and equations related to tensile and compressive action are given in US MIL-HDBK-5E. Formulas extracted from that handbook are provided on the accompanying CD-ROM, under the heading "Commonly Used Formulas in Electromechanical Reliability Studies."

MECHANICAL FAILURE MODES

This section presents mechanical failure modes that might occur and should be evaluated to ensure that the design has sufficient material strength and adequate margins of safety.

Tensile yield strength failure. This type of failure occurs under pure tension. It occurs when the applied stress exceeds the yield strength of the material. The result is permanent set or permanent deformation in the structural part. This is not generally a catastrophic condition.

Ultimate tensile strength failure. This type of failure occurs when the applied stress exceeds the ultimate tensile strength and causes total failure of the structural part at this cross-sectional point. This is a catastrophic condition.

Compressive failures. Compressive failures are similar to the preceding tensile failures only under compressive loads. They result in permanent deformation or total compressive failure causing cracking or rupturing of the material.

Failures due to shear loading. Yield and ultimate failures occur when the shear stress exceeds the strengths of the material when applying high torsion or shear loads. These failures generally occur on a 45° axis with respect to the principal axis.

Bearing failures. Bearing failures are similar to compressive failures. However, they are generally caused by a round, cylindrical surface bearing on either a flat or a concave surface like roller bearings in a race. Both yield and ultimate bearing stresses should be determined.

Creep and stress rupture failures. Long-term loads generally measured in years cause elastic materials to stretch even though they are below the normal yield strength of the material. If the load is maintained continuously and the material stretches (creeps), it will generally terminate in a rupture. Creep accelerates at elevated temperatures. This results in plastic strain because of creep even though the load is in the elastic region. Creep should always be checked under conditions where high loading for long periods of time are anticipated.

Fatigue failures. Fatigue failures result from repeated loading and unloading of a component. This is a major cause for reducing the design allowable stresses below those listed in the various design tables for static properties. The amount of reduction in the design load is based on complex curves and test data. The amount of load cycling with respect to the maximum load and the number of cycles are the predominant variables. These variables are generally described in Moody diagrams (which relate stress to number of life cycles, *S–N* curves).

Metallurgical failures. Metallurgical failures are the failures of materials because of extreme oxidation or operation in corrosive environments. Certain environmental conditions accelerate these metallurgical failures. The environmental conditions are: (1) heat, (2) erosion, (3) corrosive media, and (4) nuclear radiation. Therefore, depending on the environment, these conditions should be considered.

Brittle fracture. Certain materials that have little capacity for plastic flow and are generally brittle, such as glass, ceramics and semiconductor materials, are extremely susceptible to surface flaws and imperfections. The material is elastic until the fracture stress is reached. Then the crack propagates rapidly and completely through the component. These fractures can be analysed using fracture mechanics.

Bending failures. A bending failure is a combined failure where an outer surface is in tension and the other outer surface is in compression. The failure can be represented by tensile rupture of the outer material.

Failures from stress concentration. This occurs when there is an uneven stress "flow" through a mechanical design. This stress concentration generally takes place at abrupt transitions from thick gauges to thin gauges, at abrupt changes in loading along a structure, at right angle joints or at various attachment conditions.

Failures from flaws in materials. This is generally because of poor quality assurance or improper inspection of materials, weld and bond defects, fatigue cracks or small cracks and flaws. These reduce the allowable strength of the material and result in premature failure at the flawed location.

Instability failures. Instability failures occur in structural members, such as beams and columns particularly those made from thin material and where the loading is generally in compression. The failure may also be caused by torsion or by combined loading including bending and compression. The failure condition is generally a crippling or complete failure of the structural part.

All of the preceding failure modes should be analyzed as necessary to confirm the adequacy of the mechanical design. It should be noted that some of the analysis is very involved, particularly in the case of combined loads or complex structures. These may require the use of a finite-element computer program.

SAFETY FACTORS AND MARGINS OF SAFETY

In order to design a electromechanical component or system to perform a given function in a reliable manner, the designer must understand all the possible ways the component or system could lose its ability to perform that given function. This requires a complete understanding of how the system or component responds to externally applied loads. With the aforementioned modes of failure established, reasonable limits on load can be defined.

The *safety factor* (SF) is a strength design factor defined by the ratio of a critical design strength parameter (tensile, yield, and so forth) to the anticipated operating stress under normal operating conditions. For example, let F be the strength of the material and f_{mw} be the maximum allowable working stress. Then the factor of safety becomes

$$\mathrm{SF} = \frac{F}{f_{mw}} \tag{5}$$

The *margin of safety* (MS) is usually expressed as the maximum allowable working stress (f_{mw}) divided by the applied stress (f) minus 1, as shown in the following equation:

$$\mathrm{MS} = \frac{f_{mw}}{f} - 1.0 = \frac{F/SF}{f} - 1.0 \tag{6}$$

This is for a simple unidirectional stress. Any negative value indicates that the structural part will fail because the applied stress is in excess of the allowable material strength.

RELIABILITY PREDICTION TECHNIQUES

With electronic hardware, manufacturers produce millions of similar parts that are then incorporated into thousands of different types of electronic equipment. Billions of hours of operation are accumulated which are used to guess at electronic part failure rates. Because of the many different uses, it is possible to accumulate extensive data, with respect to failure rate, under various environmental, temperature, and electrical stress conditions.

With the exception of nuts and bolts type hardware, electromechanical equipment is designed for a specific configuration and use. There is insufficient quantity of any specific device to accumulate the hours necessary to establish a basic failure rate, or variations caused by different stress levels, environments, and temperature extremes.

A "Probabilistic Stress and Strength Analysis" (p. 3.71) provides techniques for determining structural reliability through the computation of the probability of failure. Although this technique provides a good reliability estimate based on stress and strength, it will be found that to use this method for every electromechanical part of a system would be a long and costly effort.

An alternative approach is failure mode, effects and criticality analysis (FEMCA). This procedure does not consider any failure mechanisms that are not critical to equipment performance, or will not otherwise affect the performance of a prescribed mission. Another method is to establish a reasonable safety factor to minimize the risk of part failure.

There are several methods that can be used to determine the reliability of a electromechanical system. These methods may vary from precise to simple estimates. The precise methods require extensive computations for each element that is used in the electromechanical system. The simple approach uses generic data for each of the parts to obtain an estimated reliability, and as such, does not consider stress levels, temperature extremes, and material strengths. Also, one might use accelerated life testing.

Basic Reliability Definition

In order to use standard exponential reliability formulas, it is necessary to assume that the failure rates of all parts within a given unit are constant, that is, they do not increase or decrease with time. Further, those parts that exhibit a wearout mechanism or otherwise have a limited life are replaced at regular intervals prior to any appreciable increase in failure rate.

As an example, an electronic valve has a constant failure rate of 6.5 failures per 1 million cycles. The mean life is 90,000 cycles. The valve is used in equipment that is to have a 10-year life. On an average, the valve will be activated 10 times per hour throughout the equipment's life. Therefore, wearout will occur at 9000 h. In order for the 6.5 failures per 1 million cycles for this device to be valid over the 10-year life of the equipment, this device must be scheduled for replacement every year (8760 h), or more frequently, depending on the consequences of its failure.

Methods of Determining Electromechanical System Reliability

To date, no single method or technique has been accepted as the standard for computing the reliability of electromechanical systems. The following paragraphs briefly describe procedures that are used for computing electromechanical systems reliability.

Generic Failure Rate Data

The easiest and most direct approach to computing the reliability of electromechanical hardware is to use the failure rates of generic parts. As an example, it is possible to find the generic failure rate of an electronically activated valve from several sources. There may be many different valve configurations that fit this general or generic description. However, this valve is probably only one part in a much larger electromechanical system. Therefore, by using generic failure rates for all the parts that make up the system, it is possible to determine which parts will probably be the highest contributors to the system's unreliability. This directs emphasis and design attention to the weak link in the system.

There are several sources of failure rate information with respect to electromechanical hardware. Data sources include:

GIDEP: Government Industry Data Exchange Program

NPRD-95: "Nonelectronic Parts Reliability Data," 1995, Reliability Analysis Center

MIL-HDBK-217F: "Reliability Prediction of Electronic Equipment," Feb. 1995, U.S. Military Handbook

A description of these data source is as follows:

GIDEP Failure Rate Source. GIDEP (http://www.gidep.org/gidep.htm) publishes many documents that are continually being updated to provide government and industry with an in-depth perspective of what is currently taking place in practically every technology. Members contribute test data, reports, and other nonrestricted research data and reports.

A GIDEP summary is issued on a yearly basis. The primary drawback to using GIDEP as a source of failure rate data for a part in a specific environment is the lengthy search through computer databases to obtain needed backup data. This is required when environmental conditions differ from the listed environment, or there is an appreciable difference in part application.

NPRD-95. NPRD-95, "Nonelectronic Parts Reliability Data Notebook" (http://www.rac.iitri.org), published in 1995 by the Reliability Analysis Center, is a report that is organized into four major sections. It presents reliability information based on field operation, dormant state, and test data for more than 250 major nonelectronic part types. The four sections are Background, Generic Data, Detailed Date, and Failure Modes and Mechanisms. Each device type contains reliability information in relation to specific operational environments. The data presented in this reliability publication are intended to be used as failure rate data, not part replacement data (scheduled maintenance). Only verified failures were used in the calculations of the failure rates.

It is noteworthy that: (1) no attempt has been made to develop environmental factors that allow the use of these failure rates for environmental conditions other than those indicated, (2) no attempt has been made to develop temperature history, (3) no attempt has been made to modify these failure rates for various safety factors or materials.

MIL-HDBK-217F. MIL-HDBK-217F, "Reliability Prediction of Electronic Equipment," Feb. 1995, is a U.S. Military Handbook. It includes 19 sections of various types of electronic parts including seven sections on electromechanical devices. These include: Rotating Devices, Relays, Switches, Connectors, Interconnection Assemblies, Connections, and Meters.

Example Reliability Calculation

Note. To convert failure rate per million cycles to failure rate per million hours, multiply the number of cycles per hour that the device operates times the cycle failure rate.

A pressure regulator failure rate is 34.0 failures/million cycles. It will operate at 200 cycles per hour. Therefore,

$$3.40 \text{ failures}/1{,}000{,}000 \text{ cycles} \times 200 \text{ cycles/hour}$$
$$\text{Failure rate} = 680.0 \text{ failures/million hours}$$
$$\text{MTBF} = 1/0.000680 = 1470 \text{ h}$$

For a mission time of 500 h, the probability of success (P_s) is

$$P_s = \exp(-500 \times 0.000680) = 0.712 = 71.2 \text{ percent}$$

Remember, generic failure rate sources assume that parts display a constant failure rate.

Computing Electromechanical Systems Reliability

The method of computing the reliability of a electromechanical system, either MTBF or probability of success, will depend on the situation. The reliability can normally be determined, in descending order of preference by:

1. Failure rate models (product multipliers)
2. Probabilistic stress and strength analysis for each part
3. Probabilistic stress and strength analysis for each critical part, with generic data for all others
4. Accelerated life testing
5. Parts count techniques for all parts using NPRD-95 data, GIDEP data, and/or MIL-HDBK-217F

Because of costs and time constraints, technique number 2 is rarely imposed. For critical electromechanical equipment, technique 1 should be required. For the majority of electromechanical systems, technique 1 or 3 is usually acceptable.

The use of the Parts Count Technique greatly simplifies the reliability computations of electromechanical systems. This technique, basically, is to list all electromechanical parts used, determine the generic failure rate for each part in a specific environment, add up the failure rates, and then compute the reliability, either MTBF or probability of success.

The computed value may not be within a factor of 10 of the actual test results. But the exercise will provide trend data and show which are the critical parts.

RELIABILITY PREDICTION USING PRODUCT MULTIPLIERS

Introduction

The accuracy of a reliability prediction using the generic failure rate data bank approach cannot be determined because of the wide dispersion of failure rates that occur for apparently similar components. A better method for predicting the reliability of electromechanical equipment is to use the product multiplier approach.

Variations in failure rates for electromechanical equipment are the result of the following:

Multiple Functions. Individual electromechanical components may perform more than one function.

Nonconstant Failure Rate. Failure rates of electromechanical components are not usually described by a constant failure rate because of wear, fatigue, and other stress-related failure mechanisms resulting in equipment degradation.

Stress History. Electromechanical equipment reliability is more sensitive to loading, operating mode, and utilization rate than electronic equipment reliability.

Criticality of Failure. Definition of failure for electromechanical equipment depends on its application. For example, failure owing excessive noise or leakage cannot be universally established.

The above-listed variables associated with acquiring failure rate data for electromechanical components demonstrates the need for reliability prediction models that do not rely solely on existing failure rate data banks.

Models have been developed that consider the operating environment, the effects of wear and other potential causes of degradation. The models are based on failure modes and their causes. Equations were developed for each failure mode from design information and experimental data. Failure rate models were developed from the resulting parameters in the equations and modification factors were compiled for each variable to reflect its effect on the failure rate of individual electromechanical parts.

Failure rate equations for each part, the methods used to generate the models in terms of failures per hour or failures per cycle and the limitations of the models are presented in the *Navy Handbook of Reliability Prediction Procedures for Mechanical Equipment* (NSWC-92/L01, May 1992).

Failure rate models have been developed for the following categories of electromechanical components and parts. The following are the basis for the fundamental part failures. Each of these are relatively simple. The complex equations should be simplified by holding constant the less important parameters. This will provide ease of calculation.

Beam: Based on standard beam stress equations

Shaft: Based on standard shaft stress equations

Plate: Based on standard plate stress equations

Spring: Based on standard spring stress equations

Connector, fitting: Based on standard fitting stress equations

Structural joint: Based on standard fitting stress equations

Bearing: As described on p. 3.69

Gear, spline: Based on standard gear stress equations

Seal, gasket: As described below

Solenoid: Based on standard failure rate data

Motor, pump, compressor, transducer, sensor: Based on standard failure rate data

Failure Rate Model for Seals

A seal is a device placed between two surfaces to restrict the flow of a fluid or gas from one region to another. Static seals, such as gaskets and O-rings, are used to prevent leakage through a mechanical joint when there is no relative motion of mating surfaces other than that induced by environmental changes. The effectiveness of a seal design is determined by the capability of the seal interfaces to maintain good mating over the seal operating environment.

Since the primary failure mode of a seal is leakage, the first step in determining the failure rate of a seal is to isolate each leakage path. In many cases backup seals are used in a valve design and the failure rate is not necessarily proportional to the total number of seals in the value. Each seal design must be evaluated as a potential internal or external leakage path.

A review of failure rate data suggest the following characteristics be included in the failure rate model for seals:

- Leakage requirements
- Material characteristics
- Amount of seal compression
- Surface irregularities
- Extent of pressure pulses
- Fluid viscosity
- Fluid/material compatibility
- Static versus dynamic conditions
- Fluid pressure
- Seal size
- Quality control/manufacturing processes
- Contamination level

The failure rate of a seal or gasket material will be proportional to the ratio of actual leakage to that allowable under conditions of usage. This rate can be expressed as follows:

$$\lambda_1 = \lambda_{b1} \frac{Q_a}{Q_f} \tag{7}$$

where λ_1 = failure rate of seal or gasket, failures per million cycles
λ_{b1} = base failure rate of seal or gasket, failures per million cycles
Q_a = actual leakage rate tendency, in^3/min
Q_f = leakage rate considered to be a device failure, in^3/min

The allowable leakage Q_f is determined from design drawings, specifications or knowledge of component applications. The actual leakage rate Q_a for a flat seal or gasket the leakage can be determined from the following equations:

$$Q_a = \frac{2\pi r_1 (P_s^2 - P_o^2)}{24\, vLP_o}(H^3) \tag{8}$$

where P_s = system pressure, lb/in^2
P_o = standard atmospheric pressure or downstream pressure, lb/in^2
v = absolute fluid viscosity lb—min/in^2
r_1 = inside radius
H = conduction parameter, in
L = contact length

The conduction parameter H is dependent on contact stress, material properties and surface finish. The conduction parameter is computed from the empirically devised formula:

$$H^3 = 10^{-1.1(C/M)} \times f^{1.5} \tag{9}$$

where M = Meyer hardness (or) Young's modulus for rubber resilient materials
C = apparent contact stress (psi)
f = surface finish (in)

The surface finish will deteriorate at a rate dependent on the rate of fluid flow and the number of contaminants according to the following relationship:

$$Z \geq f(\alpha\eta QdT) \tag{10}$$

where Z = seal degradation
α = contaminant wear coefficient $(in^3/\text{particle})^2$
η = number of contaminant particles per/in^3
Q = flow rate, in^3/min
d = ratio of time the seal is subjected to contaminants under pressure
t = temperature of operation, °F

The contaminant wear coefficient is a sensitivity factor for the valve assembly based on performance requirements. The number of contaminants includes those produced by wear in upstream components after the filter and those ingested by the system. Combining and simplifying terms provides the following equation for the failure rate of a seal or gasket:

$$\lambda_1 = \lambda_{b1} Xf \left\{ \frac{(P_S^2 - P_O^2) r_1}{Q_f vLP_O}(H^3)(\alpha\eta dt) \right\} \tag{11}$$

where λ_b is the base failure rate of a seal owing to random cuts, installation errors, and so forth, based on field experience data.

$$\lambda_1 = \lambda_{b1} \bullet C_P \bullet C_Q \bullet C_D \bullet C_H \bullet C_f \bullet C_V \bullet C_T \bullet C_N \bullet C_W \tag{12}$$

where λ_1 = failure rate of a seal in failures/million cycles
λ_{b1} = base failure rate of seal
C_P = multiplying factor, which considers the effect of fluid pressure on the base failure rate
C_Q = multiplying factor, which considers the effect of allowable leakage on the base failure rate
C_D = multiplying factor, which considers seal size
C_H = multiplying factor, which considers the effect of surface hardness and other conductance parameters on the base failure rate
C_f = multiplying factor, which considers seal smoothness
C_V = multiplying factor, which considers the effect of fluid viscosity on the base failure rate
C_T = multiplying factor, which considers the effect of temperature on the base failure rate
C_N = multiplying factor, which considers the effect of contaminants on the base failure rate
C_W = multiplying factor, which considers the effect of flow rate on base failure rate

Many of the parameters in the failure rate equation can be located on an engineering drawing, by knowledge of design standards or by actual measurement. Other parameters which have a minor effect on reliability are included in the base failure rate as determined from field performance data.

Reliability Prediction of Bearings

From the standpoint of reliability, bearings are by far the most important gear train components, since they are among the few components that are designed for a finite life. Bearing life is usually calculated using the Lundberg-Palmgren method. This method is a statistical technique based on the subsurface initiation of fatigue cracks through hardened air-melt bearing material. Bearing life is generally expressed as the L_{10} life, which is the number of hours at a given load that 90 percent of a set of apparently identical bearings will complete or exceed (10 percent will fail). There are a number of other factors that can be applied to the L_{10} life so that it more accurately correlates with the observed life. These factors include material, processing, lubrication film thickness, and misalignment.

The mean time to failure (MTTF) of a system of bearings can be expressed as

$$\text{MTTF}_B = \frac{L_{10}(5.45)}{N^{0.9}} \tag{13}$$

where MTTF_B = design MTTF for the system of bearings
L_{10} = design life for each individual bearing
N = number of bearings in system

Most handbooks express the L_{10} life of a bearing to be a Weibull distribution as follows (reference "Marks' Handbook," 8th ed., pp. 8–138):

$$L_{10} = \frac{16{,}700}{RPM}\left(\frac{C}{P}\right)^k \tag{14}$$

where L_{10} = rated life in revolutions
C = basic load rating, lb (look up value from handbooks)
P = equivalent radial, load, lb
K = constant, 3 for ball bearings, 10/3 for roller bearings
RPM = shaft rotating velocity, rev/min

The above expression assumes a Weibull failure distribution with the shape parameter equal to 10/9, the value generally used for the number N of rolling element applicable bearings. The life (L_{10}) given in Eq. (14) includes the effects of lubrication, material, and misalignment under design conditions in addition to the common parameters. This value will usually suffice during early design. Typical material processing factors used for vacuum-melt aircraft quality bearings are 6.0 for ball bearings and 4.0 for roller and tapered bearings.

These material processing factors should be incorporated into the basic $MTTF_B$ to determine a representative life of these type bearings. These factors should be multiplied times the $MTTF_B$ to get the expected MTTF.

If the bearings being considered are low-speed (under 100 RPM), the $MTTF_B$ should be multiplied by 2 to account for the fact that the lubrication factor is very low.

MEMS Bearings. The bearings used in a MEMS application are likely to have metal on metal sliding friction with little or no lubricant. Wear at the bearing surface should be determined by test. These bearings are, however, often used on lightly loaded gear trains. Higher bearing life is easily achieved in these applications if the loads are balanced by using dual gears or a planetary gear train.

The shaft should be a relatively hard material with the gear or armature having a low coefficient of sliding friction with the shaft. The following is a comparison of friction coefficient for different materials (reference "Marks' Handbook," 8th ed. pp. 3–27):

Nickel on nickel	0.53
Zinc on iron	0.21
Lead on iron	0.43
Aluminum on aluminum	1.40

These numbers are reduced by a factor of 3 (approximately) when a lubricant is used.

Reliability Prediction of Other Wearout Parts

Mathematical models provide the capability of predicting the reliability of all types of electromechanical components.

One can also incorporate time-varying parameters such as wear and wearout. To do this it is necessary to quantify the wear rate. This will provide a time to wearout and a probability failure rate about that wearout failure. This same technique works for fatigue life. The only difference is that strength of the material is changing as a function of stress cycles (time and magnitude of load fluctuation). This can be factored into the equation for determining the fatigue life.

Computer Programs for Determining Failure Rate of Parts

Programs for IBM-compatible PCs are available that will generate failure rate data for various electromechanical hardware as described using the product multipliers method.

Mechanical prediction programs are as follows:

MECHREL (mechanical reliability prediction program)

MRP (mechanical reliability prediction program)

RAM commander for Windows

Relex mechanical

MECHREL. Mechanical reliability prediction program performs a mechanical reliability prediction to the component level using the established reliability parameters of electromechanical parts. Environmental and material factors are used in the calculation of base failure rates for the parts. The component failure rates are then summed to give an overall system reliability prediction.

Hard/Software Re.: IBM PC

Supplier: Eagle Technology, Inc. 2300 S. Ninth St., Arlington, VA 22204; Phone: (703) 221-2154

Prices: Public Domain

MRP. Mechanical reliability prediction program addresses the increasing need for predicting failure rates of mechanical equipment. Part of PSI's family of PC-based R&M software. MRP performs reliability prediction analyses in accordance with the "Handbook of Reliability Prediction Producers for Mechanical Equipment," Document DTRC-90/010, prepared by the U.S. Navy David Taylor Research Center under sponsorship of agencies of the DoD. In addition to the mechanical handbook, PSI supports the Reliability Analysis Center document NPRD-95, nonelectronic parts reliability data LIB-91, available as an optional library. The computer program enhances your mechanical reliability predictions by including failure rate data published in NPRD-95.

Hard/Software Req.: DOS 3.0 or higher, 512 kb free RAM, 3 MB hard disk space, any printer/132 column

Interface Capabilities: Interfaces with RPP, FME, SRP, LSA, and other forms

Supplier: Powertronic Systems, Inc., 13700 Chef Menteur Hwy., New Orleans, LA 70129; Phone: (504) 254-0383

RAM Commander for Windows. It is a Windows-based reliability, availability, and maintainability (RAM) tool set that implements a wide variety of electronic, electromechanical, and nonelectronic prediction and analysis methods. Robust reliability analysis tools are based on many predictive methods, including graphics engine, large parts library, block diagram modeling, and reliability growth analysis.

Hard/Software Req.: IBM PC or Comp. Intel Pentium, Microsoft Windows

Interface Capabilities: CAD and other RAM packages

Supplier: Advanced Logistics Developments (ALD) Advanced Automation Corp., P.O. Box 4644, Rome NY 13442-4644; Phone: (800) 292-4519

Relex Mechanical. Available for Windows, Mac, and MS-DOS, Relex Mechanical performs reliability analyses on MEMS systems per the "Handbook of Reliability Prediction Procedures for Mechanical Equipment." Its main purpose is to aid the user in evaluating and improving his products' reliability. It provides a state-of-the-art user interface with extensive hypertext help, CAD interface, defaults and derating analysis, system modeling and redundancy capabilities, and much more.

Hard/Software Req.: IBM PC with Windows or Mac

Interface Capabilities: CAD, Text, ASCII, dBase, Lotus, WordPerfect, Paradox

Supplier: Innovating Software Designs, Inc., One Country Drive, Greensburg, PA 15601; Phone: (412) 836-8800

PROBABILISTIC STRESS AND STRENGTH ANALYSIS

Introduction

Probabilistic design can be determined for each type of part. This is determined by taking the basic equation that describes the stresses in the part and performing a Taylor's series expansion on the basic equation. For example, bending stress in a beam would be calculated using the equation MY/I. However, since I and Y are related (not statistically independent) one must use the section modulus (Z). It is very important to used statistically independent variables.

Probabilistic design can be a very important evaluation tool for predicting the probability of failure of an electromechanical system operating in a random loading environment. Ships, airplanes, missiles, and so forth experience random forces that must considered by the designer during the conceptual phases of the design.

The ultimate design goal would be to manufacture a product that never fails. This, however, is usually not achievable because of other considerations, such as cost and weight, which must be integral parts of any design. However, it may be possible to design a product (or system) that has a low probability of failure and not appreciably impact other design constraints.

Conventional stress analysis does not recognize that material properties, design specifications, and applied loads can be characterized as having a nominal value and some variation. Knowledge of the variables, as well as their statistical properties, will enable the designer to estimate the probability that the part will not fail.

The assumption is made in electromechanical stress and strength analysis that all random variables are statistically independent and are normally distributed. Dependent variables and distributions other than normality are complex to analyze and they usually do not enhance the validity of the analysis.

Statistical Concepts and Principles

Concepts and principles that must be understood to effectively address probabilistic design with respect to the stress-strength relationship are given on the accompanying CD-ROM. See "Concepts and Principles Useful in Electromechanical Stress/Strength Analysis."

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ON THE CD-ROM:

Commonly Used Formulas. Formulas and equations useful in electromechanical reliability analysis.

Concepts and Principles useful in probabilistic stress and strength analysis of electromechanical components.