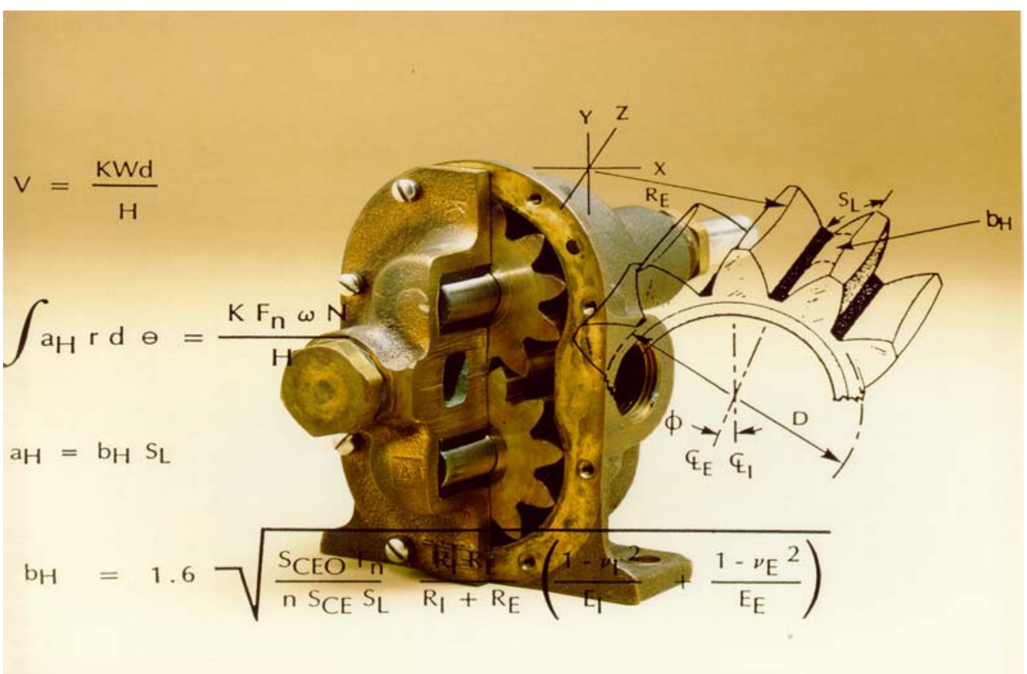
Naval Surface Warfare Center
Carderock Division
West Bethesda, Maryland 20817-5700

Handbook
of
Reliability Prediction Procedures
for
Mechanical Equipment



Logistics Technology Support

CARDEROCKDIV, NSWC-11
May 2011

Approved for Public Release; Distribution is Unlimited

PREFACE

Recognition of reliability and maintainability (R&M) as vital factors in the development, production, operation, and maintenance of today's complex systems has placed greater emphasis on the application of design evaluation techniques to logistics management. An analysis of a design for reliability and maintainability can identify critical failure modes and causes of unreliability and provide an effective tool for predicting equipment behavior and selecting appropriate logistics measures to assure satisfactory performance. Application of design evaluation techniques can provide a sound basis for determining spare parts requirements, required part improvement programs, needed redesign efforts, reallocation of resources and other logistics measures to assure that specified reliability and maintainability requirements will be met.

Many efforts have been applied toward duplicating the data bank approach or developing a new approach for mechanical equipment. The statistical analysis of equipment aging characteristics, regression techniques of equipment operating parameters related to failure rates, and analysis of field failure data have been studied in attempts to develop a methodology that can be used to evaluate a new mechanical design for R&M characteristics.

Many of the attempts to develop R&M prediction methodology have been at a system or subsystem level. The large number of variables at these levels and lack of detailed knowledge regarding operating environment have created a problem in applying the results to the design being evaluated. Attempts to collect failure rate data or develop an R&M prediction methodology at the system or subsystem level produce a wide dispersion of failure rates for apparently similar components because of the basic characteristics of mechanical components.

The Design Evaluation Techniques program was initiated by the Naval Surface Warfare Center Carderock Division (NSWCCD) and was sponsored by the Office of Naval Technology under the Logistics Exploratory Development Program, P.E. 62233N. The methodology for predicting R&M characteristics as part of this development effort does not rely solely on failure rate data. Instead, the design evaluation procedures consider the material properties, operating environment and critical failure modes at the component part level to evaluate a design for R&M. The purpose of this Handbook is to present the methodology for predicting the reliability of mechanical equipment and solicit comments as to the potential utility of a standard reference for reliability predictions of mechanical equipment.

The development of this Handbook by NSWCCD was coordinated with the military, industry and academia. Sponsors of this effort included the U. S. Army Armament Research, Development & Engineering Center (SMCAR-QAH-P), Picatinny Arsenal and the Robins AFB, WR-ALC/LVRS. These sponsors have provided valuable technical guidance in the development of the methodology and Handbook. Chapter 1 of the Handbook provides a summary of the testing program to validate the prediction methodology. Also, the Robins AFB supplied an MC-2A Air Compressor Unit for validation testing purposes. The procedures contained in this Handbook were used to predict the failure modes of the MC-2A and their frequency of occurrence. Reliability tests were then performed with a close correlation between predicted and actual reliability being achieved. Past sponsors and participants in the program include the Belvoir Research, Development, & Engineering Center; Wright-Patterson AFB; Naval Sea Systems Command; Naval Air Test Center and Louisiana Tech University.

Previous editions of this Handbook were distributed to interested engineering personnel in industry and DoD for comments as to the utility of the methodology in evaluating mechanical designs for reliability. The comments have been extremely useful in improving the prediction methodology and contents of the Handbook. The revised Handbook is available at no charge and can be downloaded by visiting the NSWCCD website. Every effort has been made to validate the equations presented in this Handbook. However, limited funding has prevented the extensive testing and application of prediction procedures to the design/procurement process for full validation of the approach. Therefore, users are cautioned that this Handbook is the result of a research program and not an official DoD document.

Several companies have chosen to produce software packages containing the material in this Handbook, the attempt being to sell a software package whereby the reliability of mechanical components can be predicted in the same way as electronic components. The Navy has not been and is not now in any way connected with the commercial ventures to produce software packages. As described previously, it is important to understand the difference between the failure rate data used to evaluate electronic equipment and the procedures used to evaluate mechanical equipment. For a company to extract equations from the Handbook without regard to the application procedures is in violation of the intent of the Handbook, the result being a potentially dangerous situation for the user in logistically relying on inaccurate results. Another result is the damaging reputation to NSWCCD and the Navy in their attempts to improve the reliability of mechanical equipment through a greater understanding of mechanical system design. To extract equations from the Handbook without regard to the procedures and parameter limits defeats the purpose of the Handbook in helping the designer of mechanical systems gain a greater insight as to the reliability of his design.

NSWCCD has developed a software package that automates the use of procedures and equations in the Handbook that can be used to evaluate the methodology. This software program called MechRel can be downloaded free of charge by visiting the NSWCCD website. In summary, the Handbook and associated software package representing many years of research and development are already available at no charge. Commercial exploitation of this work by extracting material without the full content of the evaluation procedures violates the purpose of the work being done by

CDNSWC. Any product sold using material from the Handbook or referencing the Handbook must contain a statement that CDNSWC and the Navy have not participated in the development of or approve of their product.

Interested users of the technology presented in this Handbook are urged to contact CDNSWC to obtain the latest available information on mechanical reliability. Comments and recommended changes to the Handbook should be addressed to:

Tyrone L. Jones
MechRel Program Manager
Code 2120
Naval Surface Warfare Center
9500 MacArthur Blvd
West Bethesda, MD 20817-5700
Telephone: 301-227-4383
E-mail: Tyrone.L.Jones@navy.mil

This Page Intentionally Left Blank

CONTENTS

RELIABILITY PREDICTION PROCEDURES FOR MECHANICAL EQUIPMENT

CARDEROCKDIV, NSWC-11

This Page Intentionally Left Blank

Handbook of Reliability Prediction Procedures for Mechanical Equipment

Change Record

Chapter	Revision	Page	Date	Change
Preface	A	ii,iii	02/05/06	Corrected Handbook downloading address, e-mail address and added additional disclaimers
1	A	7-11	10/07/05	Revised Table 1-1 and supporting data to reflect revisions to referenced chapters
1	B	1-6 to 1-12	12/13/09	Revised Figure 1.2 and Table 1-1 to reflect changes in Chapters 3 and 6, added illustration for example procedures and corrected definition for spring load
2	A	All	02/21/11	Added definitions reflecting chapter additions and revisions
3	A	3-24	01/12/05	Corrected equation for hardness factor
3	B	3-6	11/07/05	Added procedures for pneumatic applications, updated viscosity tables and references and corrected equation for conductance parameter
3	C	all	12/28/05	Corrected equations for gaskets added multiplying factor for gasket dimensions
3	D	3-6	07/13/06	Corrected error in equation 3-4
3	E	3-22	05/20/08	Deleted Seal Pressure Table and clarified CsubH parameter derivation. Corrected parameter identifiers for equation (3-15), added Figure 3.16 for surface finish of dynamic seals.
3	F	all	09/05/09	Expanded failure modes for dynamic seals, corrected equation 3-14 and base failure rate, and added FMECA section
3	G	all	02/01/11	Added procedures for evaluating mechanical seals. Separated procedures for dynamic seals depending on velocity of movement. Updated illustrations.

Chapter	Revision	Page	Date	Change
4	A	4-32	01/05/05	Corrected multiplying factor for spring cycle rate
4	B	4-7	11/07/05	Inserted missing constants in Equation 4-5 and corrected exponent error on various failure rate equations
4	C	all	09/15/07	Updated various tables for properties of spring materials
4	D	4-29 4-34	06/02/08	Corrected multiplying factors for beam and cantilever springs and equation 4-15 for wave washers
4	E	all	02/01/11	Added information regarding the cyclic modes of spring operation and spring life. Corrected titles of various Figures.
5	B	5-4	01/12/05	Modified definition of coil surface area. Corrected references to Figures
5	C	all	05/01/08	Corrected equation 5-1 Separated procedures for evaluating solenoids and contactors
5	D	all	10/05/08	Revised base failure rates and prediction procedures per recent research, Changed solenoid base failure rate from hours to operations
5	E	all	02/01/11	Revised failure rate equation for solenoid assembly and multiplying factor for coil temperature
6	A	6-11	01/12/05	Corrected equation 6-11 and Figure 6.3
6	B	6-6	11/07/05	Added procedures for gas valves, updated viscosity tables and references .Corrected typo in Equation 6-4
6	C	6-12	06/01/08	Simplified and corrected poppet seat stress equations and corrected Fig 6-10, seat Stress Multiplying Factor. Corrected reference to Figure 6.10

Chapter	Revision	Page	Date	Change
6	D	6-3	08/30/09	Added gas valve failure modes, expanded failure modes section, added FMECA guidelines and changed seat stress parameter to contact pressure
6	E	6-8	11/10/10	Revised procedure for determining valve seat contact pressure
7	A	all	03/01/04	Revised procedures for determining dynamic loading of bearings
7	B	all	09/15/07	Revised procedures for evaluating equivalent radial load
7	C	all	10/10/08	Modified procedures for L10 determination
7	D	7-6, 7-12	01/15/11	Revised equation and procedure for determining bearing failure rate, applied load and service factor
8	A	8-10 8-14	07/05/09	Corrected spline load factor equation. Added section on failure modes and explanation for use of Brinell hardness number. Changed title of section to Gear Loading M.F., modified equations 8-11 and 8-13. Corrected references to Section numbers and added explanation to equation 8-13
9	A	9-3, 9-4	09/10/07	Added explanation of Phase 2 wear, corrected equation 9-5 for axial loads, corrected equation 9-24 for temperature factor. Added table of failure modes
9	B	9-6	08/15/08	Added explanatory notes on side and axial loading, eliminated need for phase two calculations
9	C	all	03/01/11	Expanded failure mode list, modified temperature and contaminant modifying factors
10	A	10-9	01/24/05	Added labels to Figure 10-4

10	B	All	09/20/05	Included sections for displacement and centrifugal pumps and revised equations. Revised contaminant multiplying factor,
10	C	All	12/02/10	Expanded description of pump types, added sections on pump seals and bearings, modified equation to project failure rate of fluid driver with percent flow and service factor.
11	A	11-8 to 11-11	07/15/08	Corrected equation 11-5 and added explanatory notes on filter life. Modified Table 11-2 adding values for x for equation 11-10Revised equations 11-1, 11-7 and 11-8. Replaced equation 11-9 with reference material.
11	B	all	02/08/11	Expanded failure modes regarding filter performance, Updated equation for filter failure rate to include cyclic flow and corrected equation for pressure multiplying factor.
12	A	12-19	06/20/06	Corrected typo in Equation 12-16
12	B	all	12/20/06	Corrected nomenclatures for Equation 12-12 and typos following Equation 12-20. Added references
12	C	all	03/01/11	Revised general procedures foe determining failure rates of brake and clutch assemblies
13	A	all	09/20/05	Included procedures for various types of compressors, revised table of multiplying factors
13	B	all	03/01/11	Expanded list of failure modes to include various types of compressors, modified failure rate equation to include service multiplying factor, updated reference list.
14	A	14-6	09/15/07	Revised procedures for different classes of motors

Chapter	Revision	Page	Date	Change
14	B	all	11/12/10	Expanded failure mode table, Corrected motor failure rate equation to add base failure rate and motor load service factor. Corrected method of determining winding temperature rise and voltage unbalance. Corrected voltage and altitude correction factors.
15	A	15-3 15-6 15-12	08/30/07	Added figure to identify accumulator types. Corrected several equation subscripts and added explanation of failure distribution
15	B	15-2, 7	07/22/10	Updated illustrations
16	A	all	02/20/07	Corrected equation 16-14
16	B	all	02/01/11	Added table of failure modes, added base failure rate list for types of fasteners, modified procedure for dynamic loading, added vibration modifying factor
17	A	17-1 17-10	09/15/07	Added introductory material on various types of couplings and table of service factors
17	B	17-8	04/01/11	Modified procedure to determine coupling failure rate and added service factor to coupling failure rate equation.
18	A	all	04/15/11	Added explanatory material on sliding bearings
19	A	all	12/15/08	Corrected heading of Table 19-3. Revised base failure rates and prediction procedures per recent research
19	B	19-9	03/25/11	Revised table of failure rates for sensing elements
20	A	20-1	09/15/07	Corrected equation 20-3 removing contaminant multiplying factor. Expanded introduction to include various types of shafts and applications. Corrected equation 20-8 to agree with Figure 20.2

20	B	20-6	12/12/09	Modified stress concentration procedures
20	C	all	12/15/10	Added table of shaft failure modes. Corrected equations 20-4 and 20-7. Added table of shaft bending limites.
21	A	21-6	04/03/06	Clarified parameter for Equation 21-3 and added sections for chain drive evaluation.
21	B	21-10	02/01/11	Added section on belt load rating
22	A	all	04/05/11	Revised procedures for evaluating fluid conductor failure rates
23	A	all	05/10/08	Corrected base failure rates, added service multiplying factors
23	B	all	04/25/11	Added references to other Handbook Chapters
24	A	all	09/15/10	Revised chapter to combine FMECA, RCM and FTA into a Design Analysis of Equipment Availability
25	D	all	05/01/11	Added new references as needed

CHAPTER 1

INTRODUCTION

1.0 TABLE OF CONTENTS

1.1 PREFACE

The "Handbook of Reliability Prediction Procedures for Mechanical Equipment" has been developed by the Logistics Technology Support Group, Naval Surface Warfare Center Carderock Division (NSWCCD) in Bethesda, Maryland. The handbook presents a new approach for determining the reliability and maintainability (R&M) characteristics of mechanical equipment. It has been developed to help the user identify equipment failure modes and potential causes of unreliability in the early design phases of equipment development, and then to quantitatively evaluate the design for R&M and determine logistics support requirements.

A software program called "MechRel" has also been developed. The Handbook and MechRel software program are available free of charge from NSWCCD. Contact information is contained in Section 1.6

1.2 CURRENT METHODS OF PREDICTING RELIABILITY

A reliability prediction is performed in the early stages of a development program to support the design process. Performing a reliability prediction provides for visibility of equipment reliability requirements in the early development phase. A well done prediction also provides an awareness of potential equipment degradation during the equipment life cycle. As a result of performing a reliability prediction, equipment designs can be improved, costly over-designs prevented and development testing time optimized.

Performance of a reliability prediction for electronic equipment is well established by research and development. For example, MIL-HDBK-217 has been developed for predicting the reliability of electronic equipment. Development of this document was made possible because the standardization and mass production of electronic parts has permitted the creation of valid failure rate data banks for high population electronic devices. Such extensive sources of quality and reliability information can be used directly to predict operational reliability while the electronic design is still on the drawing board.

A commonly accepted method for predicting the reliability of mechanical equipment based on a data bank has not been possible because of the wide dispersion of failure rates which occur for apparently similar components. Inconsistencies in failure rates for mechanical equipment are the result of several basic characteristics of mechanical components:

a. Individual mechanical components such as valves and gearboxes often perform more than one function and failure data for specific applications of nonstandard components are seldom available. A hydraulic valve for example may contain a manual shut-off feature as well as an automatic control mechanism on the same valve structure.

b. Failure rates of mechanical components are not usually described by a constant failure rate distribution because of wear, fatigue and other stress-related failure mechanisms resulting in equipment degradation. Data gathering is complicated when the constant failure rate distribution can not be assumed and individual times to failure must be recorded in addition to total operating hours and total failures.

c. Mechanical equipment reliability is more sensitive to loading, operating mode and utilization rate than electronic equipment reliability. Failure rate data based on operating time alone are usually inadequate for a reliability prediction of mechanical equipment.

d. Definition of failure for mechanical equipment depends upon its application. For example, failure due to excessive noise or leakage can not be universally established. Leakage requirements for a water system are obviously different than those for a fuel system. Lack of such information in a failure rate data bank limits its usefulness.

The above deficiencies in a failure rate data base result in problems in applying published failure rates to an actual design analysis. The most commonly used tools for determining the reliability characteristics of a mechanical design can result in a useful listing of component failure modes, system level effects, critical safety related issues, and projected maintenance actions. However, estimating the design life of mechanical equipment is a difficult task for the design engineer. Many life-limiting failure modes such as corrosion, erosion, creep, and fatigue operate on the component at the same time and have a synergistic effect on reliability. Also, the loading on the component may be static, cyclic, or dynamic at different points during the life cycle and the severity of loading may also be a variable. Material variability and the inability to establish an effective data base of historical operating conditions such as operating pressure, temperature, and vibration further complicate life estimates.

Although several analytical tools such as the Failure Modes, Effects and Criticality Analysis (FMECA) are available to the engineer, they have been developed primarily for electronic equipment evaluations, and their application to mechanical equipment has had limited success. The FMECA, for example, is a very powerful technique for identifying equipment failure modes, their causes, and the effect each failure mode will have on system performance. Results of the FMECA provide the engineer with a valuable insight as to how the equipment will fail; however, the problem in completing the FMECA for mechanical components is determining the probability of occurrence for each identified failure mode.

The above listed problems associated with acquiring failure rate data for mechanical components demonstrates the need for reliability prediction models that do not rely solely on existing failure rate data banks. Predicting the reliability of mechanical equipment requires the consideration of its exposure to the environment and subjection to a wide range of stress levels such as impact loading. The approach to predicting reliability of mechanical equipment presented in this Handbook considers the intended operating environment and determines the effect of that environment at the lowest part level where the material properties can also be considered. The combination of these factors permits the use of engineering design parameters to determine the design life of the equipment in its intended operating environment and the rate and pattern of failures during the design life. The Handbook also includes a procedure for performing a design analysis of equipment availability that combines the procedures for performing a FMECA, a Fault Tree analysis (FTA) and a Reliability Centered Maintenance (RCM) analysis into one streamlined design analysis procedure for mechanical equipment.

1.3 DEVELOPMENT OF THE HANDBOOK

Useful models must provide the capability of predicting the reliability of all types of mechanical equipment by specific failure mode considering the operating environment, the effects of wear and other potential causes of degradation. The models developed for the Handbook are based upon identified failure modes and their causes. The first step in developing the models was the derivation of equations for each failure mode from design information and experimental data as contained in published technical reports and journals. These equations were simplified to retain those variables affecting reliability as indicated from field experience data. Modification factors were then compiled for each variable to reflect its quantitative impact on the failure rate of an individual component part. The total failure rate of the component is the sum of the failure rates for the component parts for a particular time period in question. Failure rate equations for each component 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 discussed in each chapter of the Handbook. The equations and procedures were validated to the extent possible with laboratory testing or engineering analysis.

The objective of the Handbook and MechRel software program is to provide procedures which can be used for the following elements of a reliability program:

- Evaluate designs for reliability in the early stages of development
- Provide management emphasis on reliability with standardized evaluation procedures
- Provide an early estimate of potential spare parts requirements
- Quantify critical failure modes for initiation of specific stress or design analyses
- Provide a relative indication of reliability for performing trade off studies, selecting an optimum design concept or evaluating a proposed design change
- Determine the degree of degradation with time for a particular component or potential failure mode
- Design accelerated testing procedures for verification of reliability performance

One of the problems any engineer can have in evaluating a design for reliability is attempting to predict performance at the system level. The problem of predicting the reliability of mechanical equipment is easier at the lower indenture levels where a clearer understanding of design details affecting reliability can be achieved. Predicting the life of a mechanical component, for example, can be accomplished by considering the specific wear, erosion, fatigue and other deteriorating failure mechanism, the lubrication being used, contaminants which may be present, loading between the surfaces in contact, sliding velocity, area of contact, hardness of the surfaces, and material properties. All of these variables would be difficult to record in a failure rate data bank; however, the derivation of such data can be achieved for individual designs and the potential operating environment can be brought down through the system level and the effects of the environmental conditions determined at the part level.

The development of design evaluation procedures for mechanical equipment includes mathematical equations to estimate the design life of mechanical components. These reliability equations consider the design parameters, environmental extremes, and operational stresses to predict the reliability parameters. The equations rely on a base failure rate derived from laboratory test data where the exact stress levels are known. Engineering equations are used to modify this failure rate to the appropriate stress/strength and environmental relationships for the equipment application. Figure 1.1 illustrates the method of considering the effects of the environment and the operating stresses at the lowest indenture level.

A component such as a valve assembly may consist of seals, springs, fittings, and the valve housing. The design life of the entire mechanical system is accomplished by evaluating the design at the component and part levels considering the material properties of each part. The operating environment of the system is included in the equations by determining its impact at the part level. Some of the component parts may not have a constant failure rate as a function of time and the total system failure rate of the system can be obtained by adding part failure rates for the time period in question.

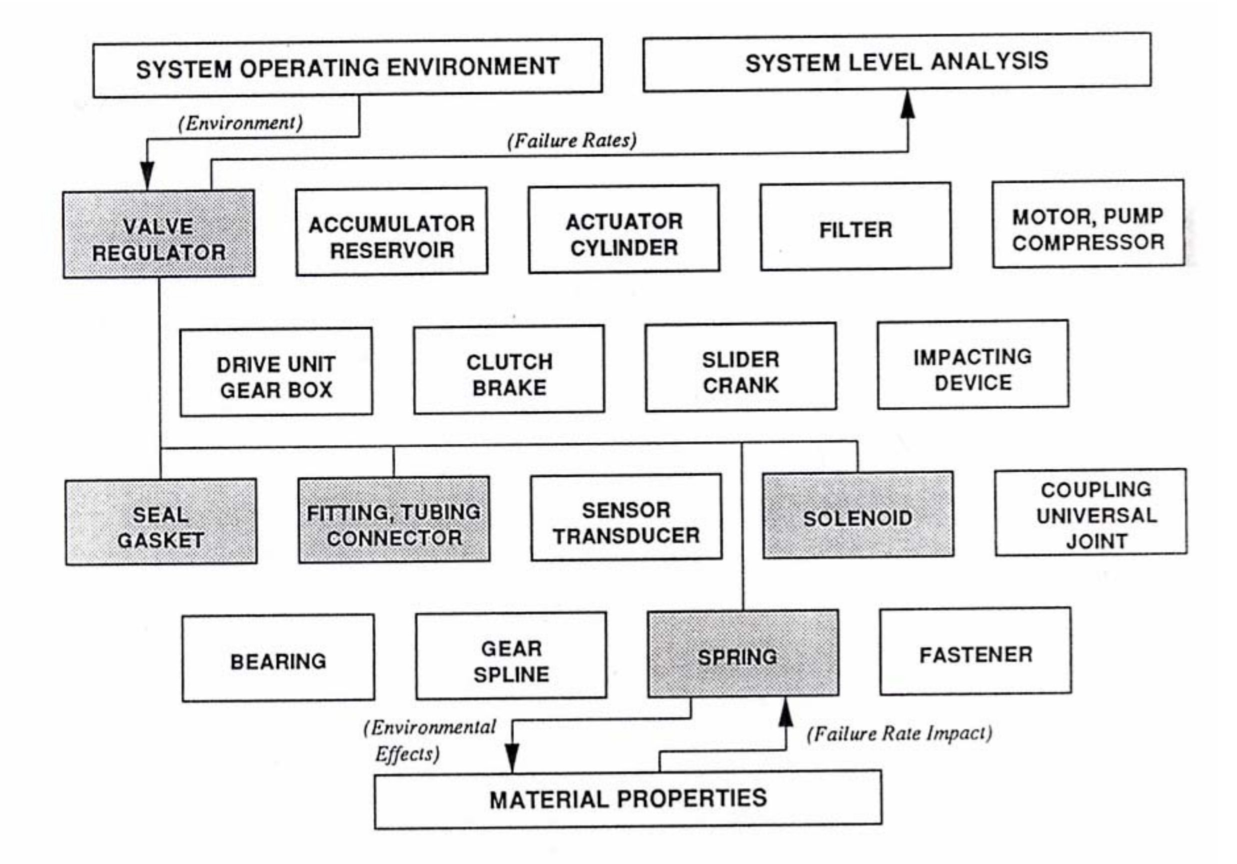


Figure 1.1 Mechanical Components and Parts

Many of the parts are subject to wear and other deteriorating type failure mechanisms and the reliability equations must include the parameters which are readily accessible to the equipment designer. As part of this research project, Louisiana Tech University was tasked to establish an engineering model for mechanical wear which is correlated to the material strength and stress imposed on the part. This model for predicting wear considers the materials involved, the lubrication properties, the stress imposed on the part and other aspects of the wear process (Reference 72). The relationship between the material properties and the wear rate was used to establish generalized wear life equations for actuator assemblies and other components subject to surface wear.

In another research project, lubricated and unlubricated spline couplings were operated under controlled angular misalignment and loading conditions to provide empirical data to verify spline coupling life prediction models. This research effort was conducted at the Naval Air Warfare Center in Patuxent River, Maryland (Reference 71). A special rotating mechanical coupling test machine was developed for use in generating reliability data under controlled operating conditions. This high-speed closed loop testbed was used to establish the relationships between the type and volume of lubricating grease employed in the spline coupling and gear life. Additional tests determined the effects of material hardness, torque, rotational speed and angular misalignment on gear life.

Results of these wear research projects were used to develop and refine the reliability equations for those components subject to wear.

1.4 EXAMPLE DESIGN EVALUATION PROCEDURE

A hydraulic valve assembly will be used to illustrate the Handbook approach to predicting the reliability of mechanical equipment. An example diagram of a valve assembly is shown in Figure 1.2. Developing reliability equations for all the different types of hydraulic valves would be an impossible task since there are over one hundred different types of valve assemblies available. For example, some valves are named for the function they perform, e.g. check valve, regulator valve and unloader valve. Others are named for a distinguishing design feature, e.g. globe valve, needle valve, solenoid valve. However, from a reliability standpoint, dropping down one indenture level provides two basic types of valve assemblies: the poppet valve and the sliding action valve.

The example assembly chosen for analysis is a poppet valve which consists of a poppet assembly, spring, seals, guide and housing.

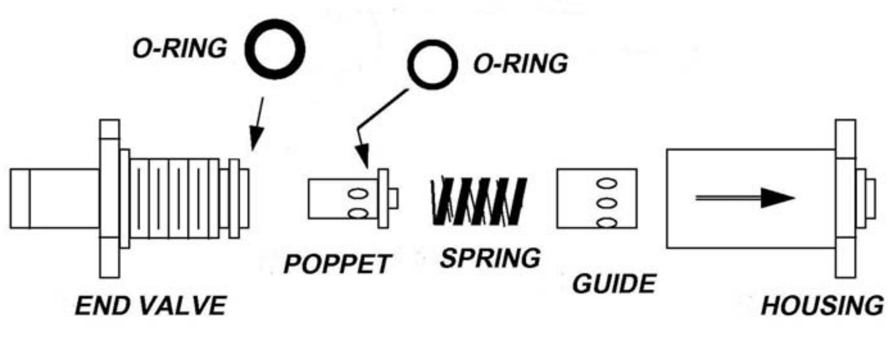


Figure 1.2 Valve Assembly

1.4.1 Poppet Assembly

The functions of the poppet valve would indicate the primary failure mode as incomplete closure of the valve resulting in leakage around the poppet seat. This failure mode can be caused by contaminants being wedged between the poppet and seat, wear of the poppet seat, and corrosion of the poppet/seat combination. External seal leakage, sticking valve stem, and damaged poppet return spring are other failure modes which must be considered in the design life of the valve.

A new poppet assembly may be expected to have a sufficiently smooth surface for the valve to meet internal leakage specifications. However, after some period of time contaminants will cause wear of the poppet assembly until leakage rate is beyond tolerance. This leakage rate, at which point the valve is considered to have failed, will depend on the application and to what extent leakage can be tolerated.

As derived in Chapter 6 of the Handbook, the following equation can be used to determine the failure rate of a poppet assembly:

$$\lambda_P = \lambda_{P,B} \; \frac{2 \, x \, 10^2 \, D_M \; f^{\,3} \left(P_1^2 - P_2^{\,2}\right) K_1}{Q_f \, \nu_a \, L_W \left(S_S\right)^{1.5}}$$

Where:

- λ_P = Failure rate of the poppet assembly, failures/million cycles
- $\lambda_{P,B}$ = Base failure rate for poppet assembly, failures/million cycles
- D_M = Mean seat diameter, in
- f = Mean surface finish of opposing surfaces, in
- P_1 = Upstream pressure, lbs/in^2
- P_2 = Downstream pressure, lbs/in^2
- K_1 = Constant which considers the impact of contaminant size, hardness and quantity of particles
- Q_f = Leakage rate considered to be a valve failure, in^3/min
- ν_a = Absolute fluid viscosity, lb-min/in^2
- L_W = Radial seat land width, in
- S_S = Apparent seat stress, lb/in^2

Values used to determine the failure rates for the parts used in this example are listed in Table 1-1. Throughout the Handbook, failure rate equations for each component and part are translated into a base failure rate with a series of multiplying factors to modify the base failure rate to the operating environment being considered. For example, as shown in Equation (6-6) of Chapter 6, the above equation can be rewritten as follows:

$$\lambda_{PO} = \lambda_{PO,B} \bullet C_P \bullet C_Q \bullet C_F \bullet C_\nu \bullet C_N \bullet C_S \bullet C_{DT} \bullet C_{SW} \bullet C_W$$

Where: λ_{PO} = Failure rate of poppet assembly in failures/million operations

$\lambda_{PO,B}$ = Base failure rate of poppet assembly, 1.40 failures/million operations

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_F = Multiplying factor which considers the effect of surface finish on the base failure rate

C_ν = Multiplying factor which considers the effect of fluid viscosity on the base failure rate

C_N = Multiplying factor which considers the effect of contaminants on the base failure rate

C_S = Multiplying factor which considers the effect of seat stress on the base failure rate

C_{DT} = Multiplying factor which considers the effect of seat diameter on the base failure rate

C_{SW} = Multiplying factor which considers the effect of seat land width on the base failure rate

C_W = Multiplying factor which considers the effect of fluid flow rate on the base failure rate

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

1.4.2 <u>Spring Assembly</u>

Depending on the application, a spring may be in a static, cyclic, or dynamic operating mode. In the current example of a valve assembly, the spring will be in a cyclic mode. The operating life of a mechanical spring arrangement is dependent upon the susceptibility of the materials to corrosion and stress levels (static, cyclic or dynamic). The most common failure modes for springs include fracture due to fatigue and excessive loss of load due to stress relaxation. Other failure mechanisms and causes may be identified for a specific application. Typical failure rate considerations include: level of loading, operating temperature, cycling rate and corrosiveness of the fluid environment. Other failure modes to be considered are listed in Chapter 4.

The failure rate of a compression spring depends upon the stress on the spring and the relaxation properties of the material. The load on the spring is equal to the spring rate multiplied by the deflection and calculated as explained in Chapter 4.

$$P_L = R\left(L_1 - L_2\right) = \frac{G_M \left(D_W\right)^4 \left(L_1 - L_2\right)}{8\left(D_C\right)^3 N_a}$$

Where: P_L = Load, lbs

R = Spring rate, lb/in

L_1 = Initial deflection of spring, in

L_2 = Final deflection of spring, in

G_M = Modulus of rigidity, lb/in^2

D_W = Mean diameter of wire, in

D_C = Mean diameter of spring, in

N_a = Number of active coils

Stress in the spring will be proportional to loading according to the following relationship:

$$S_G = \frac{8\,P_L\,D_C\,K_W}{\pi\,D_W{}^3}$$

Where: S_G = Actual stress, psi

K_W = Wahl stress correction factor

This equation permits determination of expected life of the spring by plotting the material S-N curve on a modified Goodman diagram. In the example valve application, the spring force and the failure rate remain constant. This projection is valid if the spring does not encounter temperature extremes. Corrosion is a critical factor in spring design because most springs are made of steel which is susceptible to a corrosive environment. In this example the fluid medium is assumed to be non-corrosive and the spring is always surrounded by the fluid, thus a corrosion factor need not be included in this analysis. If the valve were a safety device and subjected intermittently to a steam environment, then a corrosion factor would have to be applied consistent with any corrosion protection in the original spring design.

The failure rate of the compression spring can be estimated from the following equation:

$$\lambda_{SP} = \lambda_{SP,B} \left(\frac{S_G}{T_S} \right)^3 = \lambda_{SP,B} \left(\frac{8\ P_L\ D_C\ K_W}{\pi\ T_S\ D_W{}^3} \right)^3$$

where:

T_S = Material tensile strength, lbs/in^2

Other multiplying factors based on field performance data are detailed in Chapter 4.

1.4.3 <u>Seal Assembly</u>

The primary failure mode of a seal is leakage, and the following equation as derived in Chapter 3 uses a similar approach as developed for evaluating a poppet design:

$$\lambda_{SE} = \lambda_{SE,B}\ \frac{K_1\ (P_1^2 - P_2{}^2)}{Q_f\ \nu_a\ P_2} \bullet \frac{r_o + r_i}{r_o - r_i} \bullet H^3$$

Where: λ_{SE} = Failure rate of seal, failures/million cycles

$\lambda_{SE,B}$ = Base failure rate of seal, failures/million cycles

K_1 = Constant = 3.27 x 10^{-4}

P_1 = System pressure, lb/in^2

P_2 = Standard atmospheric pressure or downstream pressure, lb/in^2

Q_f = Allowable leakage rate under conditions of usage, in^3/min

ν_a = Absolute fluid viscosity, lb-min/in^2

r_i = Inside radius of circular interface, in

r_o = Outside radius of circular interface, in

H = Conductance parameter (Meyer hardness M; contact pressure C; surface finish f), in

The conductance parameter is a combination of Meyer hardness, contact pressure and surface finish per the following equation:

$$H = 0.23 \left(\frac{M}{C}\right)^{1.5} \bullet f^{2/3}$$

Where: M = Meyer hardness (or Young's modulus) for rubber and resilient materials, lbs/in^2

C = Contact stress, lbs/in^2

f = Surface finish, in

In the case of an O-ring seal, the failure rate will increase as a function of time because of gradual hardening of the rubber material. A typical failure rate curve for an O-ring is shown in Figure 1.2. Multiplying factors considering such parameters as fluid temperature are detailed in Chapter 3.

1.4.4 Combination of Failure Rates

The addition of failure rates to determine the total valve failure rate depends on the life of the valve and the maintenance philosophy established. If the valve is to be discarded upon the first failure, a time-to-failure can be calculated for the particular operating environment. If, on the other hand, the valve will be repaired upon failure with the failed part(s) being replaced, then the failure rates must be combined for different time phases throughout the life expectancy until the wear-out phase has been reached. The effect of part replacement and overhaul is a tendency toward a constant failure rate at the system level and will have to be considered in the prediction for the total system.

The housing will exhibit an insignificant failure rate, usually verified by experience or by finite element analysis. Typical values as assumed for the example equations are listed in Table 1-1.

After the failure rates are determined for each component part, the rates are summed to determine the failure rate of the total valve assembly. Because some of the parameters in the failure rate equation are time dependent, i.e. the failure rate changes as a function of time, the total failure rate must be determined for particular intervals of time. In the example of the poppet assembly, nickel plating was assumed with an initial surface finish of 35 μ inches. The change in surface finish over a one year time period for non-acidic fluids such as water, mild sodium chloride solutions, and hydraulic fluids will be a deterioration to 55 μ inches. In the case of the O-ring seal, the hardness of the rubber material will change with age. The anticipated failure rate as a function of time for the component parts of the valve and the total valve assembly are shown in Figure 1.3.

Table 1-1. Typical Values for Failure Rate Equations

POPPET		SPRING		SEAL	
PARAMETER	VALUE	PARAMETER	VALUE	PARAMETER	VALUE
$\lambda_{P,B}$	1.40	$\lambda_{SP,B}$	23.8	$\lambda_{SE,B}$	2.40
Q_f	0.06	L_1	3.35	Q_f	0.06
D_M	1.69	L_2	2.28	P_1	3000
F *	35 E-6	G_M	11.5 E 6	P_2	15
P_1	3000	D_C	0.58	ν_a	2 E -8
P_2	15.0	D_W	0.085	r_i	0.17
ν_a	2 E-8	N_a	14	r_o	0.35
L_W	0.85	T_S	245 E3	M/C **	0.55
S_s	4045	P_L	29.4	f	35 E-6
K_1	1.00	S_G	86.2 E 3	H	1.02 E-4
Ops/hour	0.5	K_W	1.219	K_1	3.27 E-4
TOTALS:					
λ_P	0.35	λ_{SP}	1.04	λ_{SE}	1.20

* Initial value = 35 μin; after 8,000 operating hours (4,000 operations) surface finish will equal 55 μin (Reference 5)
** Initial value = 0.55 (hardness, M = 500 psi; contact stress, C = 910 psi); after 1 year M estimated to be 575 psi (M/C = 0.63)

1.5 VALIDATION OF RELIABILITY PREDICTION EQUATIONS

A very limited budget during the development of this handbook prevented the procurement of a sufficiently large number of components to perform the necessary failure rate tests for all the possible combinations of loading roughness, operational environments, and design parameters to reach statistical conclusions as to the accuracy of the reliability equations. Instead, several test programs were conducted to verify the identity of failure modes and validate the engineering approach being taken to develop the reliability equations. For example, valve assemblies were procured and tested at the Belvoir Research, Development and Engineering Center in Ft. Belvoir, Virginia. The number of failures for each test was predicted using the equations presented in this handbook. Failure rate tests were performed for several combinations of stress levels and results compared to predictions. Typical results are shown in Table 1-2.

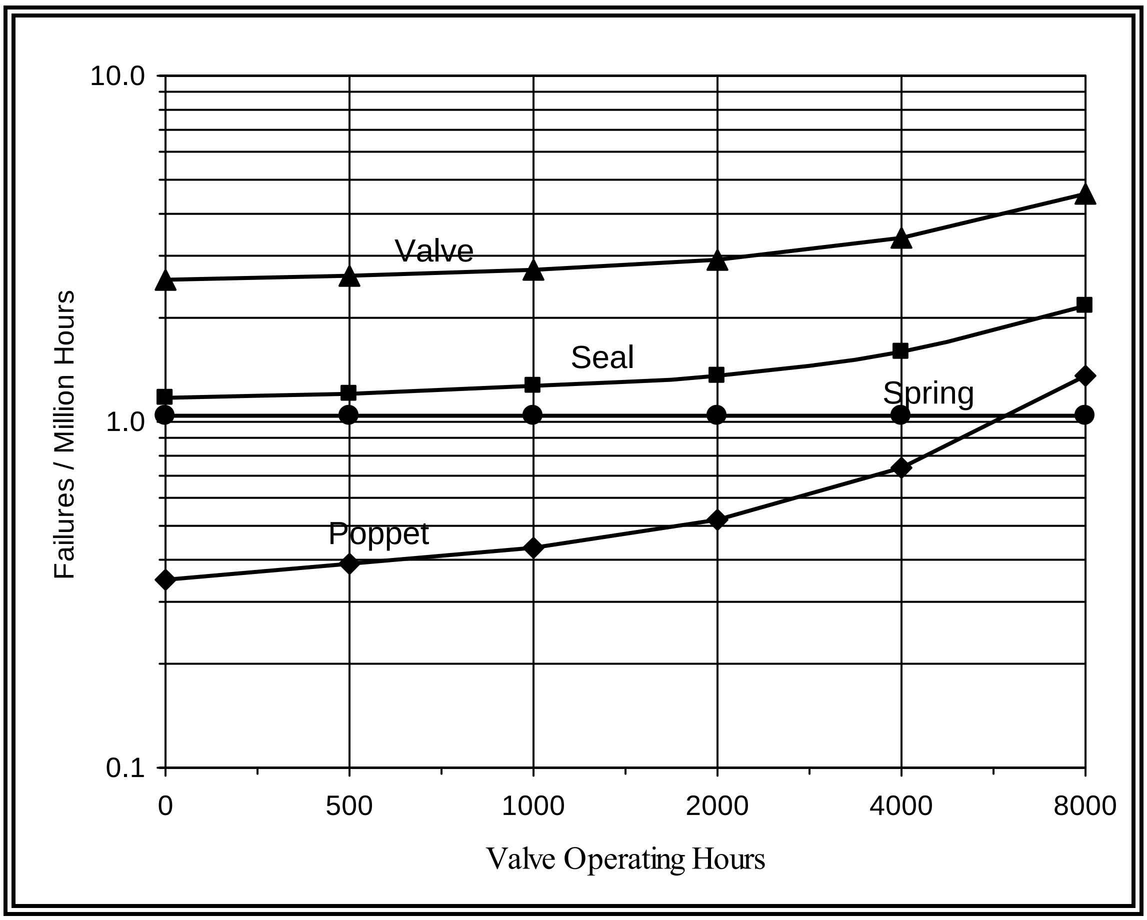


Figure 1.3 Combination of Component Failure Rates

Table 1-2. Sample Test Data for Validation of Reliability Equations for Valve Assemblies

TEST SERIES	VALVE NUMBER	TEST CYCLES TO FAILURE	ACTUAL FAILURES/ 10^6 CYCLES	AVERAGE FAILURES/ 10^6 CYCLES	PREDICTED FAILURES/ 10^6 CYCLES	FAILURE MODE #
15	11	68,322	14.64	14.64	18.02	3
24	8	257,827				1
24	9	131,126	7.63	10.15	10.82	1
24	10	81,113	12.33			1
24	11	104				2
24	12	110,488	9.05			1
24	13	86,285	11.59			1
25	14	46,879	21.33	19.67	8.45	2
25	15	300				3
25	18	55,545	18.00			1

TEST PARAMETERS:

SYSTEM PRESSURE: 3500 psi FLUID FLOW: 100% rated
FLUID TEMPERATURE: 90 C FLUID: Hydraulic, MIL-H-83282

FAILURE MODE:

1 - Spring Fatigue
2 - No Apparent
3 - Accumulated Debris

Another example of reliability tests performed during development of the handbook is the testing of gearbox assemblies at the Naval Air Warfare Center in Patuxent River, Maryland (Reference 70). A spiral-bevel right angle reducer type gearbox with 3/8 inch steel shaft was selected for the test. Two models having different speed ratios were chosen, one gearbox rated at 12 in-lbs torque at 3600 rpm and the other gearbox rated at 9.5 in-lbs torque. Prior to testing the gearboxes, failure rate calculations were made using the reliability equations from this handbook. Test results were compared with failure rate calculations and conclusions made concerning the ability of the equations to be used in calculating failure rates.

Reliability tests were also performed on stock hydraulic actuators using a special-purpose actuator wear test apparatus (Reference 72). The actuators used in this validation project had a 2.50 inch bore, a 5.0 inch stroke, and a nominal operating pressure of 3000 psig. Various loads and lubricants were used to correlate test results with Handbook prediction procedures and equations. The effect of contamination of the oil was correlated by adding 10 micron abrasive particles to the lubricant in the actuators.

Additional reliability tests were performed during development of the handbook on air compressors for 4000 hours under six different environmental conditions to correlate the effect of the environment on mechanical reliability (Reference 73). The air compressors procured for the test were small reciprocating compressors with a maximum pressure of 35 psi and a ft^3 rating of 0.35. The units were subjected to temperature extremes, blowing dust, and AC line voltage variations while operating at maximum output pressure. The data collected were used to verify the reliability equations for reciprocating compressors.

In another reliability test, a special environmentally controlled test chamber was constructed at the Naval Air Warfare Center in Patuxent River, Maryland to test gear pumps and centrifugal pumps (References 74 and 75). A series of bronze rotary gear pumps were operated for 8000 hours to collect data on operation under controlled hydraulic conditions. Tests were conducted under high temperature water, low temperature water, and water containing silicon dioxide abrasives. Data were collected on flow rates, and seal leakage while pump speed, output pressure, and fluid temperature were held constant. Similar tests were conducted on a series of centrifugal pumps.

To further evaluate wear mechanisms and their effect on mechanical reliability, fifteen impact wrenches were operated to failure with a drum brake providing frictional torque and inertial torque loading (Reference 76). The impact wrenches selected for testing were general purpose, 1/2 inch drive, pneumatic impact wrenches commonly found in Naval repair shops. This wrench is rated for 200 lb-ft of torque and uses 4 cfm at 90 psi of air. Results of these reliability tests were used to evaluate the utility of the related failure equations in the handbook.

Validation of the various reliability equations for brakes and clutches was accomplished with tests conducted at Louisiana Tech University by evaluating the wear process for the various elements used in disk and drum brakes and multiple-disk clutches (Reference 77). Two types of experimental tests were conducted in connection with development of the model: (a) abrasive wear tests and (b) measurements of the coefficient of friction. Brakes and clutches were tested while monitoring the rate of wear for various materials including asbestos-type composite, sintered resin composite, sintered bronze composite, carbon-carbon composite, cast iron, C1040 carbon steel, 17-4 PH stainless steel, and 9310 alloy steel. The number of passes required to initiate measurable wear for the various types of brakes and clutches were correlated to the models contained in this handbook.

Robins AFB, one of the sponsors of the project to develop this handbook, provided an MC-2A air compressor unit for validation testing of the handbook procedures. The

MC-2A is a diesel engine-driven, rotary vane compressor mounted in a housed mobile trailer. It is designed for general flight line activities such as operating air tools requiring air from 5 psig to 250 psig. Two objectives were established for the validation effort: (a) determine the utility of the handbook to effect significant improvements in the reliability of new mechanical designs, and (b) determine the reliability of the MC-2A in its intended operating environment and introduce any needed design modifications for reliability improvement (References 78 and 79).

An additional reliability test was performed at the Naval Air Warfare Center in Patuxent River, Maryland to verify the application of the handbook in identifying existing and impending faults in mechanical equipment. A commercial actuator assembly was purchased and its design life estimated using the equations in this handbook. The actuator was then placed on test under stress conditions and an inspection made at the minimum calculated design life taking into consideration the sensitive parameters in the reliability equations. Upon inspecting the actuator at this point in time a revised remaining life estimate of the actuator was made and the test continued until failure. Test results were then compared with estimated values. The purpose of this test was to demonstrate the use of the handbook equations to revise failure estimates based on actual operating conditions when they may be different than originally anticipated and to continually obtain a more accurate estimate of time before the next maintenance action will be required (Reference 80).

An application of the methodology included in this Handbook to a diagnostic/prognostic system was demonstrated at the Naval Surface Warfare Center in West Bethesda, Maryland. Sensors were placed on various components of a water purification system being designed and tested at the laboratory. Equations as contained in this Handbook were then loaded into a laptop computer so that a real time determination of the remaining life of critical components could be made. Results of the experiment demonstrated that the application of prognostics to cognitive-based maintenance systems achieves the goal of performing maintenance actions only when there is objective evidence that the equipment requires attention. The result is a minimally manned, low maintenance and self-sufficient platform.

1.6 SUMMARY

The procedures presented in this handbook should not be considered as the only methods for a design analysis. An engineer needs many evaluation tools in his toolbox and new methods of performing dynamic modeling, finite element analysis and other stress/strength evaluation methods must be used in combination to arrive at the best possible reliability prediction for mechanical equipment.

The examples included in this introduction are intended to illustrate the point that there are no simplistic approaches to predicting the reliability of mechanical equipment. Accurate predictions of reliability are best achieved by considering the effects of the operating environment of the system at the part level. The failure rates derived from equations as tailored to the individual application then permits an estimation of design life for any mechanical system. It is important to realize that the failure rates estimated using the equations in this handbook are time dependent and that failure rates for mechanical components must be combined for the time period in question to achieve a total equipment failure rate. Section 1.3 and specifically Figure 1.2 demonstrate this requirement.

It will be noted upon review of the equations that some of the parameters are very sensitive in terms of life expectancy. The equations and prediction procedures were developed using all known data resources. Additional research is needed to obtain needed information on some of these “cause and effect” relationships for use in continual improvement to the Handbook. In the meantime, the value of the Handbook lies in understanding these “cause and effect” relationships so that when a discrepancy does occur between predicted and actual failure rate, the cause is immediately recognized. It is hoped that users of the Handbook and the MechRel software program will communicate observed discrepancies in the Handbook and suggestions for improvement to the Naval Surface Warfare Center. Suggestions, comments and questions should be directed to:

Tyrone L. Jones
MechRel Program Manager
Code 2120
Naval Surface Warfare Center
9500 MacArthur Blvd
West Bethesda, MD 20817-5700

Telephone: 301-227-4383

E-mail: Tyrone.L. Jones@.navy.mil

1.7 REFERENCES

28. MIL-HDBK-217, "Reliability Prediction of Electronic Equipment".

70. "Validation of Gearbox Reliability Models from Test Data", Report No. 87-D-0075, October, 1987, Eagle Technology, Inc.

71. Dennis N. Pratt, "Investigation of Spline Coupling Wear", Report No. SY-51R-87, December 1987, Naval Air Warfare Center, Patuxent River, Maryland

72. Randall F. Barron, "Engineering Model for Mechanical Wear", Report No. CMLD-CR-09-88, June 1988, Louisiana Tech University

73. Dennis Pratt, "Results of Air Compressor Reliability Investigation", Report No. TM 88-38 SY, January 1989, Naval Air Warfare Center, Patuxent River, Maryland

74. D. Pratt, "Results of Gear Pump Reliability Investigation", Report No. TM 89-24 SY, February 1990, Naval Air Warfare Center, Patuxent River, Maryland

75. D. Pratt, "Results of Centrifugal Pump Reliability Investigation", Report No. TM 89-69 SY, February 1990, Naval Air Warfare Center, Patuxent River, Maryland

76. D. Pratt, "Results of Pneumatic Impact Wrench Reliability Investigation", Report No. TM 90-88 SY, December 1990, Naval Air Warfare Center, Patuxent River, Maryland

77. Randall F. Barron and Herbert G. Tull, III, "Failure Rate Model for Aircraft Brakes and Clutches", Report No. DTRC-CMLD-CR-01-90, August 1990, Louisiana Tech University

78. CDNSWC, "Interim Reliability Report on the MC-2A Compressor Unit", January, 1992

79. D. Pratt, "Results of Air Force MC-2A Air Compressor Unit Reliability Investigation", Report No. TM 92-89 SY, March 1993, Naval Air Warfare Center, Patuxent River, Maryland

80. D. Pratt, “Results of Dayton 5A701 Linear Actuator Reliability Investigation”, Report No. TM 93-89 SY, Naval Air Warfare Center, Patuxent River, Maryland (1994)

81. TD 84-3 R&M Data for Industrial Plants, A.P. Harris and Associates, 1984

CHAPTER **2**

DEFINITIONS

This chapter provides a definition of some basic engineering terms to help establish a reference for the engineering analysis of mechanical equipment.

Abrasive Wear - The removal of material from a surface by the sliding or rolling of hard particles across the surface under pressure.

Adhesive Wear - The removal of material from a surface by the welding together and subsequent shearing of two surface areas that slide across each other under pressure.

Anodizing - The forming of a conversion coating on a metal surface (usually aluminum) by anodic oxidation.

Armature - The portion of the magnetic structure of a DC or universal motor which rotates

Axial Thrust - The force or loads that are applied to a shaft in a direction parallel to the axis of the shaft (such as from a fan or pump).

B_{10} Life - The number of hours at a given load that 90 percent of a set of apparently identical bearings will complete or exceed.

Base Failure Rate - A failure rate for a component or part in failures per million hours or failures per million operations depending on the application and derived from a data base where the exact design, operational, and environmental parameters are known. Multiplying factors are then used to adjust the base failure rate to the new operating environment.

Bending Moment - The algebraic sum of the moments of the external forces to the left or right of any section on a member subjected to bending by transverse forces.

Boring - A machining method using a single point tool on internal surfaces of revolution.

Brake Lining - A frictional material used for stopping or retarding the relative movement of two surfaces.

Cathodic protection – A technique used to control the corrosion of a metal surface by making it the cathode of an electrochemical cell.

Cavitation - The formation and instantaneous collapse on innumerable tiny voids within a liquid subjected to rapid and intense pressure changes.

Cavitation Damage - Erosion of a solid surface through the formation and collapse of cavities in an adjacent liquid.

Center Distance - The distance between centers of two gears.

Coefficient of Friction - This relationship is the ratio between two measured forces. The denominator is the normal force pressing two surfaces together. The numerator is the frictional force resisting the motion of one surface over the other.

Compressive Strength - The maximum compressive stress that a material is capable of developing based on the original area of cross section.

Contamination - Foreign matter or particles in a fluid system that are transported during its operation and which may be detrimental to system performance or even cause failure of a component.

Corrosion - The slow deterioration of materials by chemical agents and/or electromechanical reactions.

Corrosion Fatigue – Cracking produced by the combined action of repeated or fluctuating stress and a corrosive environment.

Creep - Continuous increase in deformation under constant or decreasing stress.

Dependent Failure - Failure caused by failure of an associated item or by a common agent.

Diaphragm – A member made of rubber or similar material used to contain hydraulic fluid within the forming cavity and to transmit pressure to the part being formed.

Dirt lock - Complete impedance of movement caused by stray contaminant particles wedged between moving parts.

Durometer - A device used to measure the hardness of rubber compounds.

Duty Cycle - The ratio of “on-time” to “on time + off time”, usually expressed as a percentage.

Elastic Limit - The greatest stress at which a material is capable of withstanding without any permanent deformation after removal of the load

Endurance Limit - The stress level value when plotted as a function of the number of stress cycles at which point a constant stress value is reached. This is the maximum stress below which it can be assumed the material can endure an indefinite number of stress cycles.

External leakage - Leakage resulting in loss of fluid to the external environment.

Failure Mode - The indicator or symptom by which a failure is evidenced.

Failure Rate - The probable number of times that a given component will fail during a given period of operation under specified operating conditions. Failure rate may be in terms of time, cycles, revolutions, miles, etc.

Fatigue - The cracking, fracture or breakage of mechanical material due to the application of repeated, fluctuating or reversed mechanical stress less than the tensile strength of the material.

Fatigue Life - The number of stress cycles that can be sustained prior to failure under stated conditions.

Fatigue Limit - The maximum stress that presumably leads to fatigue fracture in a specified number of stress cycles.

Fatigue Strength - The maximum stress that can be sustained for a specified number of cycles without failure.

Fretting (or **Fretting Corrosion**) – Surface pitting caused by contacting asperities on mating surfaces. Corrosion damage occurs at the asperities of contact surfaces. It is caused by the combination of corrosion and the abrasive effects of debris in equipment with moving parts.

Friction Material - A product manufactured to resist sliding contact between itself and another surface in a controlled manner.

Gear - The larger of two meshed gears. If both gears are the same size they are both referred to as “gears”. See pinion

Hardness - A measure of material resistance to permanent or plastic deformation equal to a given load divided by the resulting area of indentation.

Hooke’s Law - Stress is proportional to strain. The law holds only up to the proportional limit.

Hydrogen Embrittlement - The process by which high-strength steel becomes brittle and fractures following exposure to hydrogen. Hydrogen embrittlement is often the result of unintentional introduction of hydrogen into susceptible metals during forming or finishing operations.

Hysteresis – When a ferromagnetic material is magnetized in one direction, it will not relax back to zero magnetization when the imposed magnetizing field is removed. It must be driven back to zero by a field in the opposite direction. The lack of retraceability of the magnetization curve is the property called hysteresis and it is related to the existence of magnetic domains in the material.

Independent Failure - A failure of a device which is not caused by or related to failure of another device.

Inductance - The characteristic of an electric circuit by which varying current in the circuit produces a varying magnetic field which causes voltages in the same circuit or in a nearby circuit.

Internal Leakage - Leakage resulting in loss of fluid in the direction of fluid flow past the valving unit.

Joint Efficiency - The strength of a welded joint expressed as a percentage of the strength of the unwelded base metal.

Leakage - The flow of fluid through the interconnecting voids formed when the surfaces of two materials are brought into contact.

Line of Action - The line along which the point of contact between gear teeth travels, between the first point of contact and the last.

Lubricant - A substance used to reduce friction between two surfaces in contact.

Mean Cycles Between Failure - The total number of functioning cycles of a population of parts divided by the total number of failures within the population during the same period of time. This definition is appropriate for the number of hours as well as for cycles.

Mean Cycles to Failure - The total number of functioning cycles divided by the total number of failures during the period of time. This definition is appropriate for the number of hours as well as for cycles.

Mean Stress - The algebraic mean of the maximum and minimum stress in one cycle.

Mil - One thousandth of an inch (0.001 in.)

Mild Steel - Carbon steel with a maximum of about 0.25% carbon.

Modulus of Elasticity – A measure of the rigidity of metal. The slope of the initial linear portion of the stress-strain diagram; the larger the value, the larger the stress required to produce a given strain. Also known as Young's Modulus. See Meyer Hardness.

Modulus of Rigidity - The rate of change of unit shear stress with respect to unit shear strain for the condition of pure shear within the proportional limit. Also called Shear Modulus of Elasticity.

Meyer Hardness – A measurement of the mean pressure between the surface of the indentor during hardness testing and the indention. Equal to the load divided by the projected area of the indentation.

Pinion - The smaller of two meshing gears.

Poisson's Ratio - Ratio of lateral strain to axial strain of a material when subjected to uniaxial loading.

Pressure Angle - The angle between the Line of Action in a gear tooth and a line perpendicular to the Line of Centers.

Proportional Limit - The maximum stress at which strain remains directly proportional to stress.

Random Failures - Failures that occur before wear out, are not predictable as to the exact time of similar and are not associated with any pattern of similar failures. However, the number of random failures for a given population over a period of time at a constant failure rate can be predicted.

Reliability - A quantitative measure of the ability of a product to fulfill its intended function for a specified period of time under stated operating conditions.

Shot Peening - A cold working process used to produce a compressive residual stress layer and modify mechanical properties of metals. It entails impacting a surface with shot (round metallic, glass, or ceramic particles) with force sufficient to create plastic deformation.

Silting - An accumulation and settling of particles during component inactivity.

Smearing - Surface damage resulting from unlubricated sliding contact within a bearing.

S-N diagram - A graph showing the relationship of stress (S) and the number of cycles (N) before fracture in fatigue testing.

Spalling - The cracking and flaking of particles out of a surface.

Stiction - A change in performance characteristics or complete impedance of poppet or spool movement caused by wedging of minute particles between a poppet stem and housing or between spool and sleeve.

Strain - A measure of the relative change in size or shape of a body, usually a reference to the linear strain in the direction of applied stress.

Stress - Used to indicate any agency that tends to induce “failure”. It is a measure of intensity of force acting on a definite plane passing through a given point, measured in force per unit area.

Stress-corrosion Cracking - Failure by cracking under combined action of corrosion and applied or residual stress

Stress Raiser - Change in contour or discontinuity in structure that causes a local increase in stress

Surface Finish - A measure of the roughness of a surface as a result of final treatment.

Temperature Rise - Some of the electrical energy losses inherent in motors and other components are converted to heat causing some of the component parts to heat up while running or activated. The heated parts are at a higher temperature than the surrounding air causing a rise above ambient temperature. Friction has the same effect on mechanical component parts such as actuators and shafts.

Tensile Strength - Value of nominal stress obtained when the maximum (or ultimate) load that the specimen supports is divided by the cross-sectional area of the specimen. See Ultimate Strength

Thermal Fatigue - Fracture resulting from the presence of thermal gradients producing cyclic stresses in a structure

Thrust Bearing - Special bearings used to handle higher than normal axial forces exerted on the shaft of the motor or gearmotor as is the case with some fan or pump blade mountings

Torque - Turning force delivered by a motor or gearmotor shaft usually expressed in ft-lbs derived by computing H.P. x 5250/RPM = full load torque

Triaxial Stress - A state of stress in which none of the three principal axis stresses is zero.

Ultimate Strength - The maximum stress (tensile, compressive or shear) the material will withstand. See Tensile Strength.

Viscosity - A measure of internal resistance of a fluid which tends to prevent it from flowing.

Wear-out Failure - A failure which occurs as a result of mechanical, chemical or electrical degradation.

Yield Strength - The stress that will produce a small amount of permanent deformation in a material, generally a strain equal to 0.1 or 0.2 percent of the length of the specimen.

Young's Modulus - A measure of the rigidity of metal. The slope of the initial linear portion of the stress-strain diagram; the larger the value, the larger the stress required to produce a given strain. Also known as Modulus of Elasticity.

This Page Intentionally Left Blank

CHAPTER 3

SEALS AND GASKETS

3.0 TABLE OF CONTENTS

3.1 INTRODUCTION

A seal is a device placed between two surfaces to prevent the flow of gas or liquid from one region to another. Seals are used for both static and dynamic applications. Static seals such as gaskets, bolt seals, back-up rings and sealants are used to prevent leakage through a mechanical joint when there is no relative motion of mating surfaces. Truly static seals are designed to provide a complete barrier to a potential leakage path. These seals are “zero leakage” seals (down to 10^{-11} scc/sec.helium). In a truly static seal, the mating gland parts are not subject to relative movement except for thermal expansion and movement from the application of fluid pressure. Some static seals are designed to accommodate limited movement of the surfaces being sealed due to changes in pressure, vibration or thermal cycling such as an expansion joint. These seals are sometimes referred to as semi-static seals.

A dynamic seal is a mechanical device used to control leakage of fluid from one region to another when there is rotating, oscillating or reciprocating motion between the sealing interfaces. An O-ring can be used in both static and dynamic applications. However, the employment of O-rings as primary dynamic seals is normally limited to short strokes and moderate pressures. An example of static and dynamic seal applications is shown in Figure 3.1.

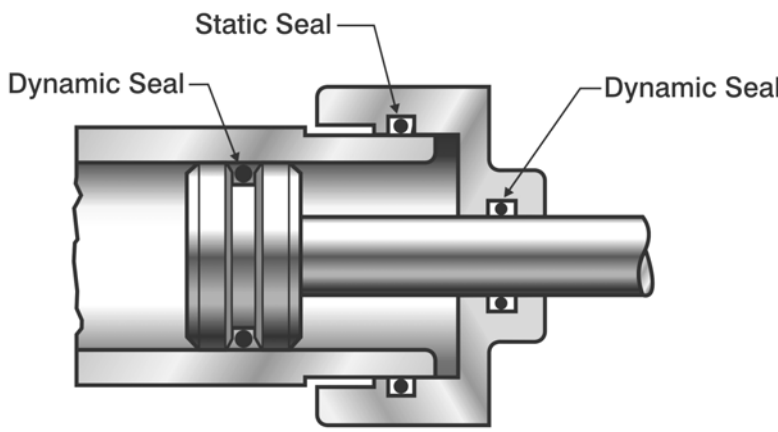


Figure 3.1 Static and Dynamic Seals

Other seal designs include reciprocating seals, oscillating seals and rotary seals where movement relative to other mechanical parts will occur. Reciprocating seals involve relative reciprocating motion along the shaft axis between the inner and outer elements. In reciprocating seal applications, the O-ring slides or rocks back and forth within its gland with the reciprocating motion. Reciprocating seals are most often seen in cylinders and linear actuators. In oscillating seal applications, the inner or outer member of the gland moves in an arc around the axis of the shaft - first in one direction and then in the opposite direction, generally intermittently with no more than a few turns in each direction. The most common application for oscillating O-ring seals is in faucet valves. Rotary seals involve motion between a shaft and a housing. Typical rotary seals include motor shafts and wheels on a fixed axle. These specific seal designs are included in the appropriate section of this chapter as either static of dynamic seals.

A mechanical seal is designed to prevent leakage between a rotating shaft and its housing under conditions of higher fluid pressure, shaft speed and temperature normally associated with dynamic seals. For purposes of this Chapter, a mechanical seal will be assumed to be rotating in contact with the fluid above 800 rpm or a sliding contact exceeding 600 feet/minute. A contact sealing face composed of a soft, sacrificial face material forms a seal against a hard material. A common design has a carbon rotating element. Failure of a mechanical seal is defined as an inoperative seal before wear-out of the sacrificial surface.

The reliability of a seal design is determined by the ability of the seal to restrict the flow of fluid from one region to another for its intended life in a prescribed operating environment. The evaluation of a seal design for reliability must include a definition of the design characteristics and the operating environment in order to estimate its design life. Section 3.2 discusses the reliability of gaskets and other static seals. Procedures for evaluating the reliability of dynamic seals are contained in Section 3.3. Procedures for evaluating mechanical seals for reliability are included in Section 3.4.

3.2 GASKETS AND STATIC SEALS

A gasket is used to develop and maintain a barrier between mating surfaces of mechanical assemblies when the surfaces do not move relative to each other. The barrier is designed to retain internal pressures, prevent liquids and gases from escaping the assembly, and prevent contaminants from entering the assembly. Gaskets can be metallic or nonmetallic. Flange pressure compresses the gasket material and causes the material to conform to surface irregularities in the flange and is developed by tightening bolts that hold the assembly together.

Gasket reliability is affected by the type of liquid or gas to be sealed, internal pressure, temperature, external contaminants, types of surfaces to be joined, surface roughness, and flange pressure developed at the joint. To achieve the barrier to a potential leakage path the seal must be sufficiently resilient to conform to cavity irregularities and imperfections, while remaining rigid enough to provide the required contact force needed to ensure a tight seal. This contact force is a function of the seal cross section, as well as the compression of the seal between the mating cavity faces. The load on the gasket must be distributed evenly over the whole area of the gasket rather than have a few points of high load with reduced stress at midpoints between the fasteners. Therefore, a larger number of small bolts is better than a few larger bolts. Use of a torque wrench during installation is always a necessity.

An O-ring is a mechanical gasket in the shape of a torus. It is a loop of elastomer with a disc-shaped cross-section, designed to be seated in a groove and compressed during assembly between two or more parts, creating a seal at the interface. The combination of the O-ring and the gland that supports the O-ring constitute the classic O-ring seal assembly. A typical O-ring configuration is shown in Figure 2.

While static seals in most cases are designed for “zero-leakage” semi-static seals for applications where there is limited movement are not designed or intended to be “zero-leakage” seals. Their contact or compression force is typically an order of magnitude lower than a static seal.

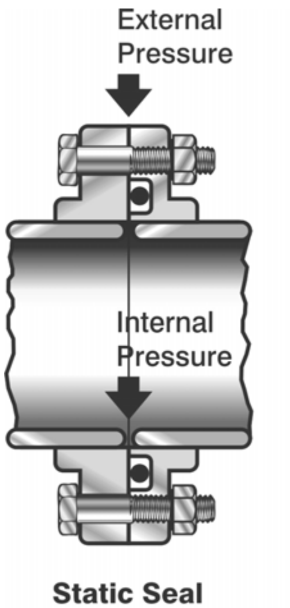


Figure 3.2 Typical O-ring Configuration

3.2.1 Static Seal Failure Modes

The primary failure mode of a gasket or static seal is leakage. In the case of an O-ring, the O-ring flows up to, but not into, the clearance gap between components under normal pressure. If the pressure is increased, both the sealing force and contact area increase. At the seal pressure limit depending on the seal material and hardness, part of the O-ring starts to extrude into the clearance gap. At this point the seal can shear creating seal leakage.

Static seals may also be subjected to high and low temperatures, chemical attack, vibration, abrasion, and movement. The integrity of a seal depends upon the compatibility of the fluid and sealing components, conditions of the sealing environment, and the applied load during application.

A failure mode especially applicable to low pressure applications is compression set. Compression set refers to the permanent deflection remaining in the seal after complete release of a squeezing load while exposed to a particular temperature level. Compression set reflects the partial loss of elastic memory due to the time effect. Operating over extreme temperatures can result in compression-type seals such as gaskets and O-rings to leak fluid at low pressures because they have deformed permanently or taken a set after used for a period of time.

All seals have an upper temperature limit determined by the type and grade of the material being used. Thermal expansion can cause misalignment and cause uneven surfaces that are designed to remain flat. Static seals also undergo plastic deformation during installation and maintenance procedures need to be reviewed for those situations where permanent set can take place before permitting the seal to be replaced. Table 3-1 contains a list of typical failure mechanisms and causes of seal leakage. Other failure mechanisms and causes should be identified for the specific product to assure that all considerations of reliability are included in any design evaluation.

3.2.2 Failure Rate Model Considerations

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

- Material characteristics
- Amount of seal compression
- Surface irregularities
- Seal size

- Fluid pressure
- Extent of pressure pulses
- Temperature
- Fluid viscosity
- Contamination level
- Fluid/material compatibility
- Leakage requirements
- Assembly/quality control procedures

Table 3-1. Typical Failure Mechanisms and Causes for Static Seals and Gaskets

FAILURE MODE	FAILURE MECHANISMS	FAILURE CAUSES
Leakage	- Wear	- Contaminants - Misalignment - Vibration - Poor surface finish
	- Elastic Deformation - Gasket/seal distortion	- Extreme temperature - Misalignment - Seal eccentricity - Extreme loading / extrusion - Compression set/overtorqued bolts
	- Surface Damage - Embrittlement	- Inadequate lubrication - Contaminants - Fluid/seal degradation - Thermal degradation - Idle periods between component use - Exposure to atmosphere, ozone - Excessive temperature
	- Creep	- Fluid pressure surges - Material degradation - Thermal expansion & contraction
	- Compression Set	- Excessive squeeze to achieve seal - Incomplete vulcanization - Hardening/high temperature
	- Installation Damage	- Insufficient lead-in chamfer - Sharp corners on mating metal parts - Inadequate protection of spares
	- Gas expansion rupture	- Absorption of gas or liquefied gas under high pressure

The failure rate of a static seal is a function of actual leakage and the allowable leakage under conditions of usage, failure occurring when the rate of leakage reaches a predetermined threshold. This rate, derived empirically, can be expressed as follows:

$$\lambda_{SE} = \lambda_{SE,B} \left(\frac{Q_a}{Q_f} \right) \tag{3-1}$$

Where: λ_{SE} = Failure rate of gasket or seal considering operating environment, failures per million hours

$\lambda_{SE,B}$ = Base failure rate of seal or gasket due to random cuts, installation errors, etc. based on field experience data, failures per million hours

Q_a = Actual leakage rate, in^3/min

Q_f = Allowable leakage rate under conditions of usage, in^3/min

Allowable leakage is dependent on the application. External leakage of a component containing water is obviously not as critical as one containing fuel. Allowable leakage, Q_f is determined from design drawings, specifications or knowledge of component applications. The actual leakage rate Q_a for a seal is determined from the standard equation for laminar flow around two curved surfaces (Reference 5):

$$Q_a = \left(\frac{\pi \left(P_1^2 - P_2^2 \right)}{25\, \nu_a P_2} \right) \left(\frac{r_o + r_i}{r_o - r_i} \right) H^3 \tag{3-2}$$

Where P_1 = System or upstream pressure, lbs/in^2

P_2 = Standard atmospheric pressure or downstream pressure, lbs/in^2

ν_a = Absolute fluid viscosity, $lb\text{-}min/in^2$

r_i = Inside radius of circular interface, in

r_o = Outside radius of circular interface, in

H = Conductance parameter, in [See Equation (3-4)]

For flat seals or gaskets the leakage can be determined from the following equation:

$$Q_a = \left(\frac{\pi \, L \left(P_1^{\,2} - P_2^{\,2} \right)}{12 \, \nu_a \, w \, P_2} \right) H^3 \tag{3-3}$$

Where: w = Width of non-circular flat seals, in

L = Contact length, in

The conductance parameter H is dependent upon contact stress of the two sealing surfaces, hardness of the softer material and surface finish of the harder material (Reference 5). First, the contact stress (load/area) is calculated and the ratio of contact stress to Meyer hardness of the softer interface material computed. The surface finish of the harder material is then determined. The conductance parameter is computed from the following empirically derived formula:

$$H = 0.23 \left(\frac{M}{C} \right)^{1.5} \bullet f^{\,2/3} \tag{3-4}$$

Where: M = Meyer hardness (or Young's modulus) for rubber and resilient materials, lbs/in^2

C = Contact stress, lbs/in^2 [See Equation (3-9)]

f = Surface finish, in

Seal wear is dependent on the finish of the surface against which the seal rubs when pressure is applied and released or pressure surges occur. The surface finish, f, will deteriorate as a function of time at a rate dependent upon several factors:

- Seal degradation
- Contaminant wear coefficient (in^3/particle)
- Number of contaminant particles per in^3
- Flow rate, in^3/min
- Ratio of time the seal is subjected to contaminants under pressure
- Temperature of operation, $^{\circ}F$

Note that surface finish and seal hardness are the two parameters in the seal reliability equation that will change as a function of time. Therefore, estimating the value of these parameters at different time intervals during the life of the product will provide an estimate of the total assembly as a function of time.

The contaminant wear coefficient is an inherent sensitivity factor for the seal or gasket based upon performance requirements. The quantity of contaminants includes those produced by wear and ingestion in components upstream of the seal and after the filter. Combining and simplifying terms provides the following equations for the failure rate of a seal.

For circular seals:

$$\lambda_{SE} = \lambda_{SE,B} \left[\frac{K_1 \left(P_1^2 - P_2^2 \right) H^3}{Q_f \, \nu_a \, P_2} \right] \bullet \left[\frac{r_o + r_i}{r_o - r_i} \right] \quad \textbf{(3-5)}$$

and, for flat seals and gaskets:

$$\lambda_{SE} = \lambda_{SE,B} \left[\frac{K_2 \left(P_1^2 - P_2^2 \right) L \, H^3}{Q_f \, \nu_a \, w \, P_2} \right] \quad \textbf{(3-6)}$$

Where K_1 and K_2 are empirically derived constants

3.2.3 Failure Rate Model for Gaskets and Static Seals

By normalizing the equation to those values for which historical failure rate data from the Navy Maintenance and Material Management (3-M) system are available, the following model can be derived:

$$\lambda_{SE} = \lambda_{SE,B} \bullet C_P \bullet C_Q \bullet C_{DL} \bullet C_H \bullet C_F \bullet C_\nu \bullet C_T \bullet C_N \quad \textbf{(3-7)}$$

Where: λ_{SE} = Failure rate of a seal in failures/million hours

$\lambda_{SE,B}$ = Base failure rate of seal, 2.4 failures/million hours

C_P = Multiplying factor which considers the effect of fluid pressure on the base failure rate (Figure 3.10)

C_Q = Multiplying factor which considers the effect of allowable leakage on the base failure rate (See Figure 3.11)

C_{DL} = Multiplying factor which considers the effect of seal size on the base failure rate (See Figure 3.12 for seals or Figure 3.13 for gaskets)

C_H = Multiplying factor which considers the effect of contact stress and seal hardness on the base failure rate (See Figure 3.14)

C_F = Multiplying factor which considers the effect of seat smoothness on the base failure rate (See Figure 3.15)

C_ν = Multiplying factor which considers the effect of fluid viscosity on the base failure rate (See Table 3-3)

C_T = Multiplying factor which considers the effect of temperature on the base failure rate (See Figure 3.16)

C_N = Multiplying factor which considers the effect of contaminants on the base failure rate (See Table 3-4)

The parameters in the failure rate equation can be located on an engineering drawing, by knowledge of design standards or by actual measurement. Design parameters other than the above multiplying factors which have a minor effect on reliability are included in the base failure rate as determined from field performance data. The following paragraphs provide background information on those parameters included in the model.

3.2.3.1 Fluid Pressure

Figure 3.10 provides fluid pressure multiplying factors for use in the model. Fluid pressure on a seal will usually be the same as the system pressure. The fluid pressure at the sealing interface required to achieve good mating depends on the resiliency of the sealing materials and their surface finish. It is the resilience of the seal which insures that adequate sealing stress is maintained while the two surfaces move in relation to one another with thermal changes, vibration, shock and other changes in the operating environment. The reliability analysis should include verification that sufficient pressure will be applied to affect a good seal.

At least three checks should be made to assure the prevention of seal leakage:

(1) One surface should remain relatively soft and compliant so that it will readily conform to the irregularities of the harder surface

(2) Sufficient sealing load should be provided to elastically deform the softer of the two sealing surfaces

(3) Sufficient smoothness of both surfaces is maintained so that proper mating can be achieved

3.2.3.2 Allowable Leakage

Figure 3.11 provides an allowable leakage multiplying factor for use in Equation 3-7. Determination of the acceptable amount of leakage which can be tolerated at a seal interface can usually be obtained from component specifications. The allowable rate is a function of operational requirements and the rate may be different for an internal or external leakage path.

3.2.3.3 Seal Size

Figure 3.5 shows a typical installation for a seal and the measurements for r_i and r_o. For a gasket, the inside perimeter dimension *w* and the contact length *L* are used in the equation. Figures 3.12 and 3.13 show the effect of seal size on reliability. The inside diameter of the seal is used in Figure 3.12 as a close approximation of the seal size.

3.2.3.4 Conductance Parameter

Three factors comprise the conductance parameter:

(1) Hardness of the softer material

(2) Contact stress of the seal interface

(3) Surface finish of the harder material

(1) Hardness of the softer material: - In the case of rubber gaskets and O-rings, the hardness of rubber is measured either by durometer or international hardness methods. Both hardness test methods are based on the measurement of the penetration of a rigid ball into a rubber specimen. Throughout the seal/gasket industry, the Shore A durometer is the standard instrument used to measure the hardness of rubber compounds. The durometer has a calibrated spring which forces an indentor point into the test specimen against the resistance of the rubber. The scale of hardness is from 0 degrees for elastic modulus of a liquid to 100 degrees for an infinite elastic modulus of a material, such as glass. Readings in International Rubber Hardness Degree (IRHD) are comparable to those given by a Shore A durometer (Reference18) when testing standard specimens per the ASTM methods. The relationship between the rigid ball penetration and durometer reading is shown in Figure 3.3.

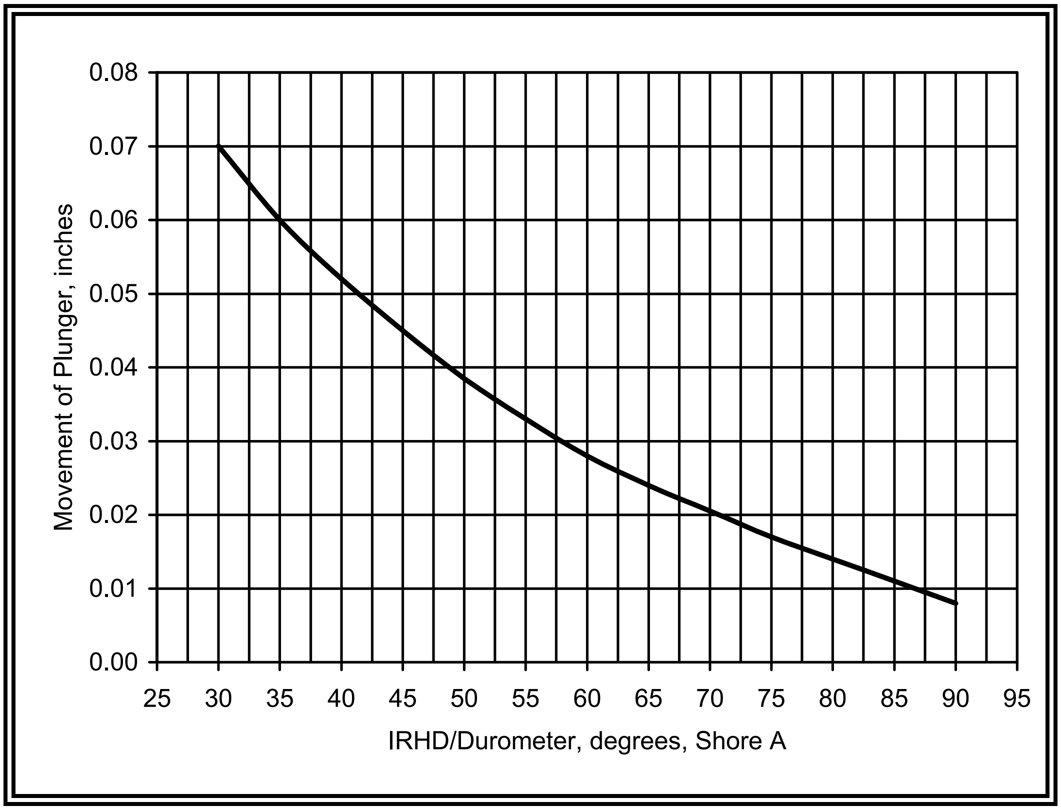


Figure 3.3 Relation Between International Rubber Hardness Degree (IRHD) and Rigid Ball Penetration

Well-vulcanized elastic isotropic materials, like rubber seals manufactured from natural rubbers and measured by IRHD methods, have a known relationship to Young's modulus. The relation between a rigid ball penetration and Young's modulus for a perfectly elastic isotropic material is (Reference 18):

$$\frac{F_1}{M_P} = 1.90\ R_p^{\ 2} \left(\frac{P_D}{R_P} \right)^{1.35} \qquad \textbf{(3-8)}$$

Where:

F_1 = Indenting force, lbf

M_p = Young's modulus, lbs/in^2

R_P = Radius of ball, in

P_D = Penetration, in

Standard IRHD testers have a ball radius of 0.047 inches with a total force on the ball of 1.243 lbf. Using these testing parameters, the relationship between seal hardness and Young's modulus is shown in Figure 3.4. Since Young's modulus is expressed in lbs/in^2 and calculated in the same manner as Meyer's hardness for rigid material; then, for rubber materials, Young's modulus and Meyer's hardness can be considered equivalent.

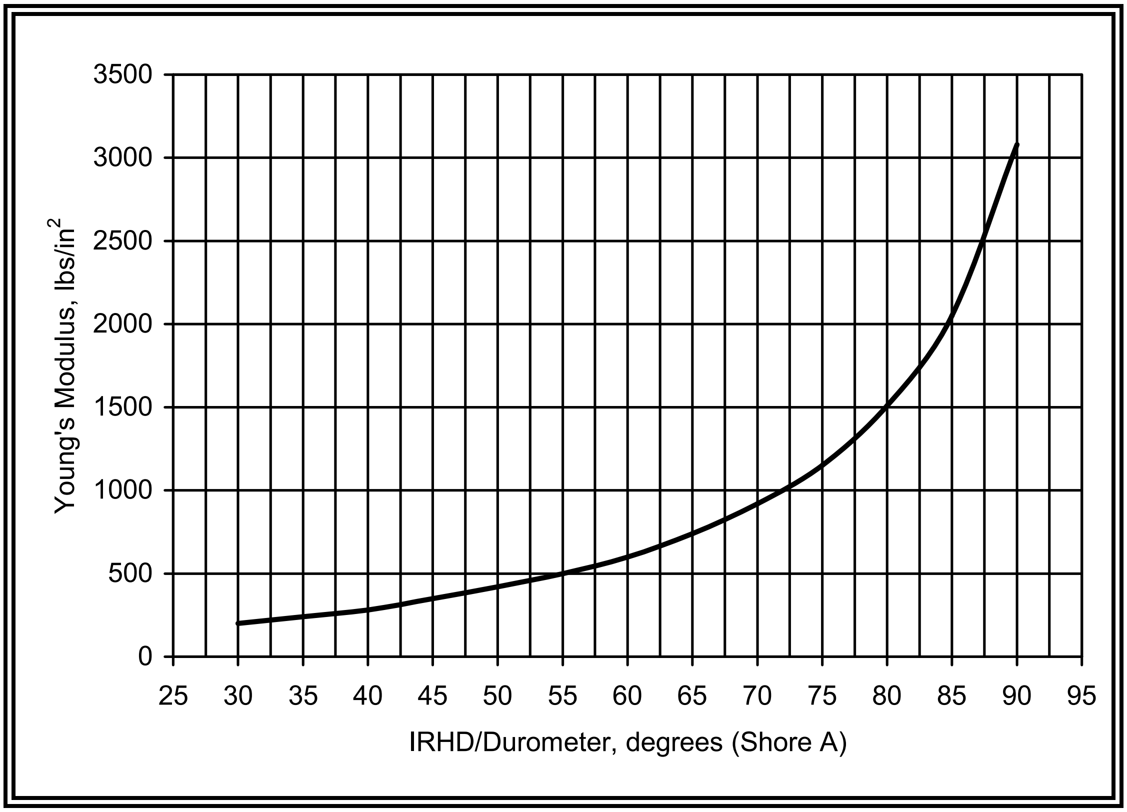


Figure 3.4 Seal Hardness and Young's Modulus

(2) <u>Surface finish of the harder material:</u> - For a gasket, the surface finish of the two surfaces containing the gasket govern the thickness and compressibility necessary for the gasket material for completing a physical barrier in the clearance gap between the flanges. Flatness of the surfaces being sealed is an important consideration. Reliability of the gasket is dependent on the type of material for the specific fluid and application. For seals contained in a gland, as pressure is applied to the fluid component, the O-ring will tend to roll in the gland and a gasket will tend to move between the retaining hardware. The surface finish on the gland will usually be about 32 microinches for elastomer seals, 16 microinches for plastic seals and 8 microinches for metals. In addition to average surface finish, the allowable number and magnitude of flaws in the gland must be considered in projecting leakage characteristics. Flaws such as surface cracks, ridges or scratches will have a detrimental effect on seal leakage. When projecting seal and gasket failure rates for different time periods of the

equipment life cycle, it is important to consider the exposure to contaminants and their effect on surface finish.

(3) Contact stress of the seal interface: - Seals deform to mate with rigid surfaces by elastic deformation. Since the deformation of the seal is almost entirely elastic, the initially applied seating load must be maintained. Thus, a load margin must be applied to allow for strain relaxation during the life of the seal yet not to the extent that permanent deformation takes place. An evaluation of cold flow characteristics is required for determining potential seal leakage of soft plastic materials. Although dependent on surface finish, mating of metal-to-metal surfaces generally requires a seating stress of two to three times the yield strength of the softer material. Figure 3.5 shows a typical installation of a gasket seal.

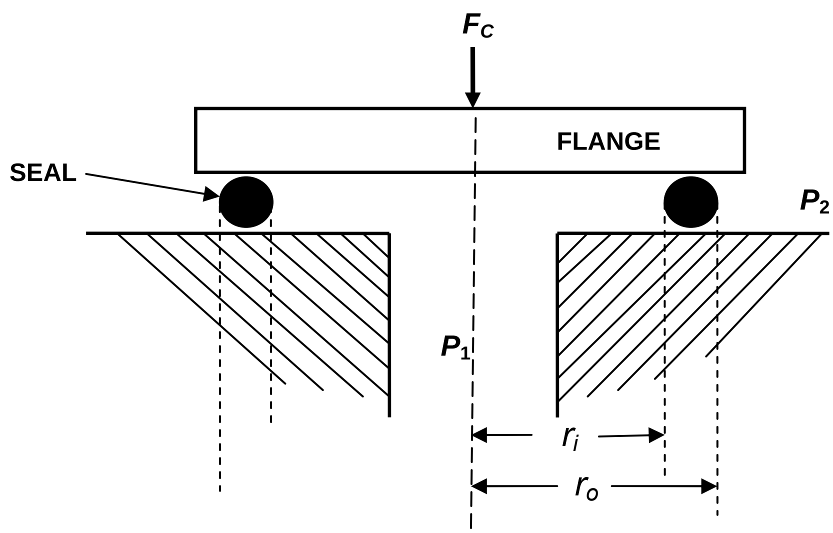


Figure 3.5 Typical Seal Installation

If the seal is pressure energized, the force F_C applied to the seal must be sufficient to balance the fluid pressure forces acting on the seal and thus, prevent separation of the interface surfaces. This requirement is determined by the maximum applied fluid pressure, geometry of the seal groove and pressure gradient at the interface due to leakage. Motion at the interface is prevented by the radial friction forces at the interface to counter the fluid pressure forces tending to radially deform the seal. Thus, the radial restraining force F_C will be greater than the radial pressure deformation forces.

The contact stress, C, in lbs/in^2 can be calculated by:

$$C = \frac{F_C}{A_{SC}} \quad \textbf{(3-9)}$$

Where: F_C = Force compressing seals, lb

A_{SC} = Area of seal contact, in^2

or:

$$C = \frac{F_C - P_1 \pi r_i^2 - (P_1 - P_2)\left(\frac{r_o + r_i}{2}\right)(r_o - r_i)}{\pi\left(r_o^2 - r_i^2\right)} \qquad \textbf{(3-10)}$$

Where: P_1 = System pressure, lbs/in^2

P_2 = Standard atmospheric pressure or downstream pressure, lbs/in^2

r_o = Outside seal radius, in

r_i = Inside seal radius, in

For most seals, the force compressing the seal F_C is normally two and one-half times the Young's modulus for the material. If too soft a material is used, the seal material will have insufficient strength to withstand the forces induced by the fluid and will rapidly fail by seal blowout. If the seal is too hard it will not sufficiently deform in the gland and immediate leakage will occur.

3.2.3.5 Fluid Viscosity

Viscosity of a fluid is much more dependent on temperature than it is on pressure. For example, when air pressure is increased from 1 atmosphere to 50, its viscosity is only increased by about 10%. In contrast, Figure 3.6 shows the dependence of viscosity on temperature for some common fluids. The graph shows how viscosity of liquids decreases with temperature while that of gases increases with temperature. Multiplying factors for the effect of fluid viscosity on the base failure rate of seals and gaskets are provided in Table 3-3. Viscosities for other fluids at the operating temperature can be found in referenced sources and the corresponding multiplying factor determined using the equation following Table 3-3. If the value located is in terms of kinematic viscosity, multiply the value by the specific gravity (density) at the desired temperature to determine the dynamic viscosity.

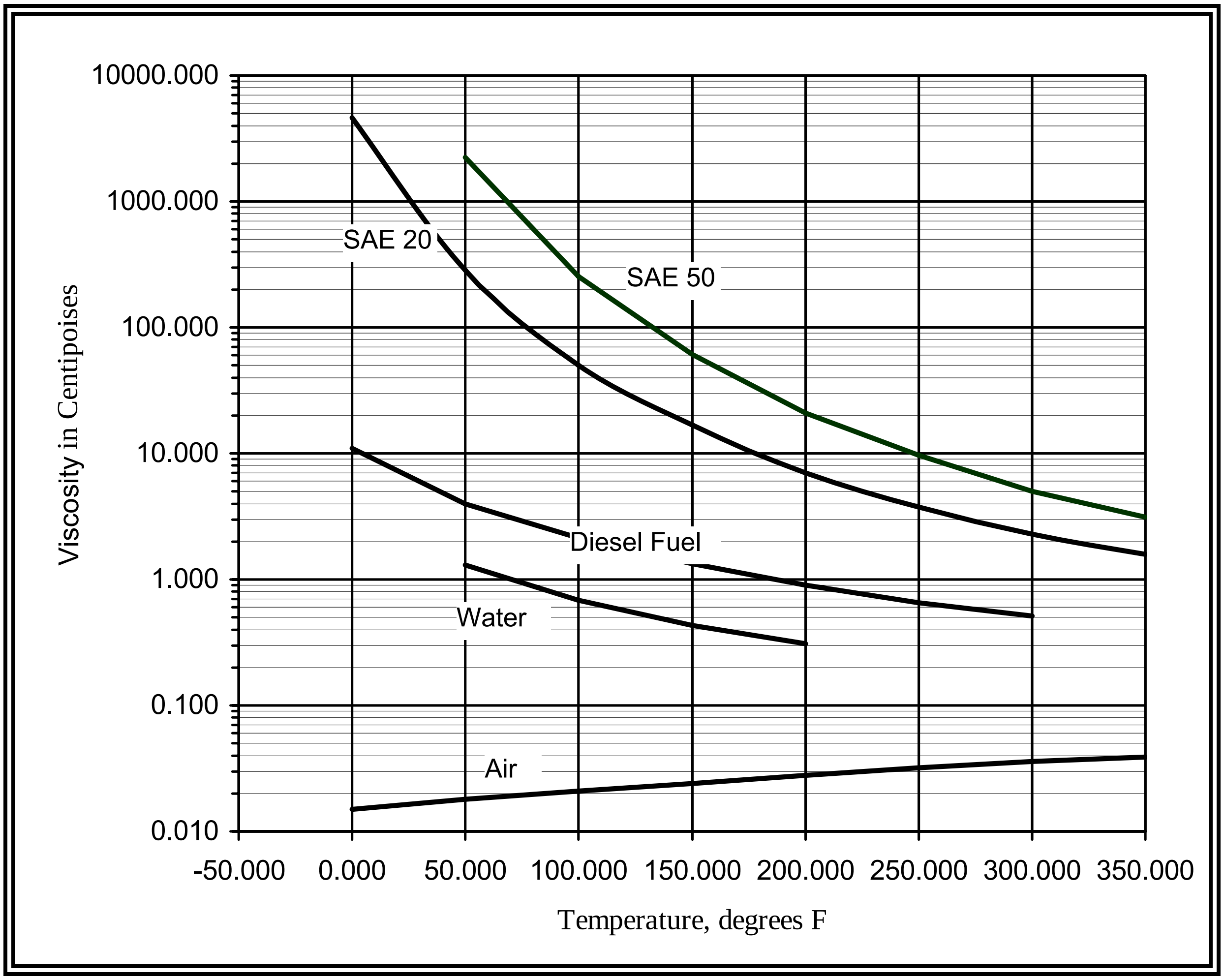


Figure 3.6 Dynamic Viscosities of Various Fluids

3.2.3.6 Fluid Operating Temperature

Operating temperature has a definite effect on the aging process of elastomer and rubber seals. Elevated temperatures, those temperatures above the published acceptable temperature limits, tend to continue the vulcanization or curing process of the materials, thereby significantly changing the original characteristics of the seal or gasket. It can cause increased hardening, brittleness, loss of resilience, cracking, and excessive wear. Since a change in these characteristics has a definite effect on the failure rate of the component, a reliability adjustment must be made.

Temperature will have a significant impact on the performance of a gasket since an increase in temperature will both degrade the physical strength of the material and deform it so that the bolt load and residual stress are modified. Manufacturers of rubber seals will specify the maximum temperature, T_R, for their products. Typical values of T_R are given in Table 3-5. An operating temperature multiplying factor can be derived as follows (Reference 22):

$$C_T = \frac{1}{2^t} \tag{3-11}$$

Where: $t = \dfrac{T_R - T_O}{18}$ for $(T_R - T_O) \leq 40\ ^oF$

T_R = Maximum rated temperature of material, ^oF

T_O = Operating temperature, ^oF

And: C_T = 0.21 for $(T_R - T_O) > 40\ ^oF$

3.2.3.7 Fluid Contaminants

The quantities of contaminants likely to be generated by upstream components are listed in Table 3-4. The number of contaminants depends upon the design, the enclosures surrounding the seal, its physical placement within the system, maintenance practices and quality control. The number of contaminants may have to be estimated from experience with similar system designs and operating conditions.

3.2.3.8 Other Design Analysis Considerations

Those failure rate considerations not specifically included in the model but rather included in the base failure rates are as follows:

- Proper selection of seal materials with appropriate coefficients of thermal expansion for the applicable fluid temperature and compatibility with fluid medium
- Space between the fasteners of a gasket must be small enough so that an even distribution of load is applied to the gasket with fluid pressure
- Potential corrosion from the gland, seal, fluid interface
- Possibility of the seal rolling in its groove when system surges are encountered
- If O-rings can not be installed or replaced easily they are subject to being cut by sharp gland edges
- Potential periods of dryness between applications of fluid

Other factors which need to be considered as a check list for reliability include:

- Chemical compatibility between fluid and seal material
- Thermal stability
- Appropriate thickness and width of the seal material
- Initial and final seating (clamping) force

3.3 DYNAMIC SEALS

In contrast to gaskets and other static seals, dynamic seals are used to control the leakage of fluid in those applications where there is motion between the mating surfaces being sealed. O-rings used in dynamic applications are subject to a sliding action against the gland. This motion introduces friction creating different designs and failure modes from those of static seals. Refer to Section 3.2 for a discussion of seals in general, the basic failure modes of seals and the parameters used in the equations to estimate the failure rate of a seal.

There are several types of dynamic seals including the contacting types such as lip seals and noncontacting types such as labyrinth seals. Assemblies with motion usually require lubrication of the O-ring to reduce wear rate. This is usually accomplished with the fluid being sealed.

Dynamic seals are further divided as follows:

- Reciprocating Seal: A seal where the rod or piston moves back and forth through or with the seal. Piston and rod seals shown in Figure 3.7 are examples of reciprocating seals.

- Rotary Seal: A seal where a shaft rotates with relation to the seal. Typical rotary seals include motor shafts and wheels on a fixed axle. O-rings are not generally used for conditions involving fluid velocities exceeding 800 rpm and/or surface speeds exceeding 600 feet/minute. See Figure 3.7.

- Oscillating Seal: A seal where a shaft turns and returns with relation to the seal. In this application the inner and outer member of the gland moves in an arc around the axis of the shaft first in one direction and then in the opposite direction with the movement usually intermittent. An example application is the faucet valve.

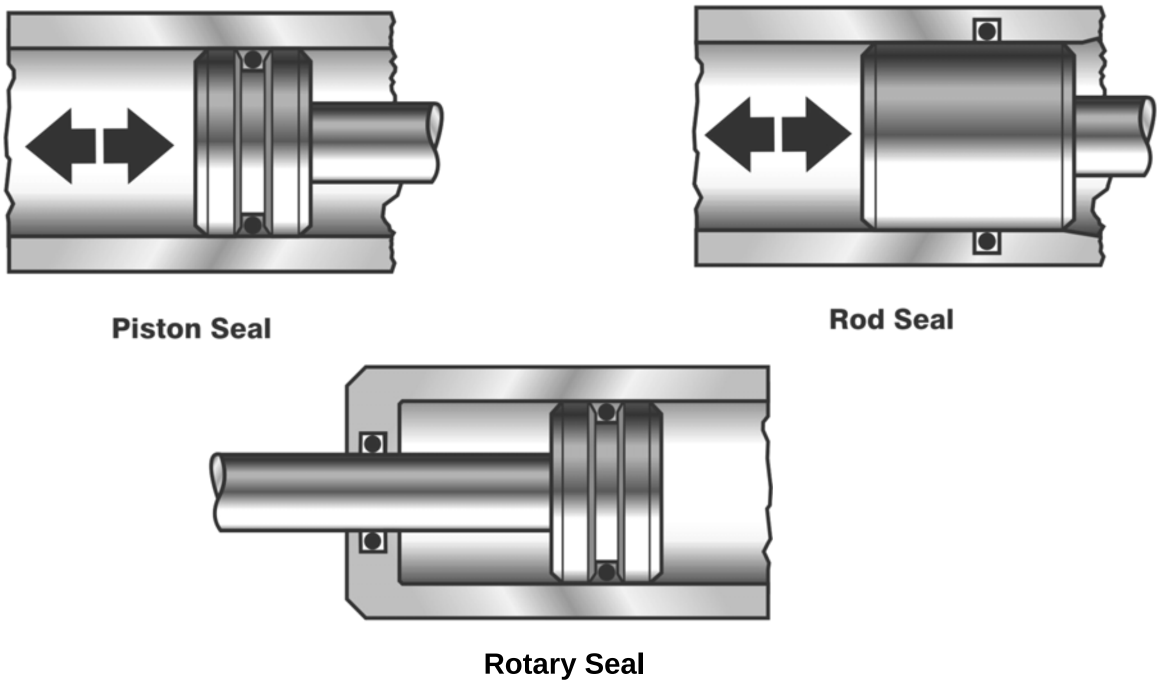


Figure 3.7 Typical Dynamic Seals

The following paragraphs discuss the specific failure modes and model parameters for dynamic seals. Mechanical seals are designed to prevent leakage between a rotating shaft and its housing. The mechanical seal is indicated as the dynamic seal faces in Figure 3.8. Section 3.4 contains specific information on mechanical seals.

3.3.1 Dynamic Seal Failure Modes

The dynamic seal may be used to seal many different liquids at various speeds, pressures, and temperatures. Dynamic seals are made of natural and synthetic rubbers, polymers and elastomers, metallic compounds, and specialty materials. Wear and sealing efficiency of fluid system seals are related to the characteristics of the surrounding operating fluid. Abrasive particles present in the fluid during operation will have a strong influence on the wear resistance of seals. Seals typically operate with sliding contact. Elastomer wear is analogous to metal degradation. However, elastomers are more sensitive to thermal deterioration than to mechanical wear. Hard particles can become embedded in soft elastomeric and metal surfaces leading to abrasion of the harder mating surfaces forming the seal, resulting in leakage.

The most common modes of seal failure are by fatigue-like surface embrittlement, abrasive removal of material, and corrosion. Wear and sealing efficiency of fluid system seals are related to the characteristics of the surrounding operating fluid. Abrasive particles present in the fluid during operation will have a strong influence on the wear resistance of seals, the wear rate of the seal increasing with the quantity of environmental contamination. A good understanding of the wear mechanism involved will help determine potential seal deterioration. For example, contaminants from the environment such as sand can enter the fluid system and become embedded in the elastomeric seals causing abrasive cutting and damage to shafts.

Compression set refers to the permanent deflection remaining in the seal after complete release of a squeezing load while exposed to a particular temperature level. Compression set reflects the partial loss of elastic memory due to the time effect. Operating over extreme temperatures can result in compression-type seals such as O-rings to leak fluid at low pressures because they have deformed permanently or taken a set after used for a period of time.

Another potential failure mode to be considered is fatigue failure caused by shaft run-out. A bent shaft can cause vibration throughout the equipment and eventual loss of seal resiliency. Typical failure mechanism and causes for dynamic seals are included in Table 3-2.

Table 3-2. Typical Failure Mechanisms and Causes For Dynamic Seals (Also see Table 3-1)

FAILURE MODE	FAILURE MECHANISMS	FAILURE CAUSES
Excessive leakage	Wear	- Misalignment - Shaft out-of-roundness - Excessive shaft end play - Excessive torque - Poor surface finish - Contaminants - Inadequate lubrication - Excessive rubbing speed
	Dynamic instability	- Shaft misalignment
	Embrittlement	- Contaminants - Fluid/seal incompatibility - Thermal degradation - Idle periods between use
	Mechanical spring Failure	- See Chapter 4, Table 4-1
	Fracture	- Stress-corrosion cracking - Excessive PV value - Excessive fluid pressure on seal
	Edge chipping	- Excessive shaft deflection - Seal faces out-of-square - Excessive shaft whip
	Axial shear	- Excessive pressure loading

Table 3-2. (continued) Typical Failure Mechanisms and Causes For Dynamic Seals (Also see Table 3-1)

FAILURE MODE	FAILURE MECHANISMS	FAILURE CAUSES
Excessive leakage	Torsional shear	- Excessive torque due to improper lubrication - Excessive fluid pressure
	Compression set	- Extreme temperature operation
	Fluid seepage	- Insufficient seal squeeze - Foreign material on rubbing surface
	Seal face distortion	- Excessive fluid pressure on seal - Foreign material trapped between faces - Excessive PV value of seal operation - Insufficient seal lubrication - Seal shrinkage
Slow mechanical response	Excessive friction	- Excessive squeeze - Excessive seal swell - Seal extrusion - Metal – to – metal contact (out of alignment)

3.3.2 Pressure Velocity

An important factor in the design of dynamic seals is the pressure velocity (PV) coefficient. The PV coefficient is defined as the product of the seal face or system pressure and the fluid velocity. This factor is useful in estimating seal reliability when compared with manufacturer's limits. If the PV limit is exceeded, a seal may wear at a rate greater than desired.

$$Q_S = 0.077 \bullet PV \bullet \mu \bullet a_o \tag{3-12}$$

Where:
Q_S = Heat input from the seal, BTU/hour
PV = Pressure-velocity coefficient [See Equation (3-13)]
μ = Coefficient of friction (See Table 3-6)
a_o = Seal face area, in^2

The following equation defines the PV factor.

$$PV = \frac{\pi}{12} \bullet DP \bullet d \bullet V \bullet k \qquad \textbf{(3-13)}$$

Where:

PV = Pressure-Velocity, lbs/in^2 $\bullet$ ft/min

DP = Pressure differential across seal, lbs/in^2

d = Diameter of seal, inches

V = Operating speed, rpm

k = Degree of seal unbalance (1.0 for unbalanced and 0.4 for balanced)

Or:

$$PV = \frac{\pi}{12} \bullet DP \bullet d \bullet V \bullet k$$

Where: V = Operating speed, ft/min

The frictional aspects of materials are not only important from a reliability viewpoint. Performance must also be considered. The more resistance a system incurs, the more power is lost and also the lower the efficiency value for the component.

There should be special consideration for tradeoffs involved with each type of seal material. For example, solid silicon carbide has excellent abrasion resistance, good corrosion resistance, and moderate thermal shock resistance. This material has better qualities than a carbon-graphite base material but has a PV value of 500,000 while carbon-graphite has a 50,000 PV value. With all other values being the same, the heat generated would be five times greater for solid silicon carbide than for carbon-graphite materials. The required cooling flow to the solid silicon carbide seal would be larger to maintain the film thickness on the dynamic seal faces. If this cooling flow can't be maintained, then an increase in wear would occur due to higher surface temperatures. A tradeoff analysis is normally performed for each candidate design to maximize reliability. Typical PV limits are shown in Table 3-7.

3.3.3 Failure Rate Model for Dynamic Seals

Most of the seal modifying factors will remain the same as the ones previously specified by Equation (3-7), the exceptions being surface finish (See Section 3.3.3.1) and the addition of the PV factor for rotational speeds greater than 800 rpm or linear speeds greater than 600 ft/min (See Section 3.3.3.3). The seal model is modified as shown in Equation (3-14).

$$\lambda_{SE} = \lambda_{SE,B} \bullet C_P \bullet C_Q \bullet C_{DL} \bullet C_H \bullet C_F \bullet C_V \bullet C_T \bullet C_N \bullet C_{PV} \qquad \textbf{(3-14)}$$

Where:

λ_{SE} = Failure rate of dynamic seal in failures/million hours

$\lambda_{SE,B}$ = Base failure rate of dynamic seal, 22.8 failures/million hours

C_P = Multiplying factor which considers the effect of fluid pressure on the base failure rate for seal movement < 800 rpm or 600ft/min (See Figure 3.10)

C_P = 1.0 for seal movement $\geq$ 800 rpm or 600ft/min

C_Q = Multiplying factor which considers the effect of allowable leakage on the base failure rate (See Figure 3.11) and Section 3.2.3.2

C_{DL} = Multiplying factor which considers the effect of seal size on the base failure rate (See Figure 3.12) for seal movement < 800 rpm or 600ft/min

C_{DL} = 1.0 for seal movement $\geq$ 800 rpm or 600 ft/min

C_H = Multiplying factor which considers the effect of contact stress and seal hardness on the base failure rate for movement < 800 rpm or 600 ft/min (See Figure 3.14)

C_H = 1.0 for seal movement $\geq$ 800 rpm or 600 ft/min

C_F = Multiplying factor which considers the effect of surface finish on the base failure rate (See Sections 3.3.3.1, 3.2.3.4 and Figure 3.17)

C_V = Multiplying factor which considers the effect of fluid viscosity on the base failure rate (See Table 3-3 and Section 3.2.3.5)

C_T = Multiplying factor which considers the effect of seal temperature on the base failure rate (See Figure 3.16 and Section 3.2.3.6)

C_N = Multiplying factor which considers the effect of contaminants on the base failure rate (See Table 3-4 and Section 3.3.3.2)

C_{PV} = Multiplying factor which considers the effect of the pressure-velocity coefficient on the base failure rate for movement ≥ 800 rpm and/or 600 ft/min (See Sections 3.3.2 and 3.3.3.3)

C_{PV} = 1.0 for seal movement < 800 rpm or 600 ft/min

3.3.3.1 Surface Finish Multiplying Factor

Surface irregularities of dynamic seals may be more pronounced than static seals. In dynamic seal applications where the seal mates with a shaft, shaft hardness, smoothness and material are factors which must be considered in the design evaluation process. Maximum seal efficiency and life are obtained with a finely finished gland surface, usually in the 10 to 20 microinch range. A metal surface finish of less than 8 microinches rms increases the total frictional drag of a compound moving against it. The degree to which the finish can be maintained in the operating range must be considered when determining the surface finish of the gland for use in the model. Figure 3.17 provides a value for the surface finish multiplying factor as a function of the surface finish.

3.3.3.2 Fluid Contaminant Multiplying Factor

One of the factors in estimating the failure rate of a dynamic seal is the number of contaminants in contact with the seal generated from other components in the system. For example, when a cylinder rod extends out into a dirty environment where it can pick up dirt, lint, metal chips and other contaminants, this foreign material can nullify the benefits of the lubricant and cause rapid abrasive wear of both the O-ring and the rod. Equipment exposed to such conditions should contain a wiper ring to prevent the foreign material from reaching the O-ring. A felt ring is usually installed between the wiper and the seal to maintain lubrication of the rod during its return stroke.

Table 3-4 provides fluid contaminant multiplying factors for various components that may be generating contaminants.

3.3.3.3 PV Multiplying Factor

C_{PV} is the multiplying factor that multiplies the base failure rate by the ratio of PV value for actual seal operation to design PV value. C_{PV} is applicable to rotary seals, lip seals and other dynamic seals that rotate with a shaft or reciprocate with a velocity greater than 5 in/sec and where a PV design factor is available from the manufacturer. The values for PV_{DS} and PV_{OP} used in Equation (3-15) will use the PV formulation in Equation (3-13).

$$C_{PV} = \frac{PV_{OP}}{PV_{DS}} \tag{3-15}$$

Where: PV_{OP} = PV factor for actual seal operation

PV_{DS} = PV factor for the original design

3.4 MECHANICAL SEALS

Mechanical seals are designed to prevent leakage between flat, rotating surfaces. A typical contacting type dynamic seal is shown in Figure 3.8. In this example, the sealing surfaces are perpendicular to the shaft, with contact between the primary and mating rings to achieve a dynamic seal. Each of the sealing surfaces is lapped flat to eliminate leakage. Wear occurs at the dynamic seal faces from sliding contact between the primary and mating rings. The rate of wear is small, as a film of the liquid sealed is maintained between the sealing faces. Preload from a spring is required to produce an initial seal, the spring pressure holding the primary and mating rings together during shutdown or when there is a lack of fluid pressure.

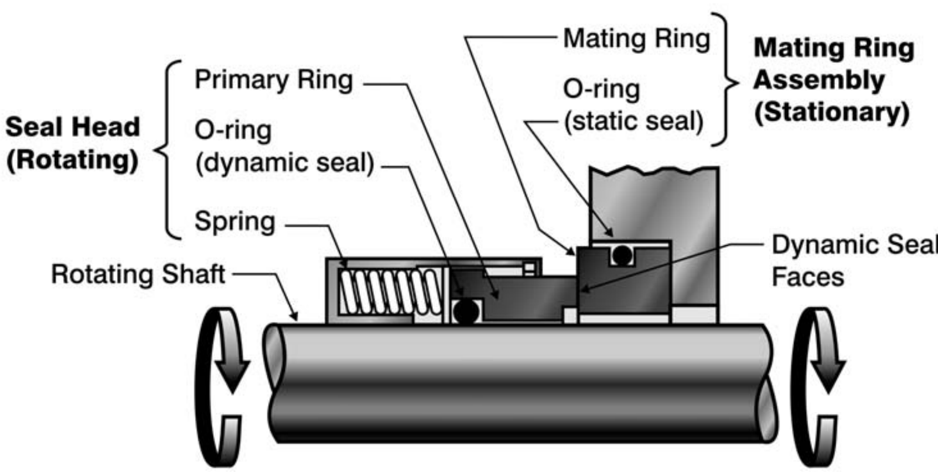


Figure 3.8 Typical Mechanical Seal

Wear occurs between the primary ring and mating ring of a mechanical seal. This surface contact is maintained by a spring. There is a film of liquid maintained between the sealing surfaces to eliminate as much friction as possible. For most mechanical seals, the three common points of sealing contact occur between the following points:

(1) Mating surfaces between primary and mating rings

(2) Between the rotating component and shaft or sleeve

(3) Between the stationary component and the gland plate

The reliability of a mechanical seal depends to a very large extent on its ability to maintain a thin fluid film in the gap between the mating faces while simultaneously minimizing the duration and extent of mechanical contact between asperities on the rubbing areas of these faces. Too much contact may overheat the materials; not enough contact may cause high leakage rates.

A mechanical seal may be an unbalanced design or a balanced design. Unbalanced seals are seal arrangements in which the hydraulic pressure of the seal chamber acts on the entire seal face without any of the force being reduced through the seal design. Unbalanced seals usually have a lower pressure limitation than balanced seals. A balanced seal design reduces the hydraulic forces acting on the seal faces through mechanical seal design. As the seal faces rub together, the amount of heat generated is determined by the amount of pressure applied, the lubricating film between the faces, the rotational speed, and the seal ring materials. Balanced seals reduce the seal ring area on which the stuffing box pressure acts. With a reduction in area, the overall closing force is diminished. This results in better lubrication and reduced heat generation and face wear compared to unbalanced seals. Unbalanced and balanced seal designs are shown in Figure 3.9. The failure rate equation assumes a balanced seal.

3.4.1 Mechanical Seal Failure Modes

Failure modes of a mechanical seal can be identified by three main causes of failure: temperature, pressure and velocity and a combination of these variables. For example, fluid pressure can create extra heat at the seal face which in turn can increase the rate of wear and other destructive failure modes such as material fracture and distortion and leakage. Elastomer seals can become extruded and damaged. As the pressure is increased, the probability of failure goes up.

Some mechanical seals wear out with use and some fail prior to wearing out. The seal face is the only part of a mechanical seal designed to wear out. Mechanical face seals should last until the carbon face wears away. If the seal starts leaking before that happens and the seal requires replacement, then the seal has failed. In some cases the seal face has opened because it became jammed on the rotating component. Another possibility is that one of the seal components such as the spring was damaged by contact, heat or corrosion.

Wear often occurs between the primary ring and mating ring. This surface contact is maintained by a spring. There is a film of liquid maintained between the sealing surfaces to eliminate as much friction as possible. For most mechanical seals the three common points of sealing contact occur between the following points:

(1) Mating surfaces between primary and mating rings

(2) Between the rotating component and shaft or sleeve

(3) Between the stationary component and the gland plate

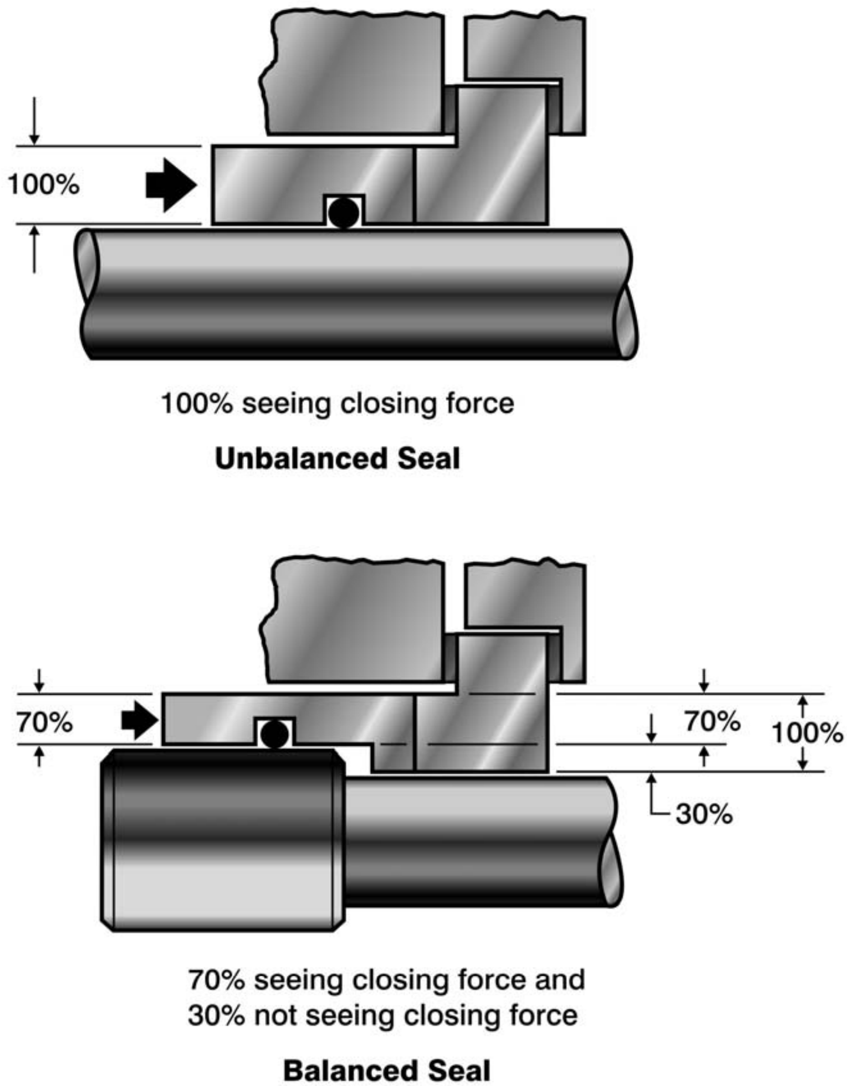


Figure 3.9 Balanced and Unbalanced Seal Designs

Seal balance is a performance characteristic that measures how effective the seal mating surfaces match. If not effectively matched, the seal load at the dynamic facing may be too high causing the liquid film to be squeezed out and vaporized, thus causing a high wear rate. The fluid pressure from one side of the primary ring causes a certain amount of force to impinge on the dynamic seal face. The dynamic facing pressure can be controlled by manipulating the hydraulic closing area with a shoulder on a sleeve or by seal hardware. By increasing the area, the sealing force is increased.

The reliability of a mechanical seal depends to a very large extent on its ability to maintain a thin film in the gap between the mating surfaces and at the same time minimizing the mechanical contact of the face surfaces. Too much contact may cause overheating of the face materials and insufficient contact may cause excessive leakage. Over time common modes of seal failure are by fatigue-like surface embrittlement, abrasive removal of material, and corrosion. Wear and sealing efficiency of fluid system seals are related to the characteristics of the surrounding operating fluid. Abrasive particles present in the fluid during operation will have a strong influence on the wear resistance of seals, the wear rate of the seal increasing with the quantity of environmental contamination. Contaminants from the environment such as sand can enter the fluid system and become embedded in the elastomeric seals causing abrasive cutting and damage to shafts as well as the mechanical seal.

There should be special consideration for tradeoffs involved with each type of seal material. For example, solid silicon carbide has excellent abrasion resistance, good corrosion resistance, and moderate thermal shock resistance. This material has better qualities than a carbon-graphite base material but has a *PV* value of 500,000 lb/in-min while carbon-graphite has a 50,000 lb/in-min *PV* value. With all other values being the same, the heat generated would be five times greater for solid silicon carbide than for carbon-graphite materials. The required cooling flow to the solid silicon carbide seal would be larger to maintain the film thickness on the dynamic seal faces. If this cooling flow can't be maintained, then an increase in wear would occur due to higher surface temperatures.

3.4.2 Failure Rate Model for Mechanical Seals

An important seal design consideration is seal balance. Seal balance refers to the difference between the pressure of the fluid being sealed and the contact pressure between the seal faces. It is the ratio of hydraulic closing area to seal face area (parameter k in Equation (3-13). A balanced seal is designed so that the effective contact pressure is always less than the fluid pressure, reducing friction at the seal faces. The result is less rubbing wear, less heat generated and higher fluid pressure capability. In an unbalanced seal, fluid pressure is not relieved by the face geometry, the seal faces withstand full system fluid pressure in addition to spring pressure and the face contact pressure is greater than or equal to fluid pressure. The failure rate equation assumes a balanced seal.

Of greatest importance with mechanical seals is a properly designed seal face. Proper mating surface materials must be matched so that excessive heat isn't generated from the dynamic motion of the seal faces. Too much heat can cause thermal distortions on the face of the seal and cause gaps which can increase the leakage rate. It can also cause material changes that can significantly increase the seal wear rate. Therefore, a careful review of the seal material should be made for each surface of the dynamic seal face. Equation (3-12) (Reference 26) includes such coefficients of friction and wear rate. Table 3-6 shows frictional values for various seal face materials.

An important factor in the design of mechanical seals is the pressure velocity (PV) coefficient. The PV coefficient is defined as the product of the seal face or system pressure and the fluid velocity. This factor is useful in estimating seal reliability when compared with manufacturer's limits. If the PV limit is exceeded, a seal may wear at a rate greater than desired. PV limits are included in manufacturer's specification sheets.

$$\lambda_{SE} = \lambda_{SE,B} \bullet C_Q \bullet C_F \bullet C_V \bullet C_T \bullet C_N \bullet C_{PV} \qquad \textbf{(3-16)}$$

Where: λ_{SE} = Failure rate of mechanical seal in failures/million hours

$\lambda_{SE,B}$ = Base failure rate of mechanical seal, 22.8 failures/million hours

C_Q = Multiplying factor which considers the effect of allowable leakage on the base failure rate (See Figure 3.11) and Section 3.2.3.2

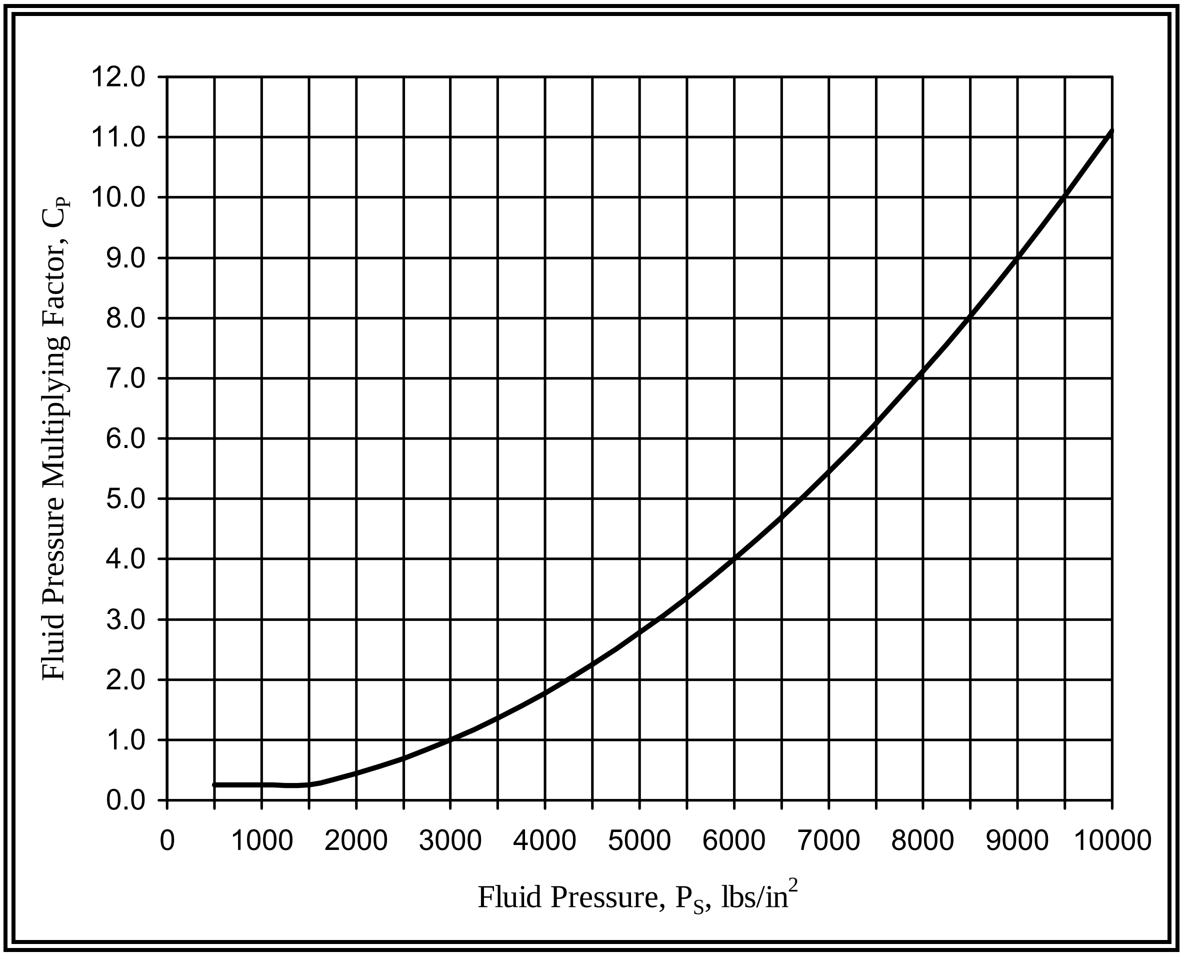
C_f = Multiplying factor which considers the effect of seal face surface finish on the base failure rate (See Sections 3.3.3.1, 3.2.3.4 and Figure 3.17)

C_V = Multiplying factor which considers the effect of fluid viscosity on the base failure rate (See Table 3-3 and Section 3.2.3.5)

C_T = Multiplying factor which considers the effect of seal face temperature on the base failure rate (See Figure 3.16 and Section 3.2.3.6)

C_N = Multiplying factor which considers the effect of contaminants on the base failure rate (See Table 3-4 and Section 3.3.3.2)

C_{PV} = Multiplying factor which considers the effect of the pressure-velocity coefficient on the base failure rate (See Sections 3.3.2 and 3.3.3.3)

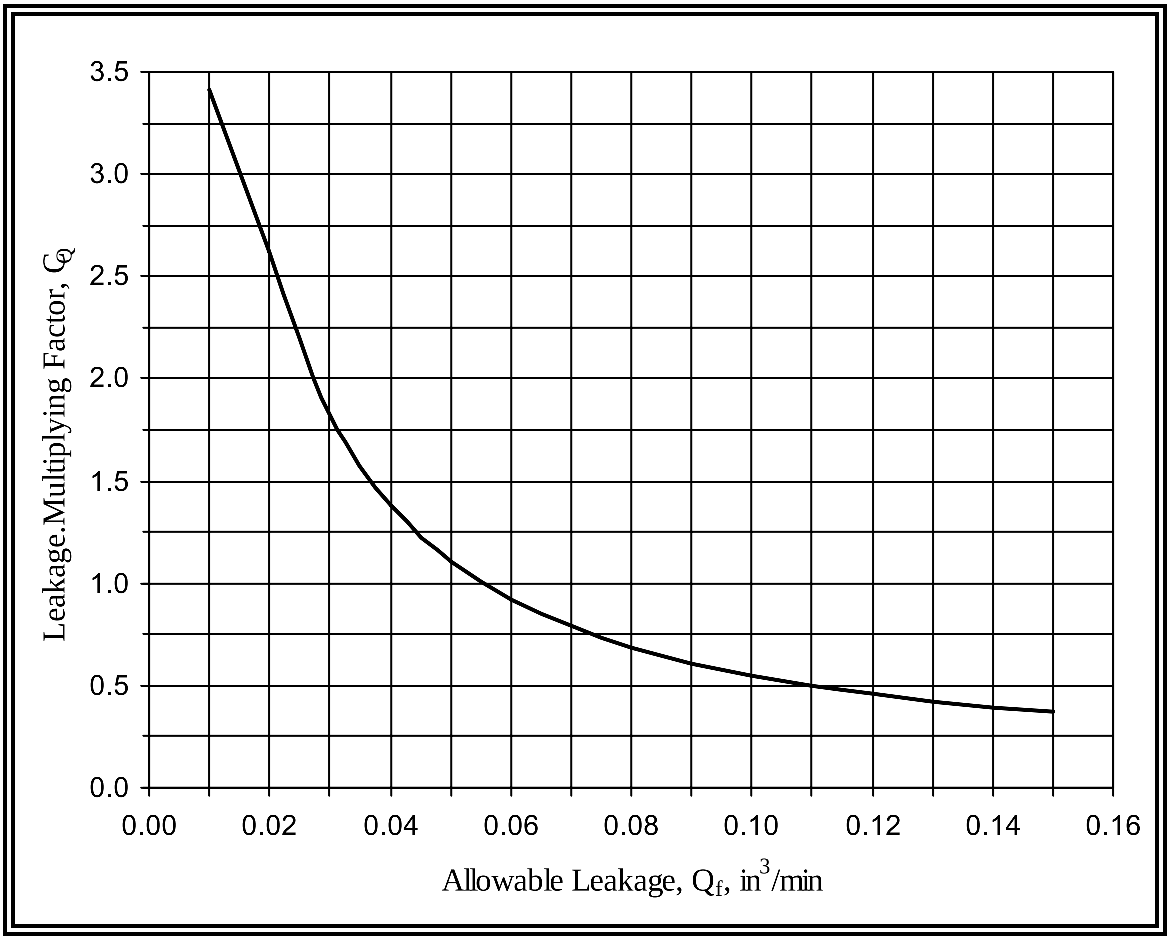


For $P_S \leq 1500$ lbs/in^2, $C_P = 0.25$

For $P_S > 1500$ lbs/in^2, $C_P = \left(\frac{P_S}{3000} \right)^2$

Where $P_S = P_1 - P_2$ (upstream – downstream pressure)

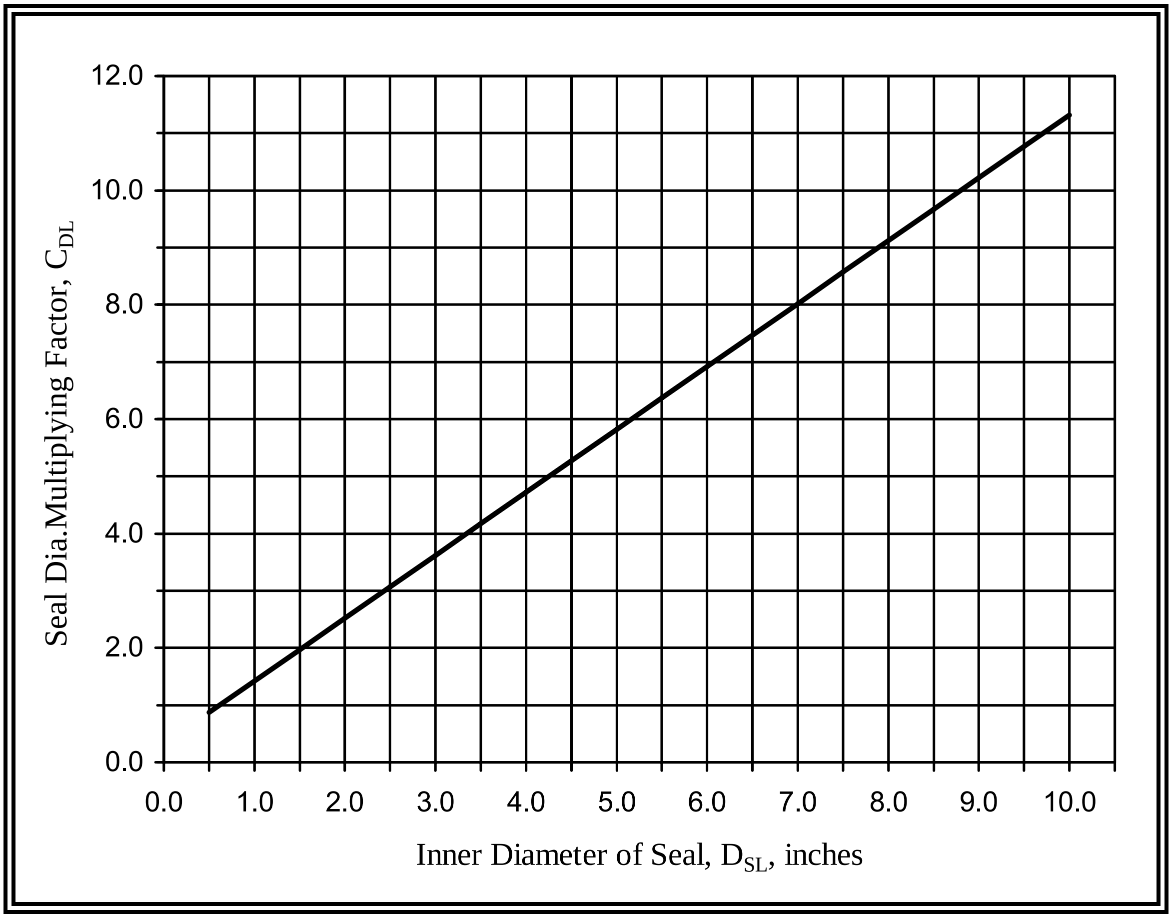
Figure 3.10 Fluid Pressure Multiplying Factor, C_P



For Leakage $>$ 0.03 in^3/min, $C_Q = 0.055/Q_f$

For Leakage $\leq$ 0.03 in^3/min, $C_Q = 4.2 - (79\ Q_f)$

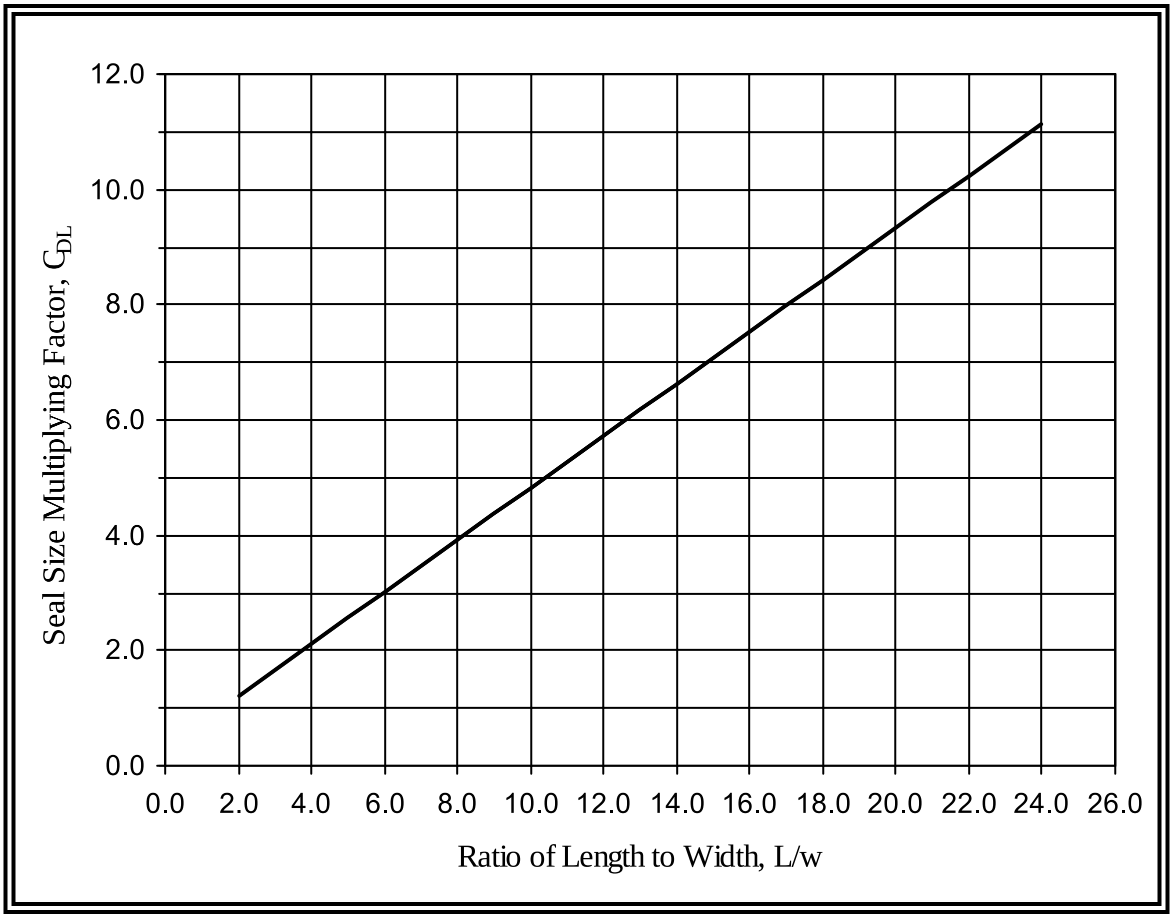
Figure 3.11 Allowable Leakage Multiplying Factor, C_Q



$$C_{DL} = 1.1\, D_{SL} + 0.32$$

Where: D_{SL} = Inner diameter of seal

Figure 3.12 Seal Diameter Multiplying Factor, C_{DL}

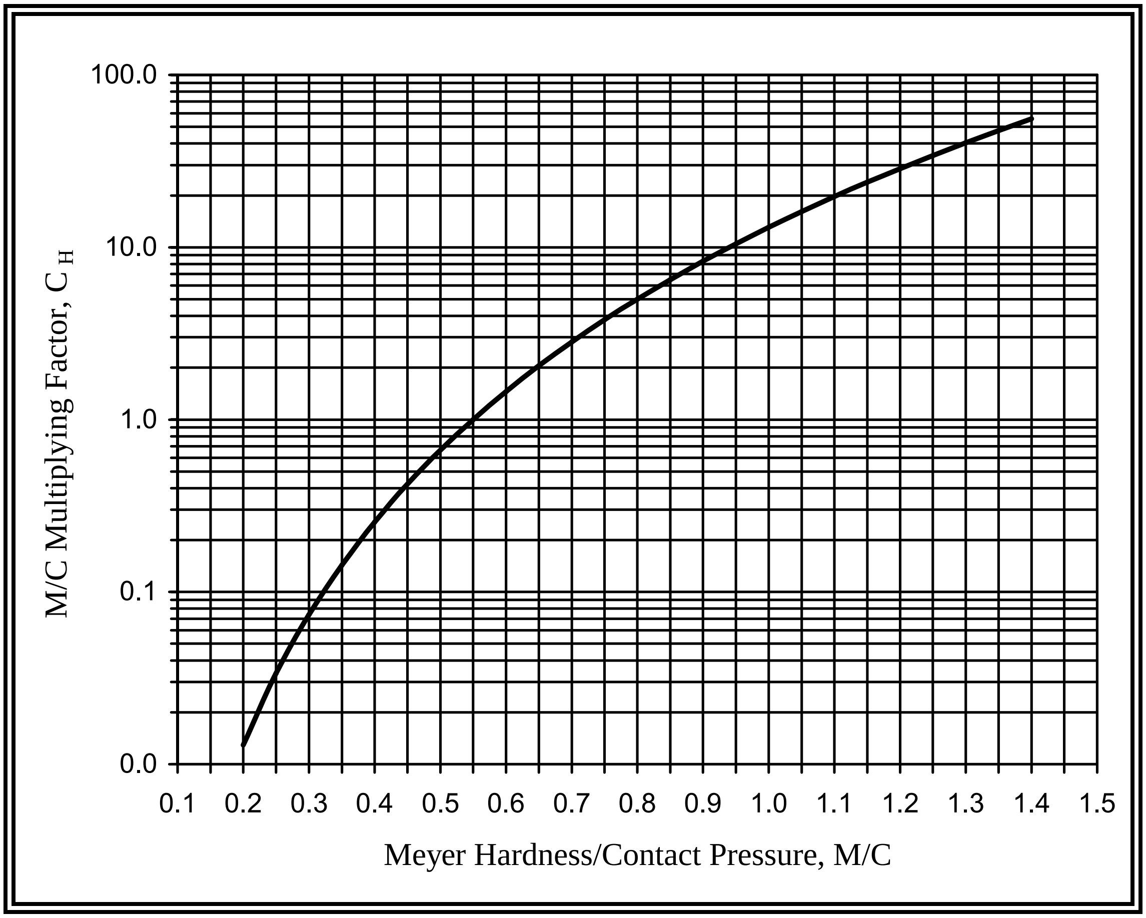


$$C_{DL} = 0.45\left(\frac{L}{w}\right) + 0.32$$

Where: L = Total linear length of gasket

w = Minimum width of gasket

Figure 3.13 Gasket Size Multiplying Factor, C_{DL}

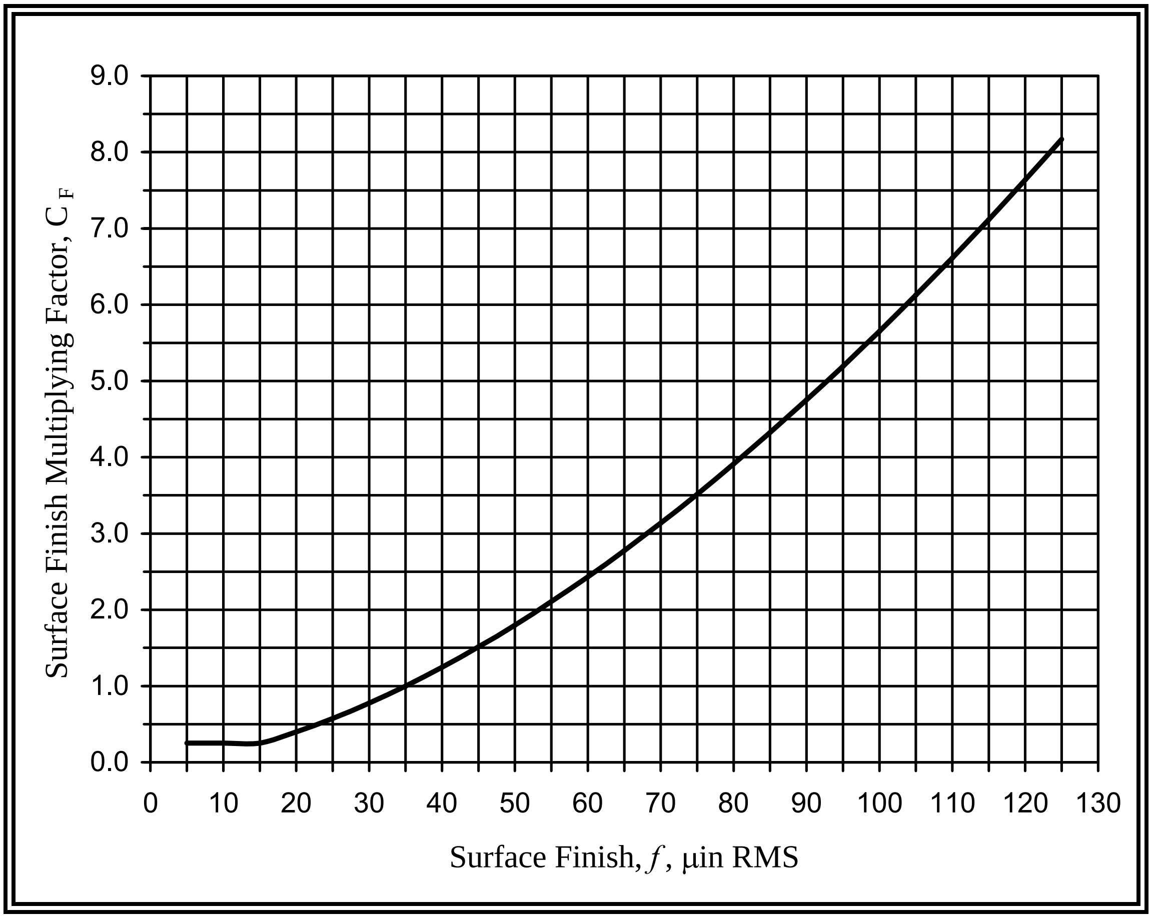


$$C_H = \left(\frac{M / C}{0.55} \right)^{4.3}$$

Where: M = Meyer Hardness, lbs/in^2

C = Contact Pressure, lbs/in^2

Figure 3.14 Material Hardness/Contact Pressure, C_H

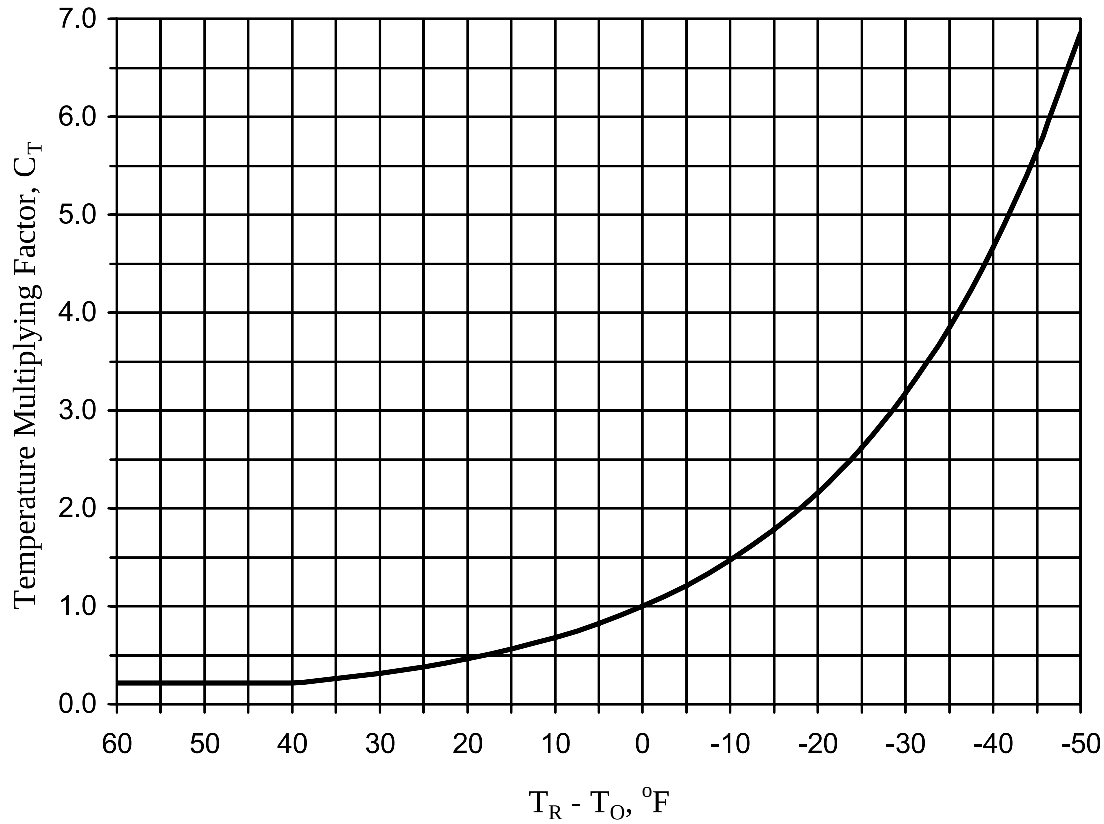


For $f \leq 15$ µin, $C_f = 0.25$

For $f > 15$ µin, $C_f = \dfrac{f^{1.65}}{353}$

Where: f = Surface Finish, µin RMS

Figure 3.15 Surface Finish Multiplying Factor, C_F (for static seals)



$$C_T = \frac{1}{2^t}$$

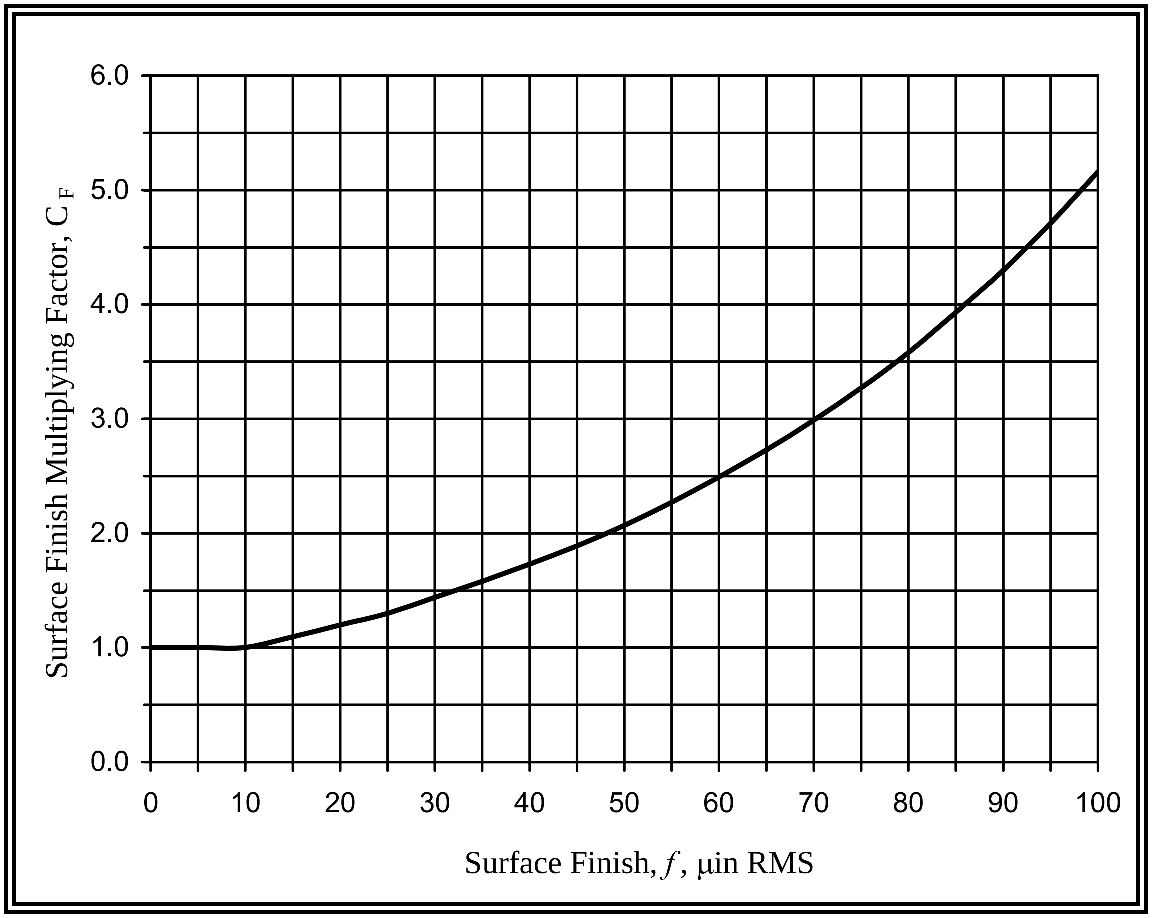
Where: $t = \frac{(T_R - T_O)}{18}$ for $(T_R - T_O) \leq 40\ ^oF$

and: $C_T = 0.21$ for $(T_R - T_O) > 40\ ^oF$

T_R = Rated Temperature of Seal, ^oF (See Table 3-6)

T_O = Operating Temperature of Seal, ^oF

Figure 3.16 Temperature Multiplying Factor, C_T



For $f \leq 10$ μin, $C_f = 1.00$

For $f > 10$ μin, $C_f = \dfrac{1}{2^{((10-f)/38)}}$

Where: f = Surface Finish, μin RMS

Figure 3.17 Surface Finish Multiplying Factor, C_F (for dynamic seals)

Table 3-3. Fluid Viscosity/Temperature Multiplying Factor, C_ν for Typical Fluids

FLUID	C_ν								
	Fluid Temperature, °F								
	-50	0	50	100	150	200	250	300	350
Air	554.0	503.4	462.9	430.1	402.6	379.4	359.5	---	---
Oxygen	504.6	457.8	420.6	390.2	365.9	343.6	325.3	---	---
Nitrogen	580.0	528.0	486.5	452.6	424.3	400.0	379.6	---	---
Carbon Dioxide	---	599.9	510.7	449.7	395.9	352.1	---	---	---
Water	---	---	6.309	12.15	19.43	27.30	---	---	---
SAE 10 Oil	---	---	0.060	0.250	0.750	1.690	2.650	---	---
SAE 20 Oil	---	---	0.0314	0.167	0.492	1.183	2.213	2.861	5.204
SAE 30 Oil	---	---	0.0297	0.1129	0.3519	0.8511	1.768	2.861	4.309
SAE 40 Oil	---	---	0.0122	0.0534	0.2462	0.6718	1.325	2.221	3.387
SAE 50 Oil	---	---	0.0037	0.0326	0.1251	0.3986	0.8509	1.657	2.654
SAE 90 Oil	---	---	0.0012	0.0189	0.0973	0.3322	0.7855	1.515	2.591
Diesel Fuel	0.1617	0.7492	2.089	3.847	6.228	9.169	12.78	16.31	---
MIL-H-83282	0.0031	0.0432	0.2137	0.6643	1.421	2.585	4.063	0.6114	0.7766
MIL-H-5606	0.0188	0.0951	0.2829	0.6228	1.108	1.783	2.719	3.628	4.880

--- Data for these temperatures determined to be unreliable

$$C_\nu = \left(\frac{\nu_O}{\nu}\right)$$

Where: ν_O = 2 x 10^{-8} lbf-min/in^2

ν = Dynamic viscosity of fluid being used, lbf-min/in^2

Table 3-4. Contaminant Multiplying Factor, C_N

TYPICAL QUANTITIES OF PARTICLES PRODUCED BY HYDRAULIC COMPONENTS	PARTICLE MATERIAL	NUMBER PARTICLES UNDER 10 MICRON PER HOUR PER RATED GPM (N10)
Piston Pump	steel	0.017
Gear Pump	steel	0.019
Vane Pump	steel	0.006
Cylinder	steel	0.008
Sliding action valve	steel	0.0004
Hose	rubber	0.0013

$$C_N = \left(\frac{C_O}{C_{10}}\right)^3 \bullet FR \bullet N_{10}$$

Where: C_o = System filter size in microns

C_{10} = Standard system filter size = 10 micron

FR = Rated flow rate, GPM

N_{10} = Particle size factor

Table 3-5. T_R Values for Typical Seal Materials (Reference 27)

SEAL MATERIAL	T_R (°F)
Natural rubber	160
Ethylene propylene	250
Neoprene	250
Nitrile	250
Polyacrylate	300
Fluorosilicon	450
Fluorocarbon	475
Silicon rubbers	450
Butyl rubber	250
Urethane	210
Fluroelastomers	500
Fluroplastics	500
Leather	200
Impregnated poromeric material	250

Table 3-6. Coefficient of Friction for Various Seal Face Materials

SLIDING MATERIALS		COEFFICIENT OF FRICTION (μ)
ROTATING (seal head)	**STATIONARY (mating ring)**	
Carbon-graphite (resin filled)	- Cast Iron - Ceramic - Tungsten Carbide - Silicon Carbide - Silicon Carbide Converted Carbon	0.07 0.07 0.07 0.02 0.015
Silicon carbide	- Tungsten Carbide - Silicon Carbide Converted Carbon - Silicon Carbide - Tungsten Carbide	0.02 0.05 0.02 0.08

Table 3-7. Typical Pressure Velocity (PV) Limits

Face Materials	PV (lb/in^2 ft/min)
Carbon vs hard faced stainless steel	543,000
Carbon vs ceramic	543,000
Carbon vs leaded bronze	992,000
Carbon vs nickel iron	1,142,000
Carbon vs tungsten carbide	2,570,000

3.5 REFERENCES

In addition to specific references cited throughout Chapter 3, other references included below are recommended in support of performing a reliability analysis of seals and gaskets.

5. Bauer, P., M. Glickmon, and F. Iwatsuki, "Analytical Techniques for the Design of Seals for Use in Rocket Propulsion Systems," Volume 1, ITT Research Institute, Technical Report AFRPL-TR-65-61 (May 1965).

18. Hauser, D.L. et al., "Hardness Tester for Polyur," NASA Tech Briefs, Vol. 11, No. 6, p. 57 (1987).

22. Howell, Glen W. and Terry M. Weathers, Aerospace Fluid Component Designers' Handbook, Volumes I and II, TRW Systems Group, Redondo Beach, CA prepared for Air Force Rocket Propulsion Laboratory, Edwards, CA, Report AD 874 542 and Report AD 874 543 (February 1970).

26. Krutzsch, W.C., Pump Handbook, McGraw-Hill Book Company, New York (1968).

27. May, K.D., "Advanced Valve Technology," National Aeronautics and Space Administration, NASA Report SP-5019 (February 1965).

83. Handbook of Chemistry and Physics, 86th Edition, CRC Press, 2005

105. OREDA Offshore Reliability Data, 5th Edition Det Norske Veritas, N-1363 Hovik, Norway 2009 ISBN 978-82-14-04830-8

123. Centrifugal Pump & Mechanical Seal Manual, William J. McNally, 2009

124. Parker O-Ring Handbook, 2001 Edition, Catalog ORD 5700/US, Parker Hannifin Corporation

125. “Improving the Reliability of Mechanical Seals”, Michael Huebner, Chemical Engineering Progress, November 2005

CHAPTER 4

SPRINGS

4.0 TABLE OF CONTENTS

4.1 INTRODUCTION

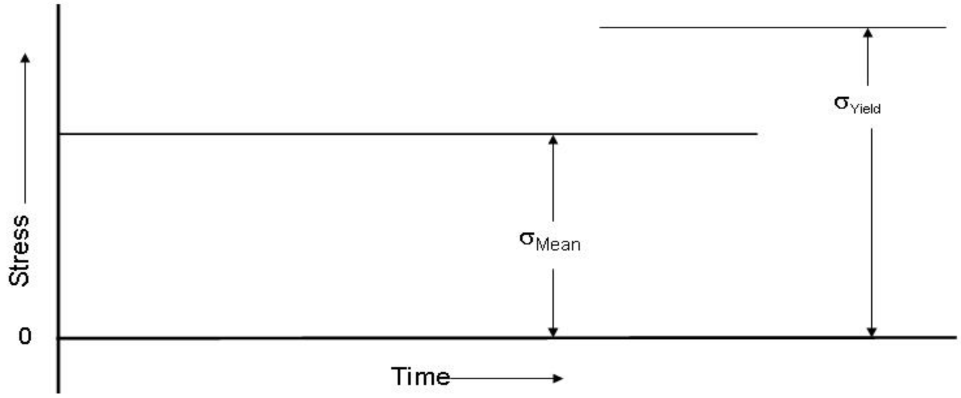
Mechanical springs are used in machine designs to exert force, provide flexibility, and to store or absorb energy. Springs are manufactured for many different applications such as compression, extension, torsion, power, and constant force. Depending on the application, a spring may be in a static, cyclic or dynamic operating mode. A spring is usually considered to be static if a change in deflection or load occurs only a few times, such as less than 10,000 cycles during the expected life of the spring. A static spring may remain loaded for very long periods of time. The failure modes of interest for static springs include spring relaxation, set and creep.

Cyclic springs are flexed repeatedly and can be expected to exhibit a higher failure rate due to fatigue. Cyclic springs may be operated in a unidirectional mode or a reversed stress mode. In one case, the stress is always applied in the same direction, while in the other, stress is applied first in one direction then in the opposite direction. Figure 4.1 shows the difference in deflection and stress between these two operating modes. For the same maximum stress and deflection between a unidirectional and reversed stress spring, the stress range for the reversed stress spring will be twice that of the unidirectional spring and therefore a shorter fatigue life would be expected.

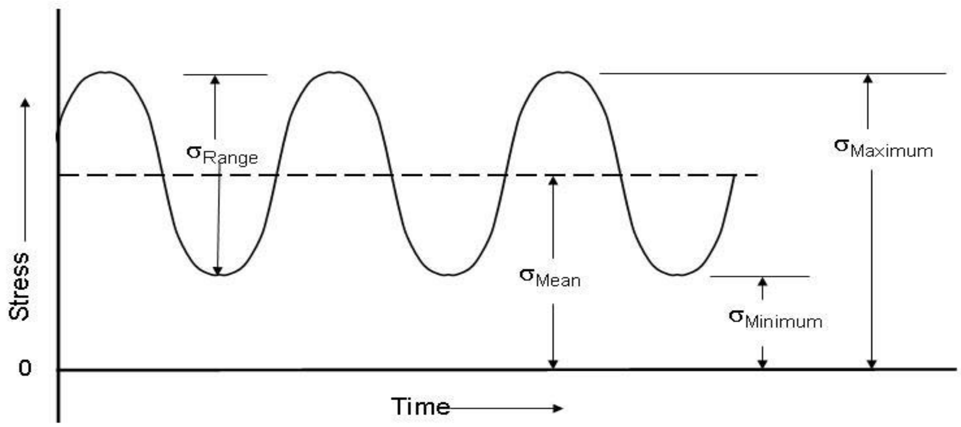
Dynamic loading refers to those intermittent occurrences of a load surge such as a shock absorber inducing higher than normal stresses on the spring. Dynamic loading of a spring falls into three main categories: shock, resonance of the spring itself, and resonance of the spring/mass system. Shock loading occurs when a load is applied with sufficient speed such that the first coils of the spring take up more of the load than would be calculated for a static or cyclic situation. This loading is due to the inertia of the spring coils. Spring resonance occurs when the operating speed is the same as the natural frequency of the spring or a harmonic of the natural frequency. Resonance can cause greatly elevated stresses and possible coil clash resulting in premature failure. Resonance of the spring/mass system occurs when the spring is required to carry a mass attached to its moving end and the combined system is subject to resonance at a much lower cycle rate than the spring alone. Failure modes for dynamic loading of a spring include fracture of the spring material due to shock pulses and resonance surging.

Springs tend to be highly stressed because they are designed to fit into small spaces with the least possible weight and lowest material cost. At the same time they are required to deliver the required force over a long period of time. The reliability of a spring is therefore related to its material strength, design characteristics, and the operating environment.

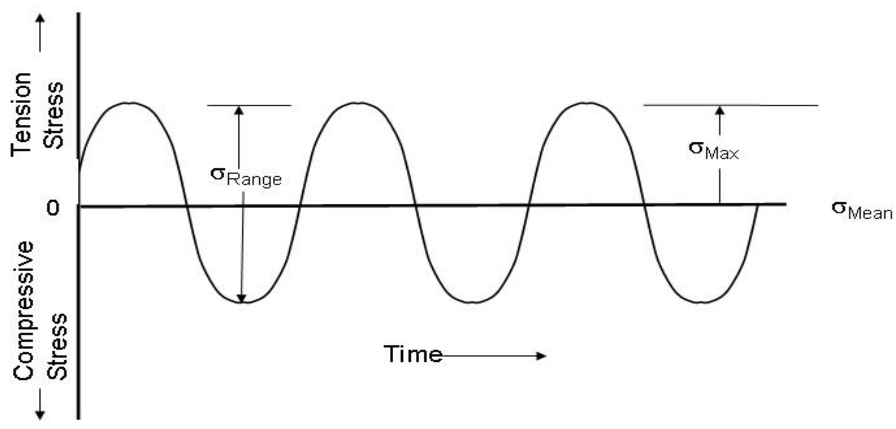
Most springs are made of steel and material strength of the spring is usually listed in terms of tensile strength in relation to the expected spring stress. Corrosion protection of the spring steel has a significant impact on reliability and therefore material properties, the processes used in the manufacturing of the spring, operating temperature, and corrosive media must all be known before any estimate of spring reliability can be made. Spring reliability is also directly related to the surface quality and the distribution, type and size of sub-surface impurities in the spring material.



Static Stress



Unidirectional Stress



Reversed Stress

Figure 4.1 Cyclic Modes of Spring Operation

Common materials of construction for springs include spring steel, stainless steel, nickel base alloy, and copper base alloy or bronze. Spring steel is any variety of steels that are normally of the high-carbon or alloy type. High carbon spring steels are probably the most commonly used material for springs except for those to be used in high or low temperature environments or for shock or high impact loads.

4.2 FAILURE MODES

Springs of all types are expected to operate over very long periods of time without significant changes in dimension, displacement or spring rate, often under changing loads. Considering these requirements, potential failure modes include yielding, fatigue, corrosion fatigue, fretting fatigue, creep, thermal relaxation, buckling, and force-induced elastic deformation. The operating life of a mechanical spring arrangement is dependent upon the susceptibility of the materials to corrosion and stress levels (static, cyclic or dynamic). The most common failure modes for springs are fracture due to fatigue and excessive loss of load due to stress relaxation. By definition, objects that are loaded under purely oscillatory loads (σ_{Mean} = 0 in Figure 4.1) fail when their stresses reach the material's fatigue limit $\sigma_{Fatigue}$. Conversely, objects that are loaded under purely static loads (σ_{Range} = 0) fail when their stresses reach the materials yield limit σ_{Yield}. For springs that have a mixture of σ_{Mean} and σ_{Alt} stresses, the Soderberg Criterion provides a way to calculate a failure limit. Mean stress is plotted on one axis and alternating stress on the other. Figure 4.2 is a typical Soderberg plot.

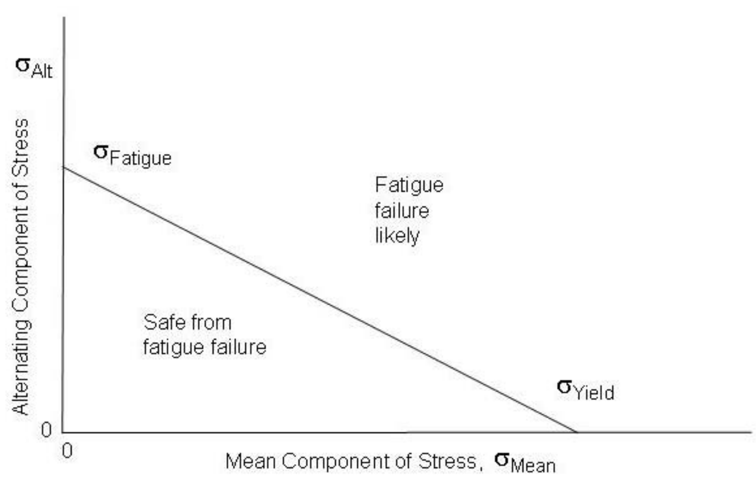


Figure 4. 2 Soderberg plot

4.2.1 Fatigue Stress

All springs have finite fatigue limits, the limit depending on fatigue stress and the degree of fluctuating loads. The four most common fatigue stress conditions include constant deflection, constant load, unidirectional stress and reversed stress. A spring inside a valve assembly is an example of a constant deflection where the spring is cycled through a specified deflection range. An example of a constant load spring is the use of vibration springs under a dead weight where the load applied to the spring does not change during operation but the deflection will. A unidirectional stress is one where the stress is always applied in the same direction such as used in the return spring of an actuator. A reversed stress is applied first in one direction then in the opposite direction such as used in a regulator valve. The three stages to a fatigue failure include crack initiation, crack propagation and finally fracture of the spring material.

Surging (resonant frequency response) can occur in high-speed cyclic applications if axial operating frequencies approach the axial natural frequency of the helical-coil spring. If the material and geometry of the axially reciprocating spring are such that its axial natural frequency is close to the operating frequency, a traveling displacement wave front is propagated and reflected along the spring with about the same frequency as the exciting force. This condition results in local compressions and rarefactions producing high stresses and/or erratic forces locally, with consequent loss of control of the spring-loaded object. Surging of a valve spring, for example may allow the valve to open erratically when it should remain closed or vice versa.

The S_{10} value for a spring is the number of cycles that 90% of the springs operating at the published stress level can be expected to complete or exceed before exhibiting the first evidence of fatigue. If an S_{10} value for the spring can be obtained, this value can be used in conjunction with the environmental multiplying factors contained in this Chapter. In order to have an average failure rate based on MTBF the S_{10} value should be multiplied by 4.55 to obtain an average life. The procedure for estimating spring failure rates contained herein is intended to be used in the absence of specific S_{10} data.

4.2.2 Spring Relaxation

Springs of all types are expected to operate over long periods of time without significant changes in dimension, displacement, or spring rates, often under fluctuating loads. If a spring is deflected under full load and the stresses induced exceed the yield strength of the material, the resulting permanent deformation may prevent the spring from providing the required force or to deliver stored energy for subsequent operations. Most springs are subject to some amount of relaxation during their life span even under benign conditions. The amount of spring relaxation is a function of the spring material and the amount of time the spring is exposed to the higher stresses and/or temperatures.

Static springs can be used in constant deflection or constant load applications. A constant deflection spring is cycled through a specified deflection range, the loads on the spring causing some set or relaxation which in turn lowers the applied stress. The spring may relax with time and reduce the applied load. Elevated temperatures can cause thermal relaxation, excess changes in spring dimension or reduced load supporting capability. Under constant load conditions, the load applied to the spring does not change during operation. Constant load springs may set or creep, but the applied stress is constant. The constant stress may result in fatigue lives shorter than those found in constant deflection applications.

In many applications, compression and extension springs are subjected to elevated temperatures at high stresses which can result in relaxation or loss of load. This condition is often referred to as "set". After the operating conditions are determined, set can be predicted and allowances made in the spring design. When no set is allowed in the application, the spring manufacturer may be able to preset the spring at temperatures and stresses higher than those to be encountered in the operating environment.

A highly stressed spring will set the first several times it is pressed. Relaxation is a function of a fairly high stress (but usually lower then that required to cause set) over a period of time. Creep in the spring may lead to unacceptable dimensional changes even under static loading (set). A spring held at a certain stress will actually relax more in a given time than a spring cycled between that stress and a lower stress because it spends more time at the higher stress. The amount of spring relaxation over a certain period of time is estimated by first determining the operating temperature, the maximum amount of stress the spring sees and how long the spring will be exposed to the maximum stress and the elevated temperature over its lifetime.

4.2.3 <u>Miscellaneous Failure Modes</u>

Most extension spring failures occur in the area of the spring end. Extension springs are designed to become longer under load and their maximum length must be controlled for long life. Their turns are normally touching in the unloaded position and they have a hook, eye or some other means of attachment. For maximum reliability, the spring wire must be smooth with a gradual flow into the end without tool marks, sharp corners or other stress risers. The spring ends should be made as an integral part of the coil winding operation and the bend radius should be at least one and one-half times the wire diameter.

The operating life of a mechanical spring arrangement is dependent upon the susceptibility of the materials to corrosion and stress levels (static, cyclic or dynamic). The most common failure modes for springs are fracture due to fatigue and excessive loss of load due to stress relaxation. Other failure mechanisms and causes need to be reviewed for a specific application to assure that all considerations of reliability are included in the prediction. Typical failure rate considerations include: level of loading, operating temperature, cycling rate and corrosive environment.

Table 4-1 is a list of failure mechanisms and causes of spring failure.

Table 4-1. Failure Modes for a Mechanical Spring

TYPE OF SPRING/STRESS CONDITION	FAILURE MODES	FAILURE CAUSES
- Static (constant deflection or constant load)	- Load loss - Creep - Set - Yielding	- Parameter change - Hydrogen embrittlement
- Cyclic (10,000 cycles or more during the life of the spring)	- Fracture - Damaged spring end - Fatigue failure - Buckling - Surging - Complex stress change as a function of time	- Excessive mean stress unidirectional operation - Material flaws - High temperature operation - Imperfection on inside diameter of the spring - Hydrogen embrittlement - Stress concentration due to tooling marks and rough finishes - Sharp bends on spring ends (extension springs) - Surface imperfections (high cycle with no shot peening) - Corrosive atmosphere - Misalignment - Excessive stress range of reverse stress - Cycling temperature - Low frequency vibration - High frequency vibration
- Dynamic (intermittent occurrences of a load surge)	- Fracture - Fatigue failure	- Maximum load ratio exceeded - Insufficient space for operation - Shock impulse - Surface defects - Excessive stress range of reverse stress - Resonance surging

4.3 FAILURE RATE CONSIDERATIONS

The following paragraphs describe the terms and parameters used in developing failure rate equations for springs.

4.3.1 Static Springs

Static springs can be used in constant deflection or constant load applications. A constant deflection spring is cycled through a specified deflection range, the loads on the spring causing some set or relaxation which in turn lowers the applied stress. The spring may relax with time and reduce the applied load. Under constant load conditions, the load applied to the spring does not change during operation. Constant load springs may set or creep, but the applied stress is constant. The constant stress may result in fatigue lives shorter than those found in constant deflection applications.

4.3.2 Cyclic Springs

As shown in Figure 4.1 cyclic springs can be classified as being unidirectional or reverse loaded. In one case, the stress is always applied in the same direction, while in the other, stress is applied first in one direction then in the opposite direction. Figure 4.20 shows the relationship between the cycle rate of a spring and its effect on failure rate.

4.3.3 Modulus of Rigidity

The modulus of rigidity (G_M) is a material property defining the resistance to shearing stresses for the spring material, the ratio of shearing stress to shear strain. Typical values of rigidity modulus are provided in Table 4-2. Modulus of Rigidity is sometimes referred to as Modulus in Torsion.

4.3.4 Modulus of Elasticity

The modulus of elasticity provides a measure of elasticity in tension for the spring material. Typical values are provided in Table 4-2.

4.3.5 Spring Index

Spring index (r) is the ratio of mean coil diameter to wire diameter. A spring with a high index will tend to tangle or buckle.

4.3.6 Spring Rate

Spring rate (R) is the change in load per unit of deflection, a measure of stiffness. Spring rate is measured in pounds / inch, the amount of weight needed to compress the spring one inch.

4.3.7 Shaped Springs

If the spring has a variable diameter such as occurs for conical, barrel and hourglass springs, the spring can be divided analytically into smaller increments and the failure rate calculated for each. The failure rate for the total spring is computed by adding the rates for the increments.

4.3.8 Number of Active Coils

For compression springs with closed ends, either ground or not ground, the number of active coils is two less than the total number of coils. There is some activity in the end coils, but during deflection, some active material comes in contact with the end coils and becomes inactive. Therefore, the total number of coils minus two is a good approximation for the number of active coils. For extension springs, all coils are active.

4.3.9 Tensile Strength

The tensile strength provides a measure of spring material deformation or set as a function of stress. Values of tensile strength are included in Table 4-3.

4.3.10 Corrosive Environment

Corrosion will reduce the load-carrying capability of a spring and its life. The precise effect of a corrosive environment on spring performance is difficult to predict. The reliability of a spring in terms of fatigue life and load-carrying ability will be affected by corrosion, the quantitative effect being very hard to predict. Springs are almost always in contact with other metal parts. If a spring is to be subjected to a corrosive environment, the use of inert materials provides the best defense against corrosion. Protective coatings can also be applied. In special situations, shot peening can be used to prevent stress corrosion and cathodic protection systems can be used to prevent general corrosion. The spring material is normally more noble (chemically resistant to corrosion) than the structural components in contact with it because the lesser noble alloy will be attacked by the electrolyte. The effects of corrosion on spring reliability must be based on experience data considering the extent of a corrosive environment. For failure rate models in Section 4.4 if corrosive protection is known to be applied to the spring during the manufacturing process, a multiplying factor, C_R, of 1.0 is used in conjunction with the base failure rate. Values of C_R greater than 1.0 are used based on the user's experience with the spring and the operating environment.

4.3.11 Manufacturing Processes

The following effects of manufacturing processes need to be considered in evaluating a design for reliability:

- Sharp corners and similar stress risers should be minimized.
- The hardness of the spring material can be sensitive to plating and baking operations. Previous experience with manufactured springs is the normal

assurance of quality control procedures. For failure rate models in Section 4.4, a multiplying factor, C_M, of 1.0 is used in conjunction with the base failure rate for known acceptable quality control procedures; otherwise a higher value for the multiplying factor is used based on previous experience with the manufacturer.

4.3.12 Other Reliability Considerations for Springs

The most common failure modes of springs include fracture due to fatigue and excessive loss of load. A reliability analysis should include a review of the following items to assure maximum possible life:

- When a spring is loaded or unloaded, a surge wave may transmit torsional stress to the point of restraint. The impact velocity should be determined to assure that the maximum load rating of the spring is not exceeded.
- Operating temperature should be determined. Both high and low temperature conditions may require consideration of specialized materials.
- Exposure to electrical fields may magnetize the spring material and cause fatigue failure.

4.4 FAILURE RATE MODELS

4.4.1 Compression Spring

The compression spring is the most commonly used spring in machine designs. An example of a compression spring is shown in Figure 4.3.

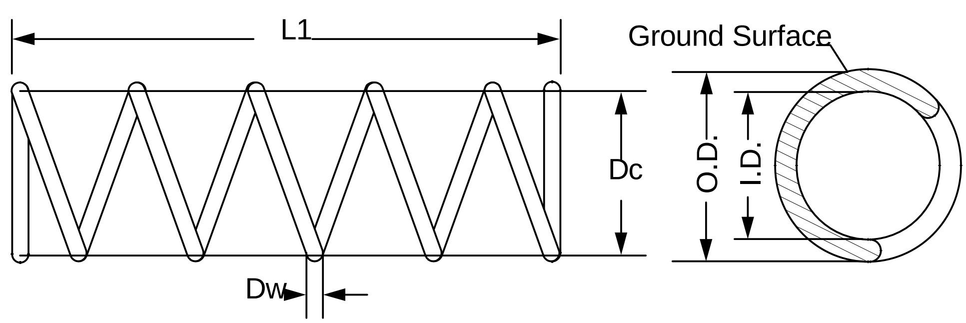


Figure 4.3 Typical Helical Compression Spring

The failure rate of a compression spring depends upon the stress on the spring and the relaxation provided by the material. This relaxation (change in load per unit deflection) is referred to as the spring rate, R, which can be defined as follows:

$$R = \frac{P_L}{L_1 - L_2} \tag{4-1}$$

Where:
R = Spring rate, lbs/in
P_L = Load, lbs
L_1 = Initial length of spring, in
L_2 = Final deflection of spring, in

The spring rate can be determined experimentally by deflecting the spring to 20% of available deflection and measuring the load (P_1) and spring length (L_1). Next, the spring is deflected to 80% of available deflection measuring the load (P_2) and spring length (L_2), being certain that no coils other than the closed ends are touching. The spring rate is then calculated as follows:

$$R = \frac{P_2 - P_1}{L_1 - L_2} = \frac{P_L}{L_1 - L_2} \tag{4-2}$$

Stress in the spring is also proportional to the load, P_L according to the following relationship:

$$S_G = \frac{8\, P_L\, D_C}{\pi\, D_W^{\,3}} K_W \tag{4-3}$$

Where:
S_G = Spring stress, lbs/in^2
K_W = Spring concentration factor (See equation 4-4)
D_c = Mean coil diameter, in
D_W = Wire Diameter, in

The spring concentration factor, K_W is a function of the Spring index (ratio of the coil diameter to wire diameter).

$$K_W = \frac{4r-1}{4r-4} + \frac{0.615}{r} \qquad \textbf{(4-4)}$$

Where: r = Spring index = D_c / D_w

P_L in Equation (4-1) can be substituted into Equation (4-3) for a stress level equation, and the spring failure rate can be determined from a ratio of stress level to the material tensile strength according to the following empirically derived relationship (Reference 14):

$$\lambda_{SP} = \lambda_{SP,B}\left(\frac{S_G}{T_S}\right)^3 = \lambda_{SP,B}\left(\frac{8\,P_L\,D_C\,K_W}{\pi\,T_S\,D_W{}^3}\right)^3 \qquad \textbf{(4-5)}$$

Where: λ_{SP} = Failure rate of spring, failures/million hours

$\lambda_{SP,B}$ = Base failure rate for spring, 23.8 failures/million hours

T_S = Material tensile strength, lbs/in^2

A generalized equation that adjusts the base failure rate of a compression spring considering anticipated operating conditions can be established:

$$\lambda_{SP} = \lambda_{SP,B} \bullet C_G \bullet C_{DW} \bullet C_{DC} \bullet C_N \bullet C_Y \bullet C_L \bullet C_K \bullet C_{CS} \bullet C_R \bullet C_M \qquad \textbf{(4-6)}$$

Where: C_G = Multiplying factor which considers the effect of the material rigidity modulus on the base failure rate (See Table 4-2)

C_{DW} = Multiplying factor which considers the effect of the wire diameter on the base failure rate (See Figure 4.10)

C_{DC} = Multiplying factor which considers the effect of coil diameter on the base failure rate (See Figure 4.11)

C_N = Multiplying factor which considers the effect of the number of active coils on the base failure rate (See Figure 4.12)

C_Y = Multiplying factor which considers the effect of material tensile strength, T_s, on the base failure rate (See Table 4-3)

C_L = Multiplying factor which considers the effect of spring deflection on the base failure rate (See Figure 4.13)

C_K = Multiplying factor which includes the spring concentration factor on the base failure rate (See Figure 4.14)

C_{CS} = Multiplying factor which considers the effect of spring cycle rate on the base failure rate (See Figure 4.20)

C_R = Multiplying factor which considers the effect of a corrosive environment on the base failure rate (See Section 4.3.10)

C_M = Multiplying factor which considers the effect of the manufacturing process on the base failure rate (See Section 4.3.11)

The parameters in the failure rate equation can be located on an engineering drawing by knowledge of design standards or by actual measurements. Other manufacturing, quality, and maintenance contributions to failure rate are included in the base failure rate as determined from field performance data.

4.4.2 Extension Spring

Helical extension springs store energy in spring tensioning devices and are used to exert a pulling force. Most helical extension springs are coiled with initial tension, equal to the minimum force required to separate adjacent coils. Extension springs require a method of attachment to other parts of the assembly. For extension springs, all coils are active and N_a will be equal to the number of coils. Otherwise, the failure rate equations for extension springs are similar to compression springs and the procedures in Section 4.4.1 should be used.

4.4.3 Torsion Spring

Helical torsion springs are used to apply a torque or store rotational energy, the most common application, the clothes pin. Torsion springs are stressed in bending as shown in Figure 4.4. A torsion spring should always be loaded in a direction that causes its body diameter to decrease because of increased stresses when the spring is loaded in a direction which increases body diameter.

The mean diameter of a helical torsion spring is equal to:

$$D_I = \frac{ID + OD}{2} \qquad \textbf{(4-7)}$$

The spring diameter will change with deflection according to the following equation:

$$D_C = \frac{D_I N_a}{N_a + \theta} \qquad \textbf{(4-8)}$$

Where: D_C = Mean diameter after deflection

D_I = Initial mean diameter, in.

θ = Angular deflection from free position, revolutions

N_a = Number of active coils

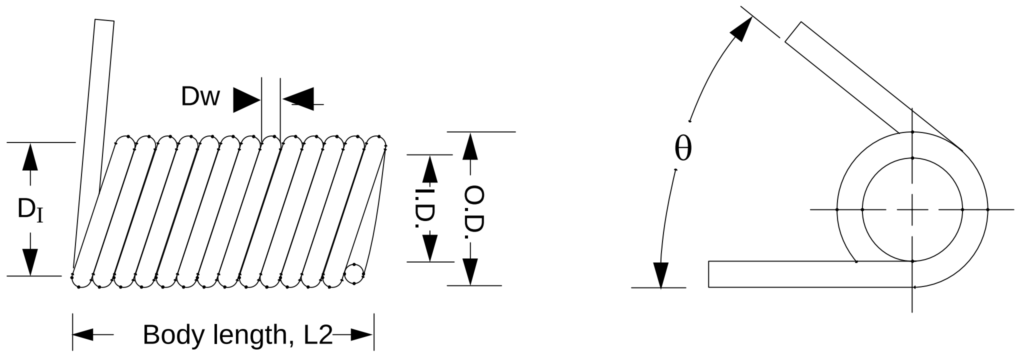


Figure 4.4 Typical Helical Torsion Spring

Most torsion springs are close-wound, with body length equal to wire diameter multiplied by the number of turns plus one. When the spring is deflected in a direction which reduces its coil diameter, body length increases to L_2 according to the following equation:

$$L_2 = D_W \left(N_a + 1 + \theta \right) \qquad \textbf{(4-9)}$$

Where: D_W = Wire diameter, in

Stress in torsion springs is due to bending and for round wire is calculated with the following equation:

$$S = \frac{3\, E_M\, D_W\, \theta}{\pi\, D_I\, N_a} \qquad \textbf{(4-10)}$$

Where: S = Bending stress, lbs/in^2

E_M = Modulus of Elasticity, lbs/in^2

D_W = Wire diameter, in

θ = Angular deflection, revolutions

D_I = Mean diameter of spring, in

N_a = Number of active coils (See Section 4.3.8)

The equation to determine the failure rate of a torsion spring can be written as follows:

$$\lambda_{SP} = \lambda_{SP,B} \left(\frac{S}{T_S} \right)^3 \bullet C_{CS} \bullet C_R \bullet C_M \tag{4-11}$$

From this equation a generalized equation can be developed containing a base failure rate with applicable multiplying factors:

$$\lambda_{SP} = \lambda_{SP,B} \bullet C_E \bullet C_{DW} \bullet C_N \bullet C_Y \bullet C_L \bullet C_{CS} \bullet C_{DC} \bullet C_R \bullet C_M \tag{4-12}$$

Where: λ_{SP} = Failure rate of torsion spring, failures/million hours

$\lambda_{SP,B}$ = Base failure rate for torsion spring, 14.3 failures/million hours

C_E = Multiplying factor which considers the effect of the material elasticity modulus on the base failure rate (See Table 4-2)

C_{DW} = Multiplying factor which considers the effect of the wire diameter on the base failure rate (See Figure 4.10)

C_N = Multiplying factor which considers the effect of the number of active coils on the base failure rate (See Figure 4.12)

C_Y = Multiplying factor which considers the effect of material tensile strength on the base failure rate (See Table 4-3)

C_L = Multiplying factor which considers the effect of spring deflection on the base failure rate (See Figure 4.13)

C_{CS} = Multiplying factor which considers the effect of spring cycle rate on the base failure rate (See Figure 4.20)

C_{DC} = Multiplying factor which considers the effect of coil diameter on the base failure rate (See Figure 4.21)

C_R = Multiplying factor which considers the effect of a corrosive environment on the base failure rate (See Section 4.3.10)

C_M = Multiplying factor which considers the effect of the manufacturing process on the base failure rate (See Section 4.3.11)

4.4.4 Curved Washer

Curved washers are used to secure fasteners, distribute loads, absorb vibrations and axial end play, and other similar applications. A typical curved washer is shown in Figure 4.5. A special type of curved washer, the Belleville washer, is discussed in Section 4.4.6. When a load is applied to a curved washer it tends to flatten causing radial and circumferential strains. This elastic deformation constitutes the spring action. Stress is not distributed uniformly in curved washers, the greatest stress occurring at the convex inner edge. Curved washers exert a relatively light thrust load. Bearing surfaces should be hard to prevent washer corners from scoring the shaft.

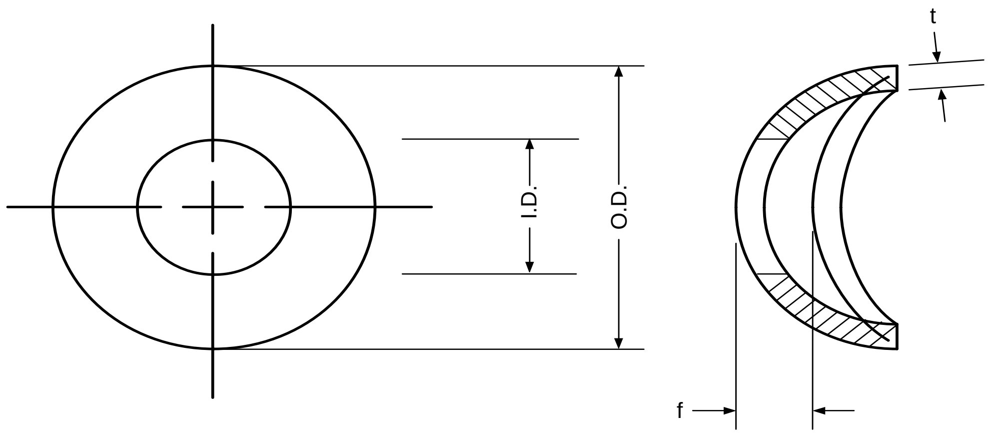


Figure 4.5 Typical Curved Washer

The stress on a curved washer is:

$$S = \frac{6\, E_M\, f\, t}{(OD)^2} \tag{4-13}$$

Where:

S = Bending stress, lb/in^2

E_M = Modulus of Elasticity, lb/in^2

f = Washer deflection, in

t = Washer thickness, in

OD = Outside Diameter, in

The failure rate of a curved washer is determined using the following equation:

$$\lambda_{SP} = \lambda_{SP,B} \left(\frac{S}{T_S} \right)^3 \bullet C_{CS} \bullet C_R \bullet C_M \quad \textbf{(4-11) ref}$$

A generalized equation that adjusts the base failure rate of a curved washer considering anticipated operating conditions can be established:

$$\lambda_{SP} = \lambda_{SP,B} \bullet C_E \bullet C_t \bullet C_D \bullet C_Y \bullet C_f \bullet C_{CS} \bullet C_R \bullet C_M \quad \textbf{(4-14)}$$

Where: λ_{SP} = Failure rate of curved washer, failures/million hours

$\lambda_{SP,B}$ = Base failure rate for curved washer, 1.1 failures/million hours

C_E = Multiplying factor which considers the effect of the material elasticity modulus on the base failure rate (See Table 4-2)

C_t = Multiplying factor which considers the effect of the material thickness on the base failure rate (See Figure 4.16)

C_D = Multiplying factor which considers the effect of washer diameter on the base failure rate (See Figure 4.17)

C_Y = Multiplying factor which considers the effect of material tensile strength on the base failure rate (See Table 4-3)

C_f = Multiplying factor which considers the effect of washer deflection on the base failure rate (See Figure 4.18)

C_{CS} = Multiplying factor which considers the effect of spring cycle rate on the base failure rate (See Figure 4.20)

C_R = Multiplying factor which considers the effect of a corrosive environment on the base failure rate (See Section 4.3.10)

C_M = Multiplying factor which considers the effect of the manufacturing process on the base failure rate (See Section 4.3.11)

4.4.5 Wave Washer

Wave washers are used to apply moderate thrust loads when radial space is limited. A typical wave washer is shown in Figure 4.6.

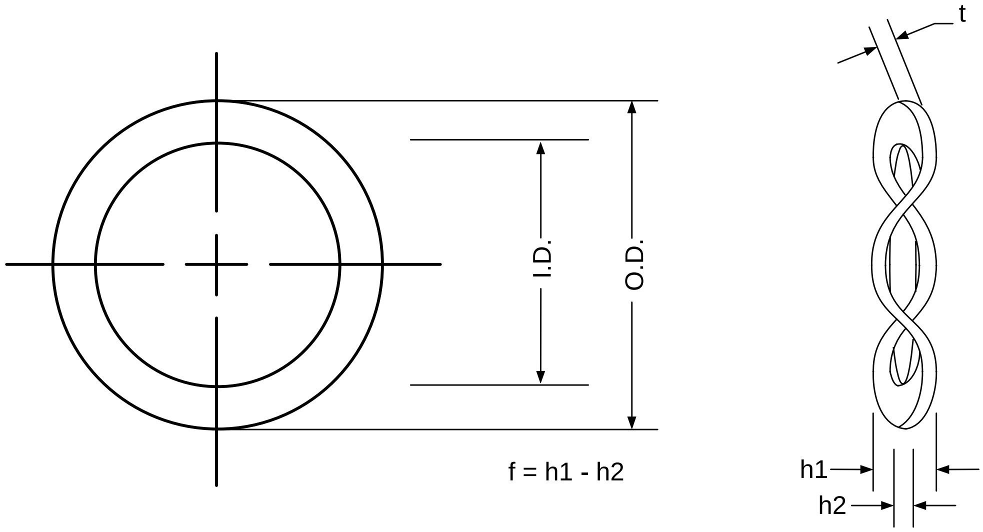


Figure 4.6 Typical Wave Washer

The stress on a wave washer is given by:

$$S = \frac{0.3 \pi E_M f t N^2}{D^2} \qquad \textbf{(4-15)}$$

Where:

Where:	S	=	Bending stress, lbs/in^2
	E_M	=	Modulus of Elasticity, lbs/in^2
	f	=	Deflection, in
	t	=	Material thickness, in
	N	=	Number of waves
	D	=	Mean diameter, in = (OD + ID)/2
	OD	=	Outside Diameter, in
	ID	=	Inside Diameter, in

The failure rate of a wave washer is determined using the following equation:

$$\lambda_{SP} = \lambda_{SP,B} \left(\frac{S}{T_S} \right)^3 \bullet C_{CS} \bullet C_R \bullet C_M \qquad \textbf{(4-11) ref}$$

A generalized equation that adjusts the base failure rate of a wave washer considering anticipated operating conditions can be established:

$$\lambda_{SP} = \lambda_{SP,B} \bullet C_E \bullet C_t \bullet C_D \bullet C_Y \bullet C_f \bullet C_{NW} \bullet C_{CS} \bullet C_R \bullet C_M \qquad \textbf{(4-16)}$$

Where: λ_{SP} = Failure rate of wave washer, failures/million hours

$\lambda_{SP,B}$ = Base failure rate for wave washer, 1.9 failures/million hours

C_E = Multiplying factor which considers the effect of the material elasticity modulus on the base failure rate (See Table 4-2)

C_t = Multiplying factor which considers the effect of the material thickness on the base failure rate (See Figure 4.16)

C_D = Multiplying factor which considers the effect of washer diameter on the base failure rate (See Figure 4.17)

C_Y = Multiplying factor which considers the effect of material tensile strength on the base failure rate (See Table 4-3)

C_f = Multiplying factor which considers the effect of washer deflection on the base failure rate (See Figure 4.18)

C_{NW} = Multiplying Factor which considers the number of waves on the base failure rate (See Table 4-4)

C_{CS} = Multiplying factor which considers the effect of cycle rate on the base failure rate (See Figure 4.20)

C_R = Multiplying factor which considers the effect of a corrosive environment on the base failure rate (See Section 4.3.10)

C_M = Multiplying factor which considers the effect of the manufacturing process on the base failure rate (See Section 4.3.11)

4.4.6 Belleville Washer

When a load is applied to a Belleville washer it tends to flatten causing radial and circumferential strains. This elastic deformation creates the spring action. A typical Belleville washer is shown in Figure 4.7.

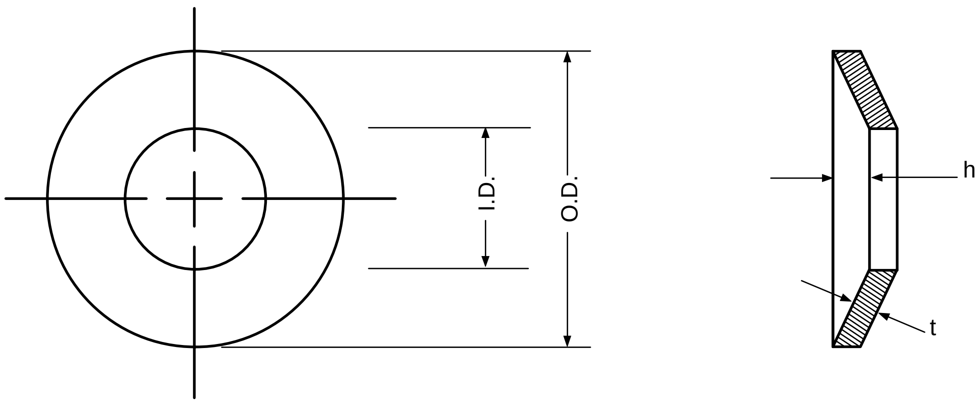


Figure 4.7 Typical Belleville Washer

Belleville washers are capable of providing very high loads at small deflections. Stress is not distributed uniformly in Belleville washers. The highest stress occurs at the top inner edge and can be estimated with the following equation:

$$S = \frac{E_M\, f\, R}{1-\mu^2} \bullet \left(\frac{t}{a^2}\right) \qquad \textbf{(4-17)}$$

Where:

S = Bending stress, lbs/in^2

E_M = Modulus of Elasticity, lbs/in^2

f = Deflection, in

μ = Poisson's Ratio

R = Dimension factor (See Figure 4.17)

t = Material thickness, in

a = O.D./2, in

The failure rate of a Belleville washer is determined using the following equation:

$$\lambda_{SP} = \lambda_{SP,B} \left(\frac{S}{T_S} \right)^3 \bullet C_{CS} \bullet C_R \bullet C_M \qquad \textbf{(4-11) ref}$$

A generalized equation that adjusts the base failure rate of a belleville washer considering anticipated operating conditions can be established:

$$\lambda_{SP} = \lambda_{SP,B} \bullet C_E \bullet C_t \bullet C_D \bullet C_f \bullet C_Y \bullet C_S \bullet C_{CS} \bullet C_R \bullet C_M \qquad \textbf{(4-18)}$$

Where: λ_{SP} = Failure rate of Belleville washer, failures/million hours

$\lambda_{SP,B}$ = Base failure rate for Belleville washer, 2.6 failures/million hours

C_E = Multiplying factor which considers the effect of the material elasticity modulus on the base failure rate (See Table 4-2)

C_t = Multiplying factor which considers the effect of material thickness on the base failure rate (See Figure 4.16)

C_D = Multiplying factor which considers the effect of washer size on the base failure rate (See Figure 4.17)

C_f = Multiplying factor which considers the effect of washer deflection under load on the base failure rate (See Figure 4.18)

C_Y = Multiplying factor which considers the effect of material tensile strength on the base failure rate (See Table 4-3)

C_S = Multiplying factor for compressive stress (See Figure 4.19)

C_{CS} = Multiplying factor which considers the effect of spring cycle rate on the base failure rate (See Figure 4.20)

C_R = Multiplying factor which considers the effect of a corrosive environment on the base failure rate (See Section 4.3.10)

C_M = Multiplying factor which considers the effect of the manufacturing process on the base failure rate (See Section 4.3.11)

4.4.7 <u>Cantilever Spring</u>

Cantilever springs are fabricated from flat strip material which stores and releases energy upon being deflected by an external load. A typical cantilever spring is shown in Figure 4.8. In complex designs, only a small part of the device may be functioning as a spring, and for analytical purposes, that portion which is active during operation may be considered as an independent device.

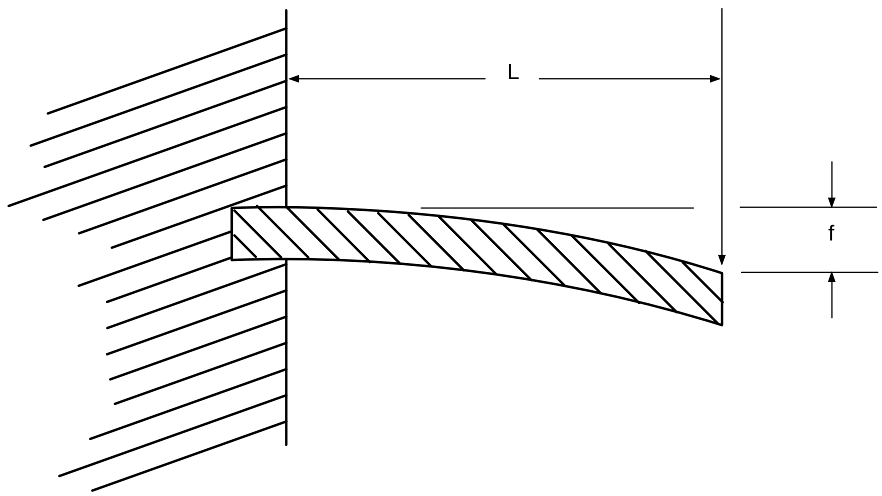


Figure 4.8 Typical Cantilever Spring

The bending stress for cantilever springs can be determined as follows:

$$S = \frac{3\,E_M\,f\,t}{2\,L^2} \qquad \textbf{(4-19)}$$

Where:

S = Bending stress, lbs/in^2

E_M = Modulus of elasticity, lbs/in^2

f = deflection, in

t = thickness, in

L = length, in

The failure rate of a cantilever spring is determined using the following equation:

$$\lambda_{SP} = \lambda_{SP,B} \left(\frac{S}{T_S} \right)^3 \bullet C_{CS} \bullet C_R \bullet C_M \qquad \textbf{(4-11) ref}$$

A generalized equation that adjusts the base failure rate of a cantilever spring considering anticipated operating conditions can be established:

$$\lambda_{SP} = \lambda_{SP,B} \bullet C_E \bullet C_t \bullet C_L \bullet C_f \bullet C_Y \bullet C_{CS} \bullet C_R \bullet C_M \qquad \textbf{(4-20)}$$

Where: λ_{SP} = Failure rate of cantilever spring, failures/million hours

$\lambda_{SP,B}$ = Base failure rate for cantilever spring, 1.1 failures/million hours

C_E = Multiplying factor which considers the effect of the material elasticity modulus on the base failure rate (See Table 4-2)

C_t = Multiplying factor which considers the effect of material thickness on the base failure rate (See Figure 4.16)

C_L = Multiplying factor which considers the effect of spring length on the base failure rate (See Figure 4.22)

C_f = Multiplying factor which considers the effect of spring deflection on the base failure rate (See Figure 4.18)

C_Y = Multiplying factor which considers the effect of material tensile strength on the base failure rate (See Table 4-3)

C_{CS} = Multiplying factor which considers the effect of spring cycle rate on the base failure rate (See Figure 4.20)

C_R = Multiplying factor which considers the effect of a corrosive environment on the base failure rate (See Section 4.3.10)

C_M = Multiplying factor which considers the effect of the manufacturing process on the base failure rate (See Section 4.3.11)

4.4.8 <u>Beam Spring</u>

Beam springs are usually rectangular in shape and formed into an arc as shown in Figure 4.9. Assuming the ends are free to laterally expand, stress can be computed as follows:

$$S = \frac{6\, E_M\, f\, t}{L^2} \qquad \textbf{(4-21)}$$

Where: S = Bending stress, lbs/in^2

E_M = Modulus of elasticity, lb/in^2

f = Spring deflection, in

t = Material thickness, in

L = Active spring length, in

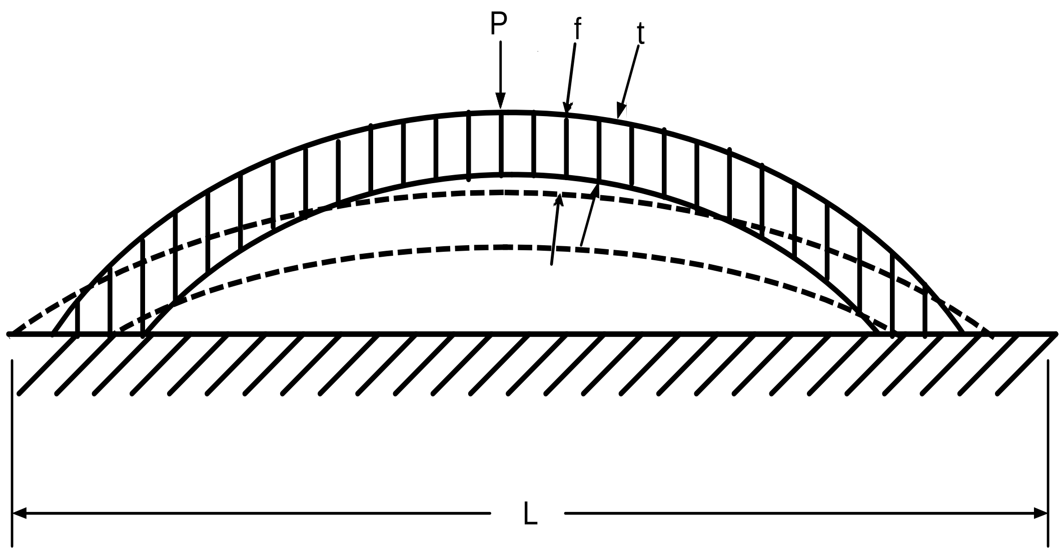


Figure 4.9 Typical Beam Spring

The failure rate of a beam spring is determined using the following equation:

$$\lambda_{SP} = \lambda_{SP,B} \left(\frac{S}{T_S} \right)^3 \bullet C_{CS} \bullet C_R \bullet C_M \qquad \textbf{(4-11) ref}$$

A generalized equation that adjusts the base failure rate of a beam spring considering anticipated operating conditions can be established:

$$\lambda_{SP} = \lambda_{SP,B} \bullet C_E \bullet C_t \bullet C_L \bullet C_f \bullet C_Y \bullet C_{CS} \bullet C_R \bullet C_M \quad \textbf{(4-22)}$$

Where:

λ_{SP} = Failure rate of beam spring, failures/million hours

$\lambda_{SP,B}$ = Base failure rate for beam spring, 4.4 failures/million hours

C_E = Multiplying factor which considers the effect of the material elasticity modulus on the base failure rate (See Table 4-2)

C_t = Multiplying factor which considers the effect of material thickness on the base failure rate (See Figure 4.16)

C_L = Multiplying factor which considers the effect of spring length on the base failure rate (See Figure 4.22)

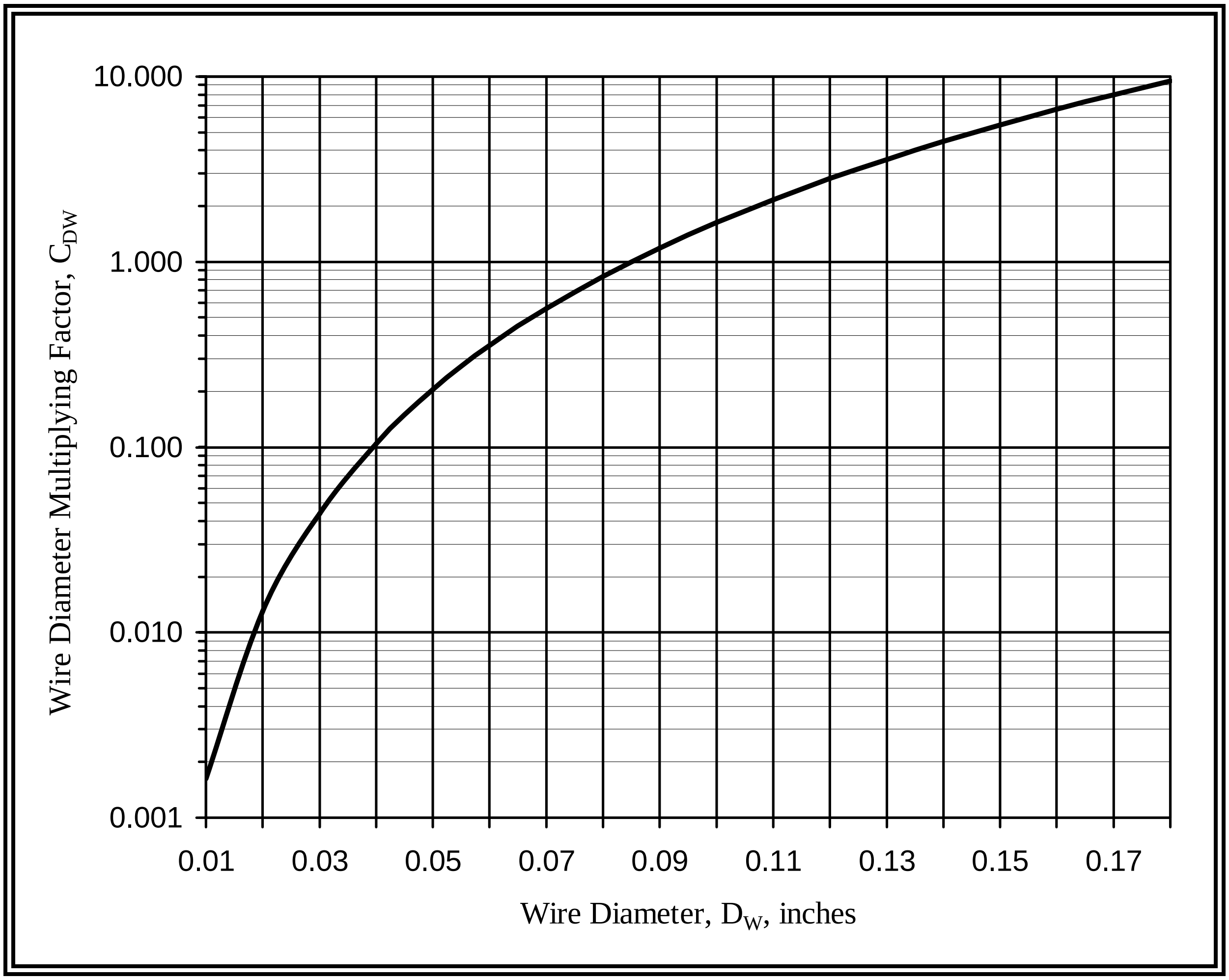
C_f = Multiplying factor which considers the effect of spring deflection on the base failure rate (See Figure 4.18)

C_Y = Multiplying factor which considers the effect of material tensile strength on the base failure rate (See Table 4-3)

C_{CS} = Multiplying factor which considers the effect of spring cycle rate on the base failure rate (See Figure 4.20

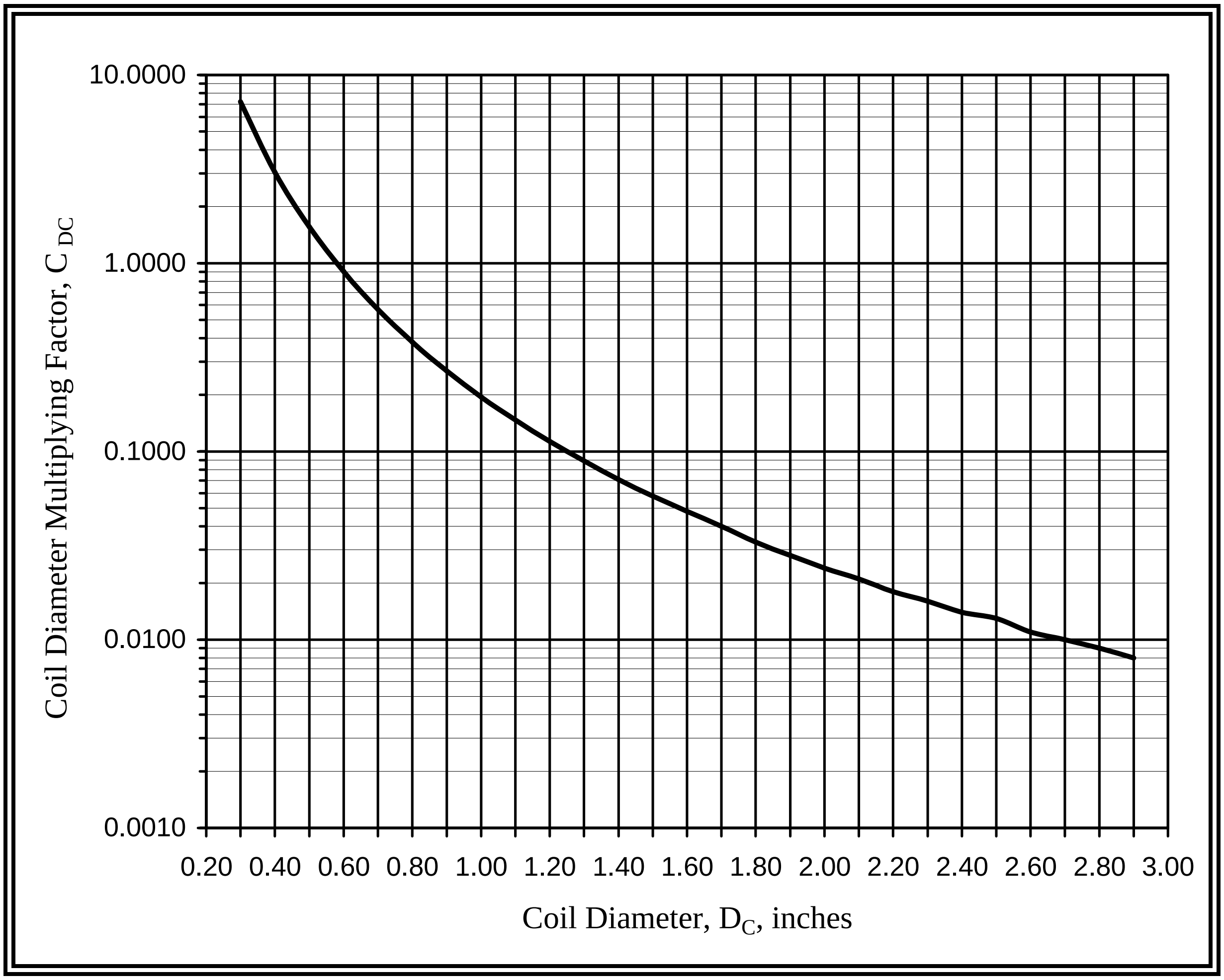
C_R = Multiplying factor which considers the effect of a corrosive environment on the base failure rate (See Section 4.3.10)

C_M = Multiplying factor which considers the effect of the manufacturing process on the base failure rate (See Section 4.3.11)



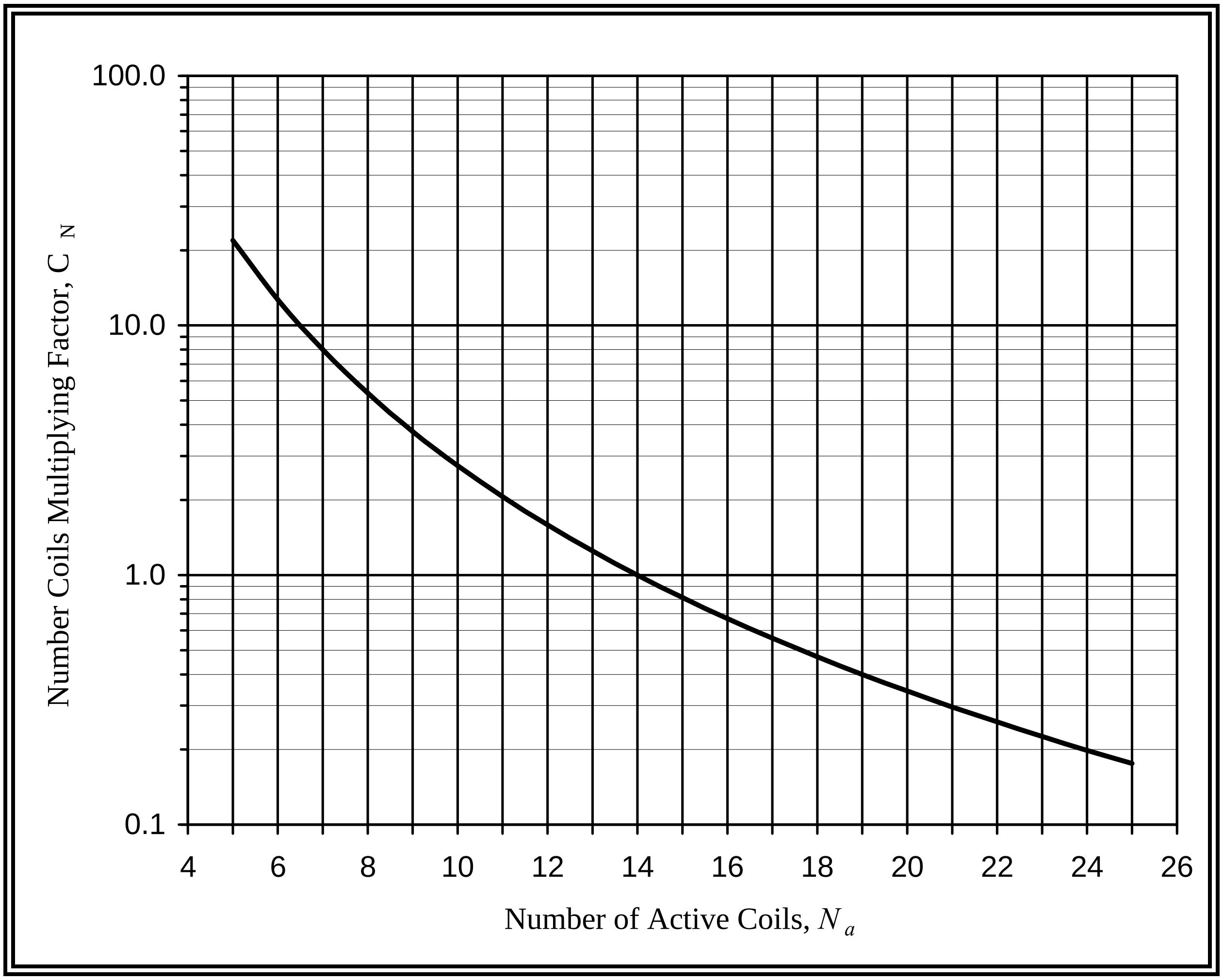
$$C_{DW} = \left(\frac{D_W}{0.085} \right)^3$$

Figure 4.10 Multiplying Factor for Wire Diameter



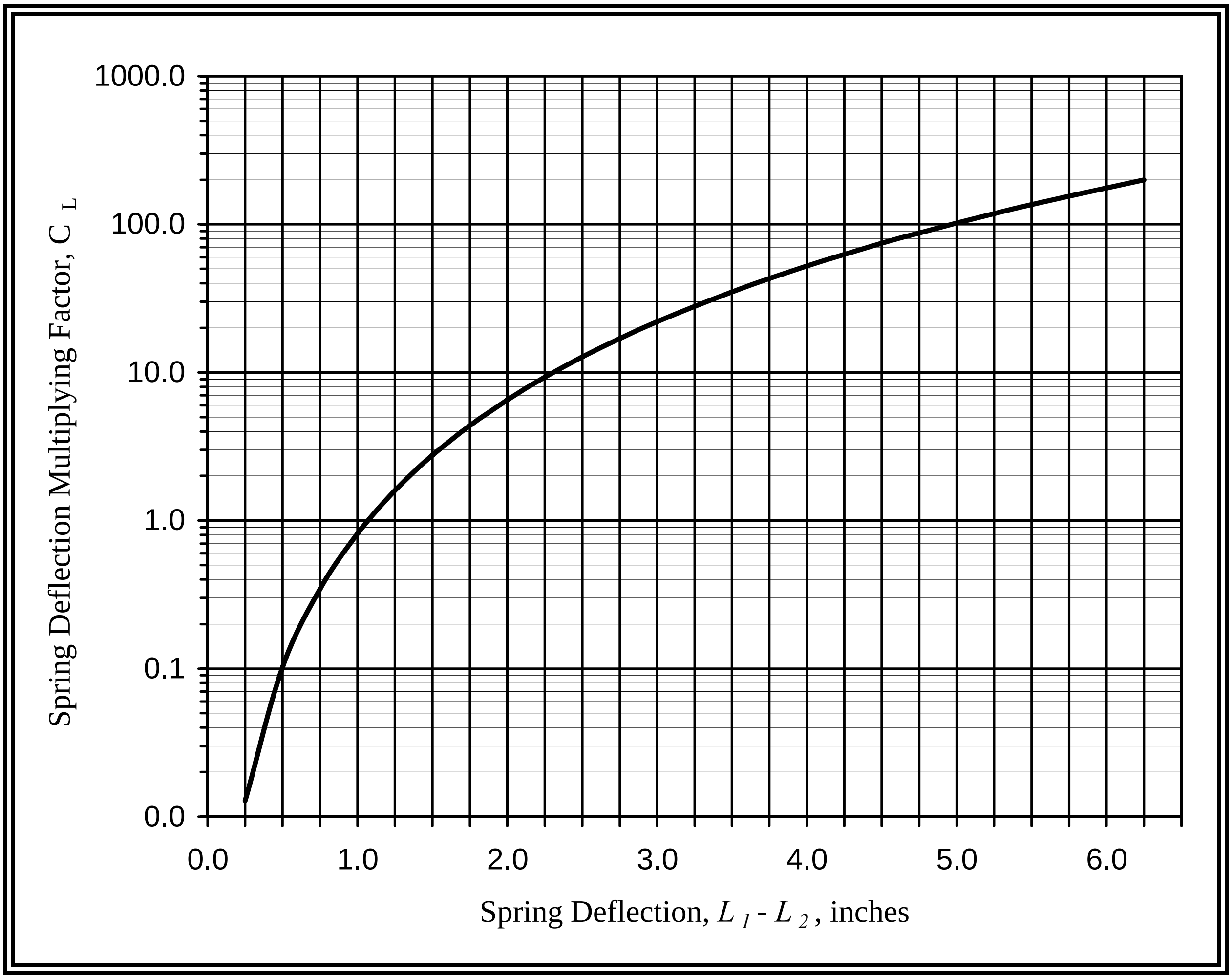
$$C_{DC} = \left(\frac{0.58}{D_C} \right)^6$$

Figure 4.11 Multiplying Factor for Spring Coil Diameter



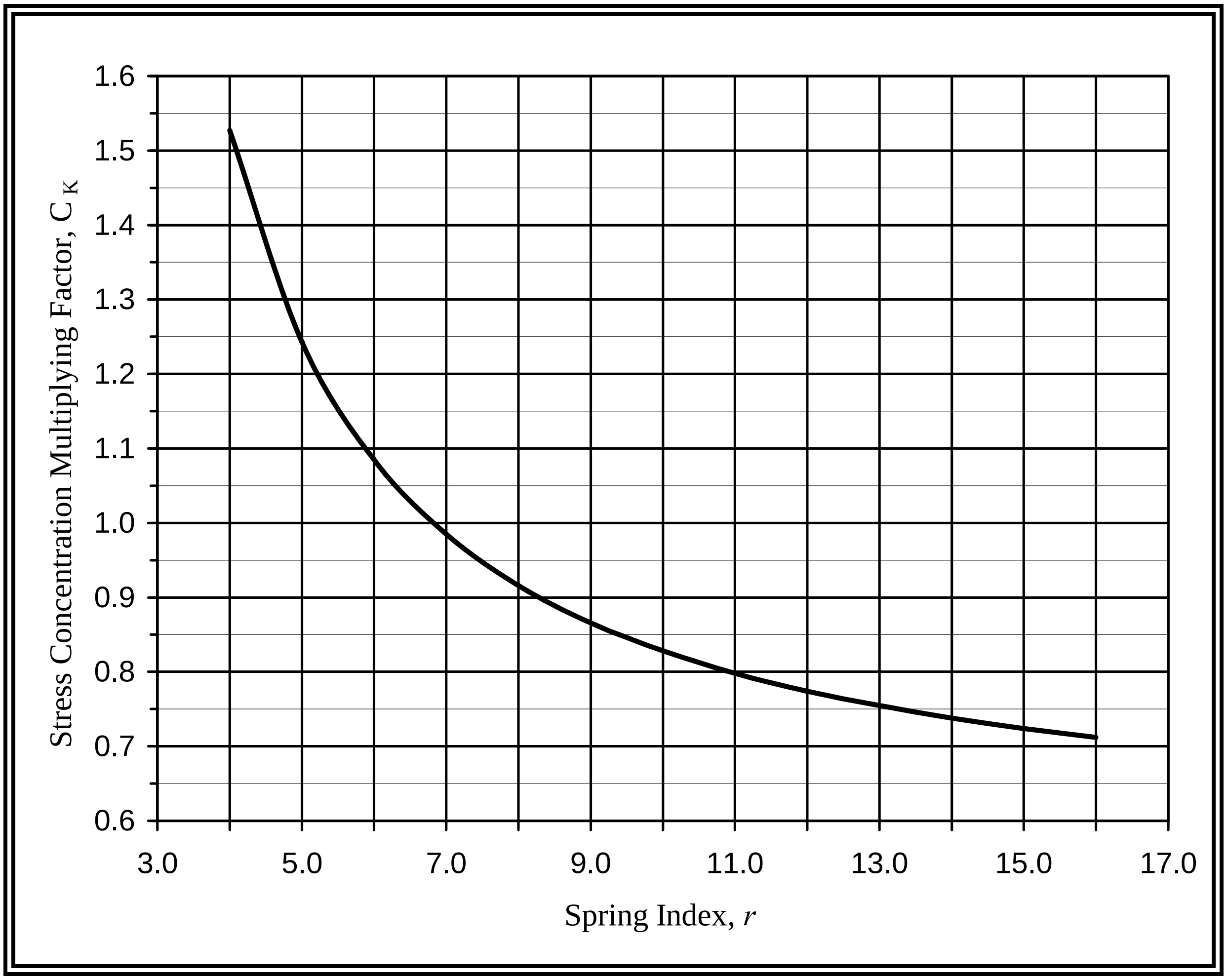
$$C_N = \left(\frac{14}{N_a} \right)^3$$

Figure 4.12 Multiplying Factor for Number of Active Coils in a Spring



$$C_L = \left(\frac{L_1 - L_2}{1.07} \right)^3$$

Figure 4.13 Multiplying Factor for Spring Deflection



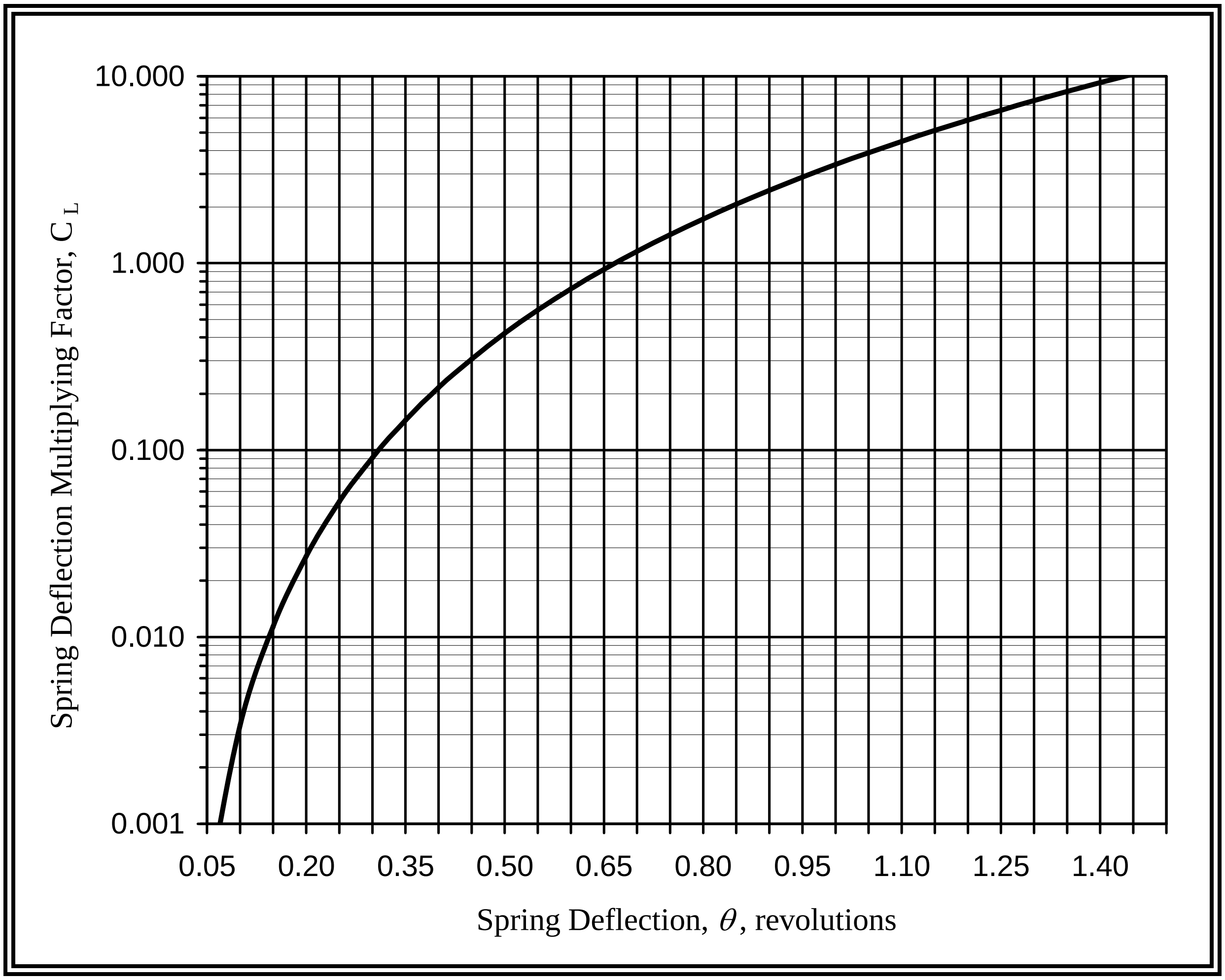
$$C_K = \left(\frac{K_W}{1.219}\right)^3$$

Where: $K_W = \frac{4r-1}{4r-4} + \frac{0.616}{r}$ and $r = \frac{D_C}{D_W}$

D_C = Coil Diameter, inches

D_W = Wire Diameter, inches

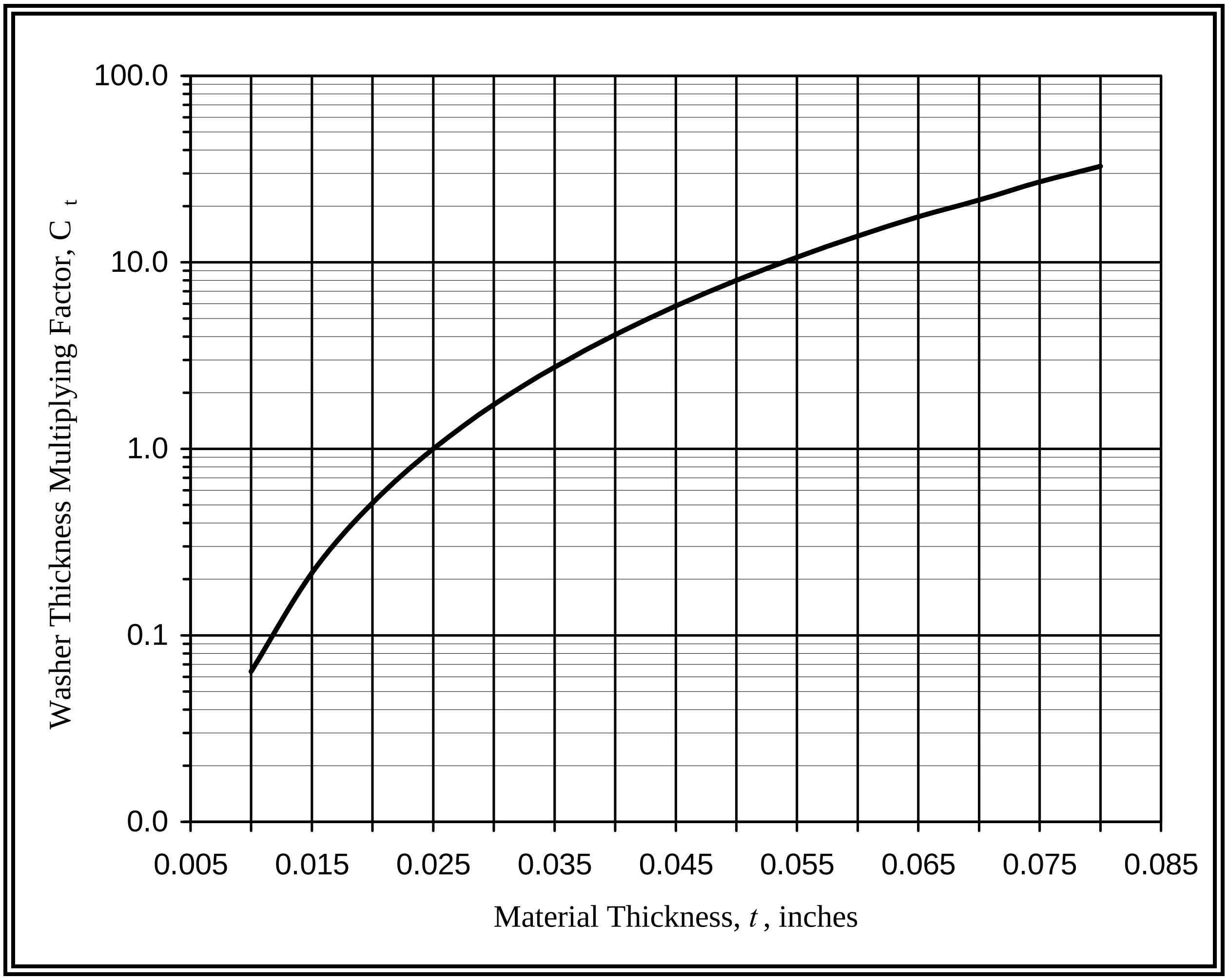
Figure 4.14 Multiplying Factor for Stress Concentration Factor



$$C_L = \left(\frac{\theta}{0.667}\right)^3$$

Where: θ = Angular rotation, revolutions

Figure 4.15 Multiplying Factor for Deflection of a Torsion Spring



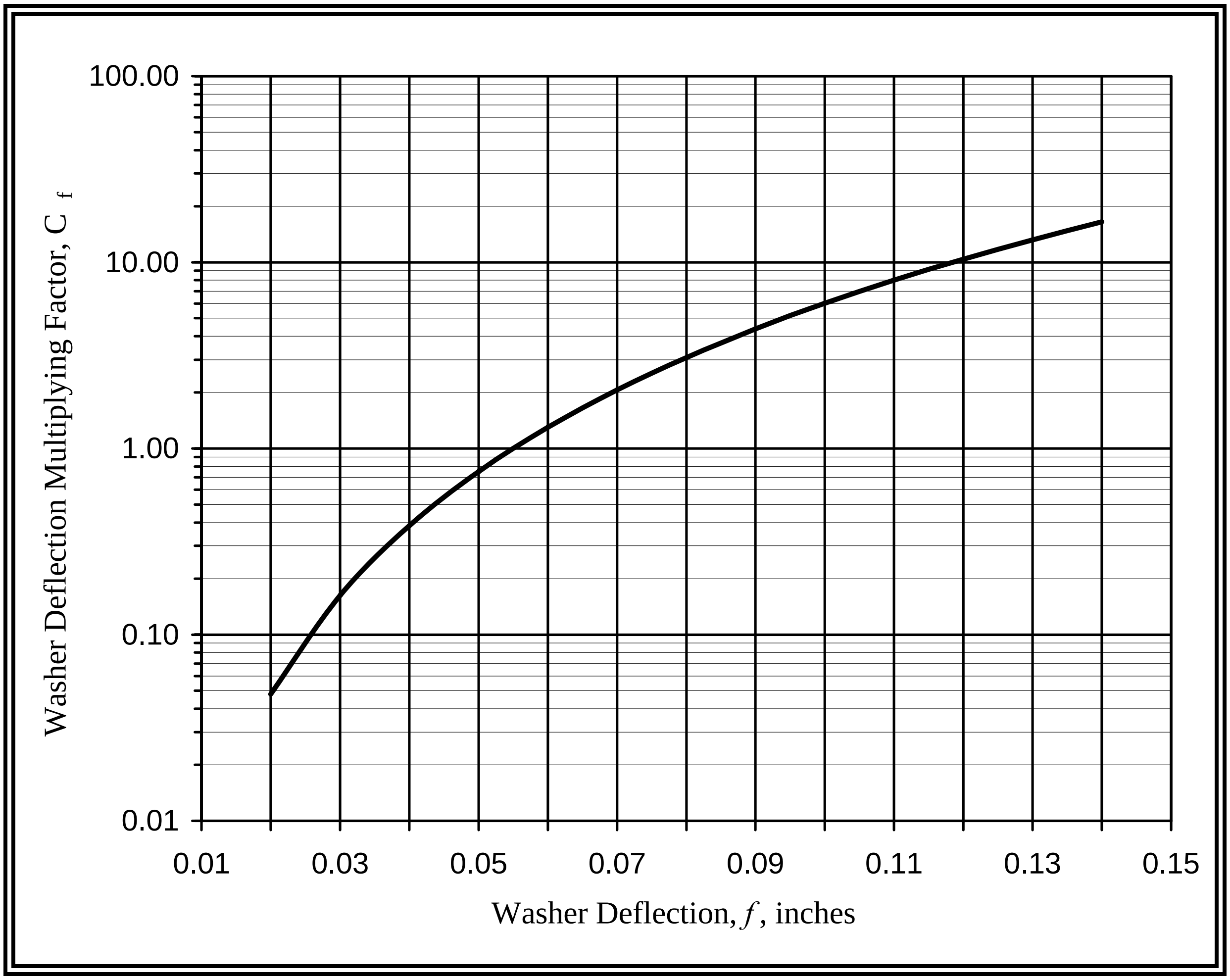
$$C_t = \left(\frac{t}{0.025} \right)^3$$

Figure 4.16 Multiplying Factor for Material Thickness

$$C_D = \left(\frac{1.20}{OD} \right)^6$$

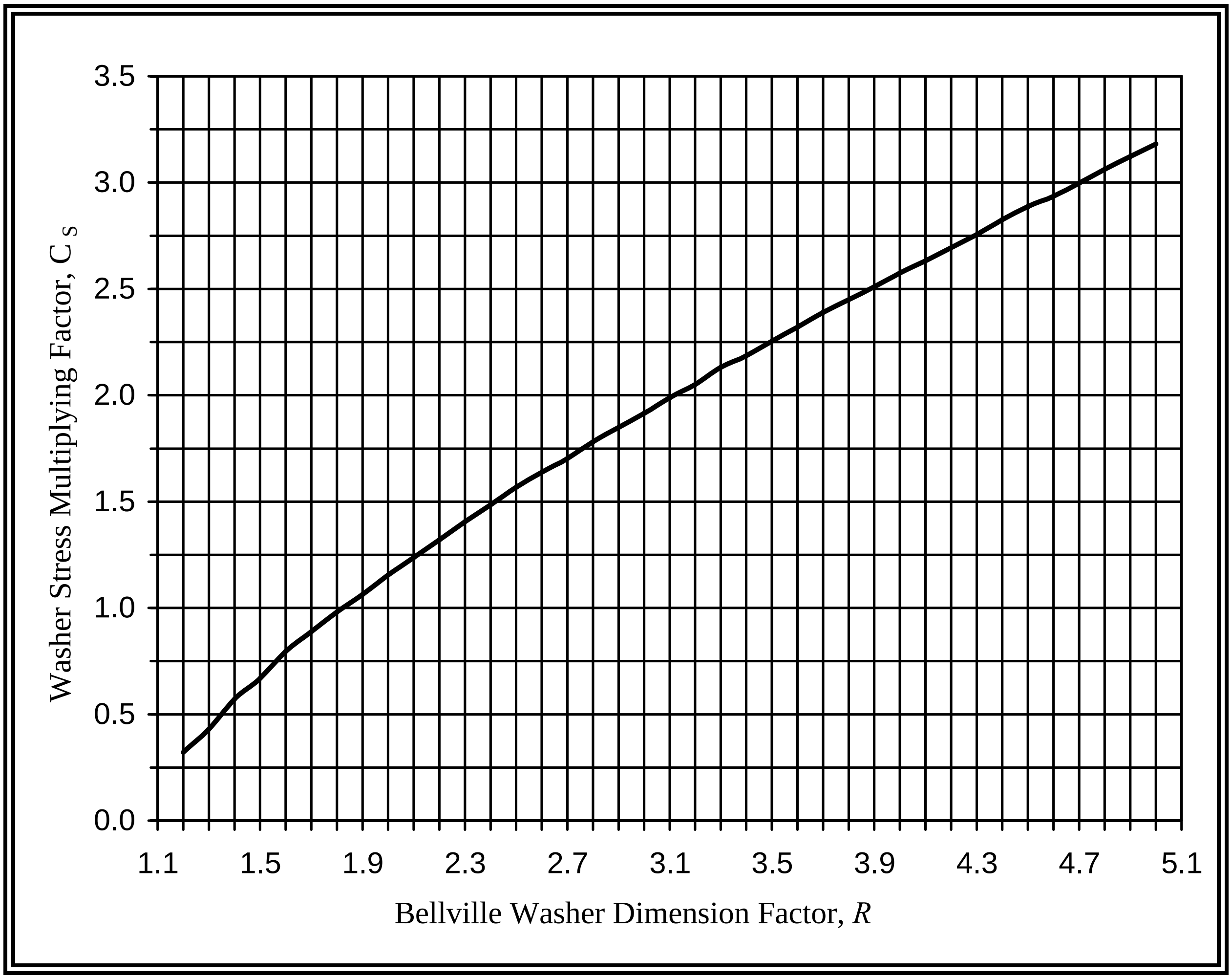
Where: OD = Outside Diameter of Belleville, Curved or Wave Washer, inches

Figure 4.17 Multiplying Factor for Washer Size



$$C_f = \left(\frac{f}{0.055} \right)^3$$

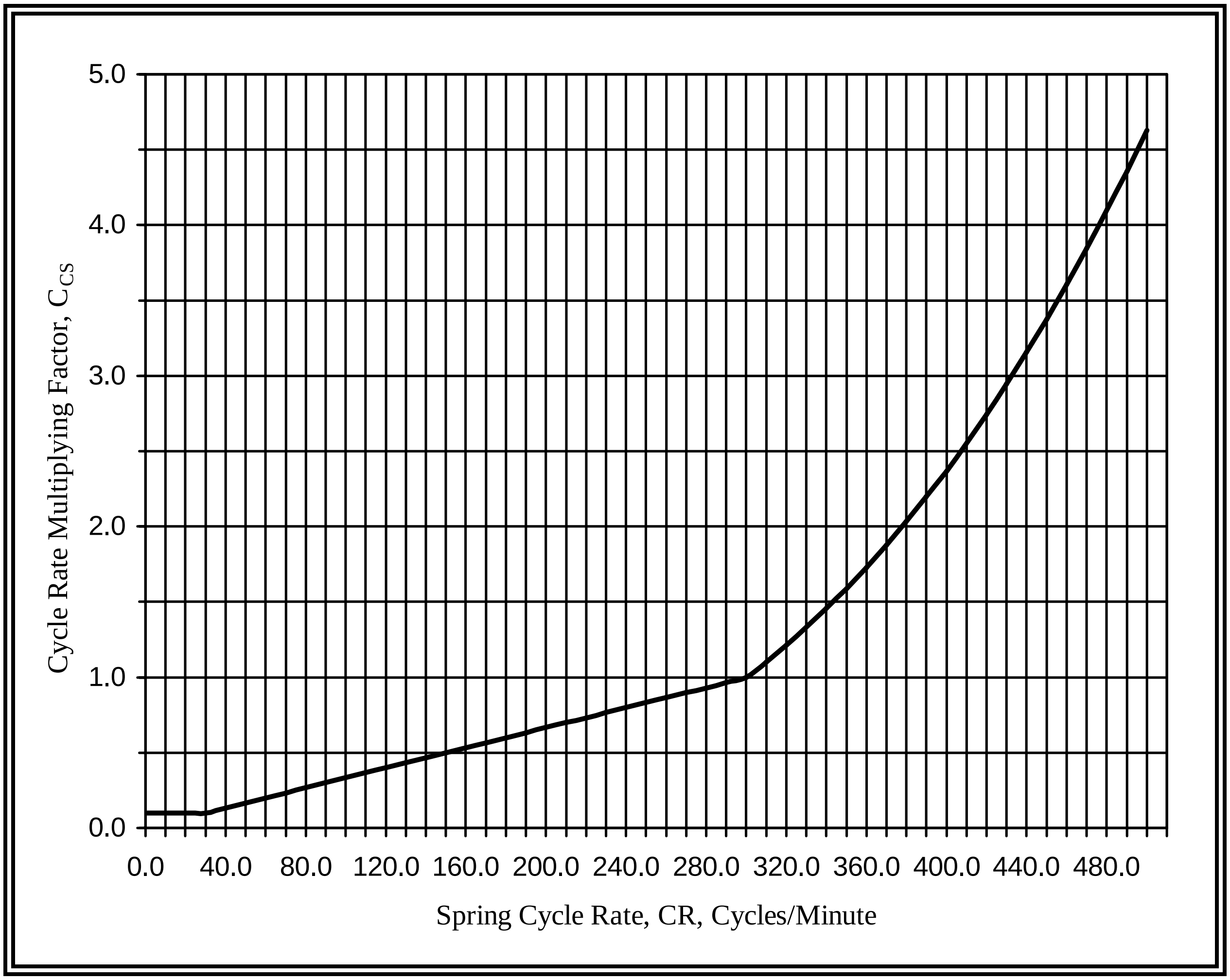
Figure 4.18 Multiplying Factor for Washer or Spring Deflection



$$C_S = \left(\frac{6}{\pi \ln R}\right)^3 \left(\frac{R-1}{\ln R} - 1\right)\left(\frac{R-1}{2}\right)\left(\frac{(R-1)^2}{R^2}\right)$$

Where: $R = \dfrac{outside\ diameter}{inside\ diameter}$

Figure 4.19 Multiplying Factor for Belleville Washer Compressive Stress



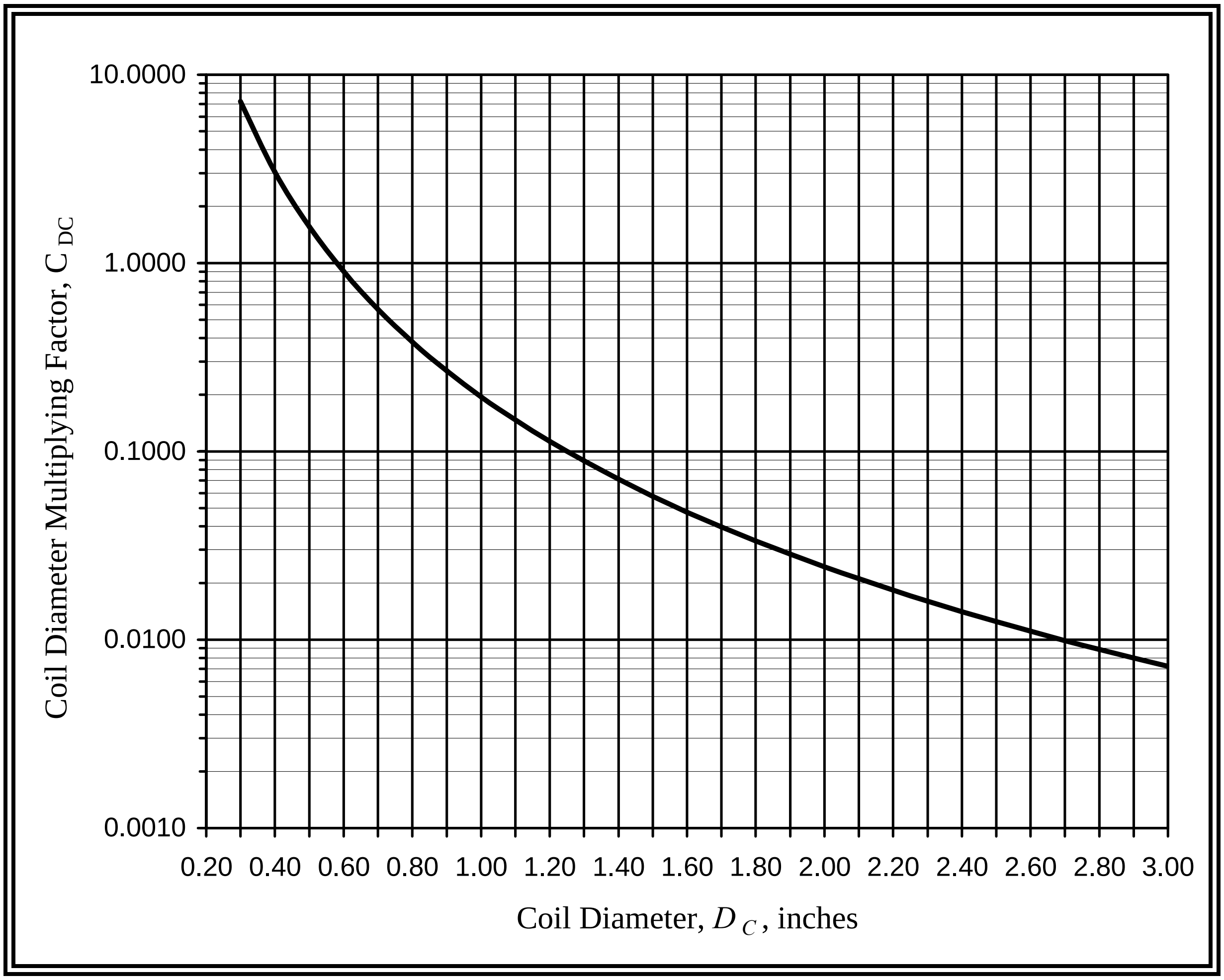
For CR $\leq$ 30 cycles/min, $C_{CS} = 0.100$

For 30 cycles/min < CR $\leq$ 300 cycles min, $C_{CS} = \frac{CR}{300}$

For CR > 300 cycles/min, $C_{CS} = \left(\frac{CR}{300}\right)^3$

Where: *CR* = Spring cycle rate, cycles/min

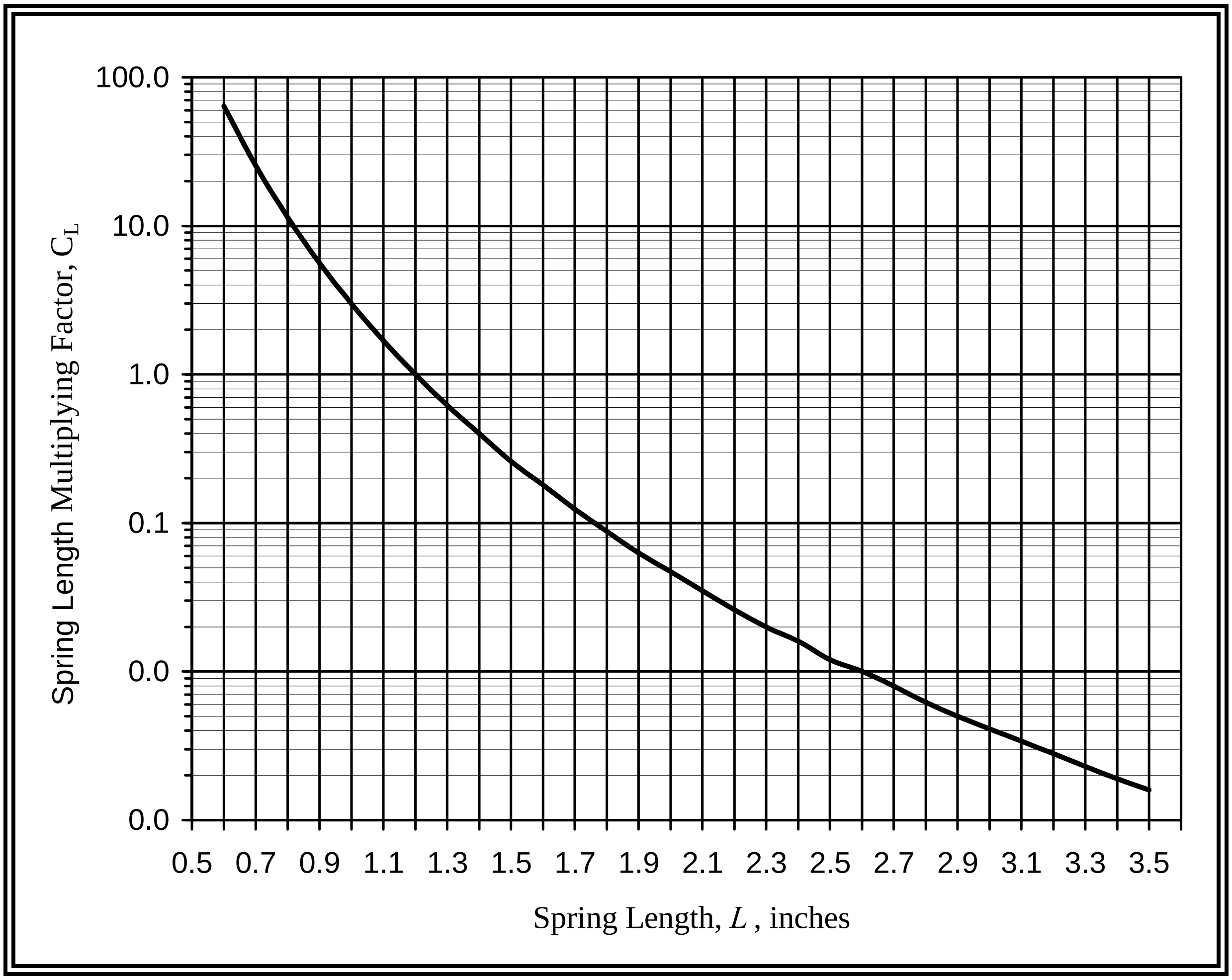
Figure 4.20 Multiplying Factor for Spring Cycle Rate



$$C_{DC} = \left(\frac{0.58}{D_C} \right)^3$$

Where D_c = Coil diameter, inches

Figure 4.21 Multiplying Factor for Spring Coil Diameter (Torsion Springs)



$$C_L = \left(\frac{1.20}{L}\right)^6$$

Figure 4.22 Multiplying Factor for Spring Length

Table 4-2. Moduli of Rigidity and Elasticity for Typical Spring Materials

MATERIAL	MODULUS OF RIGIDITY (G_M) lbs/in^2 x 10^6	C_G	MODULUS OF ELASTICITY (E_M) lbs/in^2 x 10^6	C_E
Ferrous:				
Music Wire	11.8	1.08	29.0	1.05
Hard Drawn Steel	11.5	1.00	28.5	1.00
Chrome Steel	11.2	0.92	29.0	1.05
Silicon-Manganese	10.8	0.83	29.0	1.05
Stainless, 302, 304, 316	10.0	0.67	28.0	0.98
Stainless 17-7 PH	10.5	0.76	29.5	1.04
Stainless 420	11.0	0.88	29.0	1.05
Stainless 431	11.4	0.97	29.5	1.11
Non-Ferrous:				
Spring Brass	5.0	0.08	15.0	0.15
Phosphor Bronze	6.0	0.14	15.0	0.15
Beryllium Copper	7.0	0.23	17.0	0.21
Inconel	10.5	0.76	31.0	1.09
Monel	9.5	0.56	26.0	0.76

NOTE: Modulus G_M is used for compression and extension springs; modulus E_M is used for torsion springs, flat springs and spring washers.

$$C_G = \left(\frac{G_M}{11.5\,x\,10^6} \right)^3 \qquad C_E = \left(\frac{E_M}{28.5\,x\,10^6} \right)^3$$

where: G_M = Modulus of Rigidity (lbs/in^2)

E_M = Modulus of Elasticity (lbs/in^2)

Table 4-3. Material Tensile Strength Multiplying Factor, C_Y

MATERIAL	TENSILE STRENGTH, T_S lbs/in^2 x 10^3	C_Y
Brass	110	5.15
Phosphor Bronze	125	3.51
Monel 400	145	2.25
Inconel 600	158	1.74
Monel K500	175	1.28
Copper-Beryllium	190	1.00
17-7 PH, RH 950	210	0.74
Hard Drawn Steel	216	0.68
Stainless Steel 302, 18-8	227	0.59
Spring Temper Steel	245	0.47
Chrome Silicon	268	0.36
Music Wire	295	0.27

NOTE: These are typical values based on a wire diameter of 0.1 inch. Actual values of tensile strength will vary with wire diameter.

$$C_Y = \left(\frac{190 \, x \, 10^3}{T_S} \right)^3$$

where T_S = Tensile Strength, lbs/in^2

Table 4-4. Wave Washer Multiplying Factor, C_{NW}

NUMBER OF WAVES	C_{NW}
3	2.78
4	1.56
5	1.00
6	0.69
7	0.51
8	0.39

$$C_{NW} = \left(\frac{5}{NW} \right)^2$$

4.5 REFERENCES

In addition to specific references cited throughout Chapter 4, other references included below are recommended in support of performing a reliability analysis of mechanical springs.

12. Carson, Harold, “Springs: Troubleshooting and Failure Analysis”, Marcel Dekker, Inc. New York. (1983)

14. "Engineering Guide to Spring Design" Associated Spring, Barnes Group Inc., Form No. 515 (1981).

19. Hindhede, U., et al, Machine Design Fundamentals, John Wiley & Sons, NY, 1983

35. “Optimum Design of Helical Springs”, Machine Design, (6 November 1980).

58. Parmley, R.O., Mechanical Components Handbook, McGraw-Hill Book Co., NY 1985

82. Metals Handbook, American Society for Metals, 1985, ISBN 0-87170-188-X

115. Partab Jeswani and John Bleda, “A Predictive Process for Spring Failure Rates in Automotive Parts Application”, General Motors Corporation, SAE Technical Paper 910356, February 1991

135. Handbook of Spring Design, Spring Manufacturers Institute, Inc. 2002

This Page Intentionally Left Blank

CHAPTER 5

SOLENOIDS, CONTACTORS

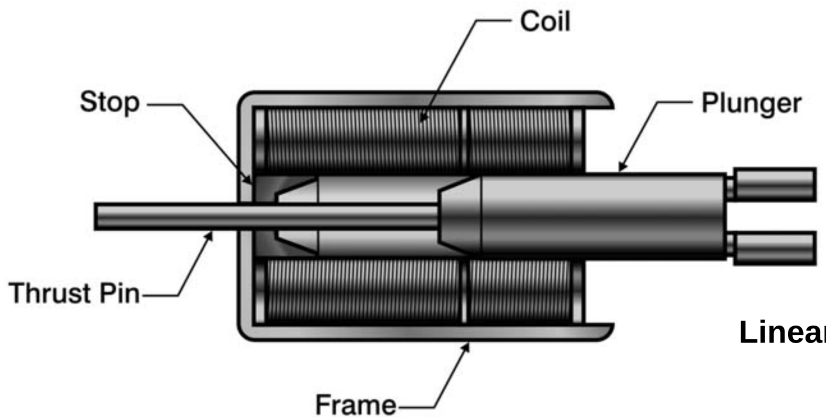
5.0 TABLE OF CONTENTS

5.1 INTRODUCTION

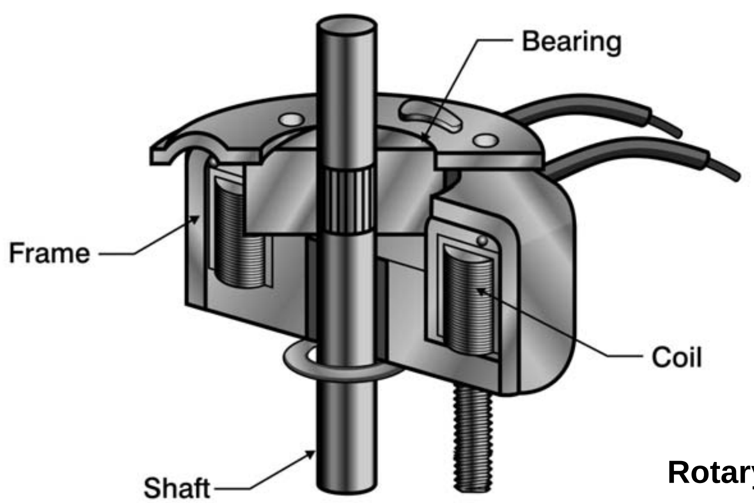
This chapter includes the procedures for estimating the reliability of solenoids and contactors in their intended operating environments. Solenoids are used to control the motion of other mechanical components such as valves and actuators. The most accurate method for predicting the failure rate of these assemblies is to sum the failure rates for the individual parts that make up the assembly. Procedures for estimating the failure rate of valves and actuators and the parts comprising these assemblies such as springs and bearings are contained in various chapters of the Handbook. In the event the solenoid is part of a relay, Section 5.4 of this chapter contains the procedures for estimating the failure rate of the contactor assembly.

Solenoids are electromechanical devices that convert electrical energy into mechanical motion. Generally this motion is used to move a load a specified distance or rotational angle within a specified time. Linear magnetic solenoids usually produce motion by pulling a plunger into the coil when energized. They can also be equipped with a push rod mounted to one end of the plunger providing a pushing motion when energized. The plunger of the solenoid assembly, also known as the armature, is made of ferrous material to increase magnetism or permeability. Rotary solenoids convert axial motion into a rotary stroke.

Component parts of a solenoid include a coil to carry current and generate ampere turns, an iron shell to provide a magnetic circuit and a movable plunger to act as the working element. Component parts of example linear and rotary solenoids are shown in Figure 5.1. Electrical current is supplied to the solenoid coil that is wound tightly enough to limit the current drain and sized large enough to provide for adequate heat dissipation. The resulting magnetic field draws the plunger from its unpowered, extended position to a seated position against a backstop or pole piece. Because the linear force on the plunger is nonlinear with position, the force is relatively high immediately adjacent to the seated position and declines rapidly with increased distance from the seated position. Return motion of the solenoid upon deenergizing the coil is provided by the load itself or a return spring. Rotary solenoids normally consist of a plunger and small ball bearings that ride on an inclined plane.



Linear Solenoid



Rotary Solenoid

Figure 5.1 Component Parts of a Solenoid

The stroke in linear solenoids or rotation in rotary solenoids is the total plunger travel or rotary shaft motion when electrical power is applied to the solenoid. The force in linear solenoids or torque in rotary solenoids is the load that the solenoid is capable of pulling, pushing, holding or rotating at the start of a specified motion. This load capability will depend on applied voltage, ambient temperature and the solenoid operating duty cycle. The solenoid force or torque will decrease as the solenoid coil temperature increases depending on coil wire resistance. Continuous duty solenoids are considerably larger than intermittent duty devices and provide more pull-and-hold strength. Solenoids that are capable of both intermittent and continuous duty will exhibit force ratings that vary widely with duty cycle.

The solenoid may be used to control the motion of various components such as a valve or contactor. Clearance between the coil assembly and the armature assembly must be maintained for proper operation and the design should be evaluated for reliability considering the operating environment. For small solenoids, the armature has minimal effect on the part failure rate. For large solenoid operated assemblies, the procedures in Chapter 20 can be used to evaluate the wear rate of the solenoid plunger. And, if the solenoid is used to control an actuator, the procedures of Chapter 9 can be used to determine the failure rate of the actuator. Chapter 6 contains procedures for evaluating a valve if the solenoid is part of a valve assembly. If a return spring is being used in the solenoid, the procedures in Chapter 4 can be used to determine the reliability of the spring in its operating environment. Chapter 7 contains procedures for evaluating the rotating bearing if a rotary solenoid is being evaluated. Section 5.4 contains the procedures for evaluating the contactor if the solenoid is part of a relay or contactor configuration.

The reliability of a typical solenoid assembly depends on the construction of the coil assembly, length of the stroke, and the environment in which it operates. The failure rate of the solenoid assembly is dependent upon manufacturing quality associated with the assembly of the coil in relation to the armature as well as upon operating environment including shock, vibration and corrosive environment. The failure rate of the solenoid assembly can be determined using the procedures in Section 5.3. The maximum specified cycling rate of the solenoid should be determined and compared with the potential operating rate to make sure the coil will not be overheated in its operating environment. When a solenoid is energized by the voltage source, heat is generated which increases the temperature of the coil. This temperature rise may have some undesired effects, since resistance of the coil winding increases with temperature, which in turn, reduces electrical current. This reduction in current reduces the force output. An extreme increase in temperature can result in damage to the winding. Usually the limiting factor for operating temperature is the rated temperature of the solenoid insulating material (see Section 5.3.1).

5.2 FAILURE MODES

The primary failure modes of a solenoid inductor (coil) assembly include one or more winding shorts or an open coil usually caused by overheating. Heat in a solenoid is a function of power and the time during which voltage is applied. Heat can be controlled by providing air flow, by mounting the solenoid on a surface (heat sink) large enough to dissipate the heat, or introduction of alternate cooling methods. Table 5-1 provides some typical failure modes of the solenoid assembly and if applicable, a contactor. Solenoids are enclosed within a steel housing to form part of the magnetic circuit and to provide structural integrity. It is important to review the mounting of the solenoid assembly to assure an adequate heat sink for high temperature operations. Usually a mounting surface area of ten times the area of the solenoid should be maintained.

Table 5-1. Typical Failure Modes of a Solenoid and Contactor

FAILURE MODE	FAILURE MECHANISM	FAILURE CAUSE
Coil burnout	Inrush current causes coil overheating and burnout	Mechanical jamming of plunger
		Insufficient heat sink area for solenoid
		Supply voltage interruption resulting in inductive surge
	Heat builds up faster than it can be dissipated	Excessive cycling rate
Failure to operate	Increase in coil resistance preventing solenoid closure	Excessive ambient temperature
	Shorted coil at lead wires	Excessive moisture
Open inductor winding	Open lead at termination	Coil voltage overload, vibration
Armature (plunger) failure	Mismatch of solenoid force and load	Excessive plunger force creating hammering
Poor response time (pull-in time)	Insufficient solenoid force with respect to load	Jammed return spring
Poor release time (drop-out time)	Insufficient load or spring force to release plunger	Damaged / jammed spring or loss of load force
Damaged contactor	Contactor arcing	Excessive load voltage

Solenoids have a maximum ON time for a given duty cycle, wattage and power input so it is important to consider the duty cycle and wattage when the solenoid is pulsed repeatedly. If the ON time of the solenoid is more than 5 minutes during a single pulse, it should be considered a continuous duty (100% duty cycle) solenoid. It is important that the solenoid reaches a sufficient cool-down time before being reenergized.

5.3 FAILURE RATE OF SOLENOID ASSEMBLY

The failure rate of the solenoid can be estimated from the following equation:

$$\lambda_{SO} = \lambda_{SO,B} \bullet C_T \bullet C_K \bullet C_S \qquad \textbf{(5-1)}$$

Where:

λ_{SO} = Failure rate of a solenoid in failures/million hours

$\lambda_{SO,B}$ = Base failure rate of solenoid, 2.77 failures/million operations

C_T = Temperature multiplying factor, See Section 5.3.1 and Figure 5.2

C_K = Application service factor, See Section 5.3.2 and Table 5-2

C_S = Use rate (operations/min x 60 min/hour)

Since an inductor consists of a number of turns of wire it will have some small amount of direct current resistance. This copper loss of the inductor can be calculated by multiplying the square of the current in the inductor by the resistance of the winding (I^2R). In addition to copper loss, an iron core coil will have hysteresis and eddy-current losses. Hysteresis loss is due to power that is consumed in reversing the magnetic field of the inductor core each time the direction of current in the inductor changes. Eddy-current loss is due to heating of the core by circulating currents that are induced in the core by the magnetic field around the turns of the coil. All these losses dissipate power in the form of heat. Inductor manufactures publish these power losses in their product specification sheets.

Manufacturers also rate their inductors in terms of insulation rating. Common ratings include:

Class A rated 105 ^oC

Class B rated 130 ^oC

Class F rated 155 ^oC

Class H rated 180 ^oC

Class C rated 220 ^oC

The rating of a solenoid by the manufacturer therefore permits the associated temperature rating to be used in Equation (5-2) in the following section and in Figure 5.2.

5.3.1 Temperature Multiplying Factor, C_T

For a specific solenoid stroke or rotation, the solenoid force or torque will decrease as the solenoid coil temperature increases. A solenoid coil will tend to achieve a stabilized coil temperature when operated at the manufacturer's rated power and duty cycle. When operated under standard operating conditions, the coil insulation will not be exceeded and the base failure rate can be assumed. It is important to determine the power dissipation and surface area of the solenoid to assure that the insulation temperature rating is not exceeded. Heat in a solenoid is a function of power and the time during which power is applied. Heat can be dissipated by controlling the air flow, by mounting the solenoid on a surface (heat sink) large enough to dissipate the energy or other cooling method.

Wear of moving parts and solenoid electrical contacts can change mechanical tolerances, resulting in increased electrical current through the solenoid coil when the solenoid is energized and possible premature failure. Energizing the solenoid at nominal design voltage does not normally impart any significant stress on the solenoid. However, after degradation from electrical stress, energizing the solenoid at nominal design voltages may result in solenoid failure. A temperature rise of the solenoid coil must be carefully considered during the analysis of design reliability. Elevated ambient temperatures must be projected such as the air flow design in enclosed relay cabinets. Duty cycle and maximum ON time have a significant effect on coil temperature. Coil temperature in turn affects the solenoid force or torque capability and its failure rate. For a specific solenoid stroke or rotation, the solenoid force or torque will decrease as the solenoid coil temperature increases.

The following equation can be used to estimate the temperature multiplying factor:

$$C_T = \left(\frac{1}{1.5^{t}}\right)^3 \qquad \textbf{(5-2)}$$

where: $$t = \frac{(T_R - T_O) - 20}{10}$$

and: T_R = Rated temperature of the solenoid, ^oC

T_O = Operating temperature of the solenoid, ^oC

The effect of temperature on the temperature multiplying factor is shown in Figure 5.2. As current flows through the coil, the wire becomes warmer and its resistance increases. The gauge of the wire in the coil determines how much self-heating occurs. The thermal mass of the solenoid helps to dissipate the heat. The following equation predicts the coil's self-heating:

$$T = T_A + 10^{(33\,S\,(I/A)^2 - 1} \bullet (234 + T_A) \qquad \textbf{(5-3)}$$

Where T = Temperature of the wire after S seconds, ^oC

T_A = Ambient temperature, ^oC

S = Application time, seconds

A = Conductor area, mils ^{2}

I = Application current, amps

When a solenoid is actuated, current is applied to the coil resulting in a temperature rise of the coil. When a solenoid is energized continuously, heating of the coil increases until a saturation level is reached which is equal to the ambient heat radiation. When a solenoid is operated in an intermittent (on-off) manner, the on-time becomes critical if higher voltages (thus currents) are applied to the coil. These higher currents cause higher coil temperatures which can exceed the cooling capacity of the solenoid through ambient radiation. This heat rise can thermally destroy the solenoid's coil. To avoid such a catastrophic failure, there is a maximum "on" time which is the longest time a solenoid can be energized without thermal damage.

$$Duty\,Cycle\,(\%) = \frac{ON\,Time}{On\,Time + Off\ Time} \bullet 100 \qquad \textbf{(5-4)}$$

Solenoids have a maximum ON time for a given duty cycle, wattage and power input. For example, if a solenoid is energized for one second out of four (25% duty cycle), its ON time of one second will cause no damage. However, if the solenoid is energized for 10 minutes out of every 40 minutes (25% duty cycle), the coil could be damaged if not rated for continuous duty.

5.3.2 Application Service Factor, C_K

The projected failure rate of a solenoid is dependent on the expected operating environment of the equipment containing the solenoid. It is also dependent on the length of stroke or degree of rotation and the side loading or torque. Table 5-2 provides multiplying factors for solenoid application in the intended operating environment.

Table 5-2. Solenoid Application Service Factor, C_K

Operating Environment	Normal load or torque, minimum stroke or rotation	Normal load or torque, extended stroke or rotation	High load or torque, minimum stroke or rotation	High load or torque, extended stroke or rotation
Uniform	1.1	1.3	1.2	1.4
Light shock	1.2	1.4	1.3	1.5
Medium shock	1.3	1.5	1.4	1.6
Heavy shock	1.4	1.6	1.5	1.7

5.4 FAILURE RATE OF CONTACTOR ASSEMBLY

Contactor life depends on their structure and the chemical and electrical operating environment. Failure of an electrical contact can be assumed when the contact resistance increases to approximately twice the initial value. The failure rate of the contactor can be written as (Reference 28):

$$\lambda_C = \lambda_{C,B} \bullet V^m \bullet I^n \qquad \textbf{(5-5)}$$

Where: λ_C = Failure rate of contactor assembly, failures/million operations

$\lambda_{C,B}$ = Base failure rate of contactor assembly, failures/million operations

V = Voltage across contactor assembly, volts

I = Current, amperes

m = Voltage constant

n = Current constant

A more general equation can be written for AC resistive loads (Reference 28):

$$\lambda_C = \lambda_{C,B} \bullet C_V \bullet C_I \qquad \textbf{(5-6)}$$

Where: λ_C = Failure rate of contactor assembly, failures/million operations

$\lambda_{C,B}$ = Base Failure of contactor assembly, resistive load, 1.10 failures/million operations

C_V = Multiplying factor considering contactor voltage (See Figure 5.3)

C_I = Multiplying factor considering contactor current (See Figure 5.4)

For AC inductive loads, the power factor must be considered, modifying Equation (5-7) as follows (Reference 28):

$$\lambda_C = \lambda_{C,B} \bullet C_V \bullet C_I \bullet C_{PF} \qquad \textbf{(5-7)}$$

Where: λ_C = Failure rate of contactor assembly, failures/million operations

$\lambda_{C,B}$ = Base failure rate of contactor assembly, inductive load, 3.60 failures/million operations

C_V = Multiplying factor considering contactor voltage (See Figure 5.3)

C_I = Multiplying factor considering contactor current (See Figure 5.4)

C_{PF} = Multiplying factor considering the power factor (See Figure 5.5)

DC loads generate greater arcing across the contacts than do AC loads. The failure rate equation for a contactor with DC loads is written as follows (Reference 28):

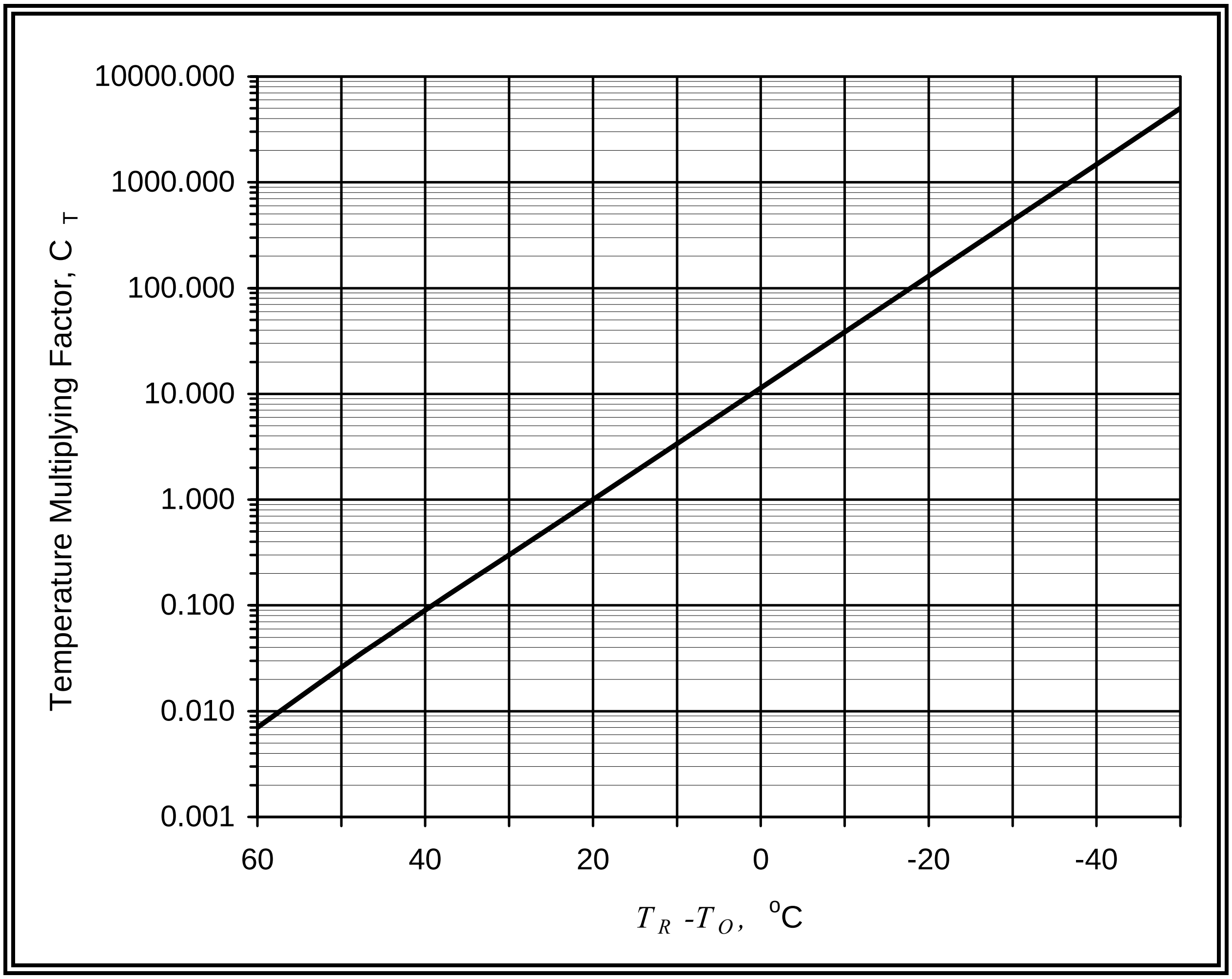
$$\lambda_C = \lambda_{C,B} \bullet C_V \bullet C_I \qquad \textbf{(5-8)}$$

Where: λ_C = Failure rate of contactor assembly, failures/million operations

$\lambda_{C,B}$ = Base Failure of contactor assembly, DC load, 2.5 failures/million operations

C_V = Multiplying factor considering contactor voltage (See Figure 5.6)

C_I = Multiplying factor considering contactor current (See Figure 5.7)



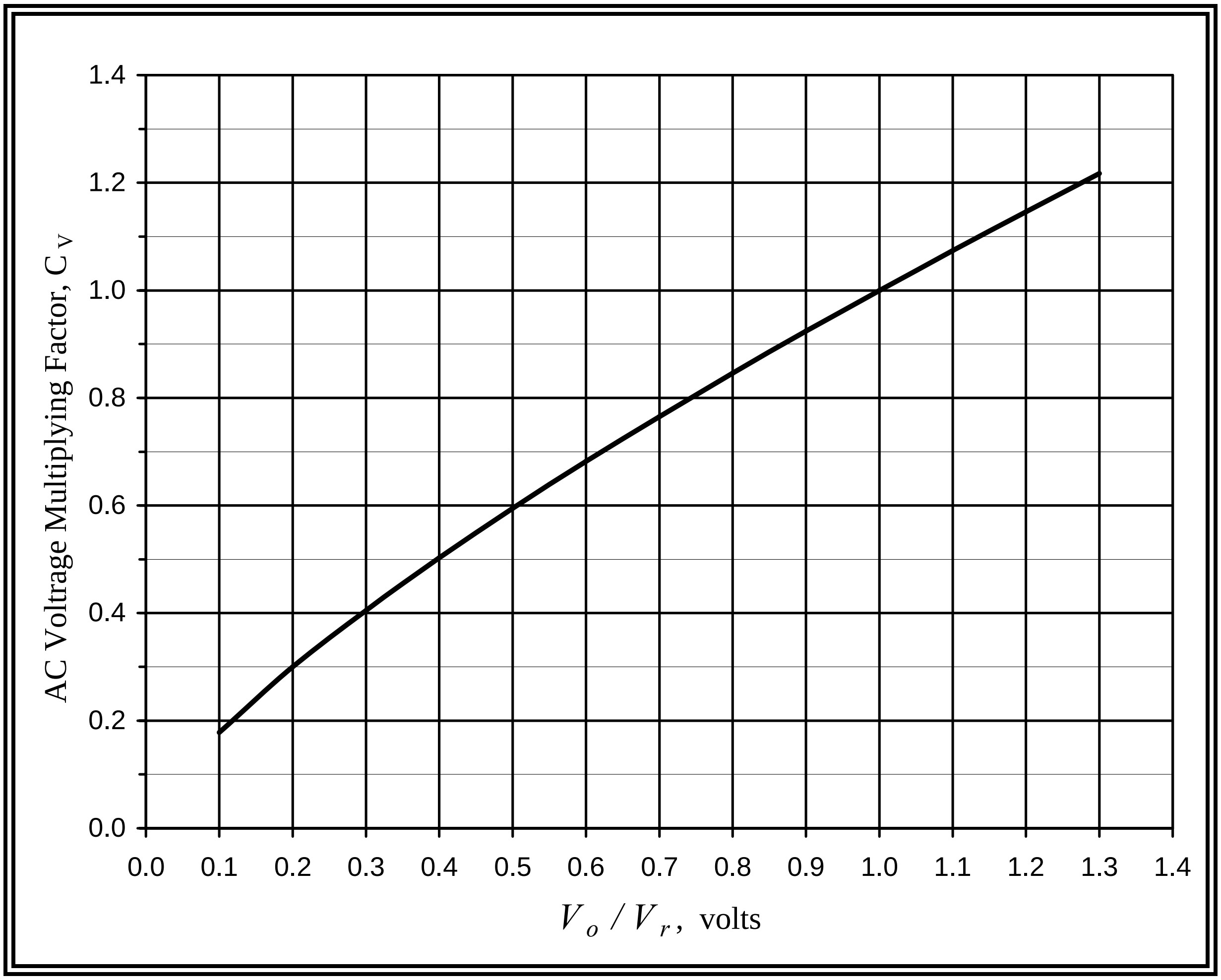
$$C_T = \left(\frac{1}{1.5^t}\right)^3$$

where: $$t = \frac{(T_R - T_O) - 20}{10}$$

T_R = Rated temperature of the solenoid, ^oC

T_O = Operating temperature of the solenoid, ^oC

Figure 5.2 Multiplying Factor for Coil Temperature

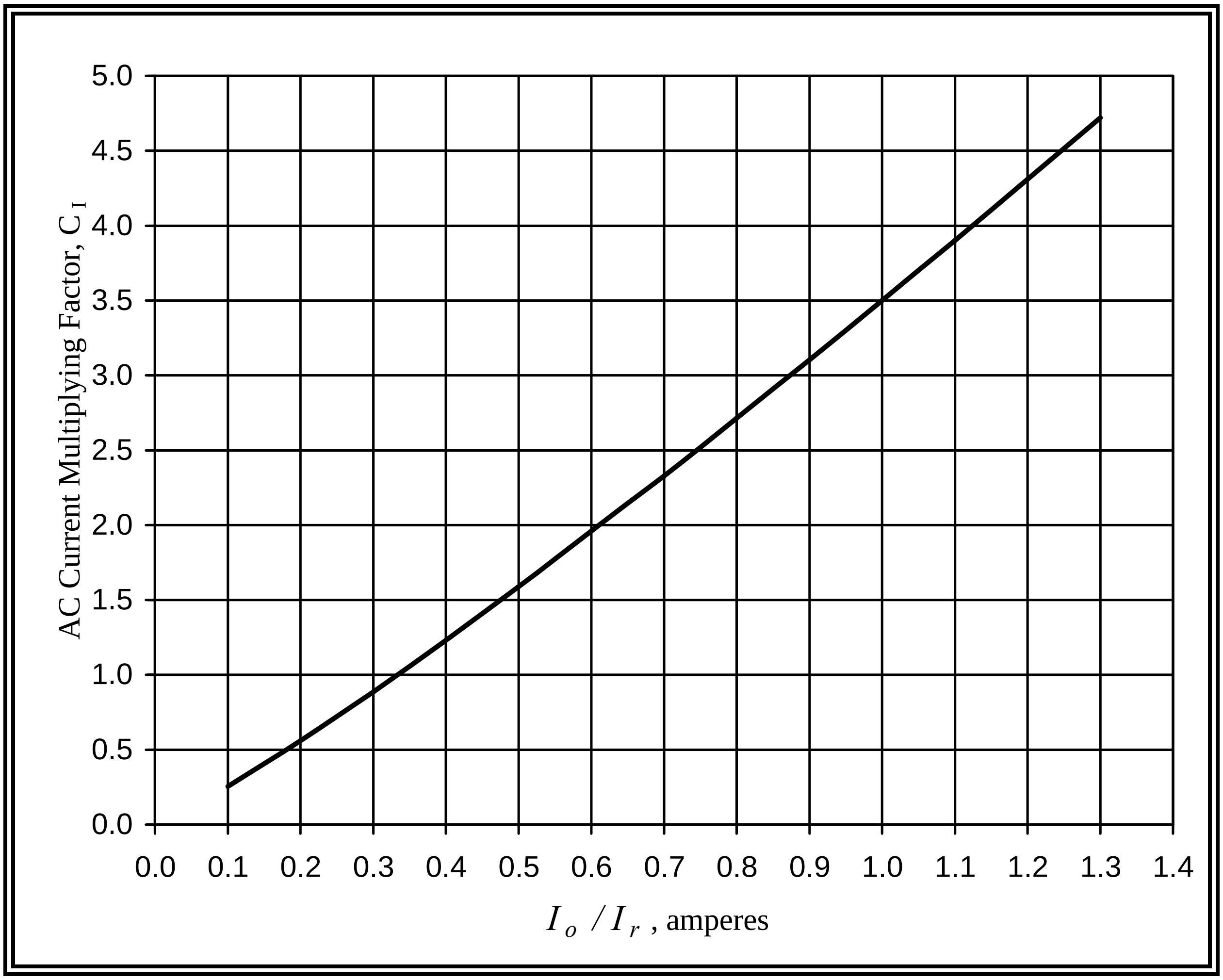


$$C_V = \left(\frac{V_o}{V_r}\right)^{0.75}$$

Where: V_o = Operating voltage, volts

V_r = Rated voltage, volts

Figure 5.3 Multiplying Factor for AC Contactor Voltage

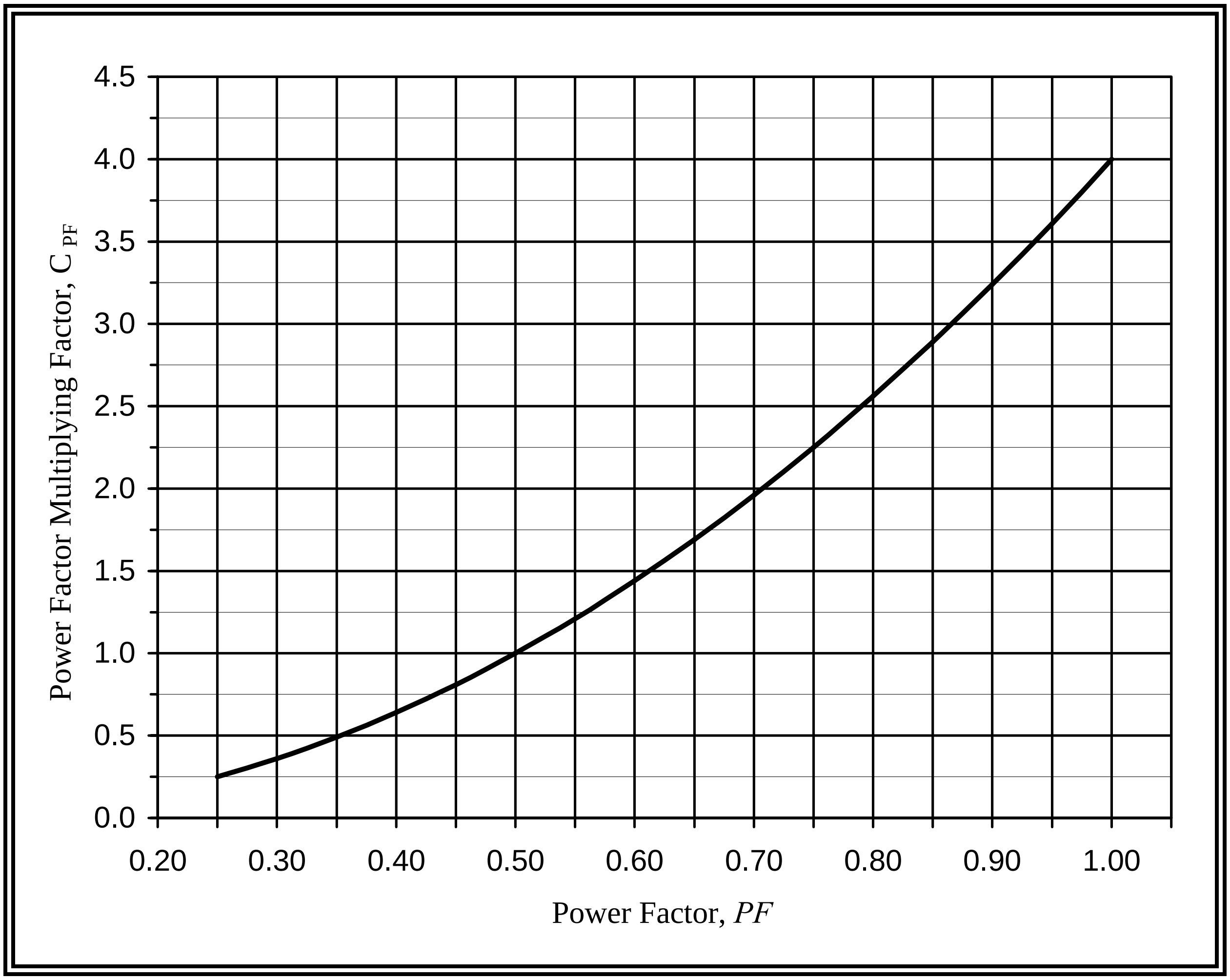


$$C_I = 3.50\left(\frac{I_o}{I_r}\right)^{1.14}$$

Where: I_o = Operating current, amperes

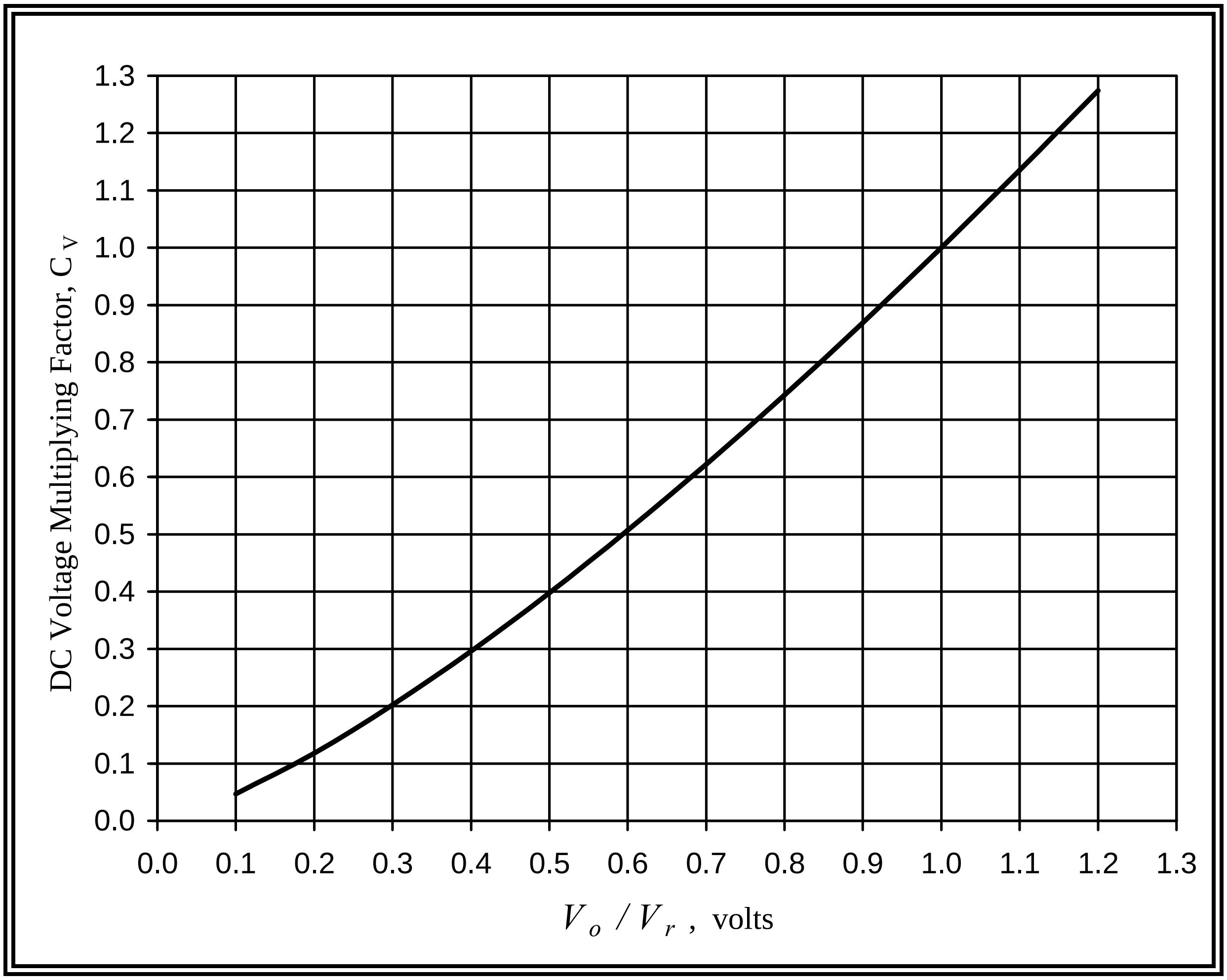
I_r = Rated current, amperes

Figure 5.4 Multiplying Factor for AC Contactor Current



$$C_{PF} = 4.0\left(PF\right)^2$$

Figure 5.5 Multiplying Factor for Power Factor

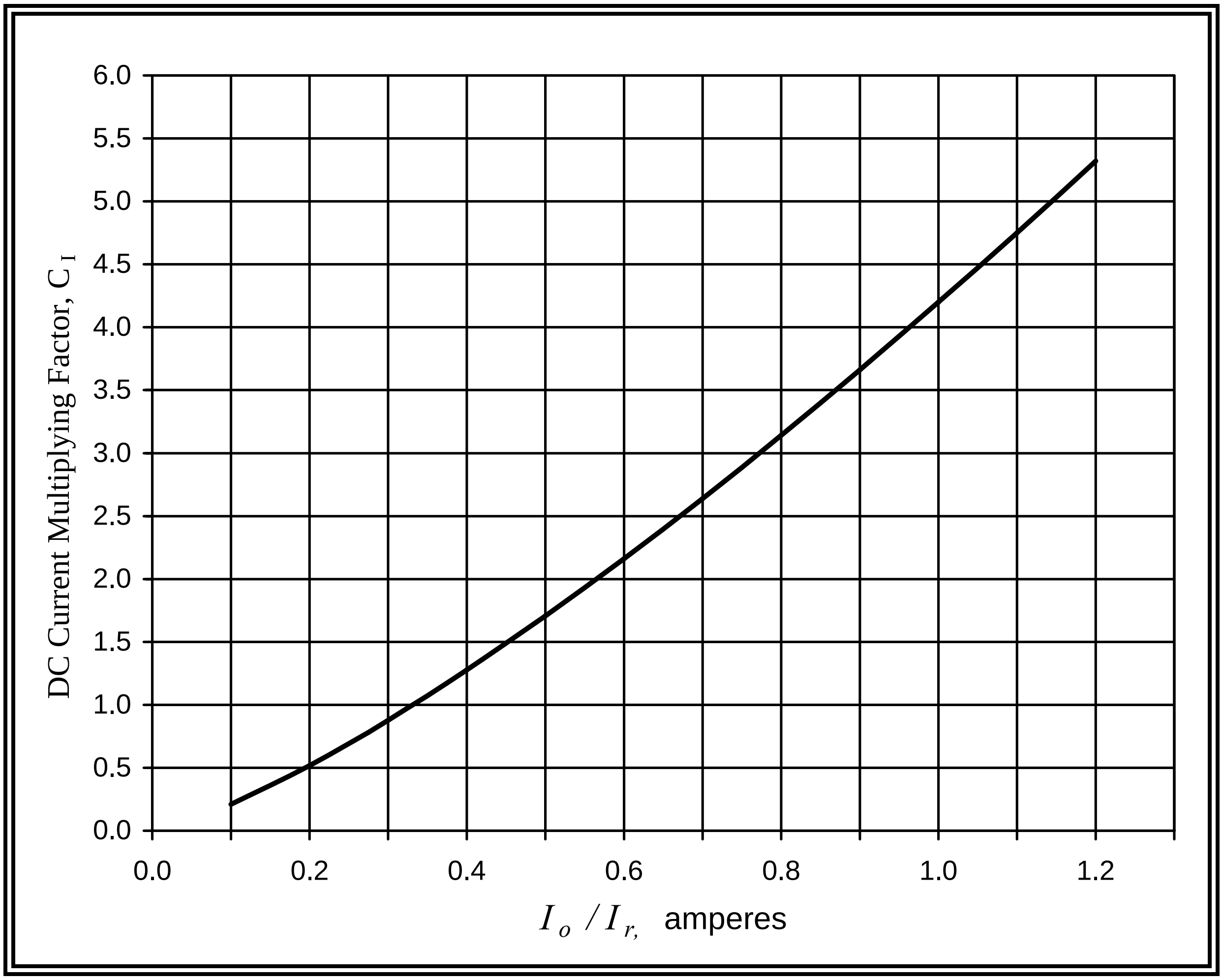


$$C_V = \left(\frac{V_o}{V_r}\right)^{1.33}$$

Where: V_o = Operating voltage, volts

V_r = Rated voltage, volts

Figure 5.6 Multiplying Factor for DC Contactor Voltage



$$C_I = 4.20 \left(\frac{I_o}{I_r} \right)^{1.30}$$

Where: I_o = Operating current, amperes

I_r = Rated current, amperes

Figure 5.7 Multiplying Factor for DC Contactor Current

5.5 REFERENCES

In addition to specific references cited throughout Chapter 5, other references included below are recommended in support of performing a reliability analysis of solenoids, relays and contactors.

28. MIL-HDBK-217, "Reliability Prediction of Electronic Equipment".

81. TD 84-3 R&M Data for Industrial Plants, A.P. Harris and Associates, 1984

111. Johnson Electric, Solenoid Technical Data, May 2008

126. Maintenance and Application Guide for Control Relays and Timers, Electric Power Research Institute, EPRI TR 102067s, December 1993

CHAPTER 6

VALVE ASSEMBLIES

6.0 TABLE OF CONTENTS

6.1 INTRODUCTION

This chapter contains procedures for evaluating fluid valve assemblies for reliability that can be used to support the development of mechanical equipment. The procedures are intended to focus attention on possible failure modes of a valve assembly and project the rate of failure for the valve in its intended operating environment.

A valve is a device that regulates the flow of a fluid including gases, liquids and fluidized solids by opening, closing or adjusting various passageways. A typical valve assembly is shown in Figure 6.1.

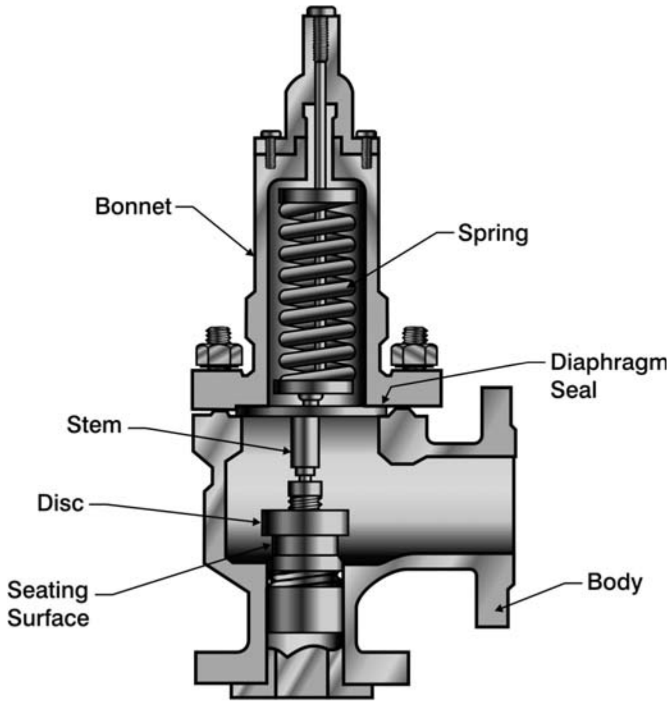


Figure 6.1 Typical Valve Configuration

Valves may be designed to be operated manually or may be automatically operated by changes in pressure, temperature or flow. These changes may act upon a diaphragm or a piston which in turn activates the valve. The most common valve to open and close passage of fluid is the poppet type valve shown in Figure 6.2. More complex control systems using valves requiring automatic control based on an external input require an actuator as a component within the valve. The actuator strokes the valve depending on its input signal and set-up, allowing the valve to be positioned accurately and allowing control over a variety of flow requirements. This type of valve called a sliding action valve is shown in Figure 6.5.

Valves are normally categorized by their function in the system design such as a ball valve, butterfly valve, check valve, control valve and gate valve. Inside the valve there are two basic designs including the poppet valve and sliding action valve described above. This chapter will describe the procedures for evaluating each of these basic designs for reliability. Basic parts of the valve include the valve seat for a poppet valve, sliding action mechanism for a sliding action valve, seals and packings to prevent the escape of fluid from the valve, a body and/or bonnet that form the valve casing, springs to maintain valve positioning and a device such a solenoid to actuate the valve.

After the failure rates are determined for each component part of the valve, the results are summed to determine the failure rate of the total valve assembly. A basic failure rate equation can be written as follows:

$$\lambda_{VA} = \lambda_{PO} + \lambda_{SE} + \lambda_{SP} + \lambda_{SO} + \lambda_{HO} \quad \textbf{(6-1)}$$

for a poppet type valve, or

$$\lambda_{VA} = \lambda_{SV} + \lambda_{SE} + \lambda_{SP} + \lambda_{SO} + \lambda_{HO} \quad \textbf{(6-2)}$$

for a sliding-action valve.

Where: λ_{VA} = Failure rate of total valve assembly in failures/million operations

λ_{PO} = Failure rate of poppet assembly in failures/million operations as derived from Section 6.3

λ_{SV} = Failure rate of sliding action valve assembly in failures/million operations as derived from Section 6.4

λ_{SE} = Failure rate of the seals in failures/million operations as derived from Chapter 3

λ_{SP} = Failure rate of spring(s) in failures/million operations as derived from Chapter 4

λ_{SO} = Failure rate of solenoid in failures/million operations as derived from Chapter 5

λ_{HO} = Failure rate of valve housing as derived from Section 6.5

6.2 FAILURE MODES OF VALVE ASSEMBLIES

Failure modes of a valve assembly are identified from the analyst's knowledge of the valve's design characteristics and the intended operating environment. Table 6-1 provides a summary of valve failure modes to be considered. Many valve assemblies are uniquely designed for special applications and a more detailed analysis is often required for those failure modes identified as critical or where results of the analysis indicate that an additional investigation is warranted. Failure rate models in this Handbook are based on the identification of failure modes with multiplying factors that modify a base failure rate considering the particular design characteristics and intended operating environment. This approach permits the estimate of an occurrence probability for individual failure modes.

Table 6-1. Failure Modes for a Valve Assembly

FAILURE MODE	FAILURE CAUSE	LOCAL EFFECT
Seal leakage	- Embrittlement - Installation damage - Wear - Surface damage - Distortion - Dynamic instability	- Internal or external valve leakage
Worn or damaged poppet seat	- Wear of poppet/seat Assembly - Contaminants	- Poppet not seating properly causing internal leakage and low/erratic pressure drop
Damaged valve stem	- Vibration, shock	- Poor valve response - Failure to open and/or close
Worn or damaged spool	- Contaminants - Misalignment	- Internal leakage
Sticking valve piston in main valve body	- Contaminants - Loss of lubrication - Air entrapment - Excessively high temperature - Structural interference	- Low/erratic pressure drop - Slow operating response - Valve immobile
Broken spring or damaged spring ends	- Fatigue	- Unable to adjust or maintain pressure
Inoperative solenoid assembly	- Open coil winding - Misalignment of solenoid with respect to spool or poppet stem	- Valve fails to open or close
External leakage	- Contaminants	- Poppet Stem Wear
Cracked fitting/housing	- Fatigue - External shock - Vibration	- External leakage

The primary failure mode of the poppet valve seating surface is wear caused by the impact of contaminants. In a gas valve, contaminants include liquids under pressure. A valve seat will wear where the sealing element contacts the seat. The local effect is insufficient seating and internal leakage. However, assuming the correct seat material for the application and minimal contaminants, the failure rate of the valve due to seating wear should be minimal. The sealing element may be a poppet, plate or ring. Possible failure modes for the sealing element include fatigue failure caused by repeated impact on the valve seat, sudden changes in differential pressure across the valve, and contaminants causing corrosion.

The failure rate of a compression spring is usually very low depending on the application. However, it is important to consider the operating environment of the valve when estimating the failure rate of the return spring. The spring material in a gas valve may not be resistant to all gas compositions. Likewise, in any valve it is important to consider the compatibility of the fluid and potential contaminants with the spring material. Failure modes of the stop plate are the same as the valve seat. The wear area of the stop plate is where the spring contacts the plate. This wear is minimal assuming correct hardness of the plate material.

For a sliding action valve the main failure modes to be considered include internal leakage caused by a worn spool and slow response of the valve caused by a sticking piston. As with a poppet valve the chief cause of a worn spool or slow response is the presence of contaminants. Assistance with evaluating the valve parts such as O-rings and solenoids for reliability can be found in the appropriate chapters of this Handbook.

After the basic parts of the valve are evaluated for reliability, the dynamics of the valve must be considered. Again, the spring is probably most critical to estimating the failure rate of the valve. For example, a sticking sealing element caused by a misaligned spring resulting in poor dynamic response is a potential failure mode that needs to be considered. Pressure differential, operating temperature, viscosity and use rate are variables that determine the failure rate of the valve. Surface finish of the poppet, allowable leakage, and a contaminant factor considering flow rate are used to estimate the failure rate of the poppet due to wear from contaminants. The failure rates of valve seals, the return spring and the solenoid driver are determined separately. See Chapters 3, 4 and 5.

Typical failure modes for a valve assembly are listed in Table 6-1. It should be noted that the failure modes, failure causes and failure effects may be interchanged depending upon the type of analysis being performed. For example, a functional analysis will tend to identify those entries in Table 6-1 under local effect as the failure mode while a very detailed hardware analysis would result in the identification of those entries under failure cause as the failure mode.

6.3 FAILURE RATE MODEL FOR POPPET ASSEMBLY

The term poppet refers to those valves in which the valve element travels perpendicular to a plane through the seating surface. The poppet valve element is used in flow control, pressure control and directional control valves. In a poppet valve, a relatively large flow area is provided with short travel of the poppet. This characteristic simplifies the actuator requirements and permits the use of solenoids and diaphragms, which are characteristically short stroke devices.

Figure 6.2 illustrates the operation of a simple poppet valve. The valve consists primarily of a movable poppet which closes against a valve seat. The valve may be actuated manually or by electrical, mechanical or pneumatic means. This chapter of the manual discusses the valve mechanism itself. Refer to Chapter 5 for procedures to evaluate the reliability of the solenoid and to other appropriate chapters of the manual to determine the reliability of other components of the actuation mechanism. In the closed position, fluid pressure on the inlet side tends to hold the valve tightly closed. A force applied to the top of the valve stem opens the poppet and allows fluid to flow through the valve.

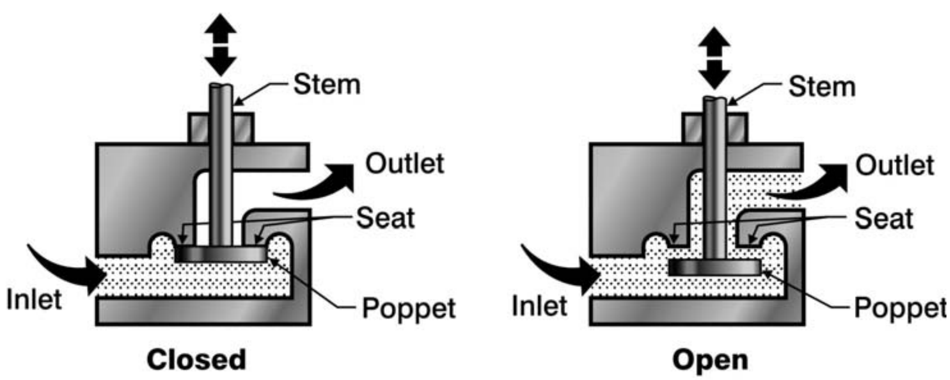


Figure 6.2 Poppet Valve Assembly

The poppet fits into the center bore of the seat. The seating surfaces of the poppet and the seat are lapped or closely machined so that the center bore will be sealed when the poppet is seated. An O-ring is usually installed on the stem of the poppet to prevent leakage past this portion of the poppet assembly.

Table 6-2 is a list of typical failure modes, mechanisms and causes for a poppet assembly. A review of failure rate data suggests the following characteristics be included in the evaluation of poppet assembly reliability:

- Leakage requirement
- Material hardness
- Surface irregularities

- Fluid viscosity
- Fluid/material compatibility
- Fluid pressure
- Physical size of poppet/seat
- Q.C./manufacturing processes
- Contamination level
- Utilization rate

Table 6-2. Failure Rate Considerations for a Poppet Assembly

FAILURE MODE	FAILURE MECHANISMS	FAILURE CAUSES
Internal Leakage	Worn poppet/seat	- Contaminants
Poor Response	Sticking/jammed poppet assembly	- Side Loading - Incorrect spring pressure - Contaminants
External Leakage	Wear of poppet stem	- Contaminants

A new poppet assembly (or fairly new if some initial deformation exists) may be expected to have a sufficiently smooth surface finish for the valve to meet internal leakage specifications. However, after some period of time contaminants will cause wear of the poppet/seat assembly until the leakage rate is beyond tolerance. This leakage rate, at which point the valve is considered to have failed, will depend upon the application.

A failure rate equation for a poppet assembly is dependent upon the ratio of actual leakage rate to that allowable under conditions of usage. This rate can be expressed as follows:

$$\lambda_{PO} = \lambda_{PO,B}\left(\frac{Q_a}{Q_f}\right) \tag{6-3}$$

Where:
λ_{PO} = Failure rate of the poppet assembly, failures/million operations

$\lambda_{PO,B}$ = Base failure rate for poppet assembly, failure/million operations

Q_a = Leakage rate, in^3/min

Q_f = Leakage rate considered to be valve 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 is determined from the following equation (Reference 22):

$$Q_a = \frac{2 \ x \ 10^2 \ D_M \ f^3 \left(P_1^2 - P_2^2\right) K_1}{\nu_a \ L_W \ (S_S)^{1.5}} \quad \textbf{(6-4)}$$

Where:

Q_a = Actual fluid leakage, in^3/min

D_M = Mean seat diameter, in

f = Mean surface finish of opposing surfaces, in

P_1 = Upstream pressure, lb/in^2

P_2 = Downstream pressure, lb/in^2

ν_a = Absolute fluid viscosity, lbf-min/in^2

L_W = Radial seat land width, in

S_S = Seat stress, lb/in^2

K_1 = Constant which considers the impact of contaminant size, hardness and quantity of particles

Failure rate of the poppet assembly will be dependent upon leakage rate and those factors which influence the deterioration of surface finish such as rate of cycling, material properties and contaminants. Deterioration of the poppet and seat by contaminants is dependent upon material properties and the number of contaminants, and that part of the time the poppet is open and subject to contaminants under fluid pressure.

A contamination factor can be derived from the following equation:

$$Z = \left(function \ of\right)\left[\alpha, n, Q, d, T\right] \quad \textbf{(6-5)}$$

Where:

Z = Poppet / seat degradation

α = Contaminant wear coefficient, in^3/particle

n = Number of contaminant particles/in^3

Q = Flow rate, in^3/min

d = Ratio of time the poppet is open to total operating time

T = Temperature of operation, °F

Table 6-5 provides typical quantities of contaminants for use in establishing a multiplying factor. By normalizing the equation to those values for which historical failure rate data are available the following model can be derived:

$$\lambda_{PO} = \lambda_{PO,B} \bullet C_P \bullet C_Q \bullet C_F \bullet C_\nu \bullet C_N \bullet C_S \bullet C_{DT} \bullet C_{SW} \bullet C_W \qquad \textbf{(6-6)}$$

Where:

λ_{PO} = Failure rate of poppet assembly in failures/million operations

$\lambda_{PO,B}$ = Base failure rate of poppet assembly, 1.40 failures/million operations (See Chapter 1 for derivation)

C_P = Multiplying factor which considers the effect of fluid pressure on the base failure rate (See Figure 6.6)

C_Q = Multiplying factor which considers the effect of allowable leakage on the base failure rate (See Figure 6.7)

C_F = Multiplying factor which considers the effect of surface finish on the base failure rate (See Figure 6.8)

C_ν = Multiplying factor which considers the effect of fluid viscosity and temperature on the base failure rate (See Table 6-6 and Figure 6.15)

C_N = Multiplying factor which considers the effect of contaminants on the base failure rate (See Table 6-5)

C_S = Multiplying factor which considers the effect of the contact pressure on the base failure rate (See Figure 6.9)

C_{DT} = Multiplying factor which considers the effect of the seat diameter on the base failure rate (See Figure 6.10)

C_{SW} = Multiplying factor which considers the effect of the seat land width on the base failure rate (See Figure 6.11)

C_W = Multiplying factor which considers the effect of flow rate on the base failure rate (See Figure 6.14)

The following paragraphs provide background information on those parameters included in the model.

6.3.1 Fluid Pressure

Figure 6.6 contains the fluid pressure multiplying factors for use in the equation. Valves having high response characteristics and consequently a high poppet velocity will incur large impact loading which tends to reduce the life expectancy of the valve. As with any piece of mechanical equipment, the higher the structural loads the shorter the life. Pressure forces arise from any net pressure unbalance acting on the valve element. Depending upon the functional design of the valve, the pressure force may increase, decrease, or virtually have no effect on the actuation force. In an unbalanced valve design such as a conventional poppet, upstream pressure normally acts in a direction to seat the valve so that an increasing upstream pressure will tend to force the valve element tighter against its seat. The use of pressure unbalance to aid in sealing requires a higher actuation force to open the valve. When the size of the valve and/or magnitude of pressure demand excessively large actuation forces, a balanced design and/or piloting are often utilized. In most cases the pressure on the poppet can be assumed to be the system upstream pressure, P_1, minus the downstream pressure, P_2.

6.3.2 Allowable Leakage

Figure 6.7 shows the allowable leakage multiplying factor for use in equation 6-6. Allowable internal leakage of a poppet design can be obtained from valve specifications. Leakage requirements vary from molecular flow for certain shutoff valves at one extreme to several cubic feet per minute in some inexpensive valves which control water or other inexpensive fluid. Allowable leakage must be evaluated with respect to total mission and operational requirements.

6.3.3 Surface Finish

Evaluation of surface finish involves both poppet and seat assemblies. Surface finishes will usually be specified on assembly drawings in terms of microinches or by a manufacturing process. Typical surface finishes for manufacturing processes are provided in Table 6-4. These values are for a finish as initially manufactured and a new valve can be expected to have a sufficiently smooth surface finish to meet internal leakage specifications. However, after some period of time contaminants will cause wear of the poppet/seat assembly until leakage rate is beyond tolerance. This deterioration of surface finish will be influenced by operating temperature and pressure, rate of cycling, loads and material properties.

6.3.4 Fluid Viscosity

Viscosity of a fluid is much more dependent on temperature than it is on pressure. For example, when air pressure is increased from 1 atmosphere to 50, its viscosity is only increased by about 10%. In contrast, Figure 6.3 shows the dependence of viscosity on temperature for some common fluids. The graph shows how viscosity of liquids decreases with temperature while that of gases increases with temperature. Multiplying factors for viscosities of typical fluids are provided in Table 6-6. Multiplying factors for other fluids are determined from the table by a knowledge of viscosity at the applicable fluid temperature. Viscosity for a specific fluid is obtainable from many reference sources. If the value located is in terms of kinematic viscosity, multiply the value by the specific gravity (density) at the desired temperature to determine the dynamic viscosity.

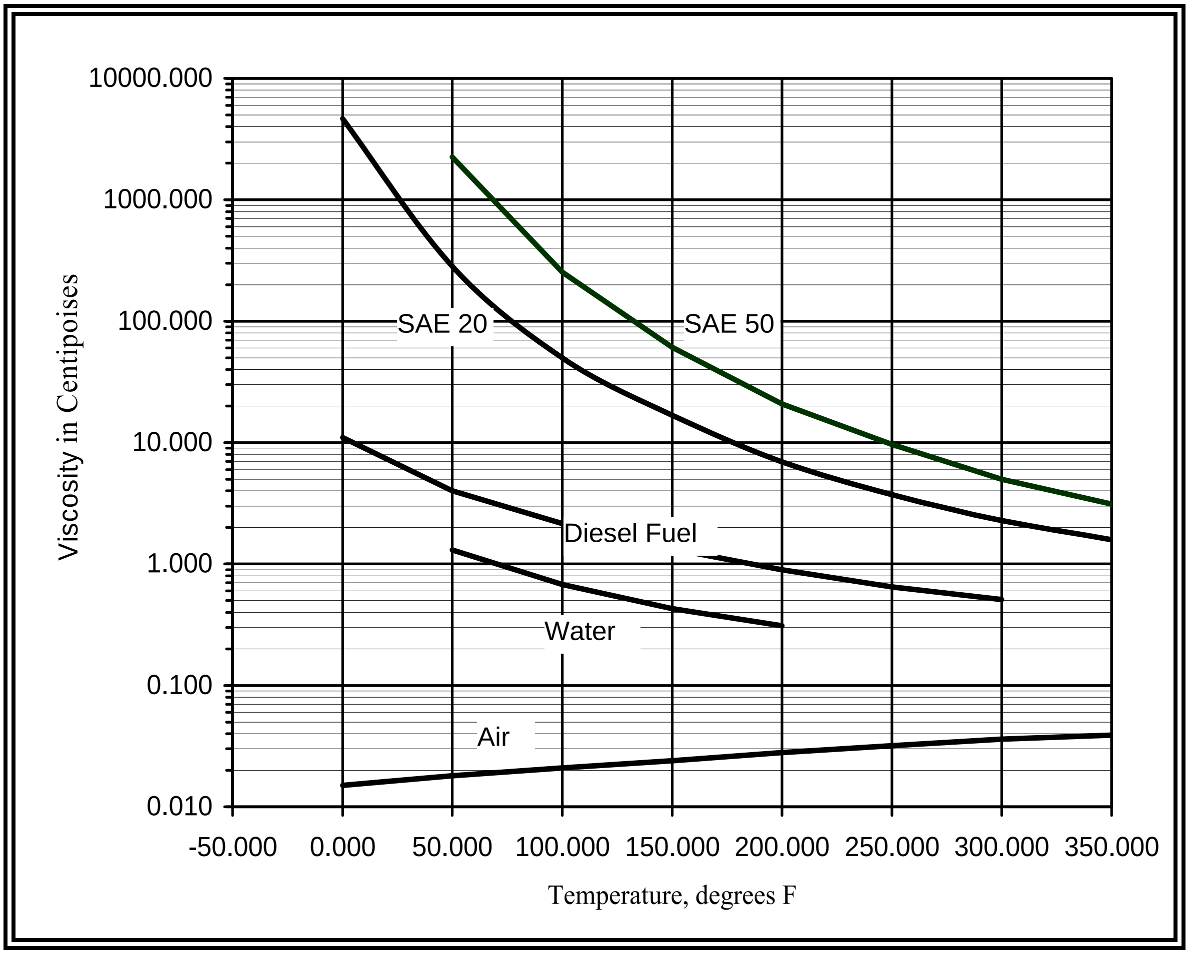


Figure 6.3 Dynamic Viscosities of Various Fluids

6.3.5 <u>Contamination Sensitivity</u>

Cleanliness of the system and of the fluid medium has a direct effect upon the operation and life of a poppet valve. Contaminants can clog or jam the poppet and cause excessive leakage in metal-to-metal seated valves. Particulate matter in gaseous media, especially in the lighter gases such as helium, can be extremely destructive to internal parts, particularly seats, because of the very high velocity that can be attained under sonic conditions.

The analysis of particle sizes includes the determination of upstream filter size, the filter maintenance schedule, the number of upstream components between the valve and filter, and the number of particles likely to be encountered at the poppet/seat assembly. Table 6-5 lists typical quantities of contaminants for use in determining the multiplying factor.

6.3.6 Contact Pressure

The force applied by the poppet against the seat is found by actual measurement or design specifications. In the typical valve seat example of Figure 6.4, the seat area A_{ST} can be computed as follows:

$$A_{ST} = \frac{\pi\left(D_1^2 - D_2^2\right)}{4} \quad \textbf{(6-7)}$$

Where: A_{ST} = Seat Area, in^2

D_1 = Outside diameter of poppet, in

D_2 = Diameter of poppet shaft, in

And the force of the poppet against the seat is:

$$F_S = A_{ST}\left(P_1 - P_2\right) \quad \textbf{(6-8)}$$

Where: F_S = Force on seat, lb

P_1 = Upstream fluid pressure, psi

P_2 = Downstream fluid pressure, psi

The seat land area acting as a seal when the valve is closed, is calculated by:

$$A_{SL} = \frac{\pi\left(D_1^2 - D_3^2\right)}{4} \quad \textbf{(6-9)}$$

Where: A_{SL} = Seat land area, in^2

D_3 = Inside diameter of valve outlet

Then the expression for contact pressure, S_S, is the force applied to the seat divided by the seat land area:

$$S_S = \frac{F_S}{A_{SL}} \tag{6-10}$$

Therefore:

$$S_S = (P_1 - P_2)\,\frac{A_{ST}}{A_{SL}} \tag{6-11}$$

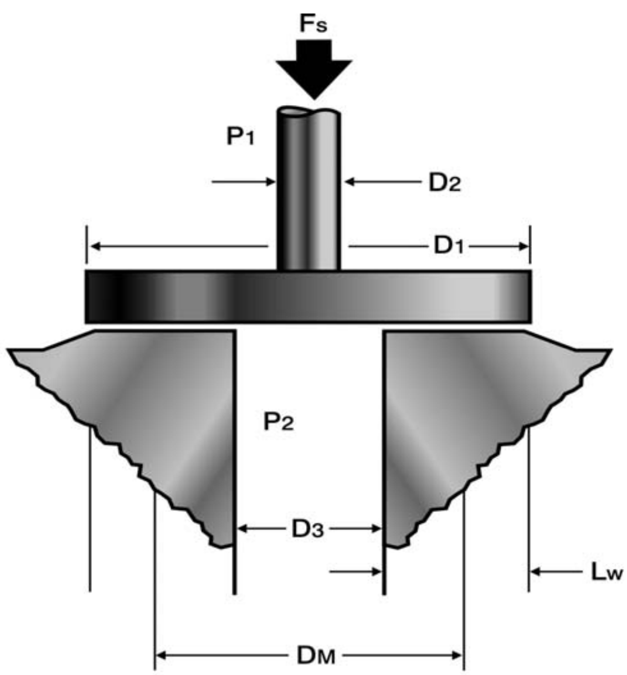


Figure 6.4 Typical Poppet Valve Seat Configuration

The minimum contact pressure to prevent leakage for most materials is approximately three times the fluid pressure. In Equation (6-4), leakage varies inversely with the seat stress raised to the 1.5 power. Therefore, a multiplying factor for the effect of contact pressure on the valve base failure rate can be derived as follows:

$$C_S = 0.26\left(\frac{9000}{S_S}\right)^{1.5} \tag{6-12}$$

Figure 6.9 provides the multiplying factors for different values of contact pressure.

6.3.7 Physical Dimensions

The poppet diameter, seat diameter, and seat land width are shown in Figure 6.4. Figures 6.10 and 6.11 provide multiplying factors for seat diameter and land width.

6.3.8 Operating Temperature

The duty cycle of a poppet valve can vary from several on-off cycles to many hundreds of cycles per hour. Multiple cycling under high pressure or operating temperature decreases the life of the valve. The rate of cycling may be important if the temperature rise, as a result of the operation, becomes significant. The effects of fluid temperature on failure rate are included in the fluid viscosity multiplying factor, C_V.

6.3.9 Other Considerations

Several failure rate considerations are not specifically included in the model but rather included in the base failure rate. The base failure rate is an average value which reflects field performance data. The following items can be used as a check list to assure that the potential failure mechanisms have been considered:

- Fluid medium considerations which are important in valve designs include the physical properties of the fluid and the compatibility of the fluid with poppet/seat materials. Corrosive fluids will rapidly change the surface finish. The state and physical properties of the fluid become particularly important in determining pressure drop and flow capacity.

- In considering maintenance, requirements for special tasks must be identified. Valve seats should be accessible and easily replaced, preferably without removing the valve from its circuit. When it is necessary to service a valve in the field, care must be exercised to insure that contamination from the work area is not introduced into the valve or system. Requirements for lubrication and adjustments should be minimized to provide high reliability in service use.

- While critical design features are usually based upon one primary fluid, consideration must also be given to secondary fluids with which the valving unit will be required to operate during cleaning and testing operations.

- The location of the valve in the circuit must be considered when considering system failure modes and failure rates. For example, in some circuits a backup control valve is used to permit continued operation in event one valve becomes stuck in the open or closed position.

6.4 FAILURE RATE MODEL FOR SLIDING ACTION VALVES

Sliding action valves are usually designed such that a spool slides longitudinally to block and uncover ports in the housing. A rotary spool is sometimes used. Sealing is accomplished by a very closely machined fit between the spool and the valve body. A typical sliding action valve is shown in Figure 6.5.

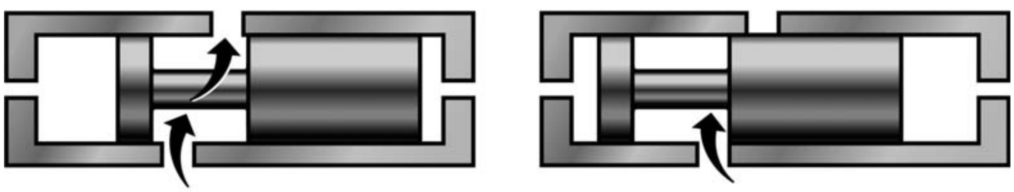


Figure 6.5 Sliding Action Valve Assembly

A primary advantage of sliding action valves is the feasibility of obtaining a pressure-balanced design. An inherent disadvantage of sliding action valves is leakage, a problem which can only be controlled by close machining or reliable dynamic sealing techniques. Diametrical spool clearances of approximately 50 microinches are common and surface finishes of 4 to 6 microinches are standard requirements for spools and sleeves. Therefore, contamination tolerance and dirt sensitivity are critical factors in the design and use of sliding action valves, and reliability will be directly affected by dirt particles. Force balances, flow rate and general mechanical operation can be influenced by the presence of contaminants within the valve. Contamination problems include wear of the spool and sleeve until the leakage rate is beyond tolerance. The steps to investigating internal leakage are the same as for the poppet type valve. Table 6-3 is a list of failure modes, mechanisms and causes for spool assemblies. Other failure modes should be identified for the specific application and evaluated to determine the applicability of the failure rate model to the analysis being performed.

Table 6-3. Failure Rate Considerations for Sliding Action Valve

FAILURE MODE	FAILURE MECHANISMS	FAILURE CAUSES
Internal leakage	Worn spool/sleeve	- Contaminants - Side loading
Poor response	Sticking sleeve assembly	- Side Loading - Incorrect spring pressure - Contaminants
External leakage	Worn gasket/seal	- Contaminants
Valve port fails to open	Jammed sleeve assembly	- Excessive side loading - Contaminants

An equation similar to that for poppet valves can be used to predict the reliability of a sliding action valve:

$$\lambda_{SV} = \lambda_{SV,B} \frac{B^2 D_{SP} \left(P_1^2 - P_2^2\right)^{1/2} \mu\, \alpha\, \eta}{Q_f\, \nu_a} \tag{6-13}$$

Where:

λ_{SV} = Failure rate of sliding action valve assembly in failures/million operations

$\lambda_{SV,B}$ = Base failure rate = 1.25 failures/million operations

B = Spool clearance, in

D_{SP} = Spool diameter, in

P_1 = Upstream pressure, lb/in^2

P_2 = Downstream pressure, lb/in^2

ν_a = Absolute fluid viscosity, lb-min/in^2

Q_f = Leakage rate considered to be device failure, in^3/min

μ = Friction coefficient

α = Contaminant wear coefficient, in^3/particle

η = Number of contaminant particles/in^3

By normalizing the characteristic equation to those values for which historical failure rate data are available, the following model can be derived:

$$\lambda_{SV} = \lambda_{SV,B} \bullet C_P \bullet C_Q \bullet C_V \bullet C_N \bullet C_B \bullet C_{DS} \bullet C_\mu \bullet C_W \qquad \textbf{(6-14)}$$

Where: $\lambda_{SV,B}$ = Base failure rate, 1.25 failures/million operations

C_P = Multiplying factor which considers the effect of fluid pressure on the base failure rate (See Figure 6.6)

C_Q = Multiplying factor which considers the effect of allowable leakage on the base failure rate (See Figure 6.7)

C_V = Multiplying factor which considers the effect of fluid viscosity/temperature on the base failure rate (See Table 6-6 and Figure 6.15)

C_N = Multiplying factor which considers the effect of fluid contaminants on the base failure rate (See Table 6-5)

C_B = Multiplying factor which considers the effect of spool clearance on the base failure rate (See Figure 6.12)

C_{DS} = Multiplying factor which considers the effect of spool diameter on the base failure rate (See Figure 6.13)

C_μ = Multiplying factor which considers the effect of friction coefficient on the base failure rate (See Table 6-7)

C_W = Multiplying factor which considers the effect of flow rate on the base failure rate (See Figure 6.14)

6.4.1 Fluid Pressure

In most sliding action valves the applied fluid pressure is the upstream pressure minus the downstream pressure. Figure 6.6 provides the multiplying factors for fluid pressure. Other factors in evaluating the effects of fluid pressure on valve reliability include the following:

Size - Structural strength becomes an increasingly important consideration with increasing valve size because pressure loads are a function of the square of the valve size.

Balance - If the valve is inherently pressure-balanced, the influence of pressure upon such parameters as size and actuation forces will be far less than in the case of an inherently unbalanced unit.

Pressure Induced Strain - Binding of certain close-tolerance sliding action valves can result with excessive pressure load on a port.

Conditions of Pressure - Circumstances under which the valve unit is subjected to high pressure must be considered. A drain valve, for example, may be required to seal against high pressure, but never be required to open until after the pressure has been relieved.

6.4.2 Allowable Leakage

Allowable internal leakage of the sliding action valve can be obtained from valve specifications usually in terms of quiescent flow or leakage flow. Quiescent flow is the internal valve flow or leakage from supply-to-return with no flow in the load ports. Allowable leakage will vary considerably according to the operational requirements. Figure 6.7 provides multiplying factors for allowable leakage.

6.4.3 Contamination Sensitivity

Cleanliness of the fluid medium and surrounding medium has a direct effect upon the occurrence of stiction, weldment and general operation of sliding valve assemblies. No fluid system is completely free of particulate contamination and sensitivity of a valve to contamination is an important consideration in reliability.

In sliding action valves there is a tradeoff between contamination sensitivity and leakage based on clearances between the spool and sleeve. If leakage is minimized by reducing the clearance between the valving element and its housing, a larger number of contaminant particles can become lodged, causing valve failure. The clearance values should be checked at both of the temperature extremes to which the valve will be subjected, in order to ensure adequate design for the largest size of contamination particle anticipated.

The analysis of particle sizes includes the determination of upstream filter sizes, the filter maintenance schedule, the number of upstream components between the valve and filter, and the number of particles likely to be encountered at the spool assembly. Table 6-5 provides typical quantities of contaminants for use in the failure rate equation.

6.4.4 Fluid Viscosity

The dependence of viscosity on temperature change is shown in Figure 6.3. The graph shows how viscosity of liquids decreases with temperature while that of gases increases with temperature. Multiplying factors for viscosities of typical fluids are provided in Table 6-6. Multiplying factors for other fluids are determined from the table by a knowledge of viscosity at the applicable fluid temperature. Viscosity for a specific fluid is obtainable from many reference sources. If the value located is in terms of kinematic viscosity, divide the value by the specific gravity (density) at the desired temperature to determine the dynamic specific gravity.

6.4.5 Spool-to-Sleeve Clearance

Highly polished and uniform surface finishes of 4-6 microinches can usually be assumed for a valve spool. The model assumes that the spool is environmentally protected. If this is not the case, a separate analysis will be required to determine the effects of aging and deterioration of the surfaces on the spool to sleeve clearance. A diametrical spool clearance of 50 microinches is typical for sliding action valves. The exact value is taken from assembly drawings. Figures 6.12 and 6.13 provide multiplying factors for the spool-to-sleeve clearance and spool diameter.

6.4.6 Friction Coefficient

A sticking valve spool is usually caused by contaminants. Particles can accumulate between the spool and sleeve as part of the silting process until the build-up is sufficient to cause stiction. Results include valve hunting, erratic regulation and eventual locking. The silting process can be aggravated by inactivity of the valve. Another failure mechanism to be considered is reduced clearance between the spool and sleeve caused by soft metal particles being wedged and burnished on the surfaces. The actual friction coefficient is used in the model. Table 6-7 contains friction coefficient values for typical materials used in valve designs.

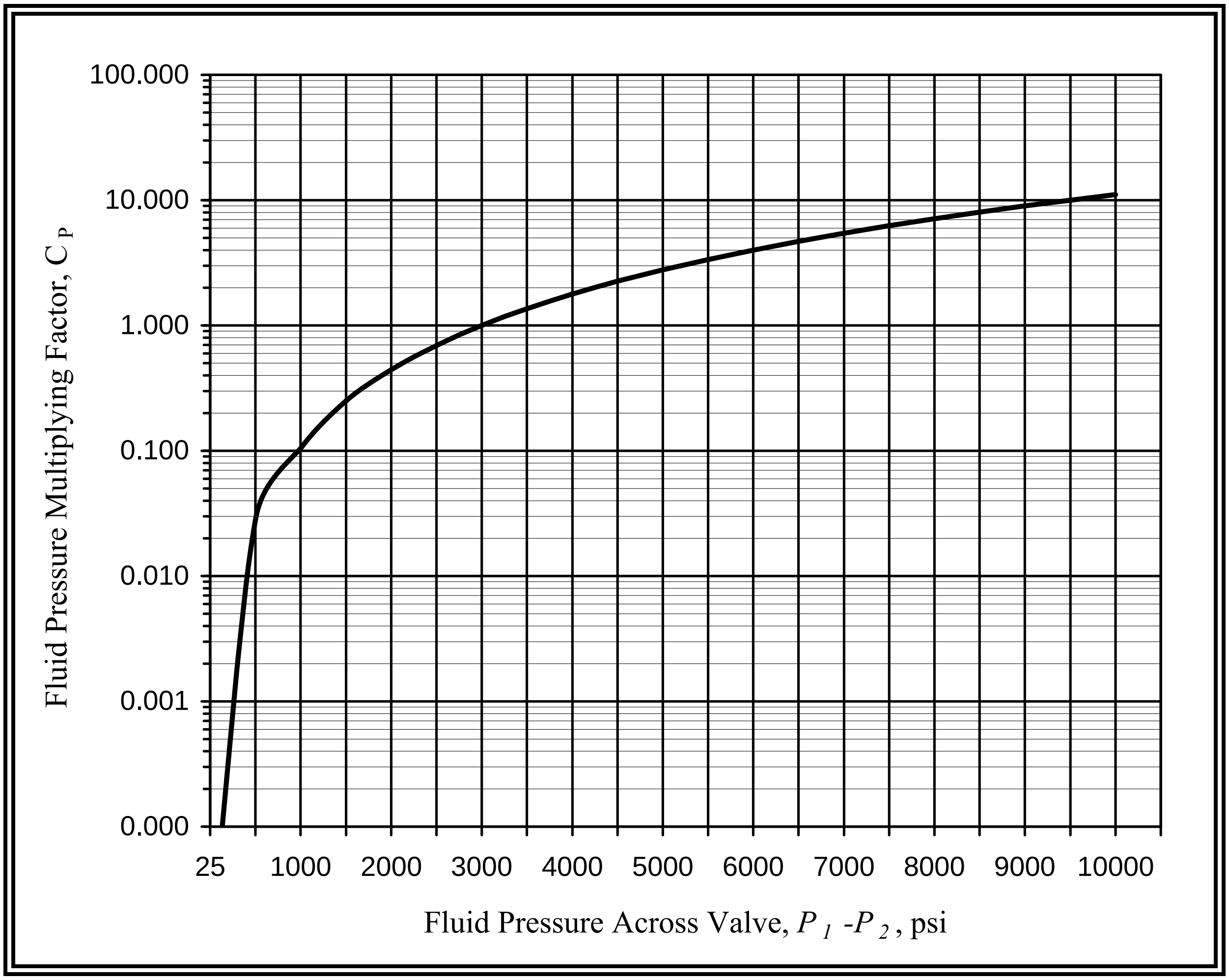
6.5 FAILURE RATE ESTIMATE FOR HOUSING ASSEMBLY

There are many factors which could be considered in determining the potential rate of fatigue failure of a valve housing including connectors. For critical safety related applications, a review of the stress analysis is warranted. Normally, the probability of a cracked housing is minimal and the failure rate is best determined from field experience data.

$$\lambda_{HO} = \lambda_{HO,B} \tag{6-15}$$

Where: λ_{HO} = Failure rate of valve housing, failures/million operating hours

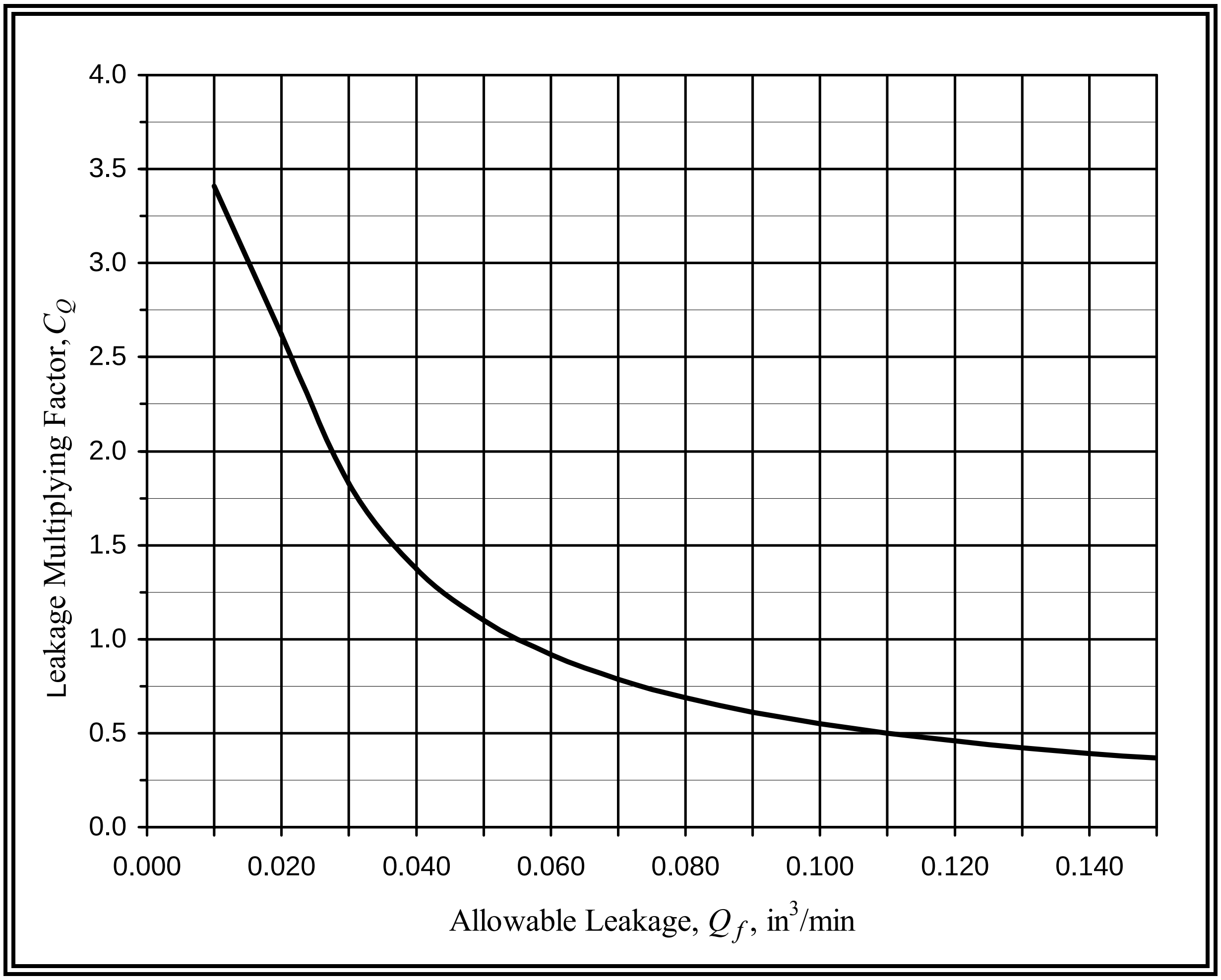
$\lambda_{HO,B}$ = Base failure rate of housing, 0.01 failures/million hours



For $P_1 - P_2 > 50$ psi:

$$C_P = \left(\frac{P_1 - P_2}{3000} \right)^2$$

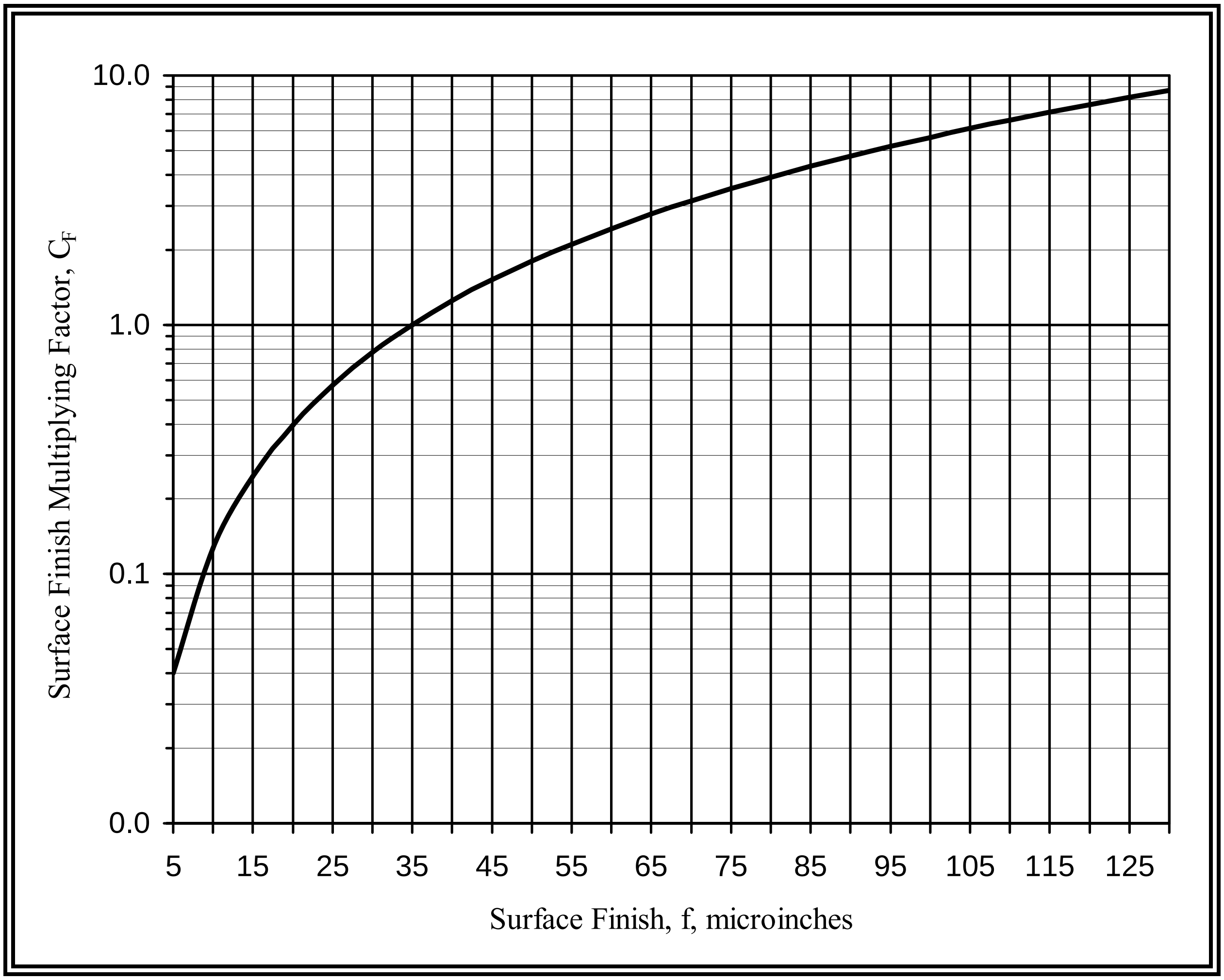
Figure 6.6 Fluid Pressure Multiplying Factor



For leakage $>$ 0.03 in^3/min, $C_Q = \dfrac{0.055}{Q_f}$

For leakage $\leq$ 0.03 in^3/min, $C_Q = 4.2 - \left(79 Q_f\right)$

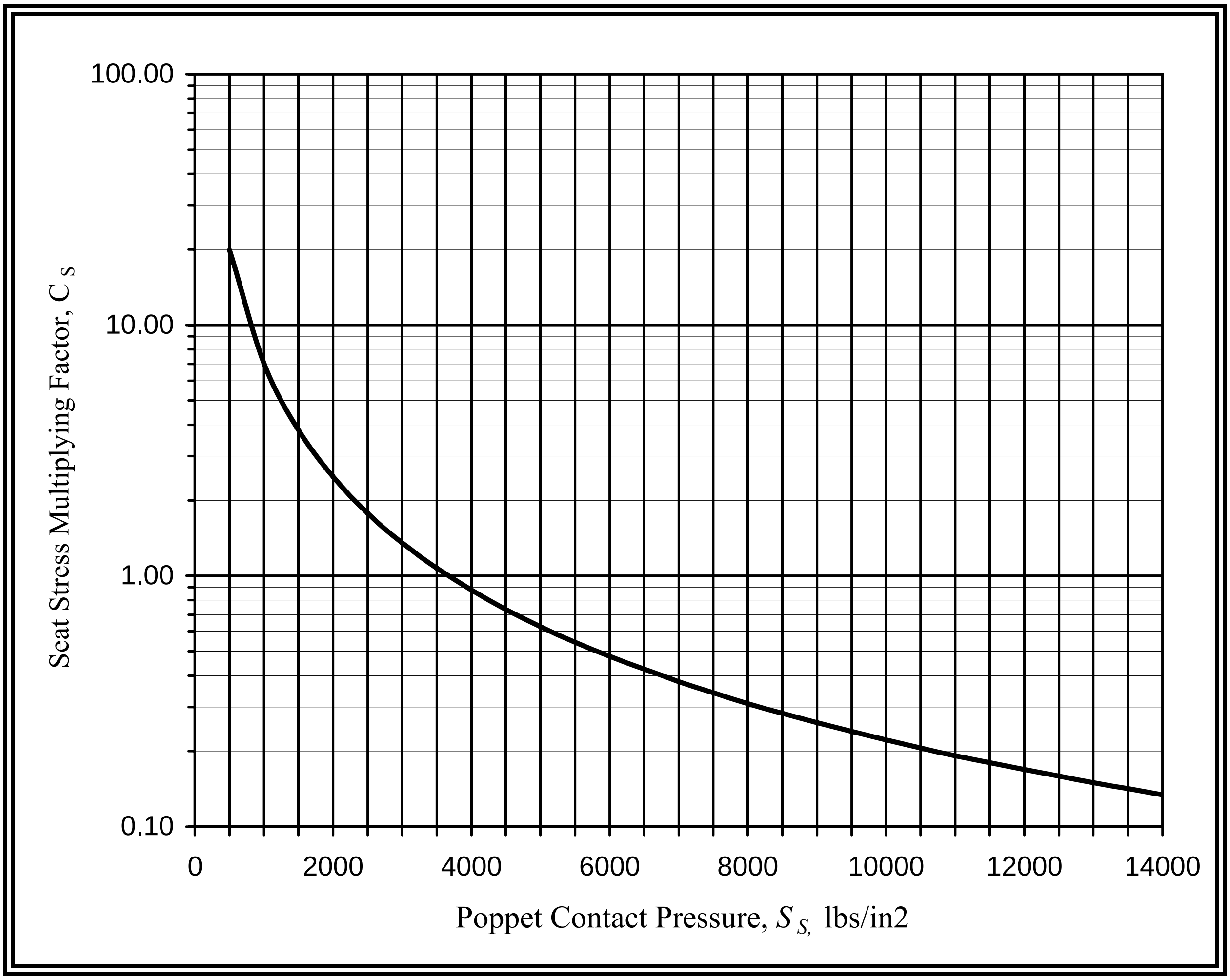
Figure 6.7 Allowable Leakage Multiplying Factor



$$C_F = \frac{f^{1.65}}{353}$$

Table 6-4 provides typical surface finishes

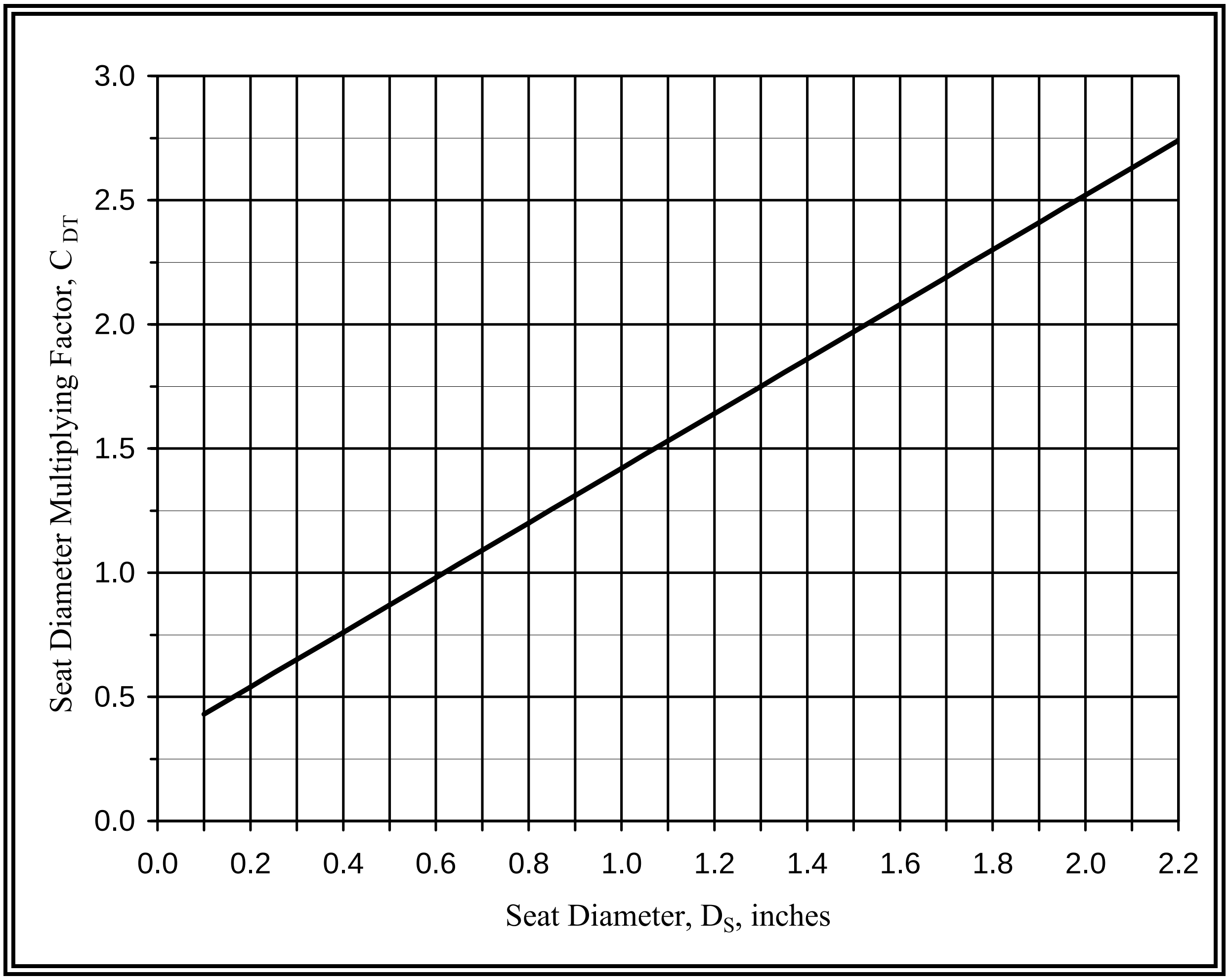
Figure 6.8 Surface Finish Multiplying Factor



$$C_S = 0.26 \left(\frac{9000}{S_S} \right)^{1.5}$$

where: S_S = poppet contact pressure, lbs/in^2

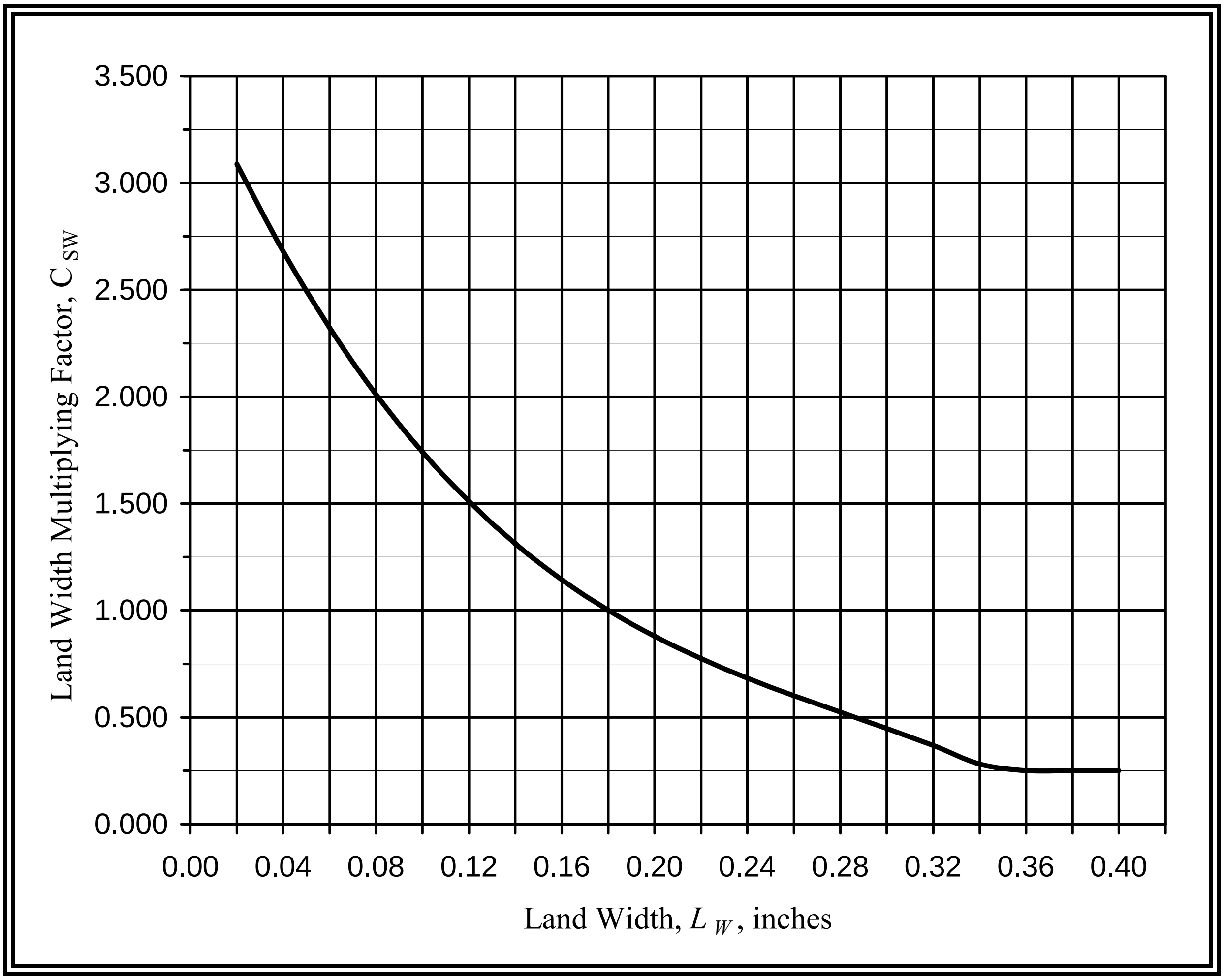
Figure 6.9 Contact Pressure Multiplying Factor



$$C_{DT} = 1.1D_S + 0.32$$

Where D_S = Seat Diameter, inches

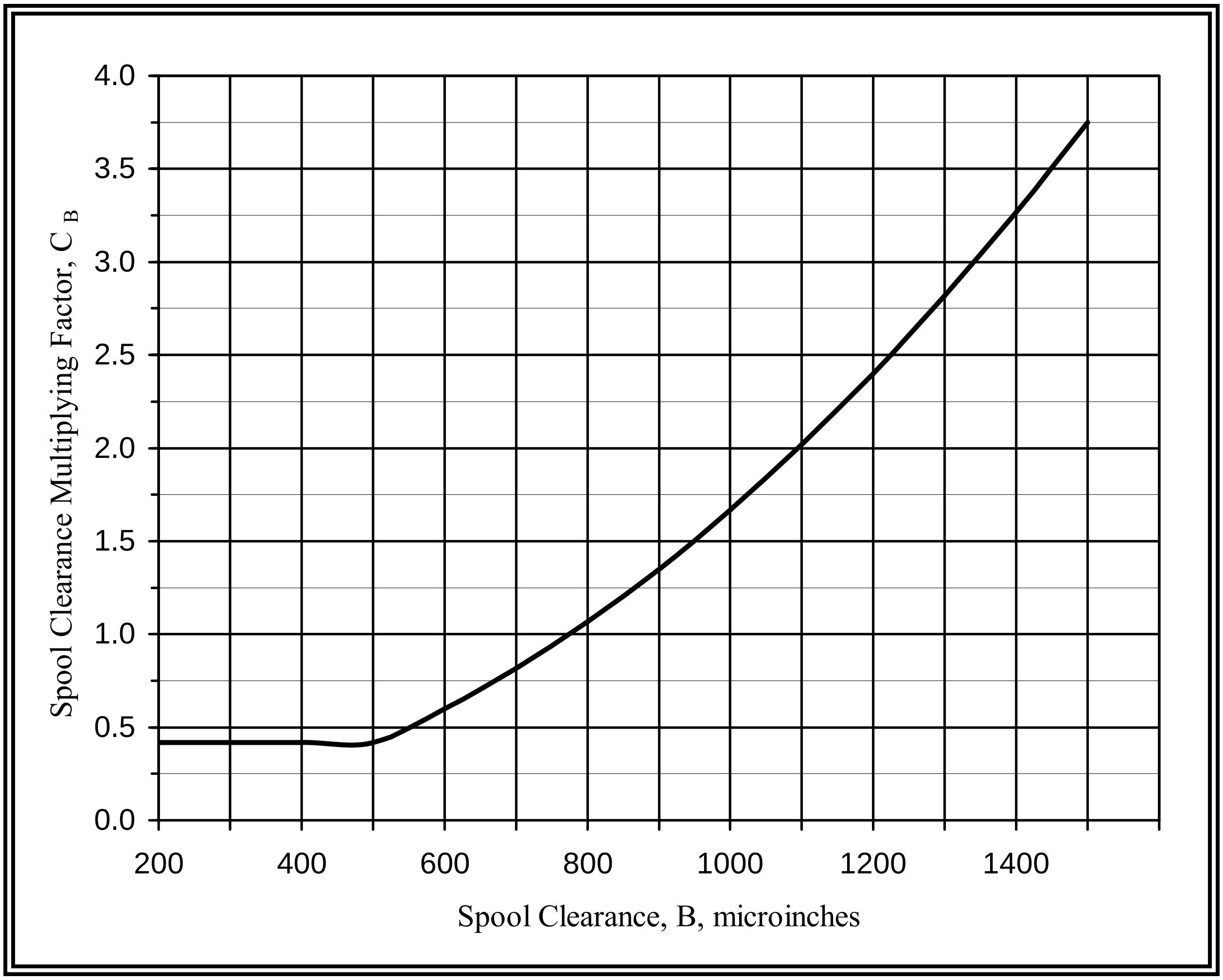
Figure 6.10 Seat Diameter Multiplying Factor



For $L_W \leq 0.34$, $\quad C_{SW} = 3.55 - 24.52\,L_W + 72.99\,L_W{}^2 - 85.75\,L_W{}^3$

For $L_W > 0.34$, $\quad C_{SW} = 0.25$

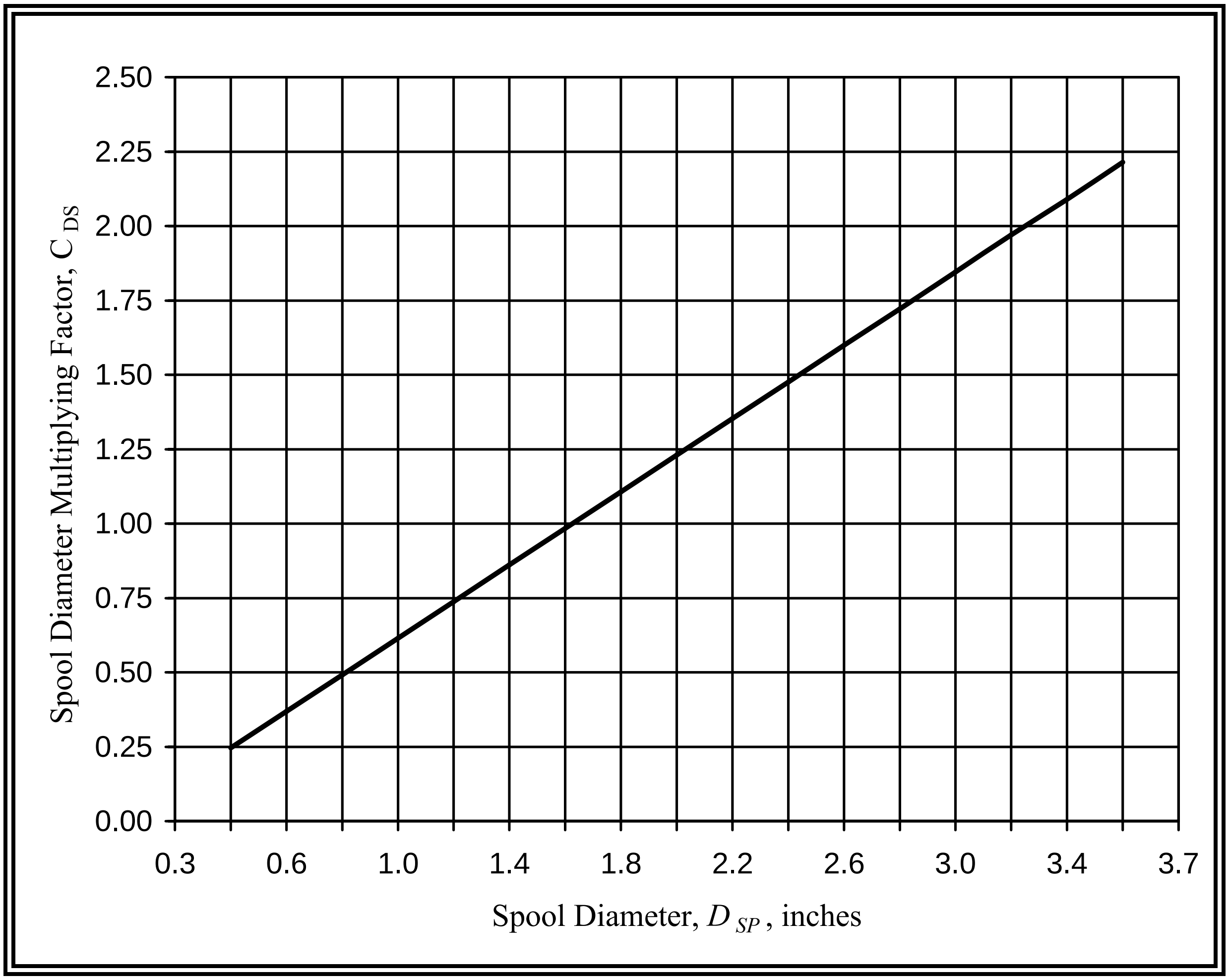
Figure 6.11 Land Width Multiplying Factor



For B < 500μin, $C_B = 0.42$

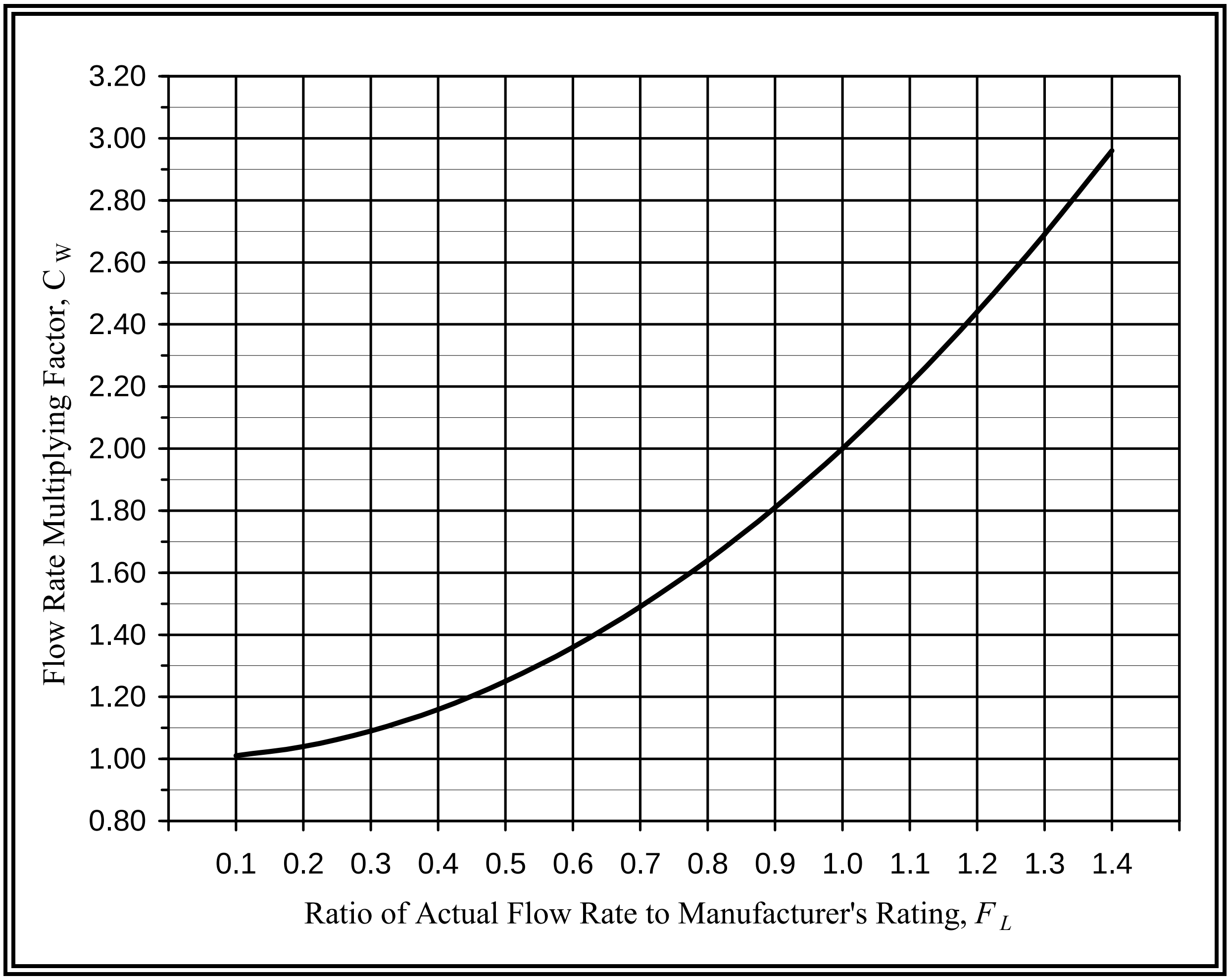
For B > 500μin, $C_B = \dfrac{B^2}{6 \ x \ 10^5}$

Figure 6.12 Spool Clearance Multiplying Factor



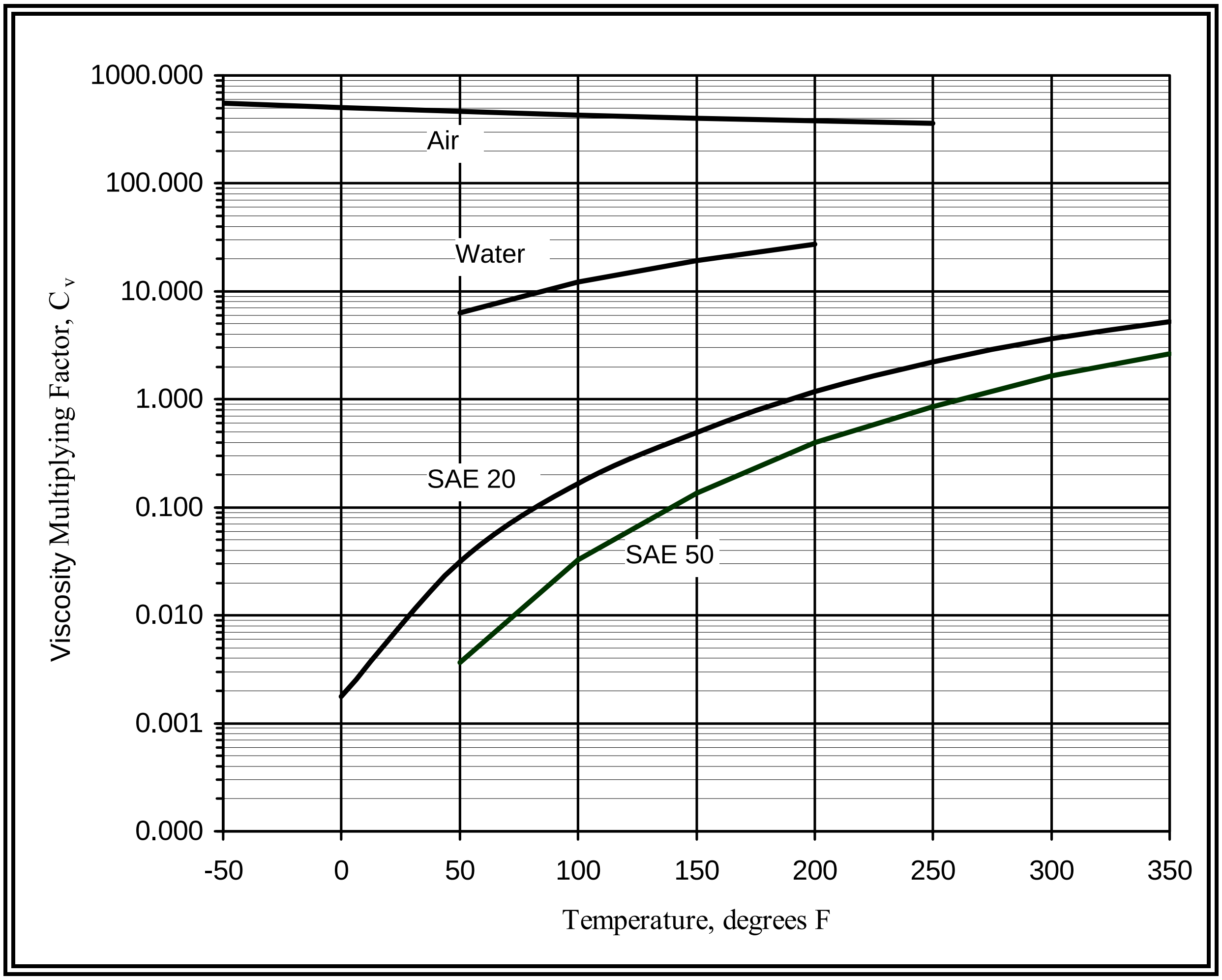
$$C_{DS} = 0.615\ D_{SP}$$

Figure 6.13 Spool Diameter Multiplying Factor



$$C_W = 1.0 + F_L^2$$

Figure 6.14 Flow Rate Multiplying Factor



See Table 6-6 for values of typical fluid viscosities

Figure 6.15 Fluid Viscosity Multiplying Factor

Table 6-4. Typical Surface Finishes for Manufacturing Processes

PROCESS	SURFACE FINISH, μin	PROCESS	SURFACE FINISH, μin
Lapping	2 - 16	Boring, turning	16 - 250
Polishing	4 - 16	Electron beam	32 - 250
Honing	4 - 32	Reaming	32 - 125
Grinding	4 - 64	Milling	32 - 250
Burnishing	8 - 16	Drilling	64 - 250

Table 6-5. Contaminant Multiplying Factor, C_N

TYPICAL QUANTITIES OF PARTICLES PRODUCED BY HYDRAULIC COMPONENTS	PARTICLE MATERIAL	NUMBER PARTICLES UNDER 10 MICRON PER HOUR PER RATED GPM (N10)
Piston Pump	steel	0.017
Gear Pump	steel	0.019
Vane Pump	steel	0.006
Cylinder	steel	0.008
Sliding action valve	steel	0.0004
Hose	rubber	0.0013

$$C_N = \left(\frac{C_O}{C_{10}}\right)^3 \bullet FR \bullet N_{10}$$

Where: C_o = System filter size in microns

C_{10} = Standard system filter size = 10 micron

FR = Rated flow rate, GPM

N_{10} = Particle size factor

Table 6-6. Fluid Viscosity/Temperature Multiplying Factor, C_ν for Typical Fluids

FLUID	C_ν								
	Fluid Temperature, °F								
	-50	0	50	100	150	200	250	300	350
Air	554.0	503.4	462.9	430.1	402.6	379.4	359.5	---	---
Oxygen	504.6	457.8	420.6	390.2	365.9	343.6	325.3	---	---
Nitrogen	580.0	528.0	486.5	452.6	424.3	400.0	379.6	---	---
Carbon Dioxide	---	---	0.7	0.8	0.9	0.9	---	---	---
Water	---	---	6.309	12.15	19.43	27.30	---	---	---
SAE 10 Oil	---	---	0.060	0.250	0.750	1.690	2.650	---	---
SAE 20 Oil	---	---	0.0314	0.167	0.492	1.183	2.213	2.861	5.204
SAE 30 Oil	---	---	0.0297	0.1129	0.3519	0.8511	1.768	2.861	4.309
SAE 40 Oil	---	---	0.0122	0.0534	0.2462	0.6718	1.325	2.221	3.387
SAE 50 Oil	---	---	0.0037	0.0326	0.1251	0.3986	0.8509	1.657	2.654
SAE 90 Oil	---	---	0.0012	0.0189	0.0973	0.3322	0.7855	1.515	2.591
Diesel Fuel	0.1617	0.7492	2.089	3.847	6.228	9.169	12.78	16.31	---
MIL-H-83282	0.0031	0.0432	0.2137	0.6643	1.421	2.585	4.063	0.6114	0.7766
MIL-H-5606	0.0188	0.0951	0.2829	0.6228	1.108	1.783	2.719	3.628	4.880

--- Data for these temperatures determined to be unreliable

$$C_\nu = \left(\frac{\nu_o}{\nu} \right)$$

Where: ν_o = 2 x 10^{-8} lbf-min/in^2

ν = Dynamic viscosity of fluid being used, lbf-min/in^2

Table 6-7. Friction Coefficient of Typical Materials used in Valve Designs

Material	Static Friction Coefficient	
	C_μ Dry	C_μ Lubricated
Steel on steel	0.8	0.5
Aluminum on steel	0.6	0.5
Copper on steel	0.5	0.4
Brass on steel	0.5	0.4
Cast iron on steel	0.4	---
Brass on nylon	0.3	---
Steel on nylon	0.3	---
Teflon on Teflon	0.05	0.04
Hard carbon on carbon	0.2	0.1
Copper on copper	1.3	0.8
Aluminum on aluminum	1.1	---
Nickel on nickel	0.7	0.3
Brass on brass	0.9	0.6

6.6 REFERENCES

In addition to specific references cited throughout Chapter 6, other references included below are recommended in support of performing a reliability analysis of valve assemblies.

5. Bauer, P., Glickmon, M., and Iwatsuki, F., “Analytical Techniques for the Design of Seals for Use in Rocket Propulsion systems”, Volume 1, ITT Research Institute, Technical Report AFRPL-TR-65-61 (May 1965)

22. Howell, Glen W. and Terry M. Weathers, Aerospace Fluid Component Designers' Handbook, Volumes I and II, TRW Systems Group, Redondo Beach, CA prepared for Air Force Rocket Propulsion Laboratory, Edwards, CA, Report AD 874 542 and Report AD 874 543 (February 1970).

27. May, K.D., “Advanced Valve Technology”, National Aeronautics and Space Administration, NASA Report SP-5019 (February 1965)

39. Shigley, J.E., Mischke, C.R., Mechanical Engineering Design, McGraw-Hill Book Co., NY, 1989

49. Kuhlmann-Wildorf, D., “Parametric Theory of Adhesive Wear in Uni-Directional Sliding”, Wear of Materials, pp. 402-413, American Society of Mechanical Engineers, New York (1983)

58. Parmley, R.O., Mechanical Components Handbook, McGraw-Hill Book Co., NY 1985

83. Handbook of Chemistry and Physics, 86th Edition, CRC Press, 2005

117. Jerry L. Lyons, Lyons’ Valve Designer’s Handbook, Van Nostrand Reinhold Company, 1982

This Page Intentionally Left Blank

CHAPTER **7**

BEARINGS

7.0 TABLE OF CONTENTS

7.1 INTRODUCTION

Bearings are used in mechanical designs to achieve a smooth, low-friction rotary motion or sliding action (linear motion) between two surfaces. Because there are so many different types of bearings in use for specific applications, it is extremely difficult to establish a base failure rate for an individual bearing design based on field performance data. In addition to the problem of locating failure rate data for an individual type of bearing, bearing analysis is also extremely difficult due to the large number of engineering parameters related to bearing design such as size, material properties, rigidity, design complexity, type of lubrication and load capacity.

Fortunately, bearings are among the few components designed for a finite life. Because of the fatigue properties of the materials used, some bearings are assigned a L_{10} life, which is the number of revolutions at a given load that 90 percent of a set of apparently identical bearings will complete or exceed before failure. To apply the L_{10} life to a specific application requires conversion of the “given load” to the equivalent radial load of the bearing for the intended application. Other factors that need to be identified in order to correlate the L_{10} life with the intended operating environment include actual lubrication properties, misalignment, velocity, type of loading, temperature and contamination levels. If L_{10} data for bearing life is available, procedures for estimating bearing reliability in this chapter utilize the manufacturer's published L_{10} life with multiplying factors to determine the failure rate for the intended operating conditions.

In many instances the manufacturer provides a rated dynamic load for the specific bearing to be used in the design. This basic dynamic load rating compared to the projected equivalent radial load for the bearing provides the L_{10} life for the bearing. Procedures in this chapter permit the projection of bearing failure rate for either source of data.

7.2 BALL BEARINGS

A ball bearing is a type of rolling element bearing that uses balls to maintain the separation between the moving parts of the bearing. Ball bearings are designed to reduce rotational friction and to support both radial and axial loads. At least two races are used in the design to contain the balls and transmit the loading through the balls. As one of the bearing races rotates, it causes the balls to rotate as well. Because the balls are rolling they produce very little friction. A typical ball bearing is shown in Figure 7.1

Ball bearings have a lower load capacity for their size than other kinds of rolling element bearings due to the smaller contact area between the balls and races. However, a ball bearing can tolerate some misalignment of the inner and outer races for higher application reliability and are generally used where there is likely to be excessive misalignment or shaft deflection.

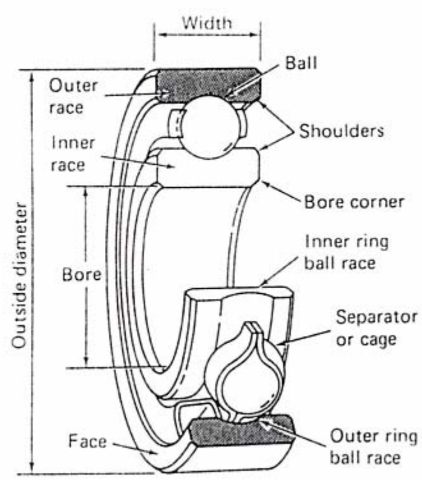
Ball bearings are usually classified as radial, thrust or angular contact. As their names imply, radial bearings are used for radial loads and thrust bearings for thrust loads. Angular contact bearings combine radial and thrust loads and are used where precise shaft location is needed. Most ball bearing designs originate from three basic types:

(1) Single-row radial - the most widely used ball bearing, a symmetrical unit capable of absorbing combined radial and thrust loads. It is not intended for pure thrust loads. Because this type of ball bearing is not self-aligning, accurate alignment between the shaft and housing bore is required.

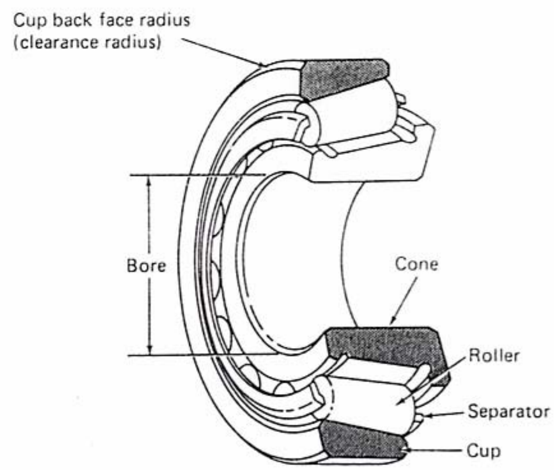
(2) Single-row angular contact - designed for combined radial and thrust loads where the thrust component may be large and axial deflection must be confined. A high

shoulder on one side of the outer ring is provided to take the thrust, and the shoulder on the other side is sufficiently high to make the bearing non-separable.

(3) Double-row angular contact - two single-row angular contact bearings built as a unit with the internal fit between balls and raceway fixed during assembly. These ball bearings have a known amount of internal preload built in for maximum resistance to deflection. They are very effective for radial loads where bearing deflection must be minimized.



Ball Bearing



Roller Bearing

Figure 7.1 Typical Bearing Configurations

All ball bearings have a life which is limited by the fatigue life of the material from which they are made and as modified by the lubricant used. In rolling contact fatigue, precise relationships between life, load, and design characteristics are difficult to predict and, therefore, the statistical L_{10} life based on a probability of survival is used with multiplying factors to adjust the L_{10} life to the actual conditions being projected.

7.3 ROLLER BEARINGS

Common roller bearings use cylinders of slightly greater length than diameter. Roller bearings typically have higher load capacity than ball bearings, but a lower capacity and higher friction under loads perpendicular to the primary supported direction. If the inner and outer races are misaligned, the bearing capacity often drops quickly compared to a ball bearing. A typical roller bearing is shown in Figure 7.1.

Because roller bearings have greater roller surface area in contact with inner and outer races, they generally support greater loads than comparably sized ball bearings. Cylindrical roller bearings are used to support pure radial loads. They are often used at one end of a highly loaded gear shaft with either tapered roller bearings or multiple-row matched ball bearings at the other end. Roller bearing life is drastically reduced by excessive misalignment or deflection; hence, when using roller bearings, the stack-up of tolerances contributing to misalignment and the shaft or housing deflections should be carefully considered. To compensate for some degree of misalignment or deflection and to carry heavy radial loads, roller bearings are crowned to prevent the phenomenon known as end loading. End loading invariably leads to a drastic reduction in bearing life. The crowning process distributes the load away from the roller ends and prevents excessive stress that could cause fatigue at the roller bearing ends.

7.4 DESIGN CONSIDERATIONS

The following paragraphs in this section describe the various features of bearing design to be considered in evaluating a mechanical assembly for reliability incorporating bearings.

7.4.1 Bearing Preload

Bearing preload is critical for the proper operation of a bearing. A bearing needs to be fitted with a shaft and there will be some clearance between the different parts of the bearing. To remove this internal clearance and create an interference fit, a preload is necessary. The preload provides a sufficient thrust load to push the bearing so that is secure in the groove and has no axial clearance. This elimination of clearance within the bearing eliminates vibration and noise of the bearing and also controls the rotational accuracy of the bearing.

7.4.2 Internal Clearance

Internal clearance, the clearance between the inner race and the shaft, is an important consideration in the design of ball and roller bearings, since improper internal clearance can drastically shorten the life of a bearing. A small internal clearance may limit the amount of misalignment that can be tolerated and can lead to heavily preloaded bearings. Excessive internal clearance will cause the load to be carried by too few rolling elements. The best practice is to ensure that under all conditions there will be a small positive internal clearance. Usually, the most significant factors to consider when determining mounted internal clearance of the bearing are the reduction of internal clearance due to shaft or housing fits and the effect of temperature on the housing/outer race interface diameters.

7.4.3 Bearing Race Creep

The creeping or spinning of bearing inner races on gear shafts is a fairly common, although not usually serious, problem in most drive systems. Lundberg and Palmgren developed fairly simple parametric calculations for the minimum fit to prevent creep with solid shafts, but there has been little if anything published on minimum press fits for hollow shafts, as are used in helicopter drive systems. Since an accurate mathematical solution to such a problem would be extremely difficult, the best approach seems to be a reliance on past experience. Sometimes it may not be possible to achieve the necessary press fit to prevent creep without introducing excessively high hoop stress in the bearing race. A common practice in this case is to use separate anti-rotation devices with a slotted bearing race. Although this practice is fairly effective with stationary races, it is seldom effective with rotating races.

7.4.4 Bearing Material

Because the wear rate of a material is proportional to the load applied to it, and inversely proportional to its hardness, one obvious way of reducing wear on bearing components is to increase the hardness at their surface. This is commonly accomplished by using hard coatings, such as electro-less nickel, hard anodised aluminum and thin dense chrome. In addition, other hard coatings, such as titanium carbide, carburising, and both carbo- and plasma nitriding are also widely used. Another advance in bearing technology has been the development of extremely clean bearing steels resulting from vacuum-melt processing. Vacuum-melt bearings have significantly increased the potential life of a bearing by one and one-half to two times the life of vacuum-degassed bearings. Bearings of such advanced materials as M-50 steel can offer even further improvement. Cost of the bearing is an important consideration and the application of the bearing considering such factors as loading and velocity must determine bearing selection.

7.4.5 Bearing Installation and Removal

The installation of bearings should be carefully considered during design not only to prevent assembly errors, but also to permit easy removal of the bearing without damaging it. Lead chamfers are often installed at bearing journals to facilitate installation. When specifying the breakout on the bearing corners, the shaft drawing should be checked to ensure that the maximum radius at the shaft shoulder will be cleared by the bearing. The height of the shaft shoulder should, if possible, be consistent with that recommended by bearing manufacturers. Where necessary, flats should be machined on the shaft shoulder so that a bearing puller can remove the bearing by contacting the inner race. Many bearings have been damaged in the past where the bearing puller could grab only the cage or rollers of the bearing. Where duplex bearings are used, the bearings should be marked so that the installer can readily determine the proper way for the bearings to be installed. Incorrectly installed duplex bearings will not properly react to the design loads. All bearings that can be separated should have the serial number clearly shown on all of the separable components. This will prevent the inadvertent mixing of components. Every assembly drawing that contains bearings should clearly explain in the drawing notes how the bearing should be installed. It is imperative that the mechanics building up this assembly have this information available.

7.5 BEARING FAILURE MODES AND MECHANISMS

The two main failure modes of a bearing are wear and fatigue. Ball and roller bearings which are well lubricated , perfectly sealed and running at moderate load and speed, will not exhibit sufficient wear that will cause a failure even after long service. In this case the bearing will eventually end its service life due to fatigue. Fatigue is the failure mode that normally creates the L_{10} bearing life. The operating conditions found in practice will almost certainly be less benign and wear must therefore be considered as a potential failure mode. Wear will be exhibited at the contact surfaces of the rings and rolling elements, at the sliding surfaces of the cage, and in roller bearings on the lip and roller faces. The process of wear begins with an increase in surface roughness of the raceway due to detached material particles. As additional material is removed from the contact area, the form of the raceway will be altered. Foreign particles may also enter the bearing through insufficient or worn seals, lubrication contaminants from other parts in a common lubrication system, or corrosion of the rolling and sliding surfaces due to water condensation as a result of temperature changes and corrosive liquids.

Roller bearings usually provide ample warning before complete failure by increasingly noisy operation and will usually fail from fatigue. Sliding bearings, on the other hand, often perform well up to moments before a catastrophic failure. It is very important to evaluate all bearing failure modes since a bearing failure emitting particles can cause severe shaft damage or other parts associated with the total design.

Common bearing failure modes, mechanisms and causes are listed in Table 7-1. One common mechanism of bearing failure is spalling, which is defined as subsurface chipping or breaking. The failure is usually caused by loading of the bearing exceeding the design load. Surface fatigue or peeling is a cracking and peeling of the surface metal. It is usually the result of poor lubrication or surface damage which interrupts the lubricant film. Scores and scratches are usually caused by hard particles being trapped in a bearing. This failure mechanism may also be caused by inadequate sealing, contaminants in the lubricant, or installation damage.

Smearing is surface damage resulting from unlubricated sliding contact within a bearing. Brinelling is the actual indentation of a rolling element under excessive load or impact that causes stresses beyond the yield point of the bearing material.

Fretting wear is usually caused by an improper fit between the bearing and the shaft or outer surface of the bearing. This allows movement of the race in relation to the housing or shaft. The surfaces then wear or score, thereby damaging the surfaces and preventing a firm, fixed contact.

Roller and tapered bearings have an additional failure mode defined as scuffing of the bearing surfaces. This failure mode is usually caused by bearing exposure to an excessive load for an extended period of time. The surfaces of the moving parts are scored or scratched, increasing the roughness of the surfaces, setting up stress concentrations and increasing friction. The scuffing also interferes with the normal lubricant film and increases the metal-to-metal contact during use.

Table 7-1. Typical Modes of Bearing Failure
(Reference 121)

FAILURE MODE	FAILURE MECHANISM	FAILURE CAUSE
Fatigue damage	- Spalling of ball/roller raceway - Brinelling - Smearing	- Heavy, prolonged load * - Excessive speed - Shock load - Excessive vibration
Noisy bearing	- Surface fatigue - Glazing - Microspalling of stressed surfaces	- Loss of lubricant - Housing bore out of round - Corrosive agents - Distorted bearing seals
Bearing seizure	- Crack formation on rings and balls or rollers - Skidding	- Inadequate heat removal capability - Loss of lubricant - High temperature - Excessive speed
Bearing vibration	- Scuffing - Fretting - Pitting of surfaces	- Misalignment - Housing bore out of round - Unbalanced/excessive load - Inadequate housing support
Presence of electric currents	- Pits on raceways and balls, corrosion	- Extensive pitting of surface caused by electric current

* Bearing failure can be caused by excessive shaft bending. See Chapter 20 to determine shaft deflection in relation to the maximum allowable.

Fatigue can occur due to cyclic loads normal to the bearing surface. Wiping occurs from surface to surface contact due to loss of sufficient lubrication film thickness. This malfunction can occur from under-rotation or from system fluid losses. Overheating is indicated by metal cracks or surface discoloration. Corrosion is frequently caused by the chemical reaction between the acids in the lubricants and the base metals in the bearing.

Severe performance requirements may affect the reliability of the bearings if there is a path of heat conduction from the machine or any friction creating components within it to the bearings (for example, brakes or clutches). This condition may cause a decrease in the bearing lubricant's operating viscosity and, consequently, a reduction in bearing life. A lubricant with a higher temperature rating should prevent leakage or excessive wear.

7.6 BEARING FAILURE RATE PREDICTION

Bearing life is usually calculated using the Lundberg-Palmgren method (Reference 53). This method is a statistical technique based on the sub-surface initiation of fatigue cracks through hardened bearing material. Most mechanical systems are not utilized precisely as the bearing manufacturer envisioned; therefore, some adjustment factors must be used to approximate the failure rate of the bearings under specific conditions.

Less than 10 percent of all bearings last long enough to fail due to normal fatigue (Reference 8). Most bearings will fail due to static overload, wear, corrosion, lubricant failure, contamination, or overheating. Experience has shown that the service life of a bearing is usually limited by either excessive wear or fatigue. Excessive wear occurs when the bearings are improperly installed or exposed to hostile operating environments. Inadequate lubrication, misalignment, contamination, shock, vibration, or extreme temperature all cause bearings to wear out prior to their estimated design life. In contrast, a bearing can be expected to perform adequately for the duration of its rated life, given proper operating conditions, until failure occurs due to fatigue.

Attempting to estimate the fatigue life of an individual bearing is not very practical because of the large number of design parameters to consider in relation to the sensitivity of the operating environment. Instead, statistical methods are used to rate bearings based on the results of large groups of the same type of bearing tested to failure under controlled laboratory conditions to establish a fatigue life rating. This rating, known as the L_{10} life, is defined as the number of hours that 90% of the bearings operating at their rated load and speed, can be expected to complete or exceed before exhibiting the first evidence of fatigue. It is important to consider the bearing application before using the published L_{10} life as a reliability estimate. For example, a bearing in a direct drive motor application may have a predicted life of 400,000 hours but the same bearing in a belt drive or pillow block application may have a life of 40,000 hours depending on loading.

Standard equations have been developed to extend the L_{10} rating to determine the statistical rated life for any given set of conditions. These equations are based on an exponential relationship of load to life.

$$L_{10} = \left(\frac{L_S}{L_A} \right)^y \quad \textbf{(7-1)}$$

where: L_{10} = Bearing life with reliability of 90%, millions of revolutions

L_S = Dynamic load rating of bearing, lbf

L_A = Equivalent radial load on bearing, lbf

y = Constant, 3.0 for ball bearings, 3.3 for roller bearings

The dynamic load rating is the dynamic load capacity of the bearing that is established during L_{10} life testing and can be found in manufacturer's catalogs. The equivalent radial load is the load the bearing will see in service and can be found in engineering drawings or calculated. Normally L_A will be approximately 0.5 L_S depending on the anticipated environmental and maintenance considerations of the design and can be used as a value for preliminary reliability estimates.

The L_{10} life can be converted to hours with the following:

$$L_{10\,\mathrm{h}} = \frac{10^6}{60n} \left(\frac{L_S}{L_A} \right)^y \quad \textbf{(7-2)}$$

where: $L_{10\,\mathrm{h}}$ = Bearing life (at 90% reliability), operating hours

n = Operating speed, revolutions/ min

In a ball or roller bearing, the rolling elements transmit the external load from one ring to the other. The external force load is generally composed of a radial load F_R and an axial load F_A and is distributed over a number of rolling elements. These two components combine to form the equivalent radial load. The equivalent radial load, L_A, is defined as the radial load producing the same theoretical fatigue life as the combined radial and thrust loads. All bearing loads are converted to an equivalent radial load. If only pure radial loads are involved, then the value for L_A is simply the radial load.

Except for the special case of pure thrust bearings, bearing ratings shown in manufacturers' catalogs are for radial loads. When thrust is present, an equivalent radial load must be determined before estimating reliability. Most bearing manufacturers provide methods of combining thrust and radial loads in accordance with ANSI standards to obtain an equivalent radial load. This relationship can be written as follows:

$$L_A = XF_R + YF_A \tag{7-3}$$

Where: L_A = Equivalent radial load, lbf

F_R = Radial load, lbf

F_A = Axial load, lbf

X = Radial factor relating to contact angle

Y = Thrust factor relating to contact angle, thrust load and the number and size of balls or rollers in the bearing

A bearing catalog will display separate tables of values to cover single-row, double-row, and angular-contact variations. X and Y can be obtained from the manufacturer of the bearing. References 44 and 83 provide design equations to calculate radial and thrust loads, and guidelines for estimating the radial and thrust factors. F_A should not exceed 30% of the radial load, F_R.

Substantial improvements in materials processing and manufacturing techniques have been made since the original development of the L_{10} concept for predicting bearing life. For instance, high-purity steels that are vacuum degassed or vacuum melted are now widely used for bearings. Also, bearing components are manufactured to tighter tolerances on geometry, and ball/raceways have finer finishes, which help to improve lubricating films. For reasons such as these, bearing manufacturers have modified their L_{10} ratings with certain adjustment factors.

Bearing life for an individual bearing or a group of identical bearings operating under the same conditions is the life associated with 90% reliability. Some bearing applications may require a consideration of reliability other than 90 percent. Bearings used in applications such as aircraft engines where safety is an issue, reliability may need to be above 90%. In a production line conveyer belt application it may be possible to use the average failure rate (L_{50}). Table 7-2 provides some typical life adjustment factors to modify the calculated failure rate in Equation (7-4). To be compatible with other components in a mechanical assembly based on MTBF values, the L_{50} value should be used.

For a specific bearing, a manufacturer may provide a dynamic load rating for individual bearings. In this case Equation (7-4) is used to determine the bearing failure rate for the intended operating environment. If the manufacturer provides an L_{10} life, that life will be based on testing at rated dynamic loading and 90% reliability. Also, the L_{10} life may be known from previous experience. In these situations the failure rate must be adjusted according to the actual dynamic load and Equation (7-5) is used to determine the bearing failure rate.

$$\lambda_{BE} = \lambda_{BE,B} \bullet C_R \bullet C_\nu \bullet C_{CW} \bullet C_t \bullet C_{SF} \bullet C_C \qquad \textbf{(7-4)}$$

Where: λ_{BE} = Failure rate of bearing, failures/million hours

$\lambda_{BE,B}$ = Base failure rate, failures/million hours

= 1 / $L_{10\,\mathrm{h}}$ where $L_{10\,\mathrm{h}} = \frac{10^6}{60n}\left(\frac{L_S}{L_A}\right)^y$ (Reference Equation (7-2))

C_R = Life adjustment factor for reliability (See Table 7-2)

C_ν = Multiplying factor for lubricant (See Section 7.6.1 and Figure 7.3)

C_{CW} = Multiplying factor for water contaminant level (See Section 7.6.2 and Figure 7.4)

C_t = Multiplying factor for operating temperature (See Section 7.6.3 and Figure 7.5)

C_{SF} = Multiplying factor for operating service conditions (See Section 7.6.4 and Table 7-3)

C_C = Multiplying factor for lubrication contamination level (See Section 7.6.5 and Table 7-4)

$$\lambda_{BE} = \lambda_{BE,B} \bullet C_Y \bullet C_R \bullet C_\nu \bullet C_{CW} \bullet C_t \bullet C_{SF} \bullet C_C \qquad \textbf{(7-5)}$$

Where: λ_{BE} = Failure rate of bearing, failures/million hours

$\lambda_{BE,B}$ = Base failure rate, failures/million hours
= 1 / $L_{10\,\mathrm{h}}$ where $L_{10\,\mathrm{h}}$ = rated life in hours (90% reliability)

C_Y = Multiplying factor for applied load (See Figure 7.2)

C_R = Life adjustment factor for reliability (See Table 7-2)

C_ν = Multiplying factor for lubricant (See Section 7.6.1 and Figure 7.3)

C_{CW} = Multiplying factor for water contaminant level (See Section 7.6.2 and Figure 7.4)

C_t = Multiplying factor for operating temperature (See Section 7.6.3 and Figure 7.5)

C_{SF} = Multiplying factor for operating service conditions (See Section 7.6.4 and Table 7-3)

C_C = Multiplying factor for lubrication contamination level (See Section 7.6.5 and Table 7-4)

The applied load will often be obtained from the bearing application such as the side loading of an actuator.

7.6.1 Lubricant Multiplying Factor

The lubricant factor, C_ν, is a function of the viscosity of the lubricant used in the bearing system at the intended operating temperature. C_ν can be expressed as:

$$C_\nu = \left(\frac{\nu_O}{\nu_L}\right)^{0.54} \qquad \textbf{(7-6)}$$

Where: ν_O = Viscosity of specification lubricant, lb-min/in^2

ν_L = Viscosity of lubricant used, lb-min/in^2

Multiplying factors for the effect of lubrication viscosity on the failure rate of a bearing are shown in Figure 7.2.

7.6.2 Water Contamination Multiplying Factor

Water contamination can have a detrimental effect on fatigue life. A water contamination multiplying factor which accounts for the reduction in fatigue life due to the leakage of water into the oil lubrication is shown in Figure 7.4. This factor is represented as C_{CW} and is represented by the following equations derived from data in Reference 19.

$$C_{CW} = 1.0 + 25.50\,CW - 16.25\,CW^2 \qquad \textbf{(7-7)}$$

Where: CW = Percentage of water in the lubricant

The C_{CW} multiplying factor will modify the base failure rate as shown in Equation (7-4) or (7-5). For bearings designed for water based lubricants $CW = 0$ and $C_{CW} = 1.00$

7.6.3 Temperature Multiplying Factor

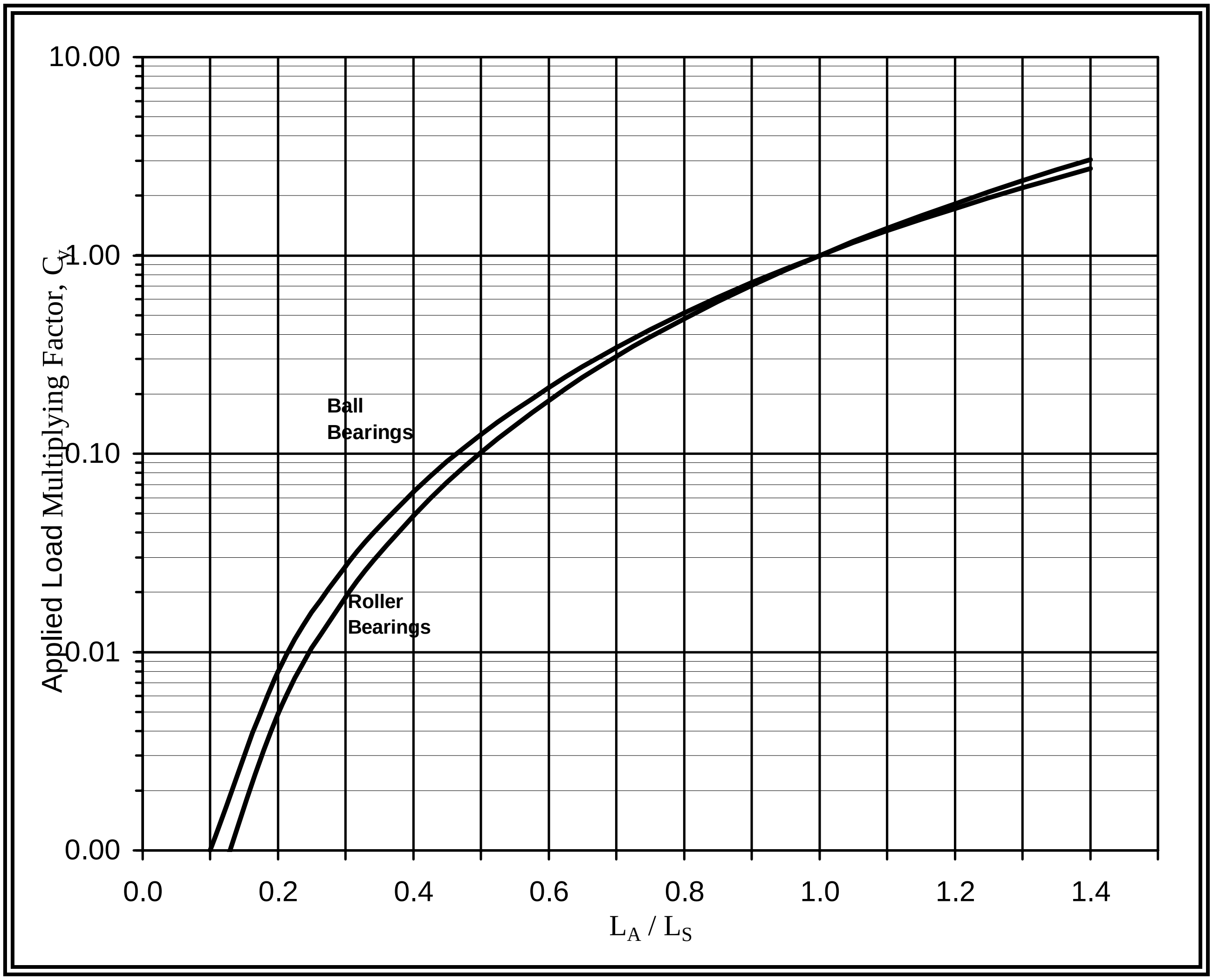
Excessive wear of a bearing is caused by exposure to hostile environments including extreme temperature. Excessive bearing heat can be generated by overloading the bearing. Heat will cause a decrease in the viscosity of the lubricant, causing more heat as it loses its ability to support the load. In addition, any residue on the bearing parts will harden at the elevated temperature destroying the ability of the grease or oil to lubricate the bearing. It will also introduce solid particles into the lubricant. Figure 7.5 provides a failure rate multiplying factor for bearing temperature.

7.6.4 Service Factor

The actual radial or axial load on the bearing may be greater than the calculated load because of vibration and shock present during operation of the equipment. A service factor can be used to adjust the failure rate for various operating conditions as shown in Table 7-3.

7.6.5 Lubricant Contamination Factor

The quality of the total equipment filtration system has a definite influence on bearing life. Hard particles in the system can induce permanent indentations, damaging the smooth surfaces of the bearing components. These rough surfaces then produce higher contact stresses resulting in shorter bearing life. Table 7-4 provides failure rate multiplying factors for the effect of lubricant contamination.



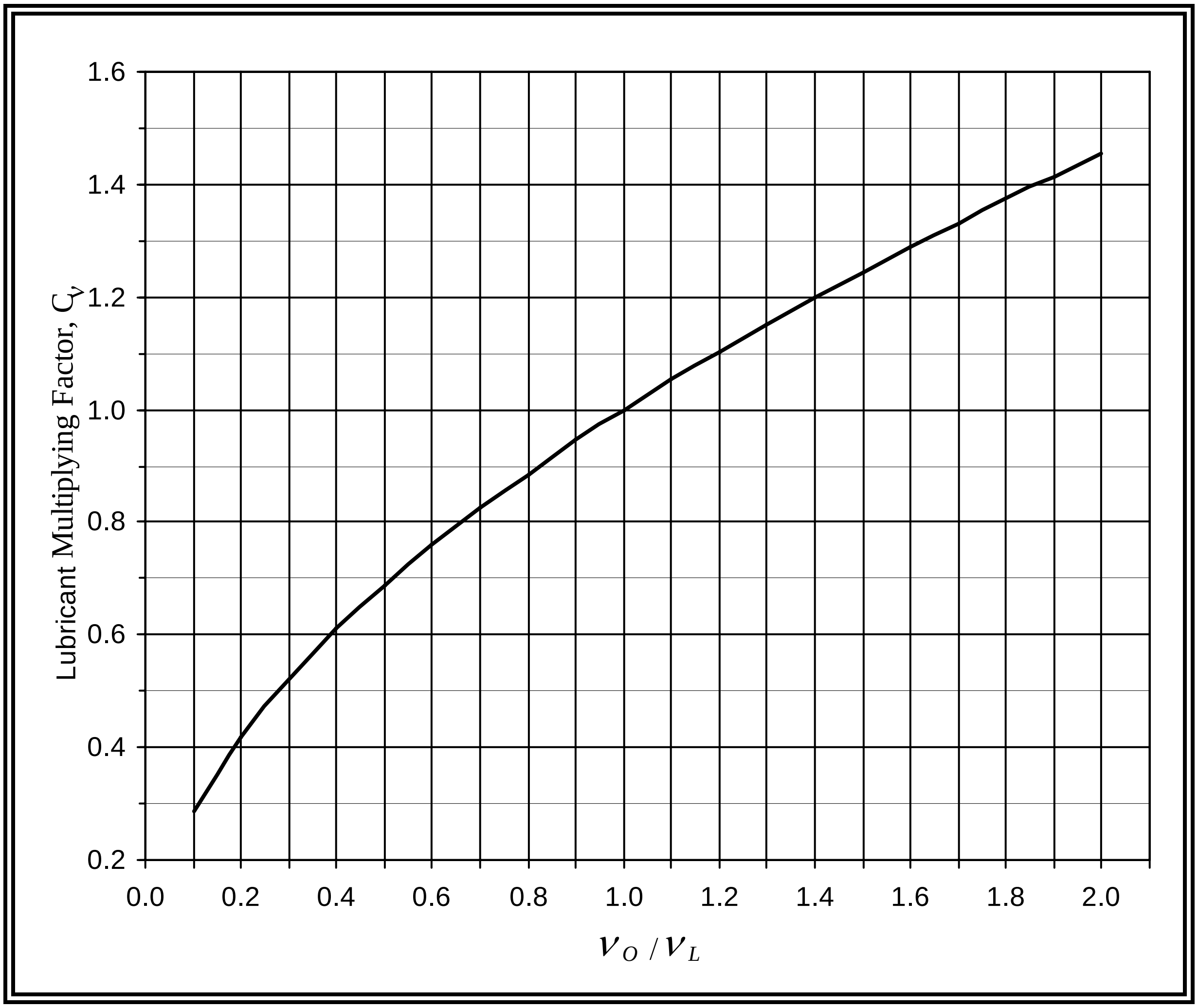
$$C_y = \left(\frac{L_A}{L_S}\right)^y$$

Where: L_A = Equivalent radial load, lbf

L_S = Dynamic load rating, lbf

y = 3.0 for ball bearings, 3.3 for roller bearings

Figure 7.2 Multiplying Factor for Applied Load

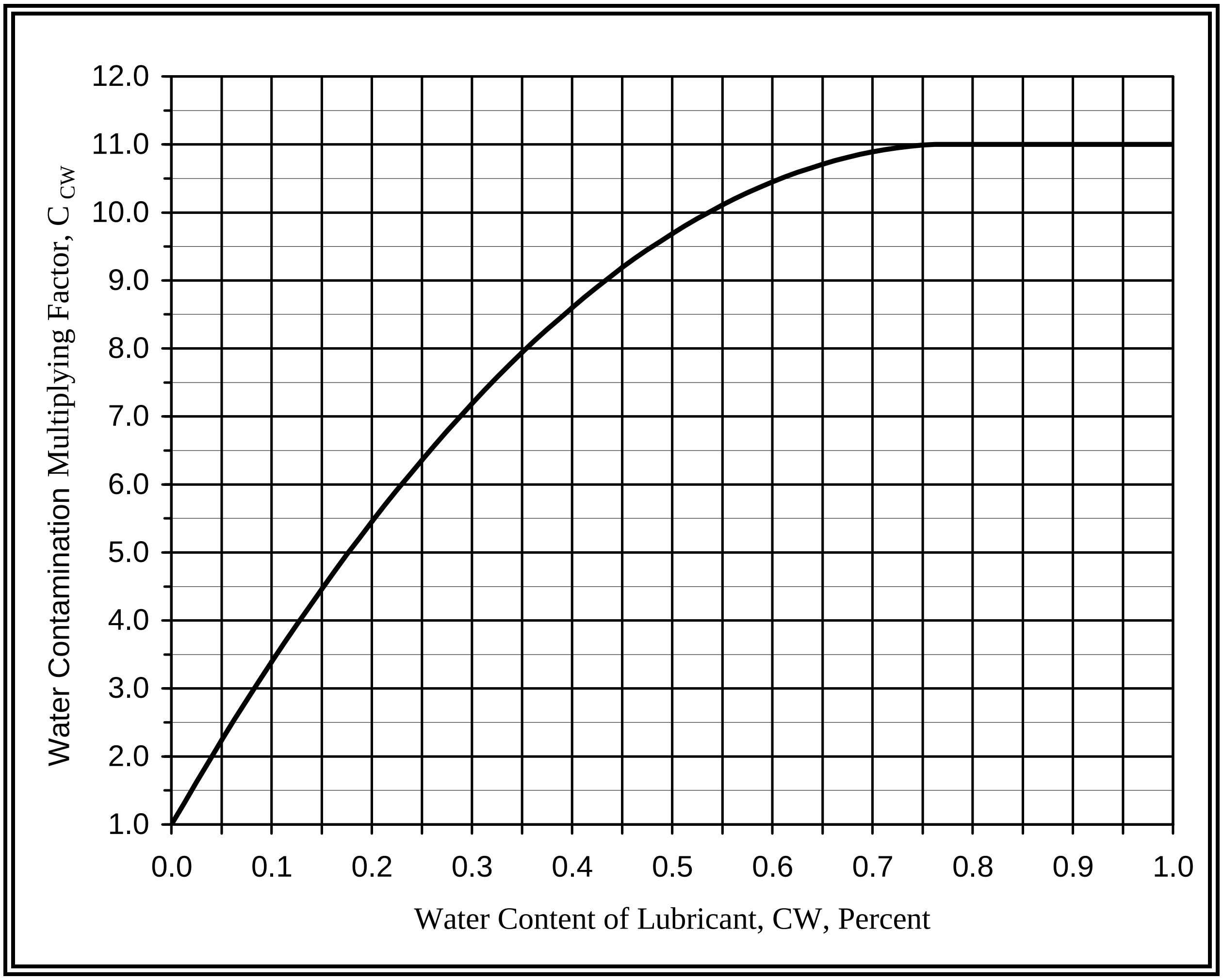


$$C_\nu = \left(\frac{\nu_O}{\nu_L}\right)^{0.54}$$

Where: ν_O = Viscosity of specification fluid

ν_L = Viscosity of lubricant used

Figure 7.3 Multiplying Factor for Bearing Lubricant

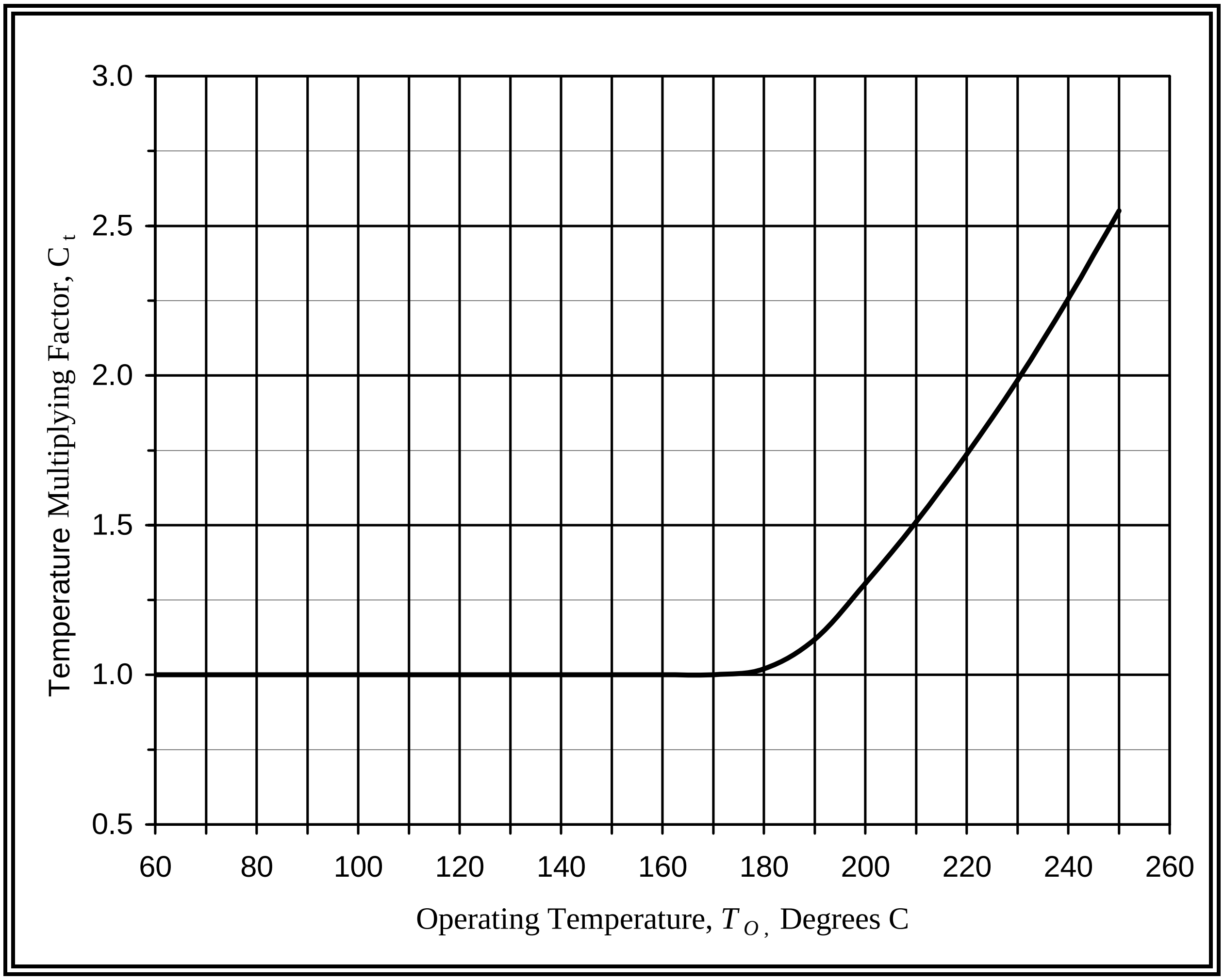


For $CW \leq 0.8$, $C_{CW} = 1.0 + 25.50\,CW - 16.25\,CW^2$

For $CW > 0.8$, $C_{CW} = 11.00$

Where: CW = Percentage of water in the lubricant

Figure 7.4 Water Contamination Multiplying Factor



$$C_t = 1.0 \text{ for } T_O < 183^{o}\text{C}$$

$$C_t = \left(\frac{T_O}{183}\right)^3 \text{ for } T_O \geq 183^{o}\text{C}$$

Where: T_O = Operating Temperature of the Bearing

Figure 7.5 Operating Temperature Multiplying Factor

Table 7-2. Life Adjustment Factor for Reliability, C_R

Reliability R %	L_a	Life adjustment factor C_R*
90	L_{10}	1.00
95	L_5	1.62
96	L_4	1.88
97	L_3	2.29
98	L_2	3.01
99	L_1	4.79
50	L_{50}	0.29

$$*C_R = \frac{0.223}{\left[\ln\left(\frac{100}{R}\right)\right]^{2/3}}$$

Table 7-3. Bearing Service Factors
(References 57 & 119)

Type of Application	Service Factor, C_{SF}	
	Ball Bearing	**Roller Bearing**
Uniform and steady load, free from shock	1.0	1.0
Normal operation, light shock load	1.5	1.0
Moderate shock load	2.0	1.3
Heavy shock load	2.5	1.7
Extreme and indeterminate shock load	3.0	2.0
Precision gearing	1.2	
Commercial gearing	1.3	
Toothed belts		1.2
Vee belts		1.8
Flat belts		3.0

Table 7-4. Bearing Contamination Level
(Reference 112)

Contamination Condition	Service Factor, C_C	
	Bearing diameter < 100 mm	**Bearing diameter > 100 mm**
Extreme cleanliness- particle size approx. lubricant film thickness (laboratory conditions)	1.0	1.0
High cleanliness – oil filtered through fine filter $\leq$ 10 micron	1.4	1.2
Normal cleanliness – slight contamination in lubricant	1.8	1.4
Slight contamination –slight contamination in lubricant – hard particles > 10 micron	2.5	2.0
Severe contamination – course filtering, no integral seals	5.0	3.3

7.7 REFERENCES

In addition to specific references cited throughout Chapter 7, other references included below are recommended in support of performing a reliability analysis of bearings.

8. Block, H. and D. Johnson, "Downtime Prompts Upgrading of Centrifugal Pumps", Chemical Engineering Magazine, pp. 35-38 (25 Nov 1985)

19. Hindhede, U., et al, Machine Design Fundamentals, John Wiley & Sons, NY, 1983

44. Sibley, L.B., "Rolling Bearings", Wear Control Handbook, M.B. Peterson and W. O. Winer, Eds., Sect. 5, pp 699-726, American Society of Mechanical Engineers, New York (1980)

50. Bentley, R.M. and D.J. Duquette, Environmental Considerations in Wear Processes, "Fundamentals of Friction and Wear of Materials", pp. 291-329, American Society of Metals, Metals Park, Ohio (1981)

53. Rumbarger, John H., “A Fatigue Life and reliability Model for Gears”, American Gear Manufacturers Association Report 229.16 (January 1972)

57. Deutschman, A.D., et al, Machine Design; Theory and Practice, MacMillan Publishing Co, NY, 1975

58. Parmley, R.O., Mechanical Components Handbook, McGraw-Hill Book Co., NY 1985

83. “Ball and Roller Bearings, Theory, Design and Application”, John Wiley & Sons, ISBN 0 471 26283 8

112. NSK Product Guide – Bearings 2008, NSK Americas, Inc.

116. Dr. Gerhard G. Antony, “How to Determine the MTBF of Gearboxes”, Power Transmission Engineering, April 2008

119. Jack A. Collins, Henry Busby and George Stabb, Mechanical Design of Machine Elements and Machines, the Ohio State University, John Wiley & Sons, 2010

120. Tyler G. Hicks, Handbook of Mechanical Engineering Calculations, McGraw-Hill, 2006

121. Oil Analysis, NAEC-92-153, Naval Air Engineering Center, Lakehurst, NJ 23 August 1982

122. “How dirt and Water Slash Bearing Life”, Richard C. Beercheck, Machine Design, July 6, 1978

131. “Tidal Current Turbine Reliability: Power Take-off Train Models and Evaluation, C. Iliev and D. Val, Third International Conference on Ocean Energy, October 2010

CHAPTER 8

GEARS AND SPLINES

8.0 TABLE OF CONTENTS

8.1 INTRODUCTION

The reliability of a gear along with the other gearbox components is an extremely important consideration in the design of any power-transmission system, ensuring that the required loads can be handled over the intended life of the system. Some general design constraints and requirements need to be given special attention because of their potential impact on the long-term reliability of the total system. One is the operating power spectrum and determining the potential requirements for growth. Another is that changing requirements may cause a configuration change where a misalignment could cause vibration that could set up stresses and lead to fatigue failure. Another example is the lubrication system if included as part of the gearbox design, assuring that the lubrication capacity and filtration system is adequate considering the operating environment of the total system. The lubricant flow should be designed so that the particles within the filtration system are removed prior to reentry into the gearbox area.

Noise and vibration can affect reliability, not only of the gearbox itself, but also of associated components within the complete power-transmission system. Hence, every effort should be made to configure a gearbox that is as quiet and as vibration-free as possible. The design analysis should also include the assurance that critical speeds and gear clash resonance frequencies, which may reinforce each other, are avoided.

In most gearbox applications, especially in airborne systems, weight is usually a constraining and, in some cases, the controlling factor. In general, overdesign means higher reliability, but in weight critical systems, overdesign in one area requires underdesign elsewhere; thereby, defeating the purpose of the overdesign. For example, bearing life should never be sacrificed in the design because bearings are likely to be the main drivers in determining gear system reliability.

When a gearbox is exposed to overstress, several conditions occur that greatly affect the failure rate. For example, bolted gear flanges will be subject to fretting and high loads will cause bevel gears to shift patterns, making tooth breakage a likely occurrence. And spur gears can develop scuffing lines increasing the roughness of the surface as loads are increased. A thorough reliability analysis includes a detailed analysis of the impact on each part or component and the assurance that the gear system used in a system does not exceed the specification load.

8.2 GEAR DESIGNS

The involute gear profile is the most commonly used system for gearing. In an involute gear, the profiles of the teeth are involutes of a circle (contour of gear teeth curves inward). A very common gear design is the spur gear. Spur gears are cylindrical in form and operate on parallel axes with the teeth straight and parallel to the axes. A typical spur gear arrangement is shown in Figure 8.1. In general, the reliability of drive train spur gears is extremely high due to present design standards. There are, however, some considerations that should be addressed because in evaluating the reliability of a gear for specific purposes.

The initial design of the spur gear mesh is normally one of standard proportions and equal tooth thickness for both pinion and gear. This initial design is then altered to achieve an optimum configuration to achieve recess action and a balanced bending stress in pinion and gear. A recess-action gear mesh, shown in Figure 8.2, has a long addendum pinion and short addendum gear. A recess-action mesh is quieter and smoother running than standard mesh and has a much lower tendency to score due to better lubrication within the mesh.

Although the advantage of having balanced bending stresses on a pinion and gear is primarily lower weight, it does have an indirect effect on reliability. As stated earlier, whenever there is an inefficient use of weight, reliability is compromised somewhat. For example, even a fraction of a pound saved in the optimization of a spur gear mesh could be applied to a bearing where the life could perhaps be doubled. While overemphasis of weight reduction can be detrimental to reliability, the carrying of excess weight can have a far-reaching effect; therefore, a balanced gear system must be the goal for efficient and reliable systems. Fortunately, it is usually a simple task to achieve recess-action and balanced bending stress in most spur gear designs. This is accomplished by experimentally shifting the length of contact up the line of action toward the driver gear, while increasing the circular tooth thickness of the pinion and decreasing that of the gear.

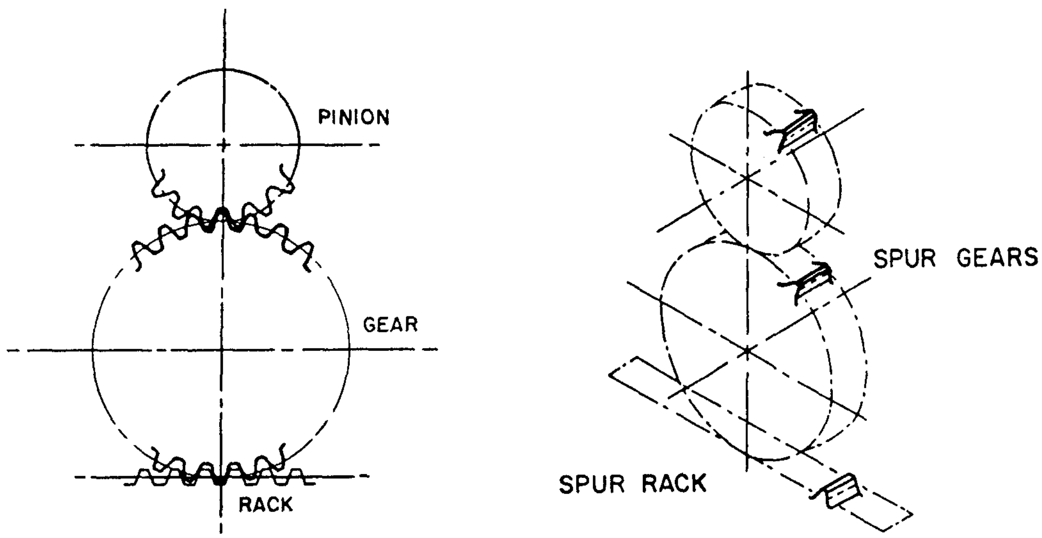


Figure 8.1 Typical Spur Gear Arrangement

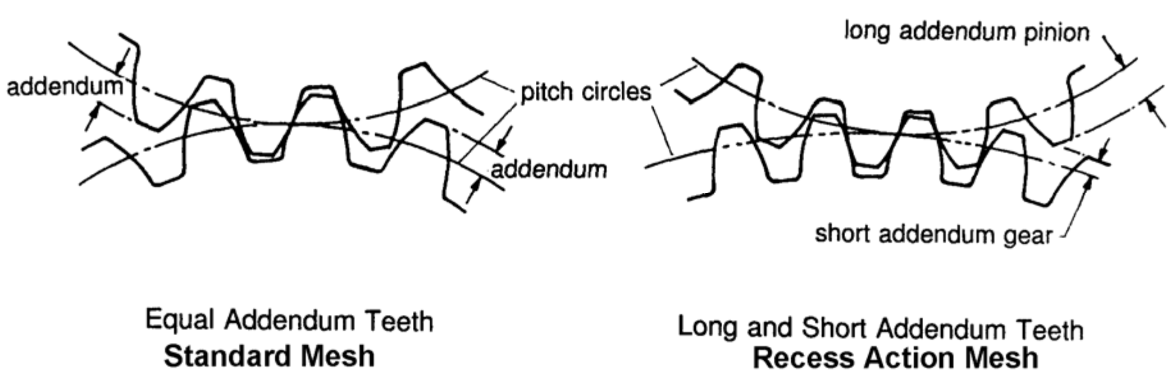


Figure 8.2 Gear Mesh Arrangements

An important parameter to evaluate lubrication effectiveness is the lubricant film thickness. The equation below is a non-dimensional expression for lubricant film thickness:

$$H_L = \frac{2.65\, G^{0.54}\, U^{0.7}}{W^{0.13}} \quad \textbf{(8-1)}$$

Where: H_L = Dimensionless film thickness factor

G = Viscosity and material parameter

U = Speed parameter

W = Load parameter

Since it is often difficult to obtain these parameters directly, this expression will only be used for a qualitative evaluation. The major impact of the formula is to establish the dependence of lubricant film thickness (H_L) from the various parameters.

To ensure smooth operation of the gear mesh under load, it is generally the practice to modify the involute profile, usually with tip relief, to correct for the deflection of the gear tooth under load. The various parameters affecting gear wear are shown in Figure 8.3. Too little tip relief will result in the gear teeth going into mesh early and going out of mesh late. This condition results in higher dynamic loads with the accompanying stress, vibration, noise and possible non-involute contact that can lead to hard-lines, scuffing or scoring of gear teeth. Too much tip relief lowers the contact ratio of the gear set and again can result in less than optimum performance with respect to stress, vibration, and noise.

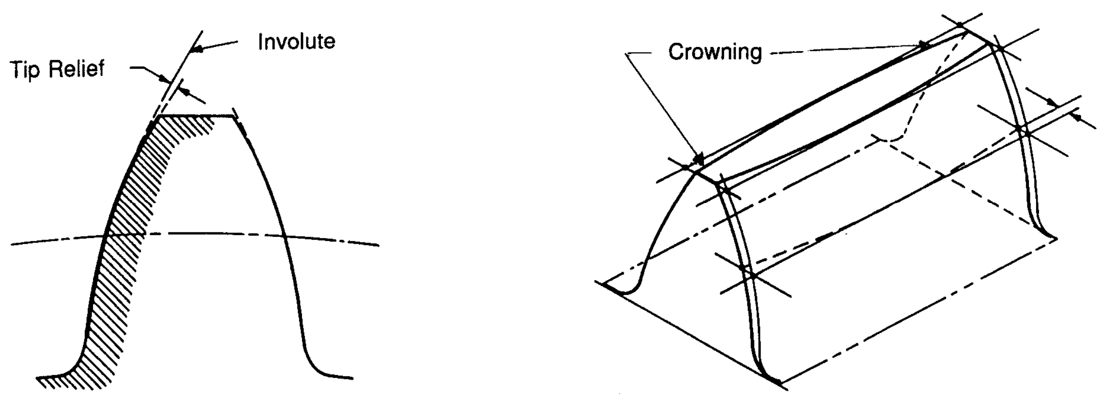


Figure 8.3 Typical Gear Tooth Designs

Crowning is generally applied to spur gears to ensure full contact across the face of the gear without end loading. With insufficient crowning, end loading will occur and result in higher than predicted vibrational stresses.

Helical gears, shown in Figure 8.4, are usually quieter and have a greater load-carrying capacity per inch of face than spur gears. The major disadvantage is that a thrust load is introduced along the gear shaft, thereby requiring larger and stronger bearings. Analysis of helical gears is very similar to that used for spur gears. The stress analysis is performed using an equivalent spur tooth. AGMA standard procedures have been developed for strength analysis of spur and helical gears.

Spline gears are used to transfer torque between shafts and flanges, gears and shafts, and shafts and shafts. A typical spline arrangement is shown in Figure 8.4. A splined shaft usually has equally spaced teeth around the circumference, which are most often parallel to the shaft's axis of rotation. These teeth can be straight sided, an involute form or included angle form (serrations). The teeth on a straight sided spline have an equal tooth thickness at any point measured radially out from the axis of rotation. Conversely, the internal parallel spline keys are integral to the shaft and equally spaced around the circumference. The involute spline has equally spaced teeth but they have an involute form like a gear tooth. The teeth do not have the same proportions as a gear tooth. They are shorter in height to provide greater strength. Involute splines provide a more smooth transition through a radius as opposed to straight sided splines decreasing the possibility of fatigue cracks. Involute splines are usually crowned. The serration type of spline has a tooth that is non-involute. The male teeth are in the form of an included angle, with the female serration having spaces of the same included angle. Serrations are generally used on smaller diameter shafts where the included angle form permits more teeth to be used on a smaller circumference, providing a greater contact area.

The most common problem associated with splines is wear due to fretting; particularly, with loose splines. It is important that the reliability analysis confirms that the bearing stress is below the allowable limit. Tight splines should have an adequate length pilot to react with bending loads. Lubrication is a particular factor in the reliability of loose splines and, if at all possible, should remain flooded with oil at all times. Crowning is usually required to prevent excessive wear.

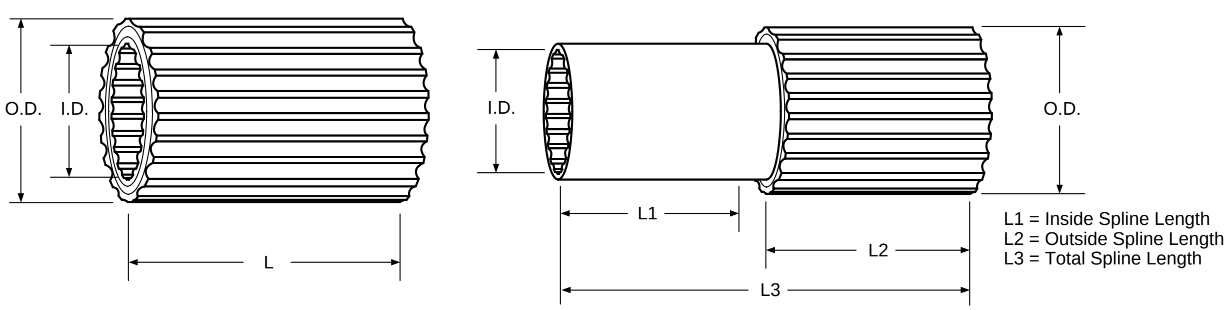


Figure 8.4 Typical Spline Gears

8.3 GEAR AND SPLINE FAILURE MODES

The definition of failure for a gear or spline is not very precise because of the wearing pattern of the gear. During the initial period of operation, minor imperfections in the gear will be smoothed out, and the working surfaces will polish up, provided that proper conditions of installation, lubrication and application are being met. Under continued normal conditions of operation, the rate of wear will be negligible. A gear has failed when it can no longer "efficiently" perform the job for which it was designed. Thus the definition of failure may be determined by the amount of vibration, noise, or results of a physical inspection.

The more common modes of gear and spline failure are wear, surface fatigue, plastic flow and breakage. The following paragraphs describe each of these failure modes. Table 8-1 provides a summary of possible failure modes for gears and splines. In the shear mode, the gear immediately ceases to transmit power while in the wear mode it degrades gradually before complete failure

8.3.1 Wear

Wear is the removal of metal, worn away normally in a uniform manner from the contacting surface of the gear teeth. The first stage of wear is the polishing phase during wear-in of the gear when asperities of the contacting surfaces are gradually worn off until very fine, smooth surfaces develop. Moderate wear of the gear occurs during its design life. Excessive wear occurs when the gear is operating in or near the boundary lubrication regime where the load is being carried by surface asperities rather than by the lubricant. Excessive wear is similar to moderate wear but the gear teeth are experiencing a considerable amount of material being removed from the surfaces. Contamination in the lubrication system can accelerate this wear. During this phase the tooth-surface profile is being destroyed so that high dynamic loads are encountered which in turn accelerates the wear rate until the gear is no longer usable.

Specific types of gear wear include abrasive wear caused by an accumulation of abrasive particles in the lubrication; corrosive wear caused by water or additives in the lubricating oil resulting in a deterioration of the gear surface from chemical action; and scoring caused by failure of the lubricant film due to overheating resulting in metal-to-metal contact and alternate welding and tearing of the surface metal.

8.3.2 Surface Fatigue

Surface fatigue is the failure of gear material as a result of repeated surface or subsurface stresses that are beyond the endurance limit of the material. Surface fatigue results in removal of metal and the formation of cavities. This pitting can be caused by the gear surfaces not properly conforming to each other due to lack of proper alignment. Spalling is similar to pitting except that the pits are larger, shallower and very irregular in shape. Spalling is usually caused by excessively high contact stress levels. The edges of the initial pits break away and large irregular voids are formed.

Table 8-1. Gear Failure Modes

FAILURE MODE	FAILURE CAUSE	FAILURE EFFECT
Pitting	Cyclic contact stress transmitted through lubrication film	Tooth surface damage
Root fillet cracking; Tooth end cracks	Tooth bending fatigue	Surface contact fatigue and tooth failure
Tooth shear	Fracture	Tooth failure
Scuffing	Lubrication breakdown	Wear and eventual tooth failure
Plastic deformation	Loading and surface yielding	Surface damage resulting in vibration, noise and eventual failure
Spalling	Fatigue	Mating surface deterioration, welding, galling, eventual tooth failure
Tooth bending fatigue	Surface contact fatigue	Tooth failure
Contact fatigue	Surface contact fatigue	Tooth failure
Thermal fatigue	Incorrect heat treatment	Tooth failure
Abrasive wear	Contaminants in the gear mesh area or lubrication system	Tooth scoring, eventual gear vibration, noise

8.3.3 Plastic Flow

Plastic flow is the cold working of the tooth surfaces, caused by heavy loads and the rolling and sliding action of the gear mesh. The result of these high contact stress levels is the yielding of the surface and subsurface material and surface deformation. This same failure mode in a slow speed operation combined with an inadequate lubricating film can result in a rippled surface. The cold working action of the gear surface leads to deteriorated gear box operation.

8.3.4 <u>Breakage</u>

Breakage is a failure caused by the fracture of a whole tooth or a substantial portion of a tooth. Gear overload or cyclic stressing of the gear tooth at the root beyond the endurance limit of the material causes bending fatigue and eventually a crack originating in the root section of the gear tooth and then the tearing away of the tooth or part of the tooth. Gear overload can be caused by a bearing seizure or sudden misalignment of a failed bearing, system dynamic loading, or contaminants entering the mesh area. Stress risers can sometimes subject the gear to higher root stress levels than originally predicted. These stress risers include such abnormalities as notches in the root fillet and small cracks from the heat treating or grinding process.

8.4 GEAR RELIABILITY PREDICTION

The previous paragraphs have provided an insight into the specific characteristics and failure modes of the more common gear types. Gears, fortunately, are designed to a specification and through the standardization of the American Gear Manufacturer's Association (AGMA), gears of various manufacturers and designs can be compared. The best approach for the calculation of failure rates for a gear system is to use the manufacturer's specification for each gear as the base failure rate, and adjust the failure rate for any difference in the actual usage from that purpose for which the gear was designed. If the manufacturer’s failure rate is not available, a gear or spline is usually designed for a life of 100 million revolutions for the particular application, the application including such factors as operating speed, temperature, lubrication and torque. Either way, the gear failure rate can be expressed as:

$$\lambda_G = \lambda_{G,B} \bullet C_{GS} \bullet C_{GP} \bullet C_{GA} \bullet C_{GL} \bullet C_{GT} \bullet C_{GV} \qquad \textbf{(8-2)}$$

Where: λ_G = Failure rate of gear under specific operation, failures/million operating hours

$\lambda_{G,B}$ = Base failure rate of gear, failures/million operating hours *

C_{GS} = Multiplying factor considering speed deviation with respect to design (See Section 8.4.1 and Figure 8.6)

C_{GP} = Multiplying factor considering actual gear loading with respect to design (See Section 8.4.2 and Figure 8.7)

C_{GA} = Multiplying factor considering misalignment (See Section 8.4.3 and Figure 8.8)

C_{GL} = Multiplying factor considering lubrication deviation with respect to design (See Section 8.4.4 and Figure 8.9)

C_{GT} = Multiplying factor considering the operating temperature (See Section 8.4.5 and Figure 8-10)

C_{GV} = Multiplying factor considering the AGMA Service Factor (See Section 8.4.6 and Table 8-1)

* $\lambda_{G,B}$ can usually be obtained from the manufacturer and it will be expressed in failures/operating hour at a specified speed, load, lubricant, and temperature. Also, a service factor will usually be provided to adjust the normal usage factor for certain specific conditions found in typical industries. These factors include such things as vibration, shock and contaminates. Failure data for similar equipment may also be available or a base failure rate of one failure/10^8 revolutions can be used:

[$\lambda_{G,B}$ = (RPM x 60) x 1 / design life (revolutions)]

.

8.4.1 Velocity Multiplying Factor

The speed deviation multiplying factor, C_{GS}, can be calculated using the relationship established in Equation (8-1) noting that the lubrication film thickness varies with speed to the 0.7 power. Therefore:

$$C_{GS} = k + \left(\frac{V_o}{V_d}\right)^{0.7} \quad \textbf{(8-3)}$$

Where: V_o = Operating Speed, RPM

V_d = Design Speed, RPM

k = Constant, 1.0

See Figure 8.6

8.4.2 Gear Loading Multiplying Factor

The gear loading multiplying factor, C_{GP}, has a lubricant and a fatigue impact. From Equation (8-1), the impact of load or torque can be expressed as:

$$\text{Change in expected life} = \frac{k}{W^{0.13}} = \frac{k}{\left(\frac{L_O}{L_D}\right)^{0.13}} \quad \text{(lubricant impact)} \quad \textbf{(8-4)}$$

Where: $W =$ Load Parameter

$L_O =$ Operating Load, lbs

$L_D =$ Design Load, lbs

$k =$ Constant

and the expression for torque or load on the fatigue rate of the component is:

$$\text{Change in expected life } = k\left(\frac{L_D}{L_O}\right)^{4.56} \quad \text{(fatigue impact)} \tag{8-5}$$

Combining Equations (8-4) and (8-5):

$$C_{GP} = \left(\frac{L_O / L_D}{k}\right)^{4.69} \tag{8-6}$$

Where: $k =$ Constant, 0.50

See Figure 8.7

8.4.3 Misalignment Multiplying Factor

The alignment of gears, bearings and shafts can be critical in the operation of a system. C_{GA}, the misalignment factor, can be expressed as:

$$C_{GA} = \left(\frac{A_E}{0.006}\right)^{2.36} \tag{8-7}$$

Where: $A_E =$ Misalignment angle in radians

See Figure 8.8

8.4.4 Lubricant Multiplying Factor

The lubricant factor C_{GL} is a function of the viscosity of the lubricant used in a gear system. C_{GL} can be expressed as:

$$C_{GL} = \left(\frac{\nu_O}{\nu_L}\right)^{0.54} \qquad \textbf{(8-8)}$$

Where: ν_O = Viscosity of specification lubricant, lb-min/in^2

ν_L = Viscosity of lubricant used, lb-min/in^2

See Figure 8.9

8.4.5 Temperature Multiplying Factor

Temperature conditions of the gear system have an impact on other parameters such as C_{GL} and C_{GP}. As the temperature increases, the lubricant viscosity decreases and the dimensions of the gears, shafts and bearings increase. This change normally causes a closer tolerance between operating units and an increase in the frictional losses in the system. To compensate for the decline in static and dynamic strengths, creep, and thermal expansion at high temperatures, the temperature factor, C_{GT}, represented by Equation (8-9) is applicable for temperatures greater than 160^{O}F (Reference 19). The multiplying factor for temperature C_{GT} can be expressed as:

$$C_{GT} = \frac{460 + T_{AT}}{620} \quad \text{for} \quad T_{AT} > 160^{O}\text{F} \qquad \textbf{(8-9)}$$

and: $C_{GT} = 1.0$ for $T_{AT} \leq 160^{O} F$

Where: T_{AT} = Operating temperature, ^{o}F

See Figure 8.10

8.4.6 AGMA Multiplying Factor

The AGMA has developed service factors for most industrial applications of gears, bearings, and gearbox designs whereby the expected extent of usage in vibration and shock environments can be taken into account when a gear system is selected for use. This service factor can be used as a multiplying factor for determining the inherent reliability or expected failure rate (C_{GV}) for a specific gearbox or bearing in a particular environment. Most manufacturers provide service factor data for each of their products. An example of a service factor for a speed-decreasing drive is shown in Table 8-1.

$$C_{GV} = \text{AGMA Service Factor} \tag{8-10}$$

8.5 SPLINE RELIABILITY PREDICTION

The failure rate in failures per million revolutions of spline gears can be calculated by:

$$\lambda_{GS,B} = \frac{10^{6}}{\theta} \tag{8-11}$$

where: λ_{GS} = Failure rate of spline gear in failures/million revolutions

θ = Life of spline gear in revolutions

An analytical expression for the spline gear life, θ, has been devised by Canterbury and Lowther (Reference 11). This equation is expressed as:

$$\theta = 7.08\,x10^{-10}\left(\frac{\phi\, G_L}{G_D}\right)^{4.56}\left(A_E\right)^{-2.36} \tag{8-12}$$

Where: G_L = Spline length, in

G_D = Spline diameter, in

A_E = Misalignment angle, radians

ϕ = Load Factor = $\dfrac{4.85\,x10^{3} G_B \left(G_D\right)^{3}}{G_T}$

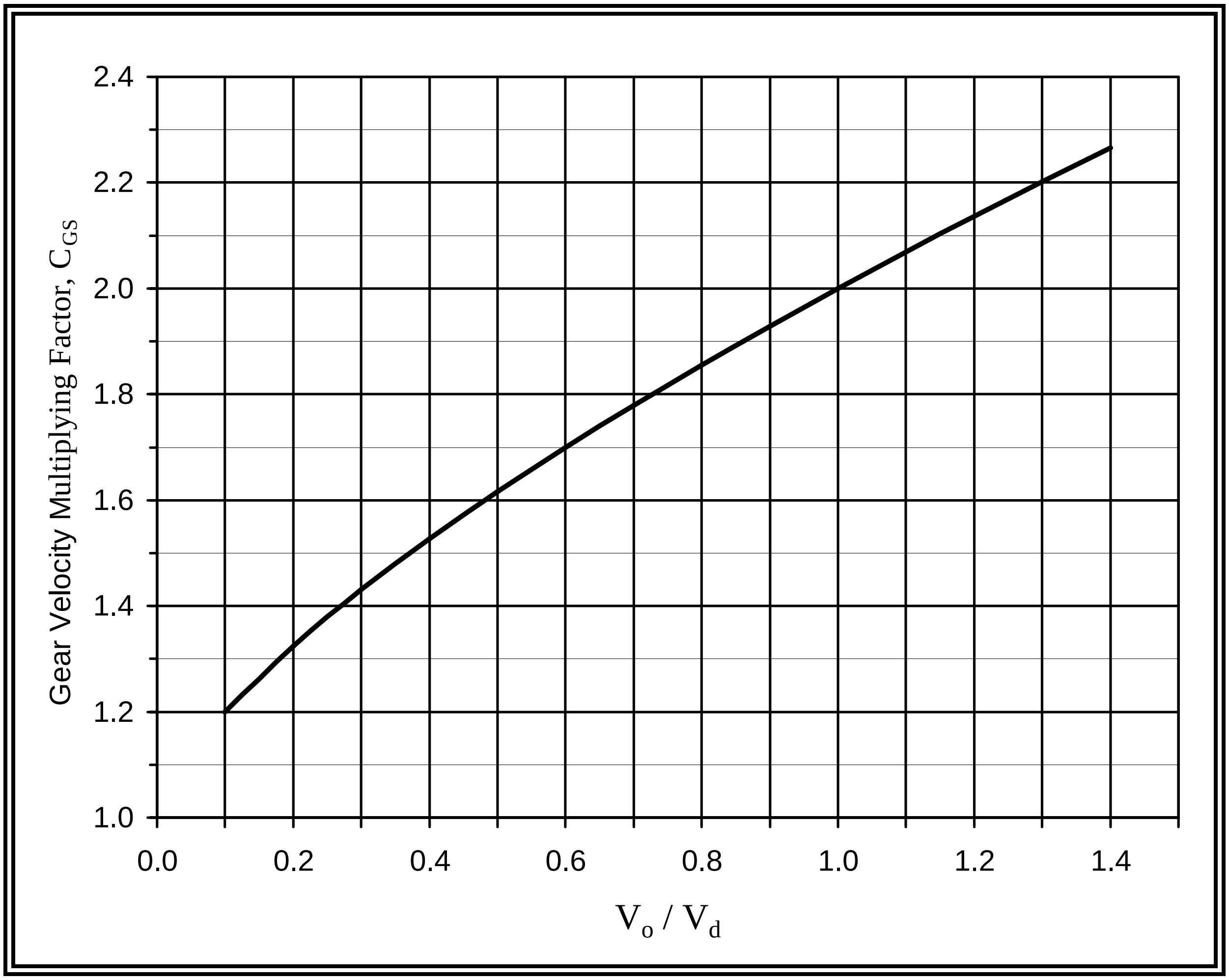
G_T = Torque, in-lbs

G_B = Tooth hardness (Brinell), lbs/in^2

Substituting the expression for the spline gear base failure rate into Equation (8-11) yields:

$$\lambda_{GS} = \lambda_{GS,B} \bullet C_{GS} \bullet C_{GL} \bullet C_{GT} \bullet C_{GV} \qquad \textbf{(8-13)}$$

where C_{GS}, C_{GL}, C_{GT}, and G_{GV} are calculated by Equations (8-3), (8-8), (8-9), and (8-10) respectively.

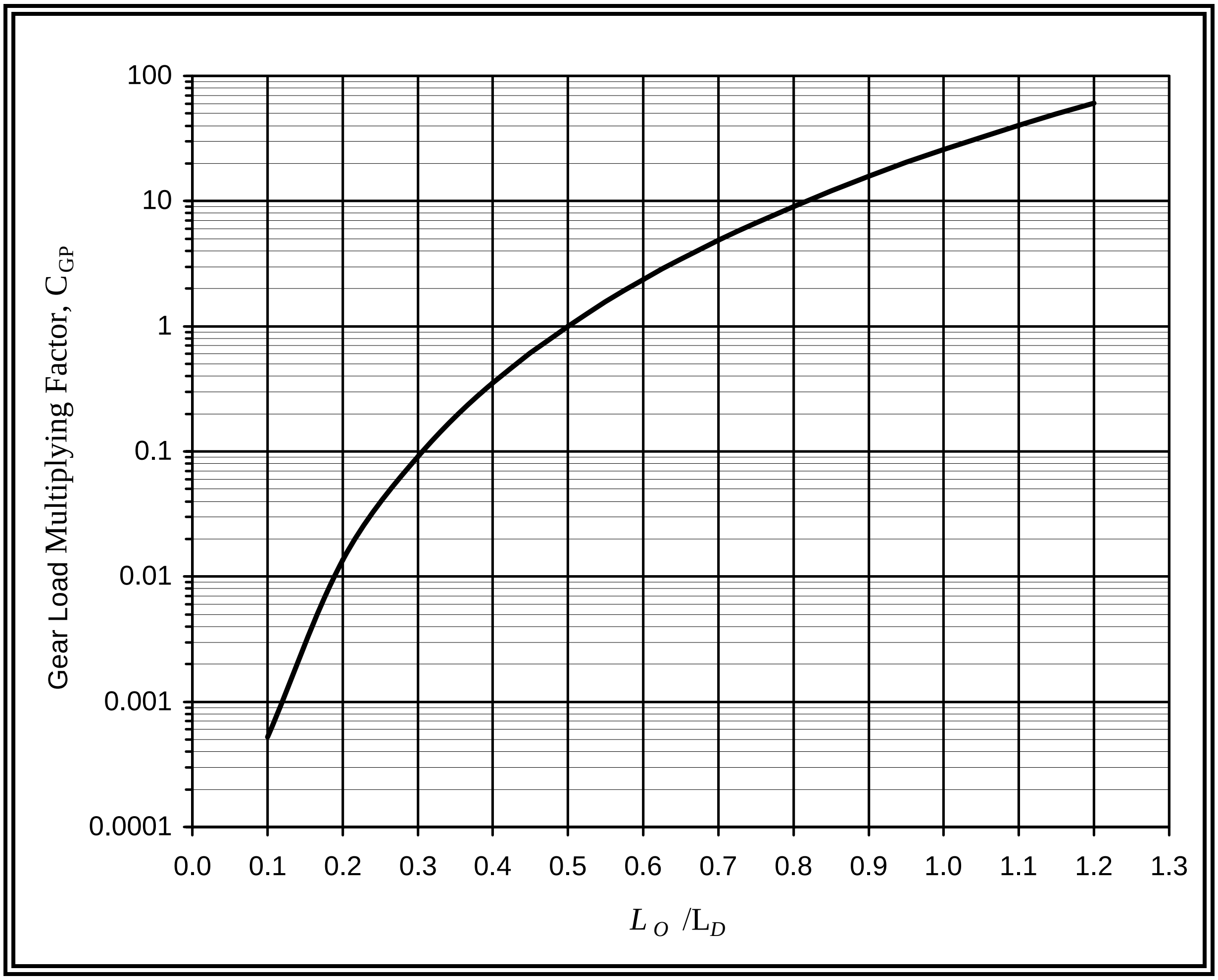


$$C_{GS} = 1.0 + \left(\frac{V_o}{V_d}\right)^{0.7}$$

Where: V_O = Operating speed, RPM

V_d = Design speed, RPM

Figure 8.6 Gear Velocity Multiplying Factor

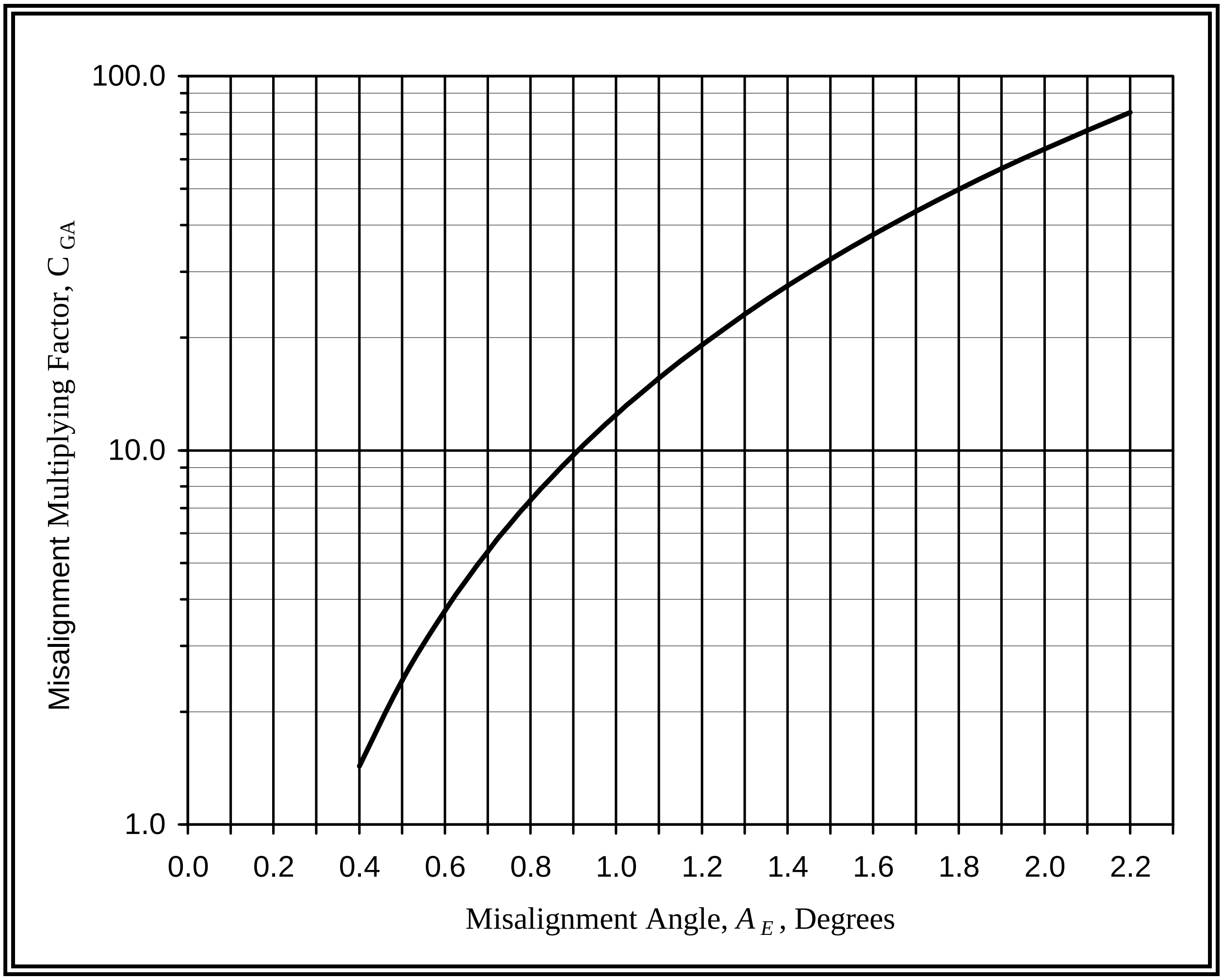


$$C_{GP} = \left(\frac{L_O / L_D}{0.5} \right)^{4.69}$$

Where: L_O = Operating load, lbs

L_D = Design load, lbs

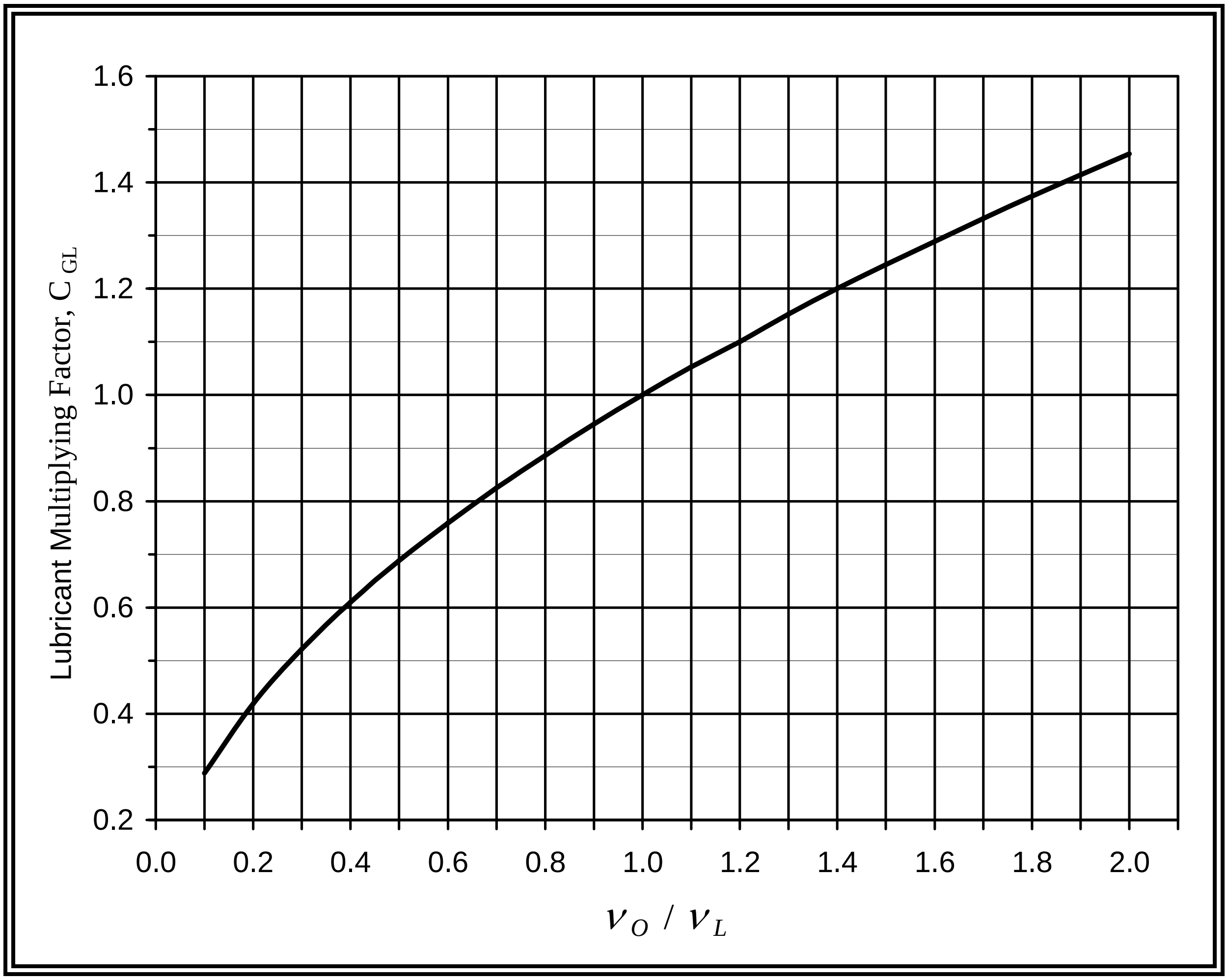
Figure 8.7 Gear Load Multiplying Factor



$$C_{GA} = 12.44\ A_E^{\ 2.36}$$

Where: A_E = Misalignment angle, degrees

Figure 8.8 Gear Misalignment Multiplying Factor

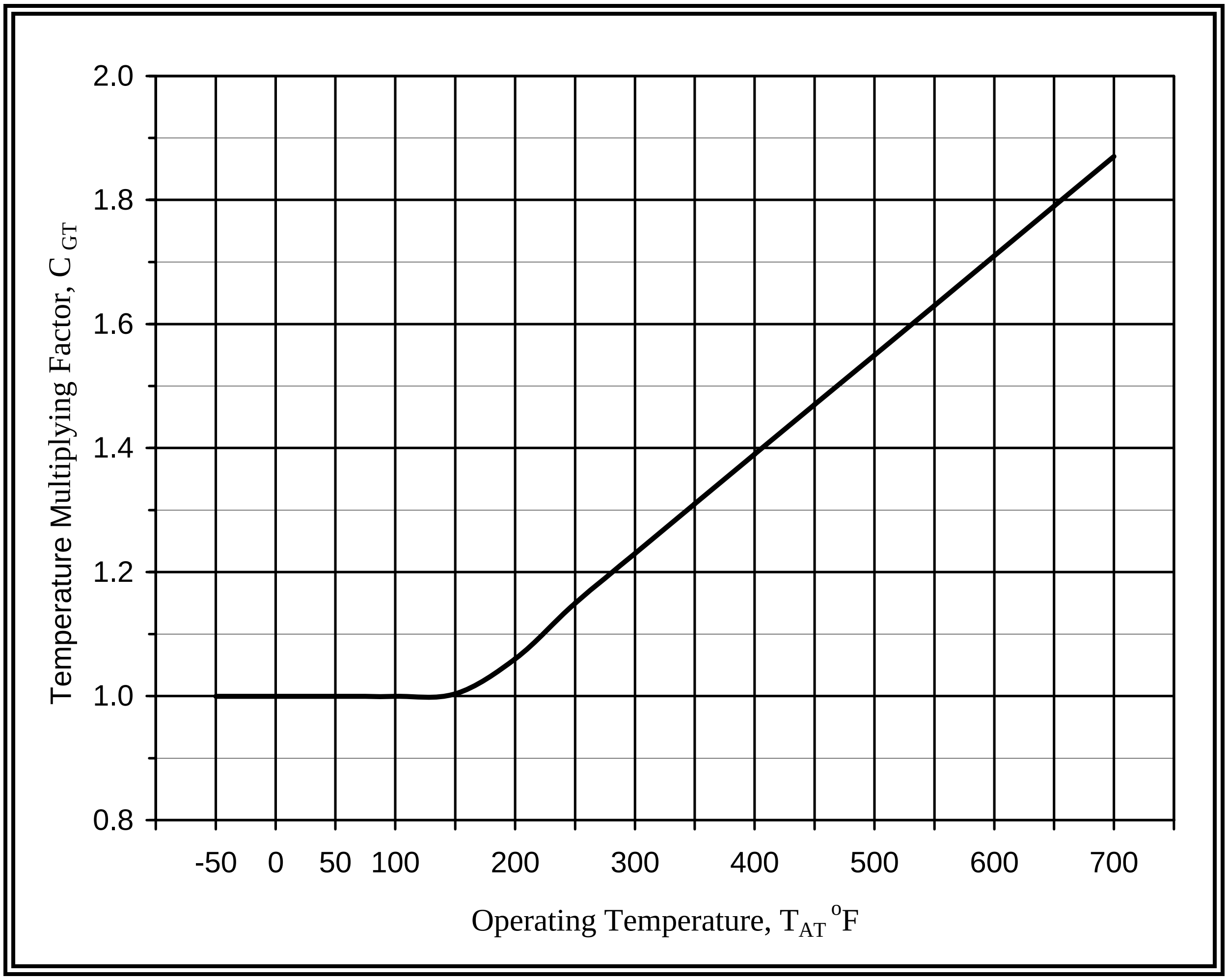


$$C_{GL} = \left(\frac{\nu_O}{\nu_L} \right)^{0.54}$$

Where: ν_O = Viscosity of specification fluid

ν_L = Viscosity of lubricant used

Figure 8.9 Gear Lubricant Multiplying Factor



$$C_{GT} = \frac{460 + T_{AT}}{620} \quad \text{for} \quad T_{AT} > 160^{O}\text{F}$$

and:

$$C_{GT} = 1.0 \quad \text{for} \quad T_{AT} \leq 160^{O} F$$

Where: T_{AT} = Operating temperature, ^{O}F

Figure 8.10 Temperature Multiplying Factor

Table 8-1. Typical AGMA Service Factor, C_{GV}

Prime Mover	Character of Load on Driven Member		
	Uniform	Medium Shock	Heavy Shock
Uniform	1.00	1.25	1.75
Medium Shock	1.25	1.50	2.00
Heavy Shock	1.50	1.75	2.25

8.6 REFERENCES

In addition to specific references cited throughout Chapter 8, other references included below are recommended in support of performing a reliability analysis of gears and splines.

10. "Boston Gear Catalogue", Catalogue 100, INCOM International Inc., Quincy, Massachusetts

11. Canterbury, Jack, and James D. Lowther, "Application of Dimensional Analysis to the Prediction of Mechanical Reliability," Naval Weapons Support Activity, Washington Navy Yard, Wash., D.C., Report ADAD35295 (September 1976).

13. Cormier, K.R., "Helicopter Drive System R&M Design Guide", United Technologies Corp., Stanford, Connecticut, Report ADAD69835 (April 1979)

19. Hindhede, U., et al, "Machine Design Fundamentals", John Wiley & Sons, NY, 1983

53 Rumbarger, John H., "A Fatigue Life and Reliability Model for Gears", American Gear Manufacturers Association, Report 229.16 (January 1972)

54. AGMA Standard for Surface Durability Formulas for Spiral Bevel Gear Teeth, American Gear Manufacturers Association, Report 216.01 (January 1964)

55. AGMA Standard Nomenclature of Gear Tooth Failure Modes, American Gear Manufacturers Association Report 110.04 (August 1980)

58. Parmley, R.O., Mechanical Components Handbook, McGraw-Hill Book Co., NY 1985

70. “Validation of Gearbox Reliability Models from Test Data”, Eagle Technology, Inc., Report No. 87-D-0075 (October 1987)

71. Dennis N. Pratt, “Investigation of Spline Coupling Wear”, Report No. SY-51R-87, Naval Air Warfare Center, Patuxent River, MD (December 1987)

98. Raymond J. Drago, “Rating the Load Capacity of Involute Splines”, Machine Design, February 12, 1976

99. David L. McCarthy, “A Better Way to Rate Gears”, Machine Design, March 7, 1996

102. Dan Seger, Niagara Gear Corporation, “Inside Splines” , Gear Solutions, January 2005

103. Mechanical Designers’ Workbook, “Gearing”, J. Shigley and C. Mischke, McGraw-Hill 1986

104. Raymond J. Drago, “Fundamentals of Gear Design”, Butterworth Publishers, 1988

CHAPTER 9

ACTUATORS

9.0 TABLE OF CONTENTS

9.1 INTRODUCTION

Actuators provide the means to apply mechanical power to systems when and where it is needed. In general, actuators take energy from pumped fluid and convert it to useful work. This conversion is accomplished by using pumped fluids under pressure to generate a differential pressure across a piston, which results in a force and motion being generated. This chapter identifies the more common failure modes and failure causes of actuators, and develops a procedure for determining the failure rate model of an actuator in its intended operating environment.

An actuator typically includes a piston and cylinder, return spring, seals and fluid connectors. Chapter 15 of this Handbook includes procedures for estimating the failure rate of a cylinder. Chapter 4 contains procedures for evaluating spring reliability, Refer to Chapter 3 for seals and Chapter 23 for connectors. Some actuators may include a valve assembly. If so, see Chapter 6.

In general, there are two types of output motions generated by actuators: linear and rotary. Within these two classifications there are many different types of actuator assemblies as discussed in the following two sections.

9.1.1 <u>Linear Motion Actuators</u>

Linear motion actuators are usually a derivative of one of the following four types:

1. Single acting
2. Double acting
3. Ram
4. Telescoping

Single acting actuators are the simplest type of the four. Pressurized fluid acts only on one side of the piston so the single acting actuator is capable of generating motion and power only in one direction and requires an external force to move the piston in the opposite direction.

Double acting actuators have fluid chambers on both sides of the piston, which allows pressurized fluid to both extend and retract the piston/rod and provide a faster response. Double acting actuators may have rods extending from either or both ends of the cylinders. Those with rods extending from both ends are balanced; that is, the piston moves at the same rate and delivers equal forces in each direction.

Ram or plunger actuators are a variation on the single acting design, but in this case, the piston rod is the same diameter as the piston. This design is useful where column loads are extremely high such as a service station lift, or when the rod hanging in a horizontally mounted cylinder has a tendency to cause sagging.

Telescoping linear actuators generate long stroke motions from a short body length such as those on a dump truck. Force output varies with rod extension: highest at the beginning, when the pressurized fluid acts on all of the multiple piston faces; and lowest at the end of the stroke, when the pressurized fluid acts only on the last extension's piston area. Telescoping cylinders may be either single or double acting.

9.1.2 Rotary Motion Actuators

Rotary actuators produce oscillating power by rotating an output shaft through a fixed arc. Rotary actuators are primarily one of two types:

1. Linear motion piston/cylinder with rotary output transmission
2. Rotary motion piston/cylinder coupled directly to output shaft

The first of the two rotary actuator types generally uses one or two linearly moving pistons to drive a transmission to convert the linear motion produced by the piston to a rotary output motion. These rotary actuators generally use crankshafts, gear rack-and-pinions or chains and sprockets as transmissions to convert the piston's linear output to rotary output. The piston/cylinder design may be single or double acting.

The second of the two rotary actuator types uses a piston designed to oscillate through a fixed arc to directly drive the output shaft. This design is simpler than the other type of rotary actuator as no transmission is required, but the unusual piston shapes required can cause sealing problems.

In a vane actuator the shaft is mounted in a cylindrical housing with one or more vanes attached to the shaft. Applying fluid pressure to the vanes produces shaft rotation. An electric motor may also be used to drive the output shaft. Torque motors can also be used as an actuator (see Chapter 14). Torque motors are direct drive induction motors which can operate indefinitely while stalled holding the actuator shaft in position, applying a steady torque to the load.

9.2 ACTUATOR FAILURE MODES

Typical modes of actuator failure are listed in Table 9-1. Forces acting upon actuators include a combination of direct forces pushing in from any axis and a twisting force that may be applied due to offset loads. The limiting factor in most actuators is the thrust and other supporting bearings. Selection of bearings and their orientation in the actuator greatly affects the maximum axial and side forces that can be applied during acceleration and movement of the actuator shaft. These forces also include twisting forces that apply an additional torque to the supporting bearings.

The primary failure mode of an actuator is a reduction in output force or stroke. This reduction in actuator output power can be caused by excessive wear of the piston/cylinder contact surfaces, which results in an increase in fluid leakage past the piston. Reduction in actuator output power can also be caused by external leakage, such as leakage through the piston rod/rod seal interface. Deterioration of the piston rod seal also permits ingestion of contaminants to the gap between the piston and cylinder increasing the rate of wear and probability of problems associated with corrosion.

Another common failure mode for actuators is jamming of the piston caused by stiction or misalignment. This failure can occur if excessive contaminants are ingested or if excessive side loads are encountered. Misalignment also increases the rate of piston/cylinder wear contributing to early failure. Temperature extremes may affect the viscosity characteristics of the pressurized fluid and increased seal wear will result from the resultant change in film lubrication.

Valve assemblies commonly use actuators to control the valve position and in some cases close the valve in event of system failure. Another use of an actuator in a valve is to determine the valve position. Depending on the equipment design, one potential failure mode requiring investigation is the loss of signal that a loss of accurate positioning of an actuator can cause to software programming or valve controls.

Table 9-1. Typical Modes of Actuator Failure

FAILURE MODE	FAILURE CAUSE	FAILURE EFFECT
Internal leakage	Side loading and piston wear, contaminants past rod seal	Loss/reduction in output force
External leakage	Seal leakage, piston/cylinder wear	Loss/reduction in output force
Damaged rod seal	Excessive side loading	Contaminants entering actuator between shaft and cylinder
Spurious position change	Stiction, binding	Loss of output control or incorrect signal transmission
Jamming, seizure	Excessive loading	Loss of load control
Aeration	Air drawn past rod seals during actuation	Damaged actuator and loss of seals
Bearing failure	Axial or side load on thrust bearing exceeding manufacturer's specification	Loss of output force, damaged piston
Lead nut failure	Axial load on thrust bearing exceeding manufacturer's specification	Loss of output force, damaged piston

9.3 FAILURE RATE MODEL FOR ACTUATOR

The reliability of an actuator is primarily influenced by its load environment which can be subdivided into external loads and internal loads. External loads are forces acting on the actuator from outside sources due to its operating environment. Conditions of storage, transportation and ground servicing as well as impact loads during operation have an effect on the rate of failure. Internal loads are caused by forces acting inside the actuator as a result of pressure variations, pressure differentials, friction forces, temperature-related expansion and contraction, and by forces developed and transmitted by the impact of external loads.

Valves often form a part of an actuator assembly and are used for primary movement control of the actuator and also for deceleration of the piston/rod assembly at the ends of their stroke. Failure rate models for valve assemblies are presented in Chapter 6 of this handbook. Chapter 3 includes failure rate models for the actuator rod seals, connector O-rings, gaskets and mechanical seals.

The complete failure rate model for the piston/cylinder actuator incorporates modifiers for contamination and temperature effects. The complete model can be expressed as follows:

$$\lambda_{AC} = \lambda_{AC,B} \bullet C_{CP} \bullet C_T \tag{9-1}$$

Where: λ_{AC} = Failure rate of actuator, failures/million cycles

$\lambda_{AC,B}$ = Base failure rate of actuator, failures/million cycles [See Section 9.3.1 and Equations (9-6) and (9-14)]

C_{CP} = Contaminant multiplying factor (See Section 9.3.2)

C_T = Temperature multiplying factor (See Section 9.3.3)

9.3.1 Base Failure Rate for Actuator

The primary failure effect of internal and external loads on an actuator is wear of the piston and cylinder which results in an increase in leakage past the piston. A criteria of actuator failure would then be a leakage rate resulting from wear which exceeds a maximum allowable leakage rate specified by the user.

Wear of the cylinder and piston will occur in two phases according to the Bayer-Ku sliding wear theory (Reference 6). The first or constant wear phase is characterized by the shearing of the surface asperities due to the sliding action of the piston within the cylinder. During this period the wear rate is practically linear as a function of the number of actuator cycles and the wear depth at the end of the constant wear phase is one half the original surface finish. During the second or severe wear phase, wear debris becomes trapped between the two sliding surfaces and gouging of the surfaces takes place. The wear rate begins to increase very rapidly and failure of the actuator is eminent. Therefore, while equations are presented in this chapter for the severe wear phase, for practical purposes the failure rate or life of the actuator can be estimated as that calculated for the constant wear phase.

The number of cycles to complete the constant wear phase can be predicted analytically by a semi-empirical modification of Palmgren's equation (Reference 6) resulting in the following equation:

$$N_O = 2000\left(\frac{\gamma F_Y}{S_C}\right)^9 \qquad \textbf{(9-2)}$$

Where: N_O = Number of cycles in constant wear phase

γ = Wear factor *

F_y = Yield strength of softer material, lbs/in^2 (See Table 9-3)

S_c = Compressive stress between the surfaces, lbs/in^2

* The wear factor, γ, will be equal to 0.20 for materials that have a high susceptibility to adhesive wear, in which the wear process involves a transfer of material from one surface to the other. The wear factor will be equal to 0.54 for materials that have little tendency to transfer material in which the material is subject to micro-gouging of the surfaces by the asperities on the material surface.

The maximum compressive stress caused by the cylinder acting on the piston is computed assuming a linear distribution of stress level along the contact area. The following equation has been derived for compressive stress of an actuator (Reference 6):

$$S_C = 0.8\left(\frac{\frac{W_S}{L}\bullet\frac{D_1 - D_2}{D_1 D_2}}{\frac{1-\eta_1^{\;2}}{E_1}+\frac{1-\eta_2^{\;2}}{E_2}}\right)^{1/2} \qquad \textbf{(9-3)}$$

Where: W_S = Side load on the actuator, lbf

L = Total linear contact between piston and cylinder, in

D_1 = Diameter of cylinder, in

D_2 = Diameter of piston, in

η_1 = Poisson's ratio, cylinder

η_2 = Poisson's ratio, piston

E_1 = Modulus of elasticity, cylinder, lbs/in^2

E_2 = Modulus of elasticity, piston, lbs/in^2

Substituting Equation (9-3) into Equation (9-2) and adding a constant for lubrication provides an equation for the number of cycles for an actuator during the initial wear phase until the severe wear period begins.

$$N_O = k_1 \left[\gamma F_Y \Bigg/ \left(\frac{\dfrac{W_S}{L} \bullet \dfrac{D_1 - D_2}{D_1 D_2}}{\dfrac{1-\eta_1^{\,2}}{E_1} + \dfrac{1-\eta_2^{\,2}}{E_2}} \right)^{1/2} \right]^9 \qquad \textbf{(9-4)}$$

Where: k_1 = 15.4 x 10^3 which includes a lubrication constant

In a similar way, if the actuator is subjected to axial stress, Equation (9-5) can be used to determine compressive stress. Which equation to use depends on the application of the actuator, axial or side loading.

$$S_C = 0.9 \left(\frac{W_A \bullet \left(\dfrac{D_1 - D_2}{D_1 D_2} \right)^2}{\left(\dfrac{1-\eta_1^{\,2}}{E_1} + \dfrac{1-\eta_2^{\,2}}{E_2} \right)^2} \right)^{1/3} \qquad \textbf{(9-5)}$$

Where: W_A = Axial Load on the actuator, lbf

And:

$$N_O = k_2 \left[\gamma F_Y \Bigg/ \left(\frac{W_A \bullet \left(\dfrac{D_1 - D_2}{D_1 D_2} \right)^2}{\left(\dfrac{1-\eta_1^{\,2}}{E_1} + \dfrac{1-\eta_2^{\,2}}{E_2} \right)^2} \right)^{1/3} \right]^9 \qquad \textbf{(9-6)}$$

Where: k_2 = 17.7 x 10^3 which includes a lubrication constant

Since the second phase of wear is severe and relatively short, it can normally be assumed that the calculated number of cycles, N_o, for the first phase of wear will be the life of the actuator. During the second or severe wear phase, the following equation can be used to determine the rate of wear (Reference 45):

$$V = \frac{K\,W\,d}{H}\left(N - N_O\right) \tag{9-7}$$

Where:

V = Volume of material removed by wear during the second phase, in^3

K = Wear coefficient (See Table 9-2)

W = Applied load, lb

d = Sliding distance, in

H = Penetration hardness, psi

N = Total number of cycles to failure

N_o = Number of cycles at the end of the initial wear phase

Solving for N results in the equation:

$$N = \frac{V\,H}{K\,W\,d} + N_O \tag{9-8}$$

This second phase of wear is characterized by rapid wear until failure of the actuator occurs usually as a result of poor response due to excessive leakage. The leakage rate past the piston within the cylinder may be modeled as laminar flow between parallel plates (Reference 5).

$$Q = \frac{\pi\,D_2\;a^3\,\Delta p}{12\,\nu\,L} \tag{9-9}$$

Where: Q = Leakage rate past piston, in^3/sec

D_2 = Piston diameter, in

a = Gap between piston and cylinder, in

Δp = Pressure differential across piston, psi

ν = Fluid viscosity, lbf-sec/in^2

L = Piston length, in

The gap between the piston and cylinder, a, as shown in Figure 9.1 is a dynamic term being a function of wear.

$$a = (D_1 - D_2) + h \tag{9-10}$$

Where: D_1 = Cylinder diameter, in

h = Depth of wear scar, in

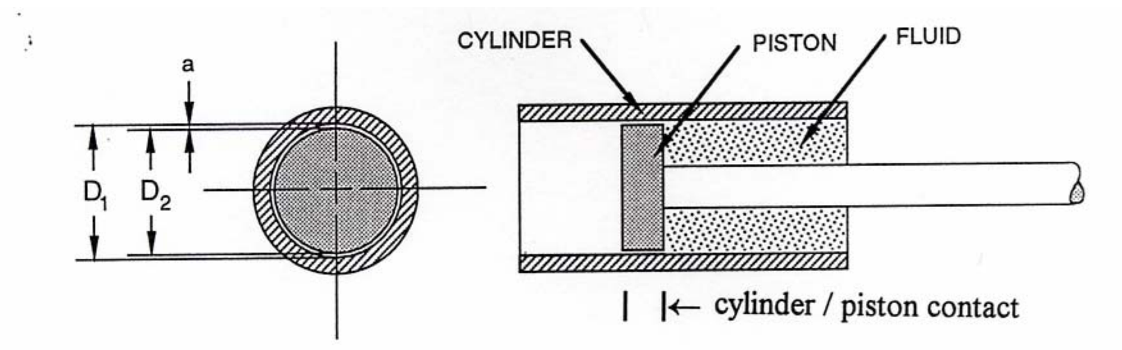


Figure 9.1 Typical Single Acting Actuator

The wear scar depth, h, will be equal to the volume of material lost due to wear, V, divided by the contact surface area, A:

$$h = \frac{V}{A} \tag{9-11}$$

Substituting Equations (9-10) and (9-11) for wear gap into Equation (9-9) results in the following equation for leakage rate between the piston and cylinder:

$$Q = \frac{\pi D_2 \left[\left(D_1 - D_2 \right) + \frac{V}{A} \right]^3 \Delta p}{12 \nu L} \tag{9-12}$$

Solving Equation (9-12) for V and substituting V in Equation (9-7) results in an equation for the number of cycles to failure.

$$N = \frac{AH}{KWd} \left[\left(\frac{12 Q \nu L}{\pi D_2 \Delta p} \right)^{1/3} + \left(D_2 - D_1 \right) \right] + N_O \tag{9-13}$$

A similar equation can be developed by combining Equations (9-13) and (9-6) for axial loading of the actuator. A typical plot of wear as a function of the number of cycles is shown in Figure 9.2.

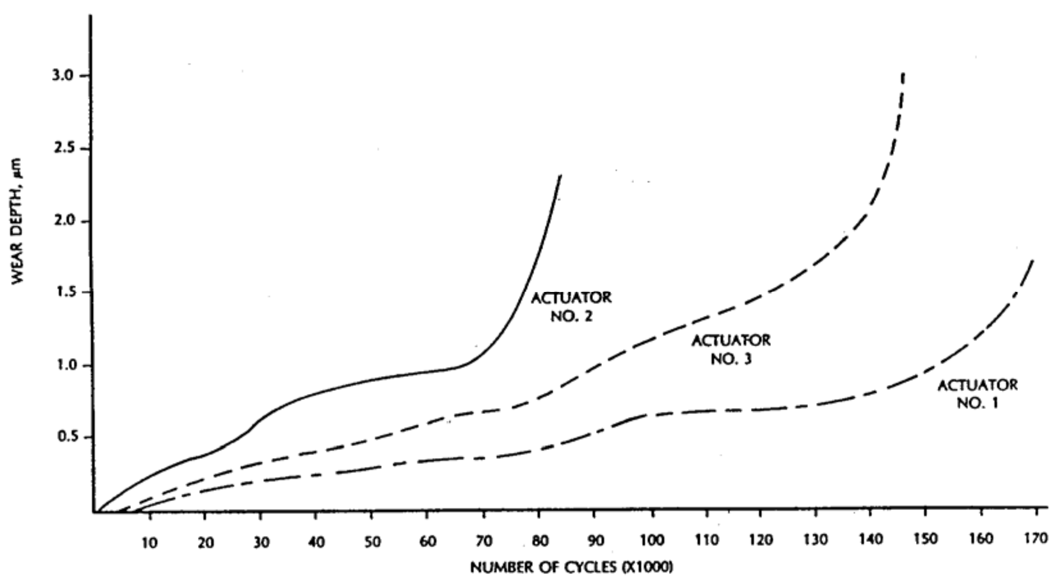


Figure 9.2 Failure Rate as a Function of Cycles for a Typical Actuator Under Different Axial and Side Loads.

Since the first phase of wear is fairly linear as a function of the number of cycles and failure will occur soon after phase one wear, the base failure rate of the actuator can be approximated as follows:

$$\lambda_{AC,B} = \frac{10^6}{N_O} \tag{9-14}$$

Where: $\lambda_{AC,B}$ = Base failure rate of actuator, failures/million cycles

N_O = Number of cycles in constant wear phase (See Equation (9-6))

As noted in Equation (9-2), the ratio of Yield strength to compressive strength is raised to the 9th power creating a wide range of resulting failure rates in Equation (9-14) for small differences in material properties. Reliability testing may be required on a sample of actuators to duplicate the curves as shown in Figure 9.2.

9.3.2 Contaminant Multiplying Factor

As established in Equation (9-1), the failure rate of the actuator can be determined as follows:

$$\lambda_{AC} = \lambda_{AC,B} \bullet C_{CP} \bullet C_T$$

Where: λ_{AC} = Failure rate of actuator, failures/million cycles

$\lambda_{AC,B}$ = Base failure rate of actuator, failures/million cycles

C_{CP} = Contaminant multiplying factor

C_T = Temperature multiplying factor (See Section 9.3.3)

During the time that the actuator is at rest, particles can work their way between the piston and cylinder. Then, when the actuator is put into motion, increased forces are needed to move the piston. This stiction phenomenon causes a loss of actuator response and in some severe cases, a completely jammed component.

Three types of wear need to be considered in determining the effects of contaminants on actuator reliability:

(1) Erosion - Particles carried in a fluid stream impact against the piston and cylinder surfaces. If the kinetic energy released upon actuator response is large

compared to forces binding the piston/cylinder walls, surface fatigue will occur. Hard particles may also cut away surface material.

(2) Abrasive Wear - A hard particle entering the gap between the piston and cylinder surfaces can cut away material of the softer surface on a single actuator engagement. The rate of wear will be proportional to the number of particles in contact with the surfaces and the particle hardness. If the hardness of the piston is significantly less than that of the cylinder, a hard particle, absorbed by the softer material causes severe abrasive wear of the harder actuator surface.

(3) Surface Fatigue - Particulate contaminants interacting with the piston and cylinder surfaces can dent a surface producing plastic deformation. Large numbers of dislocations will increase the surface roughness and deteriorate the surface material. The result is an accelerated rate of wear and a higher probability of leakage between the surfaces.

The deteriorating effects of contaminant particles on the reliability of an actuator must be equated along with the probability of the contaminants entering the gap between the actuator surfaces. The probability of contaminants entering this area will depend on the operating environment, the types and numbers of particles expected to be encountered, and the filtering system to prevent the entrance of particles. The typical actuator contains a bushing to wipe the piston on the return stroke. The life expectancy and reliability of this device must be determined as part of the overall reliability estimate of the actuator.

If the piston surface slides over a hard contaminant particle in the lubricant, the surface may be subject to pitting. The abrasive particle has edges with a characteristic radius, denoted by r. When the depth of penetration of the abrasive particle (d) reaches a certain critical value, the scratching produces additional wear particles by pitting. This elastic/plastic deformation process occurs when the maximum shear stress in the complex stress distribution beneath the contact surface exceeds the elastic limit. This maximum shear stress occurs beneath the contact at a depth equal to one half the contact radius. The value of this critical depth is given by (Reference 48).

$$d_{crit} = \frac{r}{2} - \left(\frac{1 - 2\, f_{s,max}}{F_{sy}} \right) \quad \textbf{(9-15)}$$

Where: $f_{s,max}$ = Maximum shear stress, lbs/in^2

r = Characteristic radius of particle, in

F_{sy} = Yield strength of material, lbs/in^2

If this type of wear should occur, it is so severe that actuator performance would be immediately affected and failure would occur. Actuators are designed to prevent particles of sufficient size to cause this type of failure and the probability of failure from this type of pitting is extremely low. The failure mode is presented here as a design evaluation check on the sealing technique for the piston assembly.

Fatigue wear on the microscopic wear due to contaminants is similar to that for pitting just described except that it is associated with individual asperity contacts rather than with a single large region. The additional material lost due to contaminant wear process can be estimated in the same way as the adhesive wear process was explained earlier in this chapter, the volume δV removed on an individual piston stroke proportional to a^3 where a is the radius of the individual area of contact. Similarly, the sliding distance δL is proportional to the area of contact.

$$\frac{\delta V}{\delta L} \propto \frac{\delta A}{3} \tag{9-16}$$

Where: A = Area of contact, in^2

Summing for all contacts provides the following equation:

$$\frac{V}{L_1 N} = \frac{1}{3} K_1 A = \frac{K_1 W}{3 H_V} \tag{9-17}$$

Where: V = Volume of material lost due to contaminant wear, in^3

L_1 = Sliding distance of the piston, in

N = Number of actuations

K_1 = Wear coefficient, See Table 9-2

A = Area of contact, in^2

W = Transverse load on the actuator, lbs

H_V = Vickers hardness of the piston, lbs/in^2

This expression can be rewritten in the form to include a contaminant multiplying factor, C_{CP}:

$$V = \frac{C_{CP} W L_1 N}{H_V} \tag{9-18}$$

The effect of the additional wear due to contaminant particles may be expressed as an additive term in the basic wear relationship. It will be noted from the derivation of equations for the effect of contaminant particles on actuator surface wear and the possibility of stiction problems that a probability of damaging particles entering the gap between the piston and cylinder must be estimated. The contaminant factors involved are as follows:

Hardness - The wear rate will increase with the ratio of particle hardness to actuator surface hardness. It will normally be the hardness of the piston that will be of concern. If the ratio is less than 1, negligible wear can be expected.

Number of particles - The wear rate will increase with a concentration of suspended particles of sufficient hardness.

Size - For wear of the piston or cylinder to occur, the particle must be able to enter the gap between the two surfaces. The particle must also be equal to or greater than the lubrication film thickness. With decreasing film thickness, a greater proportion of contaminant particles entering the gap will bridge the lubrication film, producing increased surface damage.

Shape - Rough edged and sharp thin particles will cause more damage to the actuator surfaces than rounded particles. As the particles remain in the gap, they will become more rounded and produce less wear. It is the more recent particles being introduced into the gap that cause the damage.

The contaminant multiplying factor can be established as follows:

$$C_{CP} = C_H \bullet C_S \bullet C_N \tag{9-19}$$

Where:

C_H = Particle hardness multiplying factor = H_P/H_C

Hp = Piston Hardness (See Table 9-1)

H_C = Cylinder Hardness (See Table 9-1)

C_S = Filtration multiplying factor = Filter size (micron) / 10

C_N = Particle size multiplying factor - For most applications the particle size factor will be equal to approximately 1.0. For severe conditions where the number of contaminants and particle size can be expected to increase, the particle size factor may be expected to approach a value of 2.0.

9.3.3 Temperature Multiplying Factor

The effect of the temperature of the surface on the wear rate is a complicated phenomenon, because the corrosion of the wear debris at different temperatures produces different oxidation products. Chemical interactions with the metal surfaces result in different wear rates as the temperature of the surface is changed (Reference 51). For example, the formation of Fe_3O_4 is likely to predominate when steel is subject to wear in the temperature range between 300 ^{O}C to 500 ^{O}C.

Wear of metals has been related to the heat of absorption of molecules of debris (Reference 51). The basic relationship in this treatment is:

$$V = \frac{K_1 W L_1}{H_V} C_O e^{\theta/T} \tag{9-20}$$

Where: C_o = Arrhenius constant

θ = Activation energy constant, K

T = Operating temperature, K

Values for the parameter θ are in the range between 1200 K and 6000 K.

The effect of variation of temperature may be determined by eliminating the Arrhenius constant in terms of the value of the exponential at ambient temperature T. Making this substitution into Equation (9-20), the following is obtained:

$$V = \frac{C_T K_1 W L_1 N}{H_V} \tag{9-21}$$

From Reference 45, the temperature multiplying factor, C_T, is given by:

$$C_T = e^{\theta/T_a[1-(T_a/T)]} \tag{9-22}$$

Where: T_a = Ambient temperature, 25.2 C

T = Operating temperature, C

It is noted that the ratio θ/T_a is in the range between 4.0 and 20.0.

For most situations θ will be equal to 1200 K and Equation (9-22) can be simplified as follows:

$$C_T = e^{4[1-(25.2/T)]} \tag{9-23}$$

Table 9-1. Material Hardness
(Use ratio of hardest particle/piston hardness for C_H)

MATERIAL	HARDNESS (H_V x 10^6)
Plain carbon steels	
- Low strength steel	140
- High strength steel	220
Low-alloy Steels	
- 4320	640
- 4340	560
Stainless Steels	
- 303	170
- 304	160
- 631 (17-7 PH hardened)	520
- 631 (17-7 PH annealed)	170
- Austenitic AISI 201 annealed)	210
- Martensitic 440C (hardened)	635
- 630 (17-4 PH hardened)	470
Nickel Alloys	
- 201	100
Nickel-copper Alloys	
- Monel (annealed)	120
- Monel K-500 (annealed)	162
Ni-Cr-Mo-Fe Alloys	
- Inconnel 625	140
- Hastelloy	200
Aluminum	
- AISI 1100 (annealed)	25
- AISI 1100 (cold worked)	45
- AISI 2024 (annealed)	50
- AISI 2024 T4 (heat treated)	125
- AISI 6061 (annealed)	32
- AISI 6061 T6 (heat treated)	100

Table 9-2. Values of Wear Coefficient (K_1) In The Severe-Wear Region
(Reference 45)

MATERIAL	K_1
4130 Alloy Steel (piston) 4130 Alloy Steel (cylinder)	0.0218 0.0221
17-4 PH Stainless Steel (piston) 4130 Alloy Steel (cylinder)	0.0262 0.0305
9310 Alloy Steel (piston) 4130 Alloy Steel (cylinder)	0.0272 0.0251

Table 9-3. Values of Yield Strength for Various Metals

Metal	Yield strength, ksi
Ordinary structural steel	30 - 40
Low alloy, high strength steel	40 - 80
Heat treated steel casting	40 - 120
Cold rolled steel	70 - 85
Stainless steel	50 - 80
Heat treated wrought aluminum	10 - 50
Aluminum alloy	35 - 42
Pure rolled aluminum	5 - 21
Cast aluminum	15 - 25
Wrought iron	25 - 35
Cast iron	8 - 40
Magnesium alloy	11 - 30

9.4 REFERENCES

In addition to specific references cited throughout Chapter 9, other references included below are recommended in support of performing a reliability analysis of an actuator assembly.

5. Bauer, P., M. Glickmon and F. Iwatsuki, "Analytical Techniques for the Design of Seals for Use in Rocket Propulsion Systems", Volume 1, ITT Research Institute, Technical Report AFROL-TR-65-61 (May 1965)

6. Bayer, R.G., A.T. Shalhey and A.R. Watson, "Designing for Zero Wear", Machine Design (January 1970)

11. Canterbury, Jack and James Lowther, "Application of Dimensional Analysis to the Prediction of Mechanical Reliability", Naval Weapons Support Activity, Washington Navy Yard, Washington, D.C., Report ADAD35295 (September 1976)

38. Roack and Young, Formulas for Stress and Strain, McGraw-Hill Book Co., NY 1989

45. Barron, Randall F., "Revision of Wear Model for Stock Actuators, Engineering Model for Mechanical Wear" (July, 1987)

48. Kragelsky, I.V. and Alisin, Friction, Wear and Lubrication, Volume 2, Pergamon Press, London (1981)

49. Kuhlmann-Hildorf, D. "Parametric Theory of Adhesive Wear in Uni-directional sliding", Wear of Materials, American Society of Mechanical Engineers, New York (1983)

50. Bently, R.M. and D.J. Duquette, "Environmental Considerations in Wear Processes", Fundamentals of Friction and Wear of Materials, American Society of Metals, Metals Park, Ohio (1981)

51. Sarkar, A.D., Wear of Metals, pp.62-68, Pergamon Press, London (1976)

80. D. Pratt, "Results of Dayton 5A701 Linear Actuator Reliability Investigation", Report No. TM 93-89 SY, Naval Air Warfare Center, Patuxent River, Maryland (1994)

120. Tyler G. Hicks, Handbook of Mechanical Engineering Calculations, McGraw-Hill, 2006

CHAPTER 10

PUMPS

10.0 TABLE OF CONTENTS

10.1 INTRODUCTION

Pumps are one of the most common types of mechanical components used by today's society, exceeded only by electric motors. Not surprisingly, there are in existence today, an almost endless number of pump types that function in systems with dissimilar operating and environmental characteristics. With so many different pump types it is impossible to establish a failure rate database based on design parameters, their use, and the materials used to construct them, or the type of fluid they move. All of these categories tend to overlap for the many different pump types. Therefore, a system to differentiate between all types of pumps is necessary. This system considers the method by which energy is added to the fluid being pumped. As seen by Figure 10.1, a pump can be classified into two general classes: Centrifugal and Positive

Displacement. These classes represent the two ways in which energy is added to the fluid.

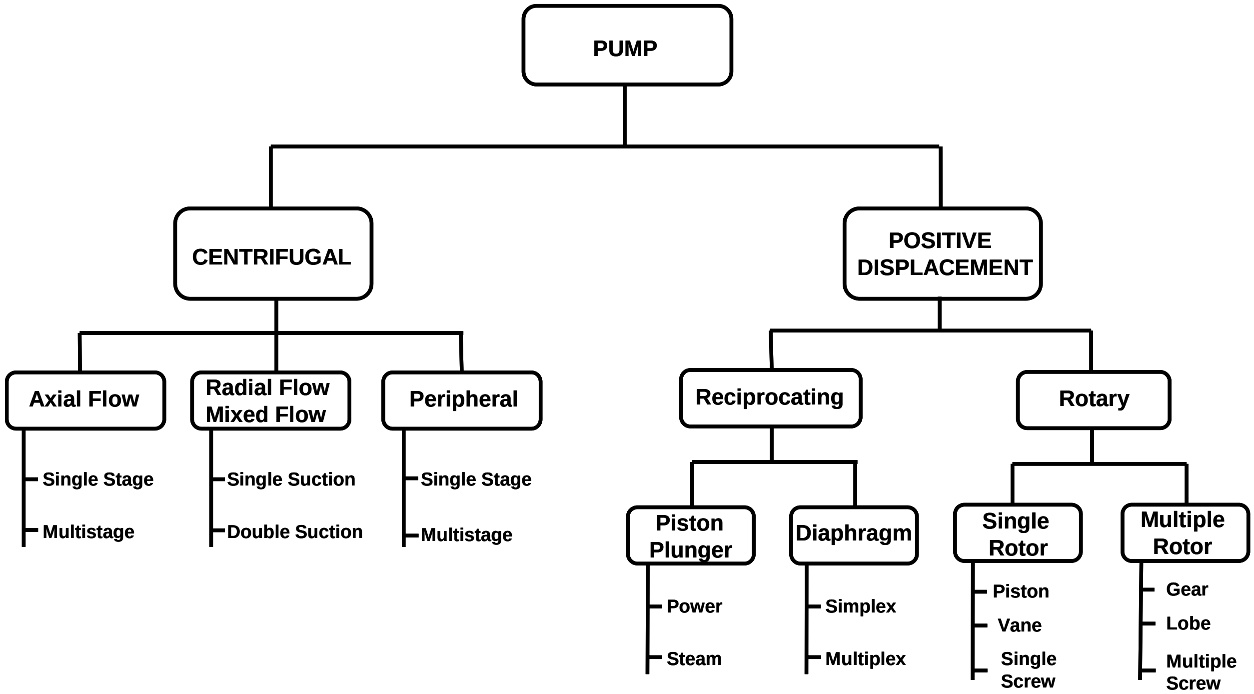


Figure 10.1 Pump Configurations

10.1.1 <u>Centrifugal Pump</u>

Centrifugal pumps consist of a set of rotating vanes, enclosed within a housing or casing, used to impart energy to a fluid through centrifugal force. The centrifugal pump has two main parts: a rotating element which includes an impeller and a shaft, and a stationary element made up of a volute casing, stuffing box, and bearings. With centrifugal pumps, the energy is added continuously by increasing the fluid velocity with a rotating impeller while reducing the flow area. This arrangement causes an increase in pressure along with the corresponding movement of the fluid. The impeller produces liquid velocity and the volute forces the liquid to discharge from the pump converting velocity to pressure. The stuffing box protects the pump from leakage at the point where the shaft passes out through the pump casing.

Centrifugal pumps can be further classified as to one of the following three designs:

- Axial Flow - In an axial flow pump, pressure is developed by the propelling or lifting action of the impeller vanes on the liquid. Axial flow pumps are sometimes referred to as propeller pumps.

- Radial or Mixed Flow – In a radial flow pump, the liquid enters at the center of the impeller and is directed out along the impeller blades in a direction at right angles to the pump shaft. The pressure is developed wholly by centrifugal force.
- Peripheral - Peripheral pumps employ a special impeller with a large number of radial blades. As the fluid is discharged from one blade, it is transferred to the root of the next blade and given additional energy.

10.1.2 Positive Displacement Pump

Positive displacement pumps differ from centrifugal pump designs in that energy is added to the fluid periodically by the movement of control mechanisms causing displacement of fluid and an increase in pressure. A positive displacement pump unlike a centrifugal pump will produce the same flow at a given rotational or cyclic speed regardless of the discharge pressure. The positive displacement pump contains an expanding cavity on the suction side of the pump and a decreasing cavity on the discharge side. Fluid enters into the pump as the cavity on the suction side expands and the fluid is forced out of the discharge as the cavity collapses. Positive displacement pumps can be subdivided into reciprocating and rotary types and this principle applies to any individual reciprocating or rotary pump design.

Positive displacement pumps deliver a definite volume of fluid for each cycle of pump operation. Therefore, the only factor that affects flow rate in an ideal positive displacement pump is the speed at which it operates. The flow resistance of the system in which the pump is operating will not affect the flow through the pump. As the discharge pressure of the pump increases, some amount of fluid leaks from the discharge of the pump back to the pump suction, thus reducing the effective flow rate of the pump. This fluid leakage from the pump discharge to the suction area is called slippage. Positive displacement pumps come in many designs and performance specifications, but they all work on the same principle. An increasing volume is opened to suction, filled, closed, moved to discharge, and displaced. The delivered capacity is nearly constant throughout the discharge pressure range.

Reciprocating pump - A reciprocating pump is characterized by a back-and-forth motion of pistons inside of cylinders that provides the flow of fluid. Each stroke of a reciprocating pump delivers a definite volume of liquid to the system. The master cylinder of the automobile brake system is an example of a simple reciprocating pump. Reciprocating pumps include piston, plunger and diaphragm.

Rotary pump - Rotary pumps operate on the principle that a rotating vane, screw, or gear traps the fluid on the suction side of the pump casing and forces it to the discharge side of the casing. A rotary displacement pump is different from a centrifugal pump in that in a centrifugal pump, the liquid displacement and pumping action depend on developed liquid velocity, while a rotary pump operates on the positive displacement principle, each revolution delivering a definite volume of liquid to the system. Rotary pumps include gear, lobe, piston, screw and vane.

10.2 FAILURE MODES

Due to the large number of pump types and applications, some failure modes are more prevalent than others for a specific category of pump. For example, with displacement pumps there is a much greater chance for cyclic fatigue to have an effect on the system than with centrifugal pumps. This is due to the inherent difference in designs. Displacement pumps have pressure transients which cause temporary unbalanced forces to be applied to the pump and its system. The displacement pump and driver shafting can experience much higher stresses during operation due to the uneven torque loading caused by this natural imbalance. A positive displacement pump does not have a shut-off head as does a centrifugal pump and cannot be operated against a closed valve on the discharge side of the pump. If a positive displacement pump is allowed to operate against a closed discharge valve or other obstruction, it will continue to produce flow, increasing the pressure in the discharge line until either the line bursts or the pump is severely damaged or both. Thus a relief valve incorporated within the pump assembly or incorporated on the discharge side of a positive displacement pump is an absolute necessity.

Centrifugal pumps tend to be more balanced than positive displacement pumps and aren't as susceptible to large stress variations. There are three main factors that affect the reliability of a centrifugal pump including operating speed, impeller diameter and flow rate. Operating speed affects the wear in the rubbing contact surfaces of shaft seals, bearing life and the heat generated by the bearings and lubricants. Impeller diameter affects reliability through the loads it imposes on the shaft and bearings. And flow rate affects reliability of a centrifugal pump when it is different than that established by the Best Efficiency Point (BEF) causing increasing and turbulent loads on the impeller. BEF operation is discussed in more detail in Section 10.8.1.

A major effect on performance of a positive displacement pump is a loss of flow due to slippage. The expanding cavity on the inlet side of a positive displacement pump creates a low pressure area that needs to be filled with fluid. This cavity can be filled with fluid from the inlet line in normal performance. However, if slippage occurs, the cavity will also be partly filled with fluid flowing back through the pump clearances from the outlet side. When a positive displacement pump is operating under a slippage condition, the pump loses the ability to deliver the volume of fluid it is theoretically capable of pumping. For a given pump and fluid, the slippage is proportional to the pressure differential from outlet to inlet. If the pump had no slippage the volume pumped would be directly proportional to the rotational speed of the pump.

Typical failure modes of pumps are shown in Table 10-1 and Table 10-2.

Table 10-1. Typical Failure Modes of Centrifugal Pump Assemblies

FAILURE MODE	FAILURE CAUSE	FAILURE EFFECT
Reduction in suction head	- Pump cavitation	- Loss of pump efficiency
Reduction in pump pressure	- Pump cavitation	- Eventual erosion of impeller, casing - Pump noise and vibration
Component corrosion	- Incorrect fluid - Excessive flow rate for fluid	- Eventual catastrophic pump failure
Shaft deflection	- High radial thrust on pump Rotor	- Eventual shaft and pump failure
Shaft unbalance	- Impeller wear	- Shaft deflection and Misalignment - Stuffing box leakage - Seal leakage - Bearing wear
Air leak thru gasket / stuffing box	- Damaged gasket	- Loss of pump head
External Leakage	- Seal failure - Worn mechanical seal - Scored shaft sleeve - Stuffing box improperly packed	- Depends on type of fluid and criticality as to time of failure
Mechanical noise	- Debris in the impeller - Impeller out of balance - Bent shaft - Worn/damaged bearing - Foundation not rigid - Cavitation	- Eventual wear of impeller and other components
Positive suction head too low	- Clogged suction pipe - Valve on suction line only partially open	- Suction cavitation - Noisy operation - Low discharge pressure - High output flow rate
Pump discharge head too high	- Clogged discharge pipe - Discharge line valve only partially open	- Discharge cavitation - Noisy operation - Low output flow rate
Suction line / impeller clogged	- Contaminants	- Loss of pump output / reduced flow
Worn / broken impeller	- Wrong flow rate, contaminants	- Loss of pump output / reduced flow
Thrust bearing failure	- Excessive axial load	- Pump failure

Table 10-2. Typical Failure Modes of Positive Displacement Pump Assemblies

FAILURE MODE	FAILURE CAUSE	FAILURE EFFECT
Pump cavitation	- Reduction in suction head	- Pump noise and vibration - Eventual erosion of rotor, casing
Component corrosion	- Incorrect fluid - Excessive flow rate for fluid	- Eventual catastrophic pump failure
Low net positive suction head (NPSH)	- Pump cavitation	- Reduced pump efficiency
Shaft unbalance	- Torsional vibration	- Shaft deflection and misalignment - Seal leakage - Bearing wear
External Leakage	- Seal failure - Worn mechanical seal - Scored shaft sleeve - Stuffing box improperly packed	- Depends on type of fluid and criticality as to time of failure
Mechanical noise	- Bent shaft - Worn/damaged bearing - Foundation not rigid - Cavitation	- Eventual wear of piston or rotor, and other components
Positive suction head too low	- Clogged suction pipe - Valve on suction line only partially open	- Suction cavitation - Noisy operation - Low discharge pressure - High output flow rate
Pump discharge head too high	- Clogged discharge pipe - Discharge line valve only partially open	- Discharge cavitation - Noisy operation - Low output flow rate
Suction line clogged	- Contaminants	- No pump output / reduced flow
Pressure surges	- Incorrect NPSH	- Cavitation damage
Increased fluid temperature	- Incorrect fluid viscosity for pump	- Misaligned pump driver

10.2.1 <u>Cavitation</u>

Cavitation occurs when the pump suction is under a low pressure/high vacuum condition creating vapor bubbles at the inlet of the pump. Dynamic motion of the pump moves the bubbles to the discharge side of the pump where the discharge pressure collapses the vapor bubbles. This imploding action attacks the pump components removing chunks of material from their surfaces and causes premature failure of the pump. In order for cavitation to occur, the local pressure must be at or below the vapor pressure of the liquid. When a fluid flows over a surface having a curvature, there is a tendency for the pressure near the surface to be lowered. There is a separation of fluid flow lines where there are different velocity regions. Between these fluid regions, turbulence can form which may cause bubbles to occur if the pressure is low enough. The collapsing of these bubbles can cause noise and vibrations. Sometimes, these pressure changes can be very dramatic and cause extensive damage to impellers, rotors, casings or shafts. If exposed for a sufficiently long time, pitting or severe erosion can occur.

Cavitation generally occurs in the first stage of a multistage centrifugal pump, although second stages have also been found to be effected when the suction head is substantially reduced. With positive displacement pumps like the rotary screw, cavitation can also occur. For these pumps it is important to understand the characteristics of entrained and dissolved air with respect to the vapor pressure of the fluid medium. The rotary screw pump shows a greater tendency for cavitation when the total available pressure at the pump inlet is below atmospheric pressure. With both displacement and centrifugal pumps, cavitation can be identified and easily remedied. Many times the inlet piping arrangement can be modified which will cause flow patterns that alleviate the problem.

10.2.2 Vortexing

Vortexing in centrifugal pumps is caused by insufficient fluid height above the suction line entrance or excess fluid velocity at the suction line entrance causing a noisy pump operation and loss of fluid flow. Vortexing of the fluid in a suction sump or pit sounds a lot like cavitation problems and will cause excessive shaft deflection and damage to mechanical seals, bearings and the pump intake structure and piping. Vortexing problems are intermittent as the vortices form as opposed to cavitation which once started tends to be a constant problem. There are several possible causes of a vortexing problem:

- The pump running at a faster speed than specified
- A change in the flow or volume to the pump inlet
- The inlet line is restricted with contaminant solids
- Excess air in the liquid

10.2.3 Operating Environment

The effect of the ambient temperature and altitude on performance is normally independent of the type of pump. Limits for satisfactory performance are established primarily by the effect of the environment on the fluid rather than by the type of pumping action. Humidity only affects requirements for the pump casing. When operating temperature extremes are specified for a hydraulic system, the operating temperature of the fluid, not the ambient temperature, is usually the critical factor.

Minimum operating temperature is normally set by the increase in fluid viscosity as operating temperature is decreased. When the fluid viscosity is increased to the point where inlet conditions can no longer keep the pump completely full, cavitation, with possible pump damage, occurs. Fire resistant fluids have a higher specific gravity than petroleum oils and higher viscosity at lower temperatures. They may also contain water which can vaporize at lower pressures or higher temperatures. Thus, pump inlet conditions are more sensitive when these fluids are used. High altitudes can produce similar effects when the fluid reservoir is not pressurized.

Because pumps are designed for specific fluids, failure rates of seals can increase if alternate fluids are used. Above allowable operating temperatures, many oils will be too thin to maintain proper lubrication at high-load points, and may progressively deteriorate a graphite seal as a result of oxidation. Under elevated temperatures, some seals may harden. See Chapter 3.

10.2.4 Interference

For rotary displacement pumps, the interference problem must be seriously addressed since very small distortions of rotors will decrease the clearance causing rubbing or direct impact between the moving parts of the pump. Thermal expansion can also pose a threat if there is no care taken in the proper selection of materials. Improper installation can also lead to interference problems. With centrifugal pumps, cavitation significantly increases the interference problem because cavitation causes vibration and imbalance. Interference can be avoided by designing the parts with appropriate elastic and thermal properties so that excessive load or temperature won't significantly deflect internal parts. Manufacturing tolerances must be carefully maintained.

10.2.5 Corrosion and Erosion

Consideration must be made for other possible failure modes such as erosion corrosion. Erosion corrosion is an acceleration of the rate of corrosion attack in metal components due to the relative motion of a corrosive fluid and a metal surface. Erosion is dependent on the rate of liquid flow through the pump and also the angle of attack at which the fluid impinges on the material. A combination of erosion and corrosion can lead to extremely high pitting rates. The analysis of pump reliability must therefore determine if there are abrasive solids in the fluid.

10.2.6 Material Fatigue

Pump shafts, bearings, gears, impellers and all other moving metal components of the pump are candidates for material fatigue failure. Material fatigue occurs with all types of pumps, but may have more of an effect on displacement pumps, which have higher fluctuating stresses. One component that often creates a material fatigue problem is an unbalanced impeller resulting in the vibration of components throughout the pump.

10.2.7 Bearing Failure

Although bearings are relatively inexpensive, they can cause costly shutdowns of complete systems. Short bearing life for centrifugal pumps, for example, can be caused by a number of problems including the following: misalignment, a bent shaft, a rotating part rubbing on a stationary part, an impeller out of balance causing vibration, excessive thrust caused by mechanical failure inside the pump, excessive bearing temperature caused by lack of lubrication, dirt or other contaminant in the fluid, excessive grease or oil in an anti-friction bearing housing, and rusting of bearings from water in housing.

Most bearing problems can be classified by the following failure modes: fatigue, wiping, overheating, corrosion, and wear. Fatigue occurs due to cyclic loads normal to the bearing surface. Wiping occurs as a result of insufficient lubrication film thickness and the resulting surface-to-surface contact. Loss of sufficient lubricant film thickness can occur from under-rotation or from system fluid losses. Overheating is shown by babbitt cracking or surface discoloration. Corrosion is frequently caused by the chemical reaction between the acids in the lubricants and the base metals in the babbitt.

10.3 MODEL DEVELOPMENT

Reliability models have been developed to address the difference between pump types. Because of the physical design differences between centrifugal and displacement pumps they have specific performance and reliability advantages and disadvantages. Centrifugal pumps are limited by pressure but can supply almost any amount of capacity desired. Displacement pumps lose capacity as the pressure increases due to the increase in slippage which occurs with an increase in pump pressure. The amount of slippage can vary from pump to pump depending on the actual manufactured clearances in the pump chamber. The slippage can also increase with time as wear increases. Slippage will be inversely proportional to the fluid viscosity. Equation (10-1) shows that since slippage “S” increases as the pressure requirements increase, the value of capacity “Q” is decreased.

$$Q = 7.48\left(DN - S\right) \qquad \textbf{(10-1)}$$

Where: Q = Capacity, gpm

D = Net fluid transferred or displaced by one cycle, ft^3

N = Rotation or cyclic speed, revolutions/minute, cycles/minute

S = Slippage, ft^3/min (The quantity of fluid that escapes the full discharge cycle through clearances or other leak paths)

The impellers, rotors, shafts, and casings are the pump components which should generally have the longer lives when compared to bearings and seals. With good designs and proper material selection, the reliability of impellers, rotors, shafts and casings should remain very high. In order to properly determine total pump reliability, failure rate models have been developed for each pump component.

Pump assemblies are comprised of many component parts including seals, shaft, bearings, casing, and fluid driver. The fluid driver can be further broken down into the various types common to pumps including the two general categories for centrifugal and displacement pumps. For displacement pumps, it will be broken down into two further categories: reciprocating and rotary. For reciprocating pumps the fluid drivers can be classified as piston/plunger type or diaphragm type. For rotary pumps the fluid driver is a vane type for single rotors and for multiple rotors it is common to find a gear, lobe, or screw type of fluid driver. The total pump failure rate is a combination of the failure rates of the individual component parts. The failure rate for centrifugal pumps and displacement pumps can be estimated using Equation (10-2).

$$\lambda_P = \lambda_{SE} + \lambda_{SH} + \lambda_{BE} + \lambda_{CA} + \lambda_{FD} \qquad \textbf{(10-2)}$$

Where: λ_P = Total failure rate of the pump

λ_{SE} = Total failure rate for all pump seals, failures/million operating hours (See Section 10.4 and Chapter 3)

λ_{SH} = Failure rate for the pump shaft, failures/million operating hours (See Section 10.5 and Chapter 20)

λ_{BE} = Total failure rate for all pump bearings, failures/million operating hours (See Section 10.6 and Chapter 7)

λ_{CA} = Failure rate for the pump casing, failures/million operating hours (See Section 10.7)

λ_{FD} = Failure rate for the pump fluid driver, failures/million operating hours (See Section 10.8)

10.4 FAILURE RATE MODEL FOR PUMP SEALS, λ_{SE}

The pump assembly will contain several types of seals such as O-rings on the fluid connectors, gaskets and mechanical seals. Failure rate equations for O-rings and gaskets are contained in Chapter 3. Mechanical seals of the pump require a specific evaluation of reliability. Mechanical seals for rotating shaft applications move the point of the seal away from the shaft to specially designed sealing faces that gradually wear down. A balanced mechanical seal is designed so that the effective contact pressure is always less than the fluid pressure, reducing friction at the seal faces. The result is less rubbing wear, less heat generated and higher system fluid pressure capability. In an unbalanced seal, fluid pressure is not relieved by the face geometry; the seal faces withstand full system fluid pressure plus surge pressures and spring pressure. Thus, the face contact pressure is greater than or equal to system fluid pressure. The balanced seal design is more expensive than the unbalanced design but provides higher reliability and longer life. Chapter 3 contains specific failure rate equations and failure modes for mechanical seals.

10.5 FAILURE RATE MODEL FOR PUMP SHAFT, λ_{SH}

A typical pump shaft assembly is shown in Figure 10.3. The reliability of the pump shaft itself is generally very high when compared to other components of the pump. Torque limits of the shaft must be compatible with requirements of the pump application. High viscosity/low speed applications produce high torque requirements. Studies have shown (Reference 26) that the average failure rate for the shaft itself is about eight times less than mechanical seals and about three times less than that of the ball bearings. The possibility that the shaft itself will fracture, or become inoperable is very unlikely when compared to the more common pump failure modes. Usually the seals or bearings will cause problems first.

The effect of the shaft on reliability of other components is of greater importance than the reliability of the shaft itself. For centrifugal pumps, there is a large difference in deflection among the types of pump casing design. In a single volute casing, there are varying amounts of fluid pressure distributed about the casing causing unequal distributions of forces on the pump shaft. This imbalance causes shaft deflection and greater seal and bearing wear.

The amount of radial thrust will vary depending on the casing design and fluid flow. The thrust load will increase from normal operation for any type of casing design when the pump is not run at its optimum flow rate speed. When the pump is not operating at its optimum rate, then the type of casing design will have a significant effect on the radial load.

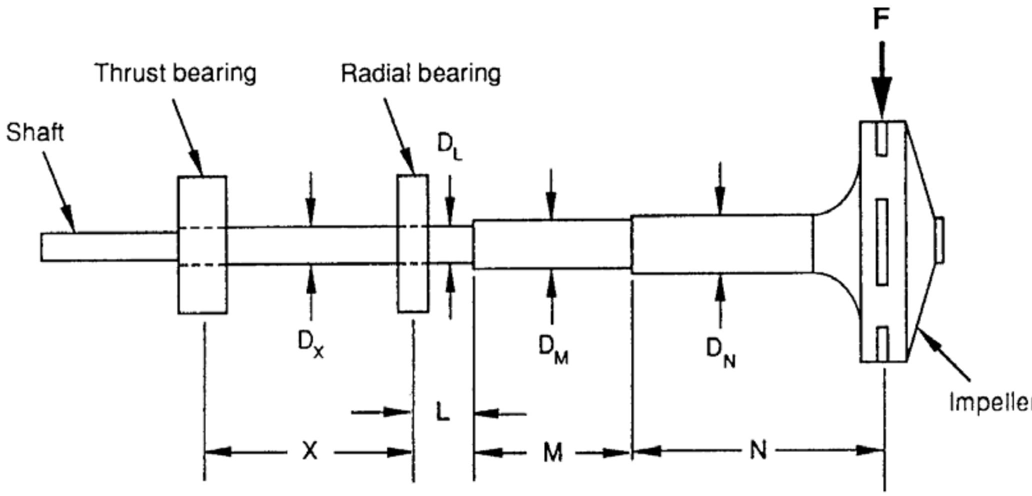


Figure 10.3 Typical Pump Shaft Assembly
(Reference 8)

The single volute type shows the greatest pressure imbalance and hence, the greatest deflection. This imbalance is decreased through different casing designs. The modified concentric casing and the double volute casing both have lower relative radial thrust because they cause a more even pressure distribution across the face of the impeller. The double volute is the most balanced and the design with the least amount of radial thrust. The maximum deflection recommended for a shaft design is approximately 0.001 inches.

The shaft diameter may be stepped up several times from the end of the coupling to its center to facilitate impeller mounting. Starting with the maximum diameter at the impeller mounting as shown in Figure 10.3, there is a step down for the shaft sleeve and another for the external shaft nut, followed by several more for the bearings and the coupling. Therefore, the shaft diameter at the impeller exceeds that required for torsional strength at the coupling by at least an amount sufficient to provide all intervening stepdowns. Chapter 20 provides the reliability model for pump shafts.

10.6 FAILURE RATE MODEL FOR PUMP BEARINGS, λ_{BE}

Bearings are used in pump assemblies to locate the rotating or linear element in its correct position relative to the stationary parts of the pump, maintain minimum friction and absorb radial and axial loads transmitted through the shaft. Bearing reliability will in turn be affected by the strength of the shaft and housing to minimize the effects of externally induced stresses or vibration. The lubricant separates the rolling elements and raceway contact surfaces and minimizes the effect of friction. Chapter 7 provides the equations and procedures for determining bearing reliability.

10.7 FAILURE RATE MODEL FOR PUMP CASING, λ_{CA}

The pump casing is normally a very reliable component. Defined as λ_{CA}, the casing failure rate will have a greater effect on total pump reliability from the standpoint of how it affects other less reliable components. For instance, for an ANSI pump, the casing may have an average life expectancy of 10 years where a seal or bearing may have only one or two years. However, the type of casing used in the pump can have a large effect on the lifetime of the bearings and seals. This is due to differing loads placed on the pump shaft by the fluid flow pattern. The fluid flow patterns are a function of the casing design. The failure rate of the pump casing (λ_{CA}) itself can be estimated at 0.01 failures/million hours.

10.8 FAILURE RATE MODEL FOR FLUID DRIVER, λ_{FD}

All pumps require some vehicle to move the fluid from the intakes and expel it through the volutes and output ports to the exhaust opening. The means by which pumps do this is what differentiates most of today's numerous types of pumps. The reliability of these fluid drivers will vary from pump to pump. Impellers will wear out long after the seals. Pump gears for rotary gear pumps will have a lower reliability than impellers due to the nature of the contact between gears and the speed they attain.

Piston-plunger displacement pumps will generally have greater wear rates for the piston walls and rings than for the impellers of centrifugal pumps. The average failure rates in Table 10-4 have been determined from data base information developed from various failure rate data sources. The equations that describe the fluid driver wear rate may vary drastically since the fluid driver varies greatly in design and application. Other chapters of this Handbook can be used to estimate the failure rates for slider-crank mechanisms, mechanical couplings, valves and other components and parts unique to the particular pump design.

In a centrifugal pump, the net positive suction head (NPSH) varies as a function of flow determined by pressure. In a positive displacement pump NPSH varies as a function of flow determined by speed. Reducing the speed of a positive displacement pump reduces the NPSH. NPSHA is the actual NPSH and NPSHR is the required NPSH obtained from the pump manufacturer. Depending on the pump design a NPSHA / NPSHR margin of 1.1 to 2.5 should be obtained.

The failure rate of a pump fluid driver can be estimated from the following equation:

$$\lambda_{FD} = \lambda_{FD,B} \bullet C_{PF} \bullet C_{PS} \bullet C_C \bullet C_{SF} \qquad \textbf{(10-3)}$$

Where: λ_{FD} = Failure rate for the pump fluid driver, failures/million operating hours

$\lambda_{FD,B}$ = Base failure rate of pump fluid driver (See Table 10-4)

C_{PF} = Percent flow multiplying factor (See Section 10.8.1)

C_{PS} = Operating speed multiplying factor (See Section 10.8.2)

C_C = Contaminant multiplying factor (See Section 10.8.3)

C_{SF} = Service factor multiplying factor (See Table 10-5)

10.8.1 Percent Flow Multiplying Factor

A centrifugal pump is designed to operate most reliably at one capacity for a given RPM and impeller diameter. This flow rate is called the Best Efficiency Point (BEP). At this flow, hydraulic loads imposed on the impeller are minimized and are steady. At flows greater than or less than the BEP the hydraulic loads increase in intensity and become unsteady due to turbulence in the casing and impeller. As a result, hydraulic loads, which are transmitted to the shaft and bearings, increase and become unsteady. Shaft deflection changes as a function of the fluid flow rate through the pump. As a pump's capacity increases or decreases, moving away from the point of maximum efficiency, fluid pressures around the impeller become unequal, tending to deflect it. Special casings, such as diffusers and double-volute and concentric casings can greatly reduce the radial thrust and, hence, the deflection.

Also, the severity of these unsteady loads can cause failures of the mechanical seal. Operation at reduced flow rates that put the pump into its recirculation mode can also lead to cavitation damage in high suction energy pumps. The effect on reliability of operating a pump too far from its maximum efficiency point is shown in Figure 10.4.

The Percent Flow multiplying factor (C_{PF}) is dependent upon the casing type and pump capacity percentage. The pump capacity percentage is the actual operating flow divided by the maximum pump specification flow. Values for C_{PF} are shown in Figure 10.4 and related equations are included in Table 10-3.

10.8.2 Operating Speed Multiplying Factor

Operating speed affects the failure rate of a pump caused by the increased rotational or cyclic speed leading to accelerated rubbing wear of shaft and mechanical seal faces, increased bearing friction and lubricant degradation. Increased operating speed also increases the energy level of the pump which can lead to cavitation damage. The effects of wear on these components are almost linear as a function of speed.

Equation 10-4 provides a multiplying factor for operating speed based on actual and design speed. See Figure 10.5.

$$C_{PS} = k \bullet \left(\frac{V_O}{V_D} \right)^{1.3} \tag{10-4}$$

Where: V_O = Operating Speed

V_D = Maximum Allowable Design Speed

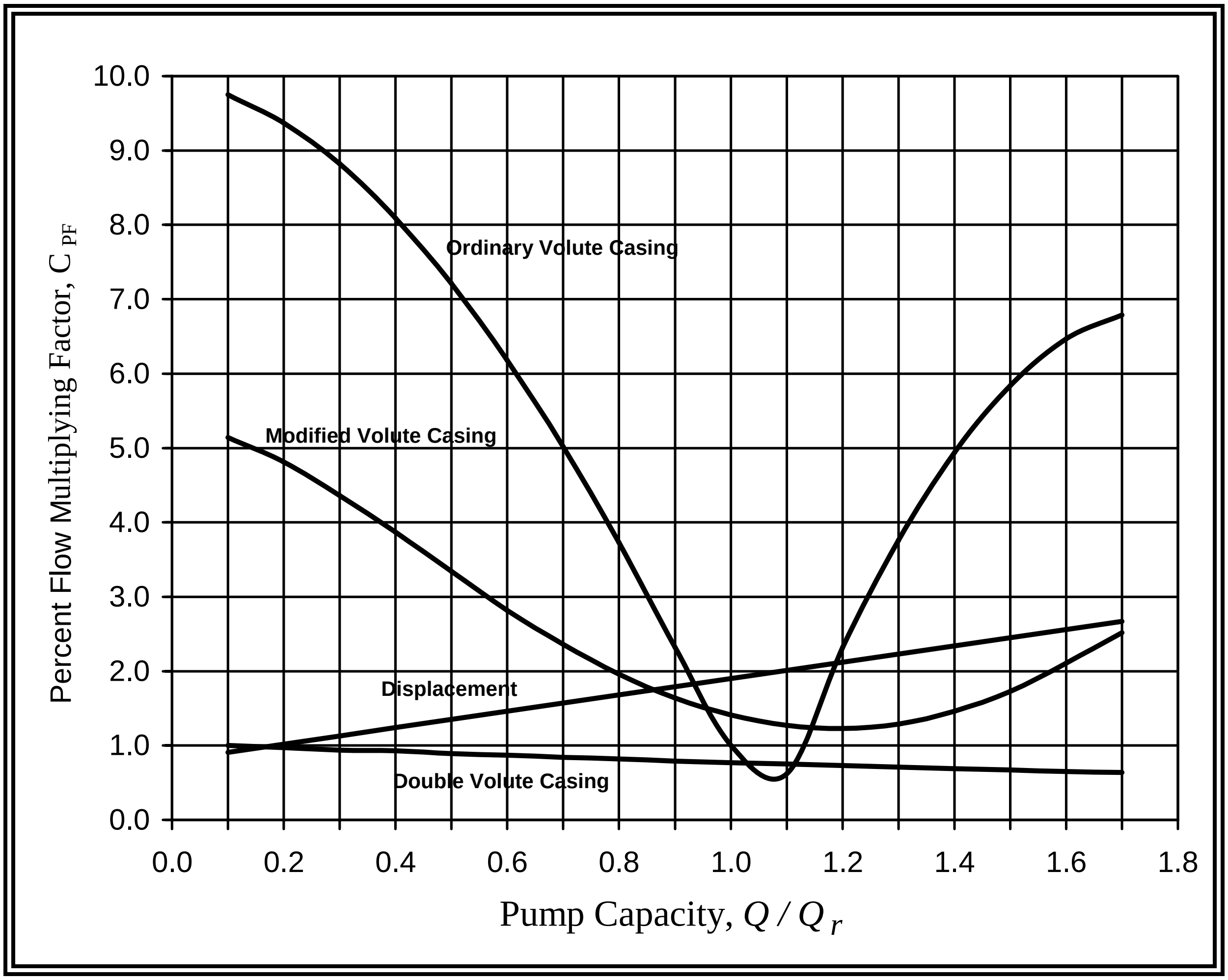
k = Constant = 5.00

10.8.3 Contaminant Multiplying Factor

The contaminant factor, C_C, was developed from research performed for the Naval Air Warfare Center in Warminster, Pennsylvania on the effect of contamination and filtration level on pump wear and performance. The contamination factor equation is based on the filtration level as follows:

$$C_C = 0.6 + 0.05 F_{AC} \tag{10-5}$$

Where: F_{AC} = Filter size, microns

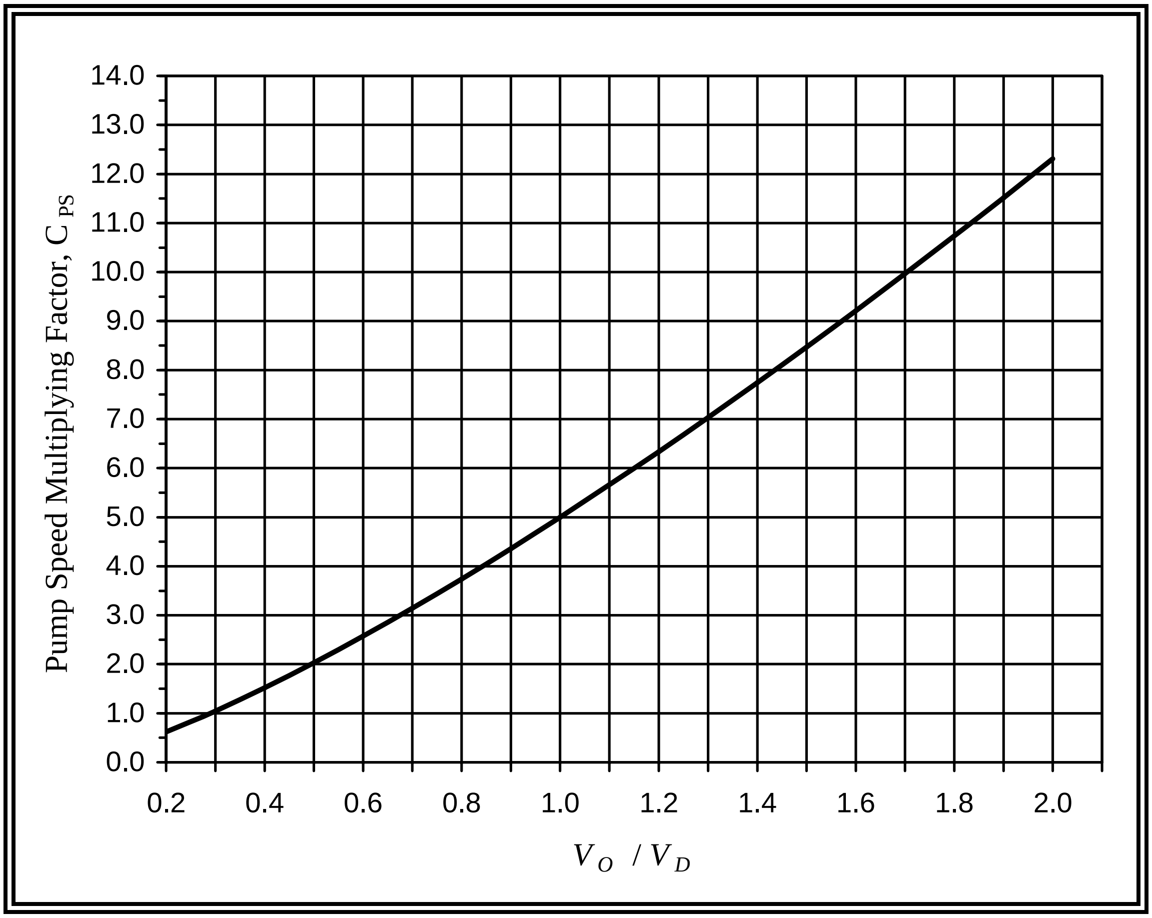


Q = Actual operating pump flow, gpm

Q_r = Specified maximum pump flow, gpm

(See equations in Table 10-3)

Figure 10.4 Percent Flow Multiplying Factor as a Function of Pump Capacity and Casing Design
(Reference 26)



$$C_{PS} = k \bullet \left(\frac{V_O}{V_D} \right)^{1.3}$$

Where: V_O = Operating speed

V_D = Maximum allowable design speed

k = Constant = 5.00

Figure 10.5 Pump Operating Speed Multiplying Factor

Table 10-3. Equations for Figure 10.5, Percent Flow Multiplying Factor, C_{PF}

Ordinary volute casings:

For 0.1 $\leq Q_r \leq$ 1.0:

$$C_{PF} = 9.94 - 0.90\left(\frac{Q}{Q_r}\right) - 10.00\left(\frac{Q}{Q_r}\right)^2 + 1.77\left(\frac{Q}{Q_r}\right)^2$$

Where: C_{PF} = Percent Flow multiplying factor
Q = Actual operating flow, gpm
Q_r = Maximum pump specified flow, gpm

For 1.0 < Q/Q_r < 1.1: $C_{PF} = 1.0$

For 1.1 $\leq Q/Q_r \leq$ 1.7: $C_{PF} = -30.60 + 36.00\left(\frac{Q}{Q_r}\right) - 4.50\left(\frac{Q}{Q_r}\right)^2 - 2.20\left(\frac{Q}{Q_r}\right)^3$

Modified volute casings:

For 0.1 $\leq Q/Q_r \leq$ 1.7:

$$C_{PF} = 5.31 - 0.55\left(\frac{Q}{Q_r}\right) - 12.00\left(\frac{Q}{Q_r}\right)^2 + 12.60\left(\frac{Q}{Q_r}\right)^3 - 4.63\left(\frac{Q}{Q_r}\right)^4 + 0.68\left(\frac{Q}{Q_r}\right)^5$$

Double volute casings:

$$C_{PF} = 1.03 - 0.30\left(\frac{Q}{Q_r}\right) + 0.04\left(\frac{Q}{Q_r}\right)^2$$

Displacement Pumps:

$$C_{PF} = 0.80 + 1.1\left(\frac{Q}{Q_r}\right)$$

Table 10-4. Base Failure Rates for Pump Fluid Drivers ($\lambda_{FD,B}$)

PUMP TYPE	FLUID DRIVER MODE	MODEL TYPE	BASE RATE*
Centrifugal	- Axial flow impeller	- Closed / open impellers	0.20
	- Mixed flow / radial flow impeller	- Open / semi-open / closed impellers	0.12
	- Peripheral	- Single stage / multi-stage	0.20
Displacement	- Reciprocating	- Piston / plunger	1.18
	- Reciprocating	- Diaphragm	0.58
	- Rotary (single rotor)	- Vane	0.4
	- Rotary (single rotor)	- Piston	1.05
	- Rotary (multiple rotor)	- Gear	0.75
	- Rotary (multiple rotor)	- Lobe	0.45
	- Rotary (multiple rotor)	- Screw	0.58

* Failures/million hours of operation

Table 10.5 C_{SF} Service Factors

Duration of Service (hours per day)	Uniform Load	Moderate Shock	Heavy Shock	Extreme Shock
Occasional (½ hour)	1.00	1.00	1.00	1.25
Less than three hours	1.00	1.00	1.25	1.50
3-10 hours	1.00	1.25	1.50	1.75
Over 10 hours	1.25	1.50	1.75	2.00

10.9 REFERENCES

In addition to specific references cited throughout Chapter 10, other references included below are recommended in support of performing a reliability analysis of a pump assembly.

8. Block, H. and D. Johnson, “Downtime Prompts Upgrading of Centrifugal Pumps”, Chemical Engineering Magazine, pp 35-38 (25 Nov 1985)

19. Hindhede, U., et al, “Machine Design Fundamentals”, John Wiley & Sons, NY, 1983

26 Igor J. Karassik et al, “Pump Handbook”, McGraw-Hill Book Company, NY (1986)

39. Shigley, J.E., Mischke, C.R. “Mechanical Engineering Design”, McGraw-Hill Book Company, NY, (1989)

62. Baumeister, T, et al, Mark's Standard Handbook for Mechanical Engineers, McGraw-Hill Book Company

85. Lev Nelik, “What Happens When a Pump No Longer Operates At Optimum Conditions”, Pumps and systems, February 2005

86. Allan Budris, Eugine Subini, R. Barry Erickson, ”Pump Reliability – Correct Hydraulic Selection Minimizes Unscheduled Maintenance”, PumpLines, Fall 2001

123. Centrifugal Pump & Mechanical Seal Manual, William J. McNally, 2009

CHAPTER 11

FLUID FILTERS

11.0 TABLE OF CONTENTS

11.1 INTRODUCTION

Fluid filtration equipment is unique in that the reliability of this equipment is more concerned with the effects of the filter on associated equipment than on the lifetime of the filter itself. This is due to severe wear of fluid system components which can occur when these components are operated with poorly filtered fluid. Proper monitoring of filter reliability can minimize maintenance costs of the overall fluid system through contamination control of fluid system components caused by abrasive wear. Other problems in the fluid system that can be prevented by a reliable filter include the following:

- Increased internal and external leakage and lower efficiency of pumps, actuators, control valves and motors.
- Sticking of actuators and control valves due to sludge and collection of fine particles in critical operational areas.
- Seizure of components caused by large amounts of contaminants getting stuck in clearances.
- Corrosion within components from acids that form as a result of mixing of incompatible particles and fluids.

The basic function of a filter is to remove particle contaminants from the fluid that reduce the service life of system components through abrasive wear. Sources of contamination that the filter is designed to remove include contaminants present from the manufacturing and installation process, maintenance procedures, replacement fluid, environmental contamination, rod seal leakage, breather caps and system component wear. Cost of a filtration system is a consideration in a fluid system design and there is an optimum level of cleanliness in any fluid system where increased filtration does not significantly affect component wear. The acceptable fluid cleanliness level must be determined for the particular system that will support the reliability requirements of the complete system.

The fluid cleanliness level required by a particular fluid system largely determines the particle blocking size and efficiency rating of the filter. Filters are rated in accordance with the size of the particles they remove and the efficiency with which they remove them. This measure of the filter's ability to remove contaminants is called the filtration ratio or beta (β) ratio. This ratio is defined as the number of particles of a specified size (and larger) upstream of the filter relative to the number of particles of the same size downstream of the filter. Beta ratio can be written as follows:

$$\beta_x = \frac{\text{number of particles of size x and larger upstream of the filter}}{\text{number of particles of size x and larger downstream of the filter}}$$

Filter performance may also be expressed as beta efficiency in %.

$$\% = \frac{\text{number of particles of size x upstream - number of particles of size x downstream}}{\text{number of particles of size x upstream}}$$

Another performance parameter to be considered in a reliability review is the particle penetration which is the percentage of particles that are not captured by the filter. These filter performance ratings will be based on laboratory tests and the expected operational reliability of the filter must therefore consider the operating environment of the system containing the filter. Filter reliability will depend on several parameters including the strength of the porous filter media with respect to contaminant loading under system fluid conditions, fluid pressure surges and cold start-ups.

This chapter will review the conditions which can lead to degradation or failure of the filter. Because a reliability review of the filter is focused on protecting other components in the system from contamination, the effect of contamination on the wear of various components is discussed. A failure rate equation for the filter assembly itself will then be developed.

11.1.1 Filtration Mechanisms

Filters consist of a porous filter media through which fluid is passed. The filter media is typically corrugated to increase the amount of filtration area within the filter volume. Filtration of gases is accomplished by absorption of suspended particles. Filtration of liquids is primarily accomplished by direct interception of the suspended particles. A typical pressure line filter is shown in Figure 11.1.

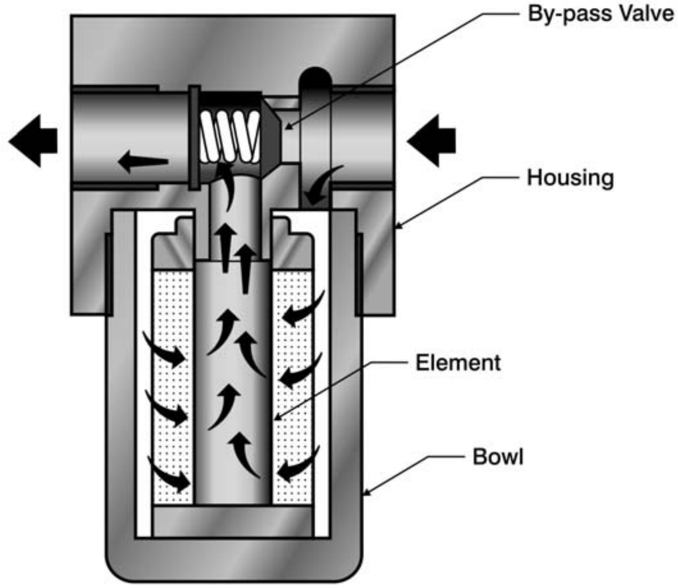


Figure 11.1 Typical Filter Construction

Filters are often placed in the return line of the fluid system as opposed to the pressure side since if the reservoir and the fluid it contains start out clean and all fluid entering the reservoir and returning fluid is adequately filtered, then fluid cleanliness will be maintained. Also the pressure across the filter will be less than that upstream of the components yet sufficient to force fluid through the filter media.

Another consideration of operational reliability is the continuous monitoring of filter pressure drop providing early warning of component failures or filter failure. For example, if the pressure across the filter suddenly increases, this could be an indication of imminent failure of a component upstream. Similarly, a sudden decrease in pressure across the filter could indicate a rupture of the filter element. Continuous monitoring of the filter can provide clues as to the performance of the filter and the condition of the fluid system.

11.1.2 Service Life

The porous structure of a filter media presents a resistance to fluid flow which causes a pressure drop across the filter. This filter differential pressure increases as captured particles or contaminants are collected and plug the porous media. Every system has a maximum differential pressure at which the filter must be cleaned or replaced. The filter service life is the time it takes the filter to reach the maximum allowable differential pressure. Use of the filter beyond its service life could result in catastrophic failure of the filter due to the high differential pressure or it could result in unfiltered fluid bypassing the filter (Reference 33).

11.1.3 Filter Failure

A filter is considered to have failed when it releases previously captured contaminants, when it allows unfiltered fluid to pass throughout the filter media, or when the filter collapses and contaminates the fluid with contents of the filter media. Plugging of the filter with contaminants, with a resulting increase in filter differential pressure, can be a normal consequence of operation. Failure due to premature plugging can occur for several reasons as described in Section 11.2. Failure of the filter can be caused by operating conditions such as high differential pressures, cyclic flow, vibration, system startups when cold, and even by the fluid that is being filtered if the fluid is incompatible with the filter. Typically, 67% of filter failures are due to leakage and 33% are due to clogging.

11.2 FILTER FAILURE MODES

Typical failure modes and causes for filter assemblies are summarized in Table 11-1. Some of the more common failure modes causing performance degradation of the filter are described as follows:

Channeling: Excessively high differential pressures can cause filter media pores to enlarge, allowing large amounts of unfiltered fluid to bypass the filter media. Enlargement of the media pores also allows previously captured contaminants to be released. Channeling can also be the result of media fatigue caused by cyclic flow conditions. Cyclic flow is caused by the periodic increase and decrease of system pressure such as from pump ripple creating a wave form of fluid flow through the filter.

Fatigue Cracks: Cyclic flow conditions in the fluid system can also cause fatigue cracks in the filter media. Such cracks may occur at the roots of pleats in corrugated filters or within the volume of loose packed media. The cracks will allow the release of contaminants from the filter and will allow some of the fluid to bypass the filter.

Table 11-1. Failure Modes of Filters

FAILURE MODE	FAILURE CAUSE	FAILURE EFFECTS
Channeling	- High differential pressures - Cyclic flow	- Release of contaminants - Circulation of unfiltered fluid
Fatigue cracks	- Cyclic flow	- Circulation of unfiltered fluid - Release of filter media
Media Migration	- Vibration - Cyclic flow - Cold starts	- Release of filter media
Filter disintegration	- Embrittlement - Cold starts - High differential pressures	- Substantial contamination of fluid with filter media
Plugging	- Condensed moisture - Oil saturated with contaminants - Absorption of oxidation products - Absorption of coolant - Accumulation of wear debris	- Increase in filter differential pressure, definition of failure dependent on system design and maintenance schedule

Media Migration: Improper bonding of the media fibers or deterioration of the bonding can result in the down stream release of media fibers. This downstream release of the filter media is termed media migration. Media migration during vibration of the filter may result from an improper fit of the filter in the filter housing or may result from the filter media abrading against the filter casing. Media migration can also occur in conjunction with fatigue cracks in the media, as caused by cyclic flow conditions. Media migration can also occur during cold temperature start-ups due to potential large differential pressure generated as a consequence of increased fluid viscosity.

Filter Disintegration: Complete disintegration of the filter can occur as a result of extremely high differential pressures. Disintegration can also be the result of embrittlement of the filter media from exposure to incompatible fluids or cold temperatures.

Plugging: Filter plugging can be caused by normal equipment operation or by filter failure, depending upon when the filter becomes plugged. Failure due to premature plugging can be attributed to several causes other than just the accumulation of wear debris such as the saturation of the fluid being filtered by contaminants or moisture combining with fluid additives to form thick filter-plugging gels.

11.3 FLUID CONTAMINATION EFFECTS

Contamination of the operating fluid with hard particles can cause progressive performance deterioration throughout the fluid system through an abrasive wear mechanism. This type of wear is characterized by a particle penetrating a softer surface and cutting away material. The rate of wear and thus the rate of performance degradation is dependent on the number of particles and the particle hardness. Particle contamination can also cause cumulative performance degradation where a rapid decline in performance follows an extended period of apparently normal operations. This type of degradation failure is caused by the creation of surface defects during operation. These surface defects may be caused by abrasion, surface fatigue or adhesion wear processes.

Fluid systems requiring filtration typically include components such as pumps, gears, control valves, ball bearings, roller bearings, journal bearings, and seals. Their potential effect on contamination of the fluid system can be described as follows (Reference 7): The following failure modes can be eliminated by effective filtering.

Pumps: In displacement-type piston pumps, the piston face can be damaged by cavitation or corrosion. Contaminant particles can enter the lubricant film between the piston and cylinder and plough the surface several times before being ejected. In swashplate controlled devices, such as variable displacement pumps and hydraulic motors, the piston shoes can cause abrasion-wear-type degradation failures as the shoes are highly loaded and are in sliding contact with the swashplate. Similar abrasive-wear-type degradation failures can occur to the sliding contact surfaces of the rotating cylinder block and the mating valve pressure plates.

Gears: Gear failures are primarily failures of the gear tooth surface. This surface is damaged by rubbing wear, scoring and pitting. Rubbing wear is generated by both adhesive and abrasive wear mechanisms. Scoring of the tooth surface is generated by the adhesive wear type mechanisms under intense local frictional heating. Pitting occurs as a result of tooth surface fatigue wear. These three wear mechanisms are all aggravated by contaminate particles.

Valves: Particle contamination can cause increased leakage in control valves by severe cutting or by milder abrasive wear mechanisms. Synthetic phosphate ester fluids have been found to cause servo valve erosion by the corrosion process. A brittle corrosion layer is formed on the valve and is abraded by fluid-borne particles, adding additional particulates to the fluid and exposing base metal, allowing further corrosion. Deterioration failures of relief valves can occur from particle contamination caused by erosion. Contamination of hydraulic fluid by water has been shown to cause rust inhibitor additive to attach to servo valve spools and prevent movement of the valve spool within its housing.

Bearings: Hard particle contamination of ball and roller bearing lubricants is the cause of two types of abrasive wear of the rolling surfaces. Hard particle contamination causes rolling surface damage that dominates the fatigue life of ball bearings under typical operating conditions. In severe circumstances, hard particle contamination causes indentations and pits which cause rapid failure of the rolling surfaces. Abrasive wear, increasing with particle concentration and hardness, can remove material from the sliding edges of a tapered rolling bearing, reducing the bearing width and allowing increased misalignment. Wear of this type does not stop until the contaminant size is reduced to less than the lubricant film thickness.

The performance of new journal bearings improves with use initially due to better surface conformity caused by wear during boundary lubrication conditions. As wear in the contact region progresses, the performance begins to gradually deteriorate. Wear of the journal bearings is caused by both abrasive and adhesive wear due to the sliding motion in the contact region. Contamination of the lubricant with water can cause the formation of a metal oxide boundary layer on the bearing which can inhibit adhesive wear. However, abrasion of this film can cause bearing failure due to rapid increases in wear, bearing corrosion, and the number of abrasive oxide particles. Maximum bearing life can be achieved by selecting a filter to filter out all particles larger than the minimum lubricating film thickness.

Seals: Seal failures are typically caused by fatigue-like surface embrittlement, abrasive removal of material, and corrosion. Elastomeric seals are more sensitive to thermal deterioration than to mechanical wear. However, hard particles can become embedded in soft elastomeric materials and sliding contact metal surfaces, causing leakage by abrasive wear of the harder mating surfaces. Abrasive particles can also plug lubricant passages which causes seal failure from the lack of lubricant.

Actuators: Clearances between the actuator piston and cylinder are critical to achieve the required axial force required upon actuation. Hard particles between this clearance will cause early abrasive wear and early failure. Small particles entering this clearance will cause the piston to stick and poor response performance of the actuator.

11.4 FILTER RELIABILITY MODEL

The filter manufacturer may provide a filter life expectancy based on various ISO multi-pass tests. These tests are conducted under steady state conditions. Use of the life expectancy based on these tests would assume the following:

(1) The rate of generation of contaminate particulates by system components and the rate of ingestion of environmental contaminates do not vary with time and the particulates are evenly mixed within the system fluid.

(2) The system fluid volume and flow rates do not change with time.

(3) The filter will not plug or become unusually restricted prior to reaching its maximum capacity.

These conditions do not exist in normal fluid system operation. While the published life expectancy of the filter can be used to determine a base failure rate, this rate needs to be modified with multiplying factors that consider a variable flow and pressure drop across the filter to adjust the base failure rate to actual operating conditions.

A typical fluid system consists of a reservoir, pump, filter, one or more control valves, and one or more fluid motors. A simplified system is shown in Figure 11.2.

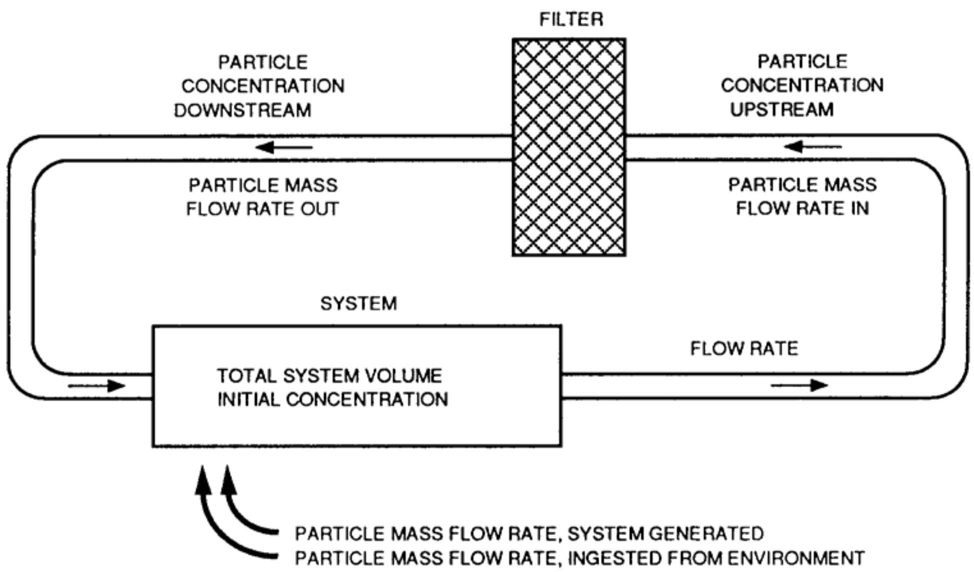


Figure 11.2 Simplified Fluid System Containing Filter

The complete filter failure rate model needs to have correction factors to modify the base failure and to account for potentially degrading effects of operating conditions. Considering the causes of failures as earlier discussed, the failure rate model can be written as:

$$\lambda_F = \lambda_{F,B} \bullet C_{DP} \bullet C_V \bullet C_{CS} \bullet C_{CF} \quad \textbf{(11-1)}$$

Where: λ_F = Failure rate of the filter in failures/million hours

$\lambda_{F,B}$ = Base failure rate of the filter in failures/million hours (See Section 11.4.1)

C_{DP} = Multiplying factor which considers the effects of the filter differential pressure on the base failure rate (See Section 11.4.2)

C_V = Multiplying factor which considers the effects of vibration on the base failure rate (See Section 11.4.3)

C_{CS} = Multiplying factor which considers the effects of cold start-up conditions on the base failure rate (See Section 11.4.4)

C_{CF} = Multiplying factor which considers the effects of cyclic flow on the base failure rate (See Section 11.4.5)

11.4.1 Base Failure Rate

The mass of contaminant particles retained by the filter determines the life of the filter. However, by definition, a clogged filter is not normally classified as a component failure. See Table 11-1 for a list of typical failure modes. The filter manufacturer may provide a filter life expectancy based on various ISO multi-pass tests. This life expectancy in hours (θ) can be converted to a base failure rate ($\lambda = 1/\theta$). A typical value for a base failure rate obtained from various data sources is given as 2.53 failures per million operating hours. Typically, the failure rate distribution will be 67% of the failures due to internal leakage and 33% of the failures due to plugging. For a more detailed analysis, the following paragraphs describe the multiplying factors to be used in conjunction with this base failure rate.

$\lambda_{F,B}$ = Base failure rate of a filter in normal operation **(11-2)**

= 2.53 failures/million hours or as obtained from manufacturer

11.4.2 Filter Pressure Multiplying Factor

The probability of particulate unloading increases with increased differential pressure. As this pressure is increased, media pores within the filter media are enlarged allowing unfiltered fluid to pass through the filter. Extremely high pressures can cause the filter to completely disintegrate adding more contaminates into the system. During a reliability review of the fluid system, the tensile strength of the filter media needs to be considered in relation to system pressure.

A multiplying factor to consider the effects of system pressure on the base failure rate can be written as follows:

$$C_{DP} = 1.25 \frac{P_O}{P_R} \qquad \textbf{(11-3)}$$

Where: P_O = Operating upstream filter pressure

P_R = Rated filter pressure

11.4.3 <u>Vibration Multiplying Factor</u>

Most filters are tested for media migration caused by vibration. A typical test is performed with the filter immersed in the system fluid and the filter is exposed to low amplitude, high frequency vibrations for about 100,000 cycles. As a result, most filters will not degrade due to vibration. However, in aircraft environments, failure of the filter housing and seals due to vibration accounts for 80 percent of the total filter failure rate (Reference 15). Thus it appears that in most systems vibration is not a problem, but in aircraft and vehicle systems excessive vibration can cause filter failure. As a result:

C_V = 1.25 for aircraft and mobile systems

C_V = 1.00 for all other systems

11.4.4 <u>Cold Start Multiplying Factor</u>

The correction factor for cold start degradation can be calculated using a ratio of the cold start fluid viscosity to the normal operating fluid viscosity. This is:

$$C_{CS} = \left(\frac{\nu_{\text{cold start}}}{\nu_{\text{normal}}} \right)^x \qquad \textbf{(11-4)}$$

Where: $\nu_{\text{cold start}}$ = Viscosity at cold start temperature, lb-min/in^2

ν_{normal} = Viscosity at normal operating conditions, lb-min/in^2

x = Exponent which varies with type of fluid

Values for ν and x can be obtained from Table 11-3. As shown in this table, values for x will range from 0.20 for light viscous fluids such as kerosene to 1.1 for heavy viscous fluids such as SAE 70 oil.

11.4.5 Cyclic Flow Multiplying Factor

Filtration performance will decrease as a function of increasing fluid flow as pressure increases and the filtration ratio (β) will be reduced in relation to the magnitude of the flow surge. Particle penetration is the percentage of particles that are not captured by the filter. Cyclic flow conditions result in an increase in particle penetration. This increase occurs after contaminants have collected on the filter. A filter that has a uniform and fixed pore size and depends on direct interception or mechanical blockage as the primary capture mechanism tends to hold on to particles during variable flow. A non-uniform filter medium, which is normally thicker and depends on adsorption (particles sticking to fibers) for contaminant capture, has a tendency to release particles in the presence of variable flow and resultant flow surges.

A multiplying factor to adjust the base failure rate to consider cyclic flow can be determined from the following table:

Table 11-2 Cyclic Flow Multiplying Factor, C_{CF}

Filter type	Surge Frequency	C_{CF}
Uniform pore size	0 – 0.1 Hz	1.0
	0.1 – 0.5 Hz	1.2
Non-uniform pore size	0 – 0.1 Hz	1.2
	0.1 – 0.5 Hz	1.5

Table 11-3. Viscosity of Fluids

Liquid	Viscosity in Centistokes, ν								X
	0 C	20 C	40 C	60 C	80 C	100 C	125 C	150 C	
Water	1.8	1.0	0.75	0.56	0.35	0.28			0.2
Sea water	1.9	1.1	0.87						0.2
Gasoline, 0.68 s.g.	0.51	0.42	0.35	0.30					0.3
Kerosene, 0.81 s.g.	3.7	2.3	1.6	1.2	0.96				0.2
Light lubricating oil, 0.91 s.g.	390	96	34	16	8.7	5.4			0.2
Heavy lubricating oil, 0.91 s.g.	3492	500	123	43	20	10			0.7
SAE 10 oil	555	122	41	14	8.7	5.4	3.3	2.2	0.5
SAE 20 oil	1141	213	65	22	11	6.8	4.4	2.8	0.6
SAE 30 oil	2282	358	101	33	15	9.4	5.5	3.6	0.7
SAE 40 oil	4640	624	137	51	26	13	7.8	5.0	0.8
SAE 50 oil	8368	1179	251	76	32	17	9.5	6.4	0.9
SAE 60 oil	15215	2206	380	107	38	20	11	7.5	1.0
SAE 70 oil	23203	2853	456	137	49	25	14	8.5	1.1

11.5 REFERENCES

In addition to specific references cited throughout Chapter 11, other references included below are recommended in support of performing a reliability analysis of fluid filter assemblies.

7. Bishop, F.E. and William M. Needleman, "The Effects of Fluid Contamination on Component Wear", Pall Corporation

15. "Fabrication and Testing of Lightweight Hydraulic System Simulator Hardware – Phase II", Report No. NADC-79024-60, prepared by Rockwell International, Columbus, Ohio for Naval Air Systems Command, Washington, DC

23. Hubert, Christopher J., John W. Beck and John H. Johnson, "A Model and the Methodology for Determining Wear Particle Generation Rate and Filter Efficiency in a Diesel Engine Using Ferrography", Society of Automotive Engineers, Paper No. 821195 (1982)

24. Hudgens, R.D. and L.B. Feldhaus, "Diesel Engine Lube Filter Life Related to Oil Chemistry", Society of Automotive Engineers, Paper No. 780974 (1978)

33. Needleman, William M., "Filtration for Wear Control", in Wear Control Handbook, M.B. Peterson and W. O. Winer, Section 4, American Society of Mechanical Engineers, New York, (1980)

39. Shigley, J.E., Mischke, C.R., Mechanical Engineering Design, McGraw-Hill Book Company, New York (1989)

This Page Intentionally Left Blank

CHAPTER **12**

BRAKES AND CLUTCHES

12.0 TABLE OF CONTENTS

12.1 INTRODUCTION

The principal function of a brake or clutch assembly is to convert kinetic energy to heat and then either to absorb or dissipate heat while simultaneously (through energy transfer) reducing the relative movement between the friction material and the part to which it is engaged. Reliability models for brakes and clutches are presented together in this Handbook because of similar design and operational characteristics; and because one of the most important functional parts of each of these components is the friction material. A discussion of friction materials to achieve the energy transfer common to both brakes and clutches is included in Section 12.2. Section 12.3 addresses the brake model, which includes actuators, springs, friction linings, bearings, seals and housings. An analysis of the energy transfer materials which are common to both brakes and clutches is included in the brake model. Section 12.4 outlines and describes the reliability model for clutches, which includes the following components: actuators, bearings, friction linings, seals and springs.

12.2 FRICTION MATERIALS

As stated in the introduction, the major functional components of brake equipment and clutch equipment are the friction materials. The reliability of brakes and clutches is concerned with the wear of these friction materials. For brake assemblies, the friction lining provides the friction necessary to slow down or stop a vehicle or other machinery. Friction materials used in clutches are placed in the power-transmission system to couple it together so it rotates as one unit. Table 12-1 provides a list of friction material types and applications.

Table 12-1. Friction Material Types And Applications
(Reference 32)

TYPE	MANUFACTURE	APPLICATION
Woven cotton	Closely woven belt of fabric is impregnated with resins which are then polymerized	Industrial drum brakes, mine equipment, cranes, lifts
Woven asbestos	Open woven belt of fabric is impregnated with resins which are then polymerized. May contain wire to scour the surface	Industrial band and drum brakes, cranes, lifts, excavators, winches, concrete mixers, mine equipment
Molded flexible Semi-flexible rigid	Asbestos fiber and friction modifiers mixed with thermo-setting polymer and mixture heated under pressure	Industrial drum brakes; heavy duty brakes, excavators, tractors, presses
Sintered metal	Iron and/or copper powders mixed with friction modifiers	Heavy duty brakes and clutches, press brakes, earthmoving equipment
Cermets	Similar to sintered metal pads, but large portion of ceramic material present	Heavy duty brakes and clutches, press brakes, earthmoving equipment

Friction materials that are used in brake and clutch linings have severe performance requirements. The necessary energy conversion must be accomplished with a minimum of wear on the contacting parts. For a particular type of brake or clutch, the amount of heat and friction generated varies according to 5 conditions: (1) the amount of pressure applied between the sliding surfaces, (2) the operating environment, (3) the roughness of the surfaces, (4) the material from which the surfaces are made, and (5) the frequency of application.

The reliability of these high energy components is important for a variety of reasons including economy, operational readiness and, most important, safety. In today's modern machinery and equipment, a vast number of friction materials have become available to fulfill the very diverse requirements of this equipment group.

In design it is necessary to have equations for the prediction of the wear life of clutches and brakes. Lining wear properties are generally considered in terms of system life under several different conditions of use severity. Consequently, lining life is often the last performance character to be quantified. Thus, knowledge of lining wear behavior from laboratory testing can be of great value.

Friction modifier additives, such as cashew resin, graphite, etc. have been used for many years in order to control friction properties in brake and clutch composites. Friction composites are composed of a balanced mixture of resin plus additives and generally contain over a dozen ingredients in order to achieve desired characteristics.

Friction materials containing conventional organic binding agents exhibit poor frictional stability under varying temperature conditions. The thermal degradation of such binders results in inferior frictional characteristics, giving rise to fade and often resulting in increased wear. Furthermore, organic materials, particularly resins, tend to have a short shelf life, and are not always easy to reproduce.

Friction materials commonly consist of sintered lead bronzes and iron powders with additions of dry lubricants and so-called friction reinforcers. Graphite and molybdenum disulfide, for example, are suitable as dry lubricants. However, ceramic additives and minerals, such as quartz and corundum, may be used to increase the coefficient of friction. By appropriate variation in the additives it is possible to make adaptations for all applications, particularly concerning the coefficient of friction.

Semi-metallics rely heavily on iron, steel, and graphite substitutions for the organic and asbestos materials. Some organic components are, however, used to obtain desirable properties. The use of abrasives must be minimized to maintain acceptable mating surface compatibility. Semi-metallics have distinct advantages over conventional organics including:

- Improved frictional stability and fade resistance

- Excellent compatibility with rotors and high temperature wear resistance
- High performance with minimal noise

The cost of raw material mix represents the major factor in the premium prices of semi-metallics, and as such, widespread use of semi-metallics is not found. Metallic linings withstand more severe loads, higher temperatures, and have less tendency to fade. Sintered metallic-ceramic friction materials have successfully been used for specialized applications such as jet aircraft.

Reliability of the total brake of clutch assembly is directly related to performance of the friction material used for the particular design. Table 12-2 is a summary of failure modes for the brake or clutch friction material surface.

Table 12-2. Friction Material Failure Modes (Reference 32)

FAILURE MODE	CHARACTERISTICS	CAUSES
Heat spotting	Heavy gouging resulting in rapid lining wear	Material rubbing against a heat spotted metal member
Crazing	Randomly oriented cracks on the friction material, resulting in a high wear rate	Overheating of the braking surface
Scoring	Grooves formed on the friction material, resulting in a reduction of life	Metal member needs regrinding
Fade	Material degrades or flows at the friction surface, resulting in a temporary loss of performance	Overheating caused by excessive braking
Metal pick-up	Metal from the mating member embedded in the lining	Unsuitable combination of materials
Grab	Lining contacting at ends only giving high servo effect and erratic performance	Incorrect radiusing of lining
Strip braking	Braking over a small strip of the rubbing path giving localized heating and preferential wear at those areas	Distortion of the brake path
Neglect	Material completely worn off the shoe reducing performance	Failure to provide required maintenance
Misalignment	Excessive grooving wear at preferential areas of lining surface	Lining not fitted correctly to the shoe platform

12.3 BRAKES

12.3.1 Brake Assemblies

The reliability of a brake system is dependent on the reliability of its component parts, which may include: actuators, bearings, friction linings, housings, seals, and springs. With the exception of friction linings, all of these component parts are addressed in earlier sections of this handbook. The characteristics of these parts that are peculiar to the braking environment will be discussed in this chapter. The reliability of brake components is sensitive to friction materials used in their assembly and a review of brake reliability should consider the friction material failure modes addressed in the previous section.

Brakes are called upon to convert large amounts of kinetic energy to thermal energy in a very short time. The life of currently used brake lining materials is determined by wear, which in turn is strongly dependent on the temperature experienced by these materials during sliding. This temperature dependence is due largely to softening of the metal binder (usually copper or iron) present in brake lining composite materials.

Some of the systems which use brakes include passenger cars, light trucks, tractors, buses, agricultural equipment, construction equipment, industrial equipment, railroad trains and aircraft. Brake lining materials used in passenger cars and light trucks fall into two categories: drum brake segments, which are less than 3/4" thick, and disk brake pads. Brake systems used by trucks, buses, truck tractors and trailer combinations are air assisted hydraulic (air brake) systems.

Industrial equipment encompasses all equipment used in fixed facility or buildings such as overhead cranes or hoists. Hydraulic brake systems used in agricultural and construction equipment are of either the dry or the wet brake type. Dry brakes are the conventional types of a drum or disk system. Wet brakes use drum and disk brake assemblies but the friction material is in a fluid environment. This type of brake exhibits decreased heat build up and subsequently less fade, reduced lining and drum or rotor wear and improved reliability.

Most railroad trains rely on two braking systems, a dynamic brake and a friction brake. Most self-propelled rail cars have a dynamic brake, which is used either independently or together with the train’s friction braking system down to about 5-10 mph, using complete friction braking for the last distance to a complete stop.

The use of organic friction materials in aircraft brakes is currently limited primarily to small general aviation aircraft. The trend in larger aircraft brake materials has been toward higher energy absorption per unit mass of brake materials. On larger aircraft organic friction materials have been replaced by more expensive copper and iron-based metallics. The use of disk brakes, with one brake for each of the main landing gears is common.

12.3.2 Brake Varieties

There are numerous brake system types, each with their own parts and reliability characteristics. The various types of brake systems and methods of actuation are listed in Table 12-3.

Table 12-3. Methods of Actuation
(Reference 32)

TYPE	ADVANTAGES	DISADVANTAGES
Mechanical	- Robust, simple operation provides good control	- Large leverage needed - Potential frictional losses at pins and pivots
Pneumatic	- Large forces available	- Compressed air supply needed - Brake chambers may be bulky
Hydraulic	- Compact - Large forces available - Quick response and good control	- Special fluid needed - Temperatures must not be high enough to vaporize fluid - Potential seal problems
Electrical	- Suitable for automatic control - Quick response	- Limited to On-off operations

Specific design examples of brake systems include the following:

Band Brakes – Band brakes are simpler and less expensive than most other braking devices. Component parts include a friction band element and the actuation levers. Band brakes are characterized by uneven lining wear and poor heat dissipation.

Externally and Internally Pivoted Drum Brakes – Internal expanding and external contracting drum brakes are simple designs requiring relatively little maintenance. They may become self-locking with extreme wear if not properly designed. Internal types offer more protection from foreign material.

Linearly Acting External and Internal Drum Brakes - These brakes are fitted with shoes that, when activated, approach the drum by moving parallel to a radius through the center of the shoe. Springs between the friction materials may separate both shoes when the brake is released. Lining wear is more uniform in comparison with internal drum brakes.

Dry and Wet Disk Brakes - Disk brakes have two main advantages over drum brakes: better heat dissipation and more uniform braking action. However, disk brakes require a larger actuation force due to higher moments of inertia. Both annular and pad type disk brakes include friction materials, springs, actuators, housings, seals, and bearings. Wet disk brakes may operate in an oil bath. Thus, these brakes are isolated from dirt and water, and the circulation of the oil through a heat exchanger usually provides greater heat dissipation than direct air cooling.

Magnetic Particle, Hysteresis, and Eddy-Current Brakes - In all three of these brake types the braking torque is developed from electromagnetic reactions rather than mechanical friction, and therefore requires a source of electrical power.

12.3.3 Brake Assembly Failure Modes

A list of failure modes for a typical brake system is included in Table 12-4. The brake system friction materials are sacrificial replacements. Because friction linings are designed to wear out before the life of the vehicle or machinery, brake lining service life is a better measure of its durability with lining replacement counted as a maintenance action as opposed to an equipment failure. However, if replacement is required prior to service life, then the friction lining would be determined a failure.

Table 12-4. Brake System Failure Modes
(Reference 9)

FAILURE MODE	FAILURE CAUSE	FAILURE EFFECT
Sticking piston	Contamination	Low output pressure
Leaking cylinder	Contamination	Low output pressure
Broken/weak spring	Fatigue activation	Unable to adjust pressure
Sticking bleeder valve	Contamination	Inadequate dissipation of air
Deteriorated lining	Aged/heat	Exposed metal-on-metal contact reduces arresting capability
Worn bearing	Lack of lubrication	Low rotary motion
Worn seals	Aged	External leakage
Cracked housing	Vibration, fatigue	External leakage

In brake systems the rubbing elements include the friction material and a countersurface. The friction material is the sacrificial element, although the essence of good brake design is to minimize wear. The countersurface is usually metallic, to provide structure and to dissipate the frictional heat. Most countersurfaces are a cast iron drum or disc. In a few applications, steel rubbing surfaces are used. The countersurface is also nominally a non-wearing surface. Countersurfaces typically wear less than ten percent of the total volume worn from the friction interface (Reference 16). Descriptions of the countersurface failure characteristics are included in Table 12-5.

Table 12-5. Metal Countersurface Failures
(Reference 32)

FAILURE MODE	CHARACTERISTICS	CAUSES
Heat spotting	Often cracks are formed in these regions owing to structural changes in the metal.	Friction material not sufficiently conformable to the metal member
Crazing	Randomly oriented cracks	Overheating and repeated stress cycling
Scoring	Scratches in the line of movement	Metal too soft for friction material. Abrasive debris embedded in the lining material.

12.3.4 Brake Failure Rate Model

To determine the failure rate for a particular brake system, the brake system needs to be reduced to its component parts. Brake systems will normally contain some combination of the following components:

- Actuators
- Springs
- Brake friction linings
- Bearings
- Seals
- Housings

The total brake system failure rate is the sum of the failure rates of each of the above component parts in the system:

$$\lambda_{BR} = \lambda_{AC} + \lambda_{SP} + \lambda_{FR} + \lambda_{BE} + \lambda_{SE} + \lambda_{HO} \qquad \textbf{(12-1)}$$

Where:

λ_{BR} = Total failure rate for the brake system, failures/million hours

λ_{AC} = Total failure rate for actuators, failures/million hours (See Chapter 9)

λ_{SP} = Total failure rate for springs, failures/million hours (See Chapter 4)

λ_{FR} = Total failure rate for brake friction materials, failures/million hours (See Section 12.3.5)

λ_{BE} = Total failure rate for bearings, failures/million hours (See Chapter 7)

λ_{SE} = Total failure rate for seals, failure/million hours (See Chapter 3)

λ_{HO} = Total failure rate for brake housing, 3.0 failures/million hours, from Navy Maintenance and Material Management Information System and miscellaneous data sources

In the hydraulic drives of brake systems, seals are used to prevent leakage of brake fluid. The hardness and swelling of the seals, when exposed to brake fluid, must remain within limits such that the seals will give reliable operation.

The reliability of springs associated with brake systems is generally very high when compared to other components. Some of the spring assemblies in a brake system may be static, maintaining a constant tension on a part, other springs may be cyclic or dynamic depending on their function.

Severe performance requirements may affect the reliability of the bearings if there is a path of heat conduction from the friction surface to the bearings. This conduction may cause a decrease in the operating viscosity of the bearing lubricant and, consequently, a reduction in bearing life. A lubricant with a sufficiently high temperature rating should prevent leakage or excessive wear.

The reliability of brake actuators normally is very high. Under severe brake performance, conditions of increased temperature and excessive vibration may decrease the reliability of the actuator.

Refer to the appropriate chapters of this Handbook for the reliability models for individual parts comprising the brake assembly. In some cases the result in failure/million cycles will need to be converted to failures/million hours by multiplying by the number of cycles per hour.

12.3.5 Disk / Brake Lining Reliability Model

There are several factors that affect the wear rate of friction components of brakes, including:

(a) nominal pressure
(b) elastic properties of the materials
(c) strength properties of the materials
(d) surface roughness of the mating surfaces
(e) temperature of the material
(f) compatibility of the lining/drum or disk/pad materials

The wear of the brake lining or disk pad material can be determined with the following equation (Reference 77):

$$V = k_o \; P s \qquad \textbf{(12-2)}$$

Where: V = Volume of material lost by wear, in^3

k_o = Wear coefficient, $(lb/in^2)^{-1}$ (See Table 12-8)

P = Applied load, lbf

s = Sliding distance during braking, in = $v_s \bullet t_b$

v_s = Sliding velocity, in/sec

t_b = Braking time, sec

If the effective thickness of the lining or pad is d (inches), pad life is commonly given by the following linear relationship:

$$\text{Life} = \frac{d}{W_p} \qquad \textbf{(12-3)}$$

Where: Life = Number of applications before friction material is completely worn

d = Lining thickness, in

W_p = Pad wear per application, in

Equation (12-2) can be written in terms of surface pad/lining wear per application:

$$W_p = \frac{k_o \; P \; v_S \; t_b}{A} \tag{12-4}$$

where:

A = Brake lining area, in^2

and:

$$\lambda_{FR,B} = \frac{1}{\text{Life}} = \frac{W_p}{d} = \frac{k_o \; P \; v_S \; t_b}{d \; A} \tag{12-5}$$

For the purpose of compatibility with the other models developed for mechanical components, the lining life will be converted to a rate of failure. By normalizing Equation (12-5) to those values for which historical failure rate data are available, the following failure rate model can be derived:

$$\lambda_{FR} = \lambda_{FR,B} \bullet C_{BT} \bullet C_{RD} \bullet C_T \tag{12-6}$$

Where: λ_{FR} = Failure rate of the brake friction material in failures/million hours

$\lambda_{FR,B}$ = Base failure rate of the brake friction material, failures/million hours (See Section 12.3.5.1)

C_{BT} = Multiplying factor which considers the effect of brake type on the base failure rate (See Section 12.3.5.2)

C_{RD} = Multiplying factor which considers the effect of dust contaminants on the base failure rate (See Section 12.3.5.3)

C_T = Multiplying factor which considers the effect of ambient temperature on the base failure rate (See Section 12.3.5.4)

12.3.5.1 Base Failure Rate for Brake Lining / Disk Material

The brake friction material base failure rate, $\lambda_{FR,B}$, may be provided by the lining manufacturer. If not, the base rate can be calculated from Equation (12-5). Wear coefficients are included in Table 12-8. It should be noted that $\lambda_{FR,B}$ from a manufacturer may be in terms of brake actuations. During a single brake actuation the vehicle wheel or industrial machine will rotate several rotations during the stopping or slowing process.

12.3.5.2 Brake Type Multiplying Factor

A typical disk brake will wear better than a drum type due to the disk brakes ability to dissipate heat more quickly. The friction material for the annular brake is in the shape of an annulus and is bonded to both sides of the rotor disk. The slotted annular brake is nearly the same as the annular brake described above, the only exception being the presence of slots cut through the friction material on both sides of the rotor. The purpose of the slots is to decrease surface temperature and wear rate during braking. The pad brake configuration employs pads of friction material on the brake stators. Multiplying factors for the specific type of brake design are as follows, based on field performance data (Reference 20):

C_{BT} = 1.25 for drum type brakes

C_{BT} = 1.25 for slotted annular disk type brakes

C_{BT} = 1.00 for pad disk type brakes

C_{BT} = 0.90 for annulus disk type brakes

12.3.5.3 Dust Contaminant Multiplying Factor

Operating conditions with high amounts of dust contaminants affects lining wear depending on the binder resin used in formulating the friction material. The correction factor for dust conditions is shown in Table 12-6.

Table 12-6. Dust Contamination Multiplying Factor
(Reference 42)

Binder Resin	C_{RD}
Phenolic	3.5
Oil-modified phenolic	1.2
Rubber phenolic	1.1
Cashew	1.1
Oil-phenolic	1.1

12.3.5.4 Temperature Multiplying Factor

Wear of the friction material will be influenced by the ambient temperature in which the vehicle is operating. The correction factor for temperature is (Reference 3):

$$C_T = 1.42 - 0.00154X + 0.00000138X^2 \qquad \textbf{(12-7)}$$

(for sintered metallic truck linings)

$$C_T = 2.79 - 0.0109X + 0.0000124X^2 \qquad \textbf{(12-8)}$$

(for resin-asbestos linings used in automotive and moderate duty industrial brakes)

$$C_T = 3.80 - 0.00759X + 0.00000507X^2 \qquad \textbf{(12-9)}$$

(for carbon-carbon linings)

$$C_T = 17.59 - 0.0603X + 0.0000534X^2 \qquad \textbf{(12-10)}$$

(for resin-asbestos truck linings)

Where: X = 590 + T

T = Ambient temperature, ^{o}F

12.4 CLUTCHES

12.4.1 <u>Clutch Assemblies</u>

The reliability of a clutch system is generally very high and is the result of the low failure rate of its parts, which may include actuators, bearings, clutch friction linings, seals and springs. With the exception of clutch friction linings, these component parts are addressed in other chapters of this handbook. The general characteristics of friction materials are addressed in Section 12.2. Those characteristics of friction materials peculiar to clutches will be discussed in the following paragraphs.

The principal function of a friction clutch is to convert kinetic energy to heat and then either to absorb or otherwise dissipate the heat while simultaneously, through friction, reducing the relative movement between the friction material and the part to which it is engaged. In order to achieve these objectives the necessary energy conversion must be accomplished with a minimum of wear on the contacting parts.

12.4.2 Clutch Varieties

Clutches are made up of two basic components, pressure plate and disc. The pressure plate supplies sufficient force or pressure to the disc so enough friction is developed to transmit the required torque.

Friction clutches, although available in many different forms tend to be of the axial or rim type. Axial clutches operate where the movement is parallel to the axis of the shaft. Rim types operate where the movement is radial. Examples of the former are the plate and cone clutches. Examples of the latter include coil or wrap spring and chain clutches.

Plate clutches are divided into two designs, single and multiplate. The single plate design is used for transmission in light to medium power applications. The single plate is normally provided with a friction lining on each side of the disc. Multiplate designs employ a number of discs lined on both sides which serve to distribute the load over a large area. These types are used for high torque and high load applications. They require only moderate clamping pressures, and are suitable for high speed operation because their relatively small size generates lower centrifugal forces.

Cone clutches are used for smaller, medium power, low speed transmission systems which may be subjected to rough usage. These devices cope well with such treatment because of their simple robust construction, and due to the fact that heat is dissipated more readily than with plate clutches.

Rim and block clutches employ various means of engaging the stationary half of the assembly through radial movement against the rim of the driving member. The action is similar to that of an internally expanding brake shoe.

Centrifugal clutches are often used with squirrel cage motors. The fabric facing may be fitted to shoes or blocks mounted to a spider which is keyed onto the driving shaft. The shoes or blocks are thrown outward by centrifugal force, engagement being automatic when a predetermined speed has been reached.

Coil or wrap spring clutches operate on the principle of a spring mounted on a drum being tightened. The action is much like that of a rope tightening around a revolving capstan. The design is compact, simple in construction and is used where high torques are required from low power.

Chain clutches employ inner and outer friction rings in an oil filled housing actuated by cams bearing on chain toggles which force the rings together.

Sprag clutches consist of a number of specially shaped steel springs or wedges which jam inner and outer races in one direction only. This action leads to their use for applications in over-running (where the clutch acts as a free-wheel) and back-stopping. This design is particularly useful for intermittent rotary motion involving, for example, indexing or inching (Reference 34).

Materials classification divides the friction materials into organic and metallic groups. The organic group includes all materials composed of both asbestos and non-asbestos fibers and bound by some resin binder. The metallic group consists of all friction materials containing iron, copper, ceramic bronze, graphite, carbon or other metallic material as the base material.

12.4.3 Clutch Assembly Failure Modes

Failure modes of a clutch assembly are very similar to those of a brake assembly. Hydraulic clutch failures are almost without exception due to a lack of fluid pressure. Since a hydraulic clutch system uses pressure to engage and disengage components, any loss of system pressure will result in unintended operation. These fluid leaks can be external, but are more often internal leakages caused by seal failure. A list of failure modes for a typical clutch assembly is shown in Table 12-7.

12.4.4 Clutch Failure Rate Model

A typical clutch assembly will contain the following component parts:

- Actuators
- Bearings
- Clutch friction linings
- Seals
- Springs

The total clutch system failure rate is the sum of the failure rates of each of the above component parts in the system:

$$\lambda_{CL} = \lambda_{AC} + \lambda_{BE} + \lambda_{CF} + \lambda_{SE} + \lambda_{SP} \qquad \textbf{(12-11)}$$

Where: λ_{CL} = Total failure rate for the clutch system, failures/million hours

λ_{AC} = Total failure rate for actuators, failures/million hours (See Chapter 9)

λ_{BE} = Total failure rate for bearings, failures/million hours (See Chapter 7)

λ_{CF} = Total failure rate for clutch friction materials, failures/million hours (See Section 12.4.5)

λ_{SE} = Total failure rate for seals, failures/million hours (See Chapter 3)

λ_{SP} = Total failure rate for springs, failures/million hours (See Chapter 4)

The failure rates obtained from other chapters of the Handbook may have to be converted from failures/million cycles to failures/million hours by multiplying by the number of cycles per hour. The failure rate model for clutch friction materials is presented in Section 12.4.5.

Table 12-7. Clutch Friction Surface Failure Modes
(Reference 32)

PROBLEM	CHARACTERISTICS	CAUSES
Dishing	Clutch plates distorted into a conical shape	Lack of conformability. The temp. of the outer region of the plate is higher than the inner region.
Waviness or Buckling	Clutch plates become buckled into a wavy platter	Lack of conformability. The inner area is hotter than the outer area.
Banding or Crushing	Loss of friction material at the ends of a band	Crushing and excessive wear of the friction material
Material Transfer	Friction material adhering to opposing plate, often giving rise to excessive wear	Overheating and unsuitable friction material
Bond Failure	Material parting at the bond to the core plate causing loss of performance	Poor bonding or overheating, the high temperature affecting bonding agent
Burst Failure	Material splitting and removed from the spinner plate	High stresses on a facing when working at high speeds
Grooving	Grooving of the facing material on the line of movement	Material transfer to opposing plate
Reduced Performance	Decrease in coefficient of friction giving a permanent loss in performance	Excess oil or grease on friction material or on the opposing surface
Distortion	Facings out of flatness after high operating temperature	Unsuitable friction material

12.4.5 Clutch Friction Material Reliability Model

By using the clutch system beyond the life of the friction material a drastic reduction of friction coefficient can occur. This rapid deterioration can result in a catastrophic failure of the clutch.

Under normal operating conditions, the friction materials used in clutches are reliable mechanical components. Like brake friction materials, the wear of clutch materials is dependent on the amount of accumulated energy dissipated by the mechanical component.

$$h = k_o \, p \, s \tag{12-12}$$

Where: h = Change in thickness of the clutch friction material caused by wear, in

k_o = Wear coefficient, $(\text{lb/in}^2)^{-1}$ See Table 12-8

p = Nominal pressure between the clutch wear plates, $\text{lbf/in}^2 = P/A$

P = Applied load, lbf

A = Wear plate area, in^2

s = Sliding distance during clutch actuation, in = $v_s \bullet t_a$

v_s = Sliding velocity, in/sec

t_a = Cutch actuation time, sec

If the effective thickness of the clutch lining is d (inches), life of the clutch friction material is given by the following equation:

$$\text{Life} = \frac{d}{W_p} \tag{12-13}$$

Where: Life = Number of applications before friction material is completely worn

d = Lining thickness, in

W_p = Friction material wear per application, in

$$W_p = \frac{k_o\, P\, v_S\, t_a}{A} \qquad \textbf{(12-14)}$$

where:

A = Clutch lining area, in^2

and:

$$\lambda_{FR,B} = \frac{1}{\text{Life}} = \frac{W_p}{d} = \frac{k_o\, P\, v_S\, t_a}{d\, A} \qquad \textbf{(12-15)}$$

By normalizing Equation (12-15) to those values for which historical failure rate data is available, the following failure rate model can be derived:

$$\lambda_{CF} = \lambda_{CF,B} \bullet C_{NP} \bullet C_T \qquad \textbf{(12-16)}$$

Where:

λ_{CF} = Failure rate of the clutch friction material in failures/million hours

$\lambda_{CF,B}$ = Base failure rate of the clutch friction material, failures/million hours (See Section 12.4.5.1)

C_{NP} = Multiplying factor which considers the effect of multiple plates on on the base failure rate (See Section 12.4.5.2)

C_T = Multiplying factor which considers the effect of ambient temperature on the base failure rate (See Section 12.4.5.3)

12.4.5.1 Base Failure Rate for Clutch Lining / Disk Material

The clutch friction material base failure rate, $\lambda_{FR,B}$, may be provided by the manufacturer of the clutch assembly. If not, then the base rate can be calculated from Equation (12-13).

12.4.5.2 Clutch Plate Quantity Multiplying Factor

As noted previously, clutches can be divided into two design groups: single and multiple plate. Multiplate designs use a number of discs which distribute the load, and will therefore increase the reliability of the system.

The correction factor for the number of plates is given by:

C_{NP} = Number of disks in the clutch

12.4.5.3 Temperature Multiplying Factor

Because the temperature of the friction material affects the wear of the material, the ambient temperature to which the clutch is exposed will affect the wear of the friction lining (Reference 3). As a result:

$$C_T = 1.42 - 0.00154X + 0.00000138X^2 \tag{12-17}$$

(for sintered metallic linings)

$$C_T = 2.79 - 0.0109X + 0.0000124X^2 \tag{12-18}$$

(for resin-asbestos linings used in automotive and moderate duty industrial brakes)

$$C_T = 3.80 - 0.00758X + 0.00000507X^2 \tag{12-19}$$

(for carbon-carbon linings)

$$C_T = 17.59 - 0.0603X + 0.0000534X^2 \tag{12-20}$$

(for resin-asbestos truck linings)

Where: X = 590 + T

T = Ambient temperature, ^{o}F

Table 12-8. Typical Wear Coefficients for Brake and Clutch Linings
(Against cast iron or steel)

Lining (pad) Material	Wear Coefficient, k_o (psi^{-1})
Asbestos-type I Composite	6.46×10^{-11}
Asbestos-type II Composite	8.09×10^{-11}
Carbon-Carbon Composite	2..24 x 10^{-11}
Sintered bronze (dry)	2.42×10^{-10}
Non-asbestos composite (dry)	9.90×10^{-10}
Sintered bronze (wet)	5.02×10^{-13}
Sintered bronze composite	9.31×10^{-11}
Sintered resin composite	3.03×10^{-11}

12.5 REFERENCES

In addition to specific references cited throughout Chapter 12, other references included below are recommended in support of performing a reliability analysis of brakes and clutches.

1. Abadzheva, R.N. et al., "Effects of Brake Fluid Components on Rubber", Khimiya i Tekhnologiya Topliv i Masel, No. 8, pp. 18-19 (Aug 1982).

3. Anderson, A.E., "Wear of Brake Materials", in: Wear Control Handbook, M.B. Peterson and W.O. Winer, Eds., pp. 843-857, Am. Soc. Mech. Eng., New York (1980).

9. Boone, Tony D., "Reliability Prediction Analysis for Mechanical Brake Systems", NAVAIR-SYSCOM Report (Aug 1981).

12. Carson, Harold, Springs: Troubleshooting and Failure Analysis, Marcel Dekker, Inc., New York. (1983).

16. Ferodo Limited, Friction Materials for Engineers, Stockport, England (1969).

20. Ho, T.L., F.E. Kennedy and M.B. Peterson, "Evaluation of Materials and Design Modifications for Aircraft Brakes", NASA Report CR134896 (Jan 1975).

29. Minegishi, H. et al., "Prediction of Brake Pad Wear/Life by Means of Brake Severity Factor as Measured on a Data Logging System", SAE Paper 840358 (1984).

32. Neale, M.J., Tribology Handbook, Butterworths, London.

36. Orthwein, William C., Clutches and Brakes: Design and Selection, Marcel Dekker, Inc., New York (1986).

37. Rhee, S.K. and P.A. Thesier, "Effects of Surface Roughness of Brake Drums on Coefficient of Friction and Lining Wear", SAE Paper 720449 (1972).

40. Spokas, R.B., "Clutch Friction Material Evaluation Procedures", SAE Paper 841066 (1984).

58. Parmley, R.O., Mechanical Components Handbook, McGraw-Hill Book Co., NY, 1985.

77. Randall F. Barron and Herbert G. Tull, III, "Failure Rate Model for Aircraft Brakes and Clutches", Report No. DTRC-CMLD-CR-01-90, August 1990, Louisiana Tech University

78. Randall F. Barron and Herbert G. Tull, III, "Failure Rate Model for Aircraft Brakes and Clutches", Report No. NSWC-92/LO2, August 1992, Louisiana Tech University

This Page Intentionally Left Blank

CHAPTER 13

COMPRESSORS

13.0 TABLE OF CONTENTS

13.1 INTRODUCTION

A compressor is a machine for compressing gas from an initial intake pressure to a higher exhaust pressure through a reduction in volume. A compressor consists of a driving unit, the compression unit and accessory equipment. The driving unit provides power to operate the compressor and may be an electric motor or a gasoline or diesel engine. Types of gases compressed include air for compressed tool and instrument air systems; hydrogen, oxygen, etc. for chemical processing and various gases for storage or transmission. A compressed air system consists of one or more compressors, each with the necessary power source, air regulator, intake air filter, aftercooler, air receiver, and connecting piping, together with a distribution system to carry the air to points of use.

Compressors can be classified, in their broadest sense, in two categories: (1) positive displacement and (2) centrifugal. The positive-displacement classification can generally be described as a "volume reducing" type. In essence, an increase in gas pressure can be achieved by simultaneously reducing the volume enclosing the gas. In all positive displacement compressors, a measured volume of inlet gas is confined in a given space and then compressed by reducing this confined volume. Next, the gas at this now elevated pressure is discharged into the system piping.

The centrifugal classification refers to the type of velocity increase for centrifugal action. In a centrifugal compressor the gas is forced through the impeller by rapidly rotating impeller blades. The kinetic velocity energy from the rotating impeller is converted to pressure energy, partially in the impeller and partially in the stationary diffuser. The stationary diffuser converts the velocity head into pressure.

Each type of compressor is designed for specific applications and requirements. A reliability analysis therefore requires an investigation of the design features for the particular compressor. It is important to know what is inside the compressor not only to know the failure rate, but also how to logistically support the compressor in terms of spare parts and maintenance philosophy. The following section provides a basic description of the different types of compressors.

13.2 POSITIVE DISPLACEMENT COMPRESSORS

Positive displacement compressors include a wide spectrum of design configurations. As shown in Figure 13.1, positive displacement machines can be further defined by two sub classifications: rotary and reciprocating.

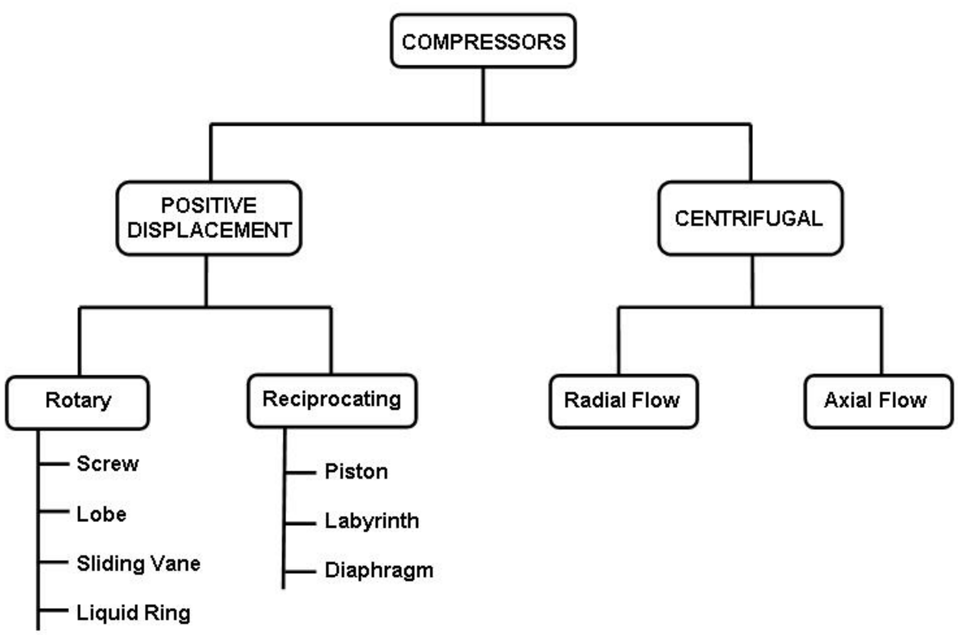


Figure 13.1 Common Classifications for Compressors

13.2.1 Rotary Compressors

Rotary positive displacement compressors incorporate a rotating element to displace a fixed volume of gas during each machine revolution. The following paragraphs provide a brief description of the different types of rotary compressors.

Rotary Screw - A common rotary compressor is the rotary screw. Rotary screw compressors produce compressed gas by filling the void between two helical mated screws and their housing. As the two helical screws are turned, the volume of gas is reduced resulting in an increase of gas pressure. Cooling and lubrication are obtained by injecting oil into the bearing and compression area. After the compression cycle, the oil and gas are separated before the gas is exhausted from the compressor.

Lobe - The rotary lobe compressor is typically constructed with two or three figure eight-shaped rotors, meshed together, and driven through timing gears attached to each shaft. It is a relatively low pressure machine (normally 5 to 7 psig and up to 25 psig for special types) and is well suited for applications with vacuum pressures. A lobe compressor provides a large throughput capability with little or no flow pulsation.

Sliding Vane - The sliding vane rotary compressor has a rotor construction which is offset, containing slots for vanes to slide in and out during each revolution. As the rotor turns during a single revolution, compression is achieved as the volume goes from a maximum at the intake ports to a minimum at the exhaust port. The vanes are forced outward from within the rotor slots and held against the stator wall by rotational acceleration. Oil is injected into the gas intake and along the stator walls to cool the gas, lubricate the bearings and vanes, and provide a seal between the vanes and the stator wall. After the compression cycle, the oil and gas are separated prior to the gas being transferred from the compressor.

Liquid Ring – In a liquid ring compressor the rotor is positioned centrally in an oval-shaped casing. During rotation, which happens without metal-to-metal contact, a ring of liquid is formed which moves with the rotor and follows the shape of the casing. During rotation, the liquid completely fills the chambers of the rotor and as the rotation continues, the liquid follows the contour of the casing and recedes again, leaving spaces to be filled by the incoming gas. As a result of the suction action thus created, gas is pulled into the compressor. As the rotation progresses, the liquid is forced back into the chambers, compressing the gas. This gas is forced out of the discharge port through an outlet flange. The compressor is fed continuously with liquid which maintains a seal between the inlet and discharge ports and at the same time removes the heat of compression. This liquid leaves the compressor together with the compressed gas and is separated from the gas in a discharge separator.

13.2.2 Reciprocating Compressors

Reciprocating air compressors are positive displacement machines in that they increase the pressure of the air by reducing its volume. As shown in Figure 13.1, there are various reciprocating compressor designs, the most common being the piston.

Piston - In this design, successive volumes of air are taken into the compressor and a piston within a cylinder compresses the air to a higher pressure. Air is released by mechanical valves that typically operate automatically by differential pressures. Inlet valves open when the pressure in the cylinder is slightly below the intake pressure. Discharge valves open when the pressure in the cylinder is slightly above the discharge pressure. Depending on the system design, cylinders may have one or multiple suction and discharge valves. Single stage compressors are commonly available for pressures in the range of 70 psi to 100 psi and two stage compressors are generally used for higher pressures in the range of 100 psi to 250 psi. A reciprocating air compressor is single acting when the compression is accomplished using one side of the piston and double acting if compression is accomplished using both sides of the piston during the advancing and retreating stroke.

Compression to high pressures in a reciprocating compressor may result in a temperature rise too great to permit the compression to be carried to completion in one cylinder, even though it is cooled. In such cases, the compression is carried out in stages, with a partial increase of pressure in each stage, and cooling of the gas between stages. Two and three-stage compression is common where pressures of 300-1000 psi are needed. In determining the number of stages (pistons) within a reciprocating compressor, the change in temperature across a stage, loading of the piston rod, and change in pressure across a stage are among the parameters taken into consideration.

Labyrinth - The labyrinth compressor is a vertical type reciprocating machine. In this type of compressor, rider rings and piston rings are not used as in the case of a horizontal type design. In labyrinth piston compressors, an extremely large number of throttling points provide the sealing effect around pistons and piston rods. No contact seals are used. The piston contains a labyrinth type piece at the center called a skirt. The cylinder also contains serration-like labyrinths on its inside surface. The piston is not in direct contact with the cylinder and close clearance is maintained between the piston and cylinder.

Labyrinth compressors are used where total dry operation is required and where lubricants are not allowed in the cylinders such as an oxygen compressor where safety is extremely important. Labyrinth compressors are also employed in applications where the process gas is heavily contaminated with impurities.

Diaphragm - The diaphragm compressor is a unique design employing a flexible diaphragm to compress the gas. The back and forth moving membrane is driven by a rod and a crankshaft mechanism. Only the membrane and the compressor box come in contact with the pumped gas and thus the diaphragm compressor is often used for pumping explosive and toxic gases. The membrane has to be reliable enough to take the strain of pumped gas. It must also have adequate chemical properties and sufficient temperature resistance.

Piston and diaphragm compressors possess many of the same components: crankcase, crankshaft, piston, and connecting rods. The primary difference between the two compressor designs lies in how the gas is compressed. In a piston compressor, the piston is the primary gas displacing element. However, in diaphragm compressors compression is achieved by the flexing of a thin metal, rubber or fabricated disk which is caused by the hydraulic system and operated by the motion of a reciprocating piston in a cylinder under the diaphragm. The diaphragm completely isolates the gas from the piston during the compression cycle. A hydraulic fluid transmits the motion of the piston to the diaphragm.

Diaphragms, in general, are round flexible plates which undergo an elastic deflection when subjected to an axial loading. In the application of compressors, this axial loading and elastic deflection creates a reduction in volume of the space adjacent to the diaphragm. The gas is compressed and a pressure builds. The diaphragm can be designed in many different ways with variations in such parameters as materials, size and shape.

13.3 CENTRIFUGAL COMPRESSORS

Centrifugal compressors depend on the transfer of energy from a rotating impeller to a gas discharge. The centrifugal force utilized by a centrifugal compressor is similar to a centrifugal pump. As gas enters the eye of the impeller the rotating impeller presses the gas against the compressor casing. The high speed spinning impellers accelerate the gas as additional gas is pressed against the casing by the impeller blades. A liquid ring (or piston) rotary is constructed of circular vanes, turning inside a casing sealed with a liquid. Centrifugal forces cause the liquid to form a ring around the periphery of the casing interior, while forcing the gas inward toward the center of the vaned rotor. The gradual decrease in volume increases the pressure of the gas. Any liquid entrained in the gas is separated out. This type of compressor is characteristically used in low pressure and vacuum applications. Centrifugal compressors are normally designed for higher capacity than positive displacement machines because flow through the compressor is continuous. Typical applications include aircraft engines.

Centrifugal compressors can be divided into two subcategories based on the direction of flow of the product gas: radial flow and axial flow machines. The characteristic curves of these machines offer a wide range in flow with a corresponding small change in head pressure. The centrifugal compressor is a continuous duty compressor with few moving parts making it well suited to high volume applications. The lack of rubbing parts in the compressed gas stream is a particularly desirable feature of these machines from a reliability standpoint.

Radial Flow - In radial compressors, velocity is imparted to a gas stream through centrifugal forces acting in a radial direction to the shaft. The simplest style of radial centrifugal compressor is the single-stage overhung design. The conventional closed or shrouded impeller is used for adiabatic heads to about 12,000 ft-lb/lb. The open, radial-bladed impeller develops more head for the same diameter and speed.

Axial Flow - In axial flow machines, the gas flow remains parallel to the shaft, without a direction change. These machines are typically used for higher capacities than radial flow machines, but generate much lower head pressure per stage. As a result, these machines are usually built with many stages. The characteristic performance curve is steeper than that of radial flow machines, with a more narrow stability range.

In summary, the different types and designs of compressors will result in different failure modes and failure rates. The next section provides some failure modes, causes and effects that need to be considered prior to estimating the total failure rate of the compressor in its operating environment.

13.4 COMPRESSOR FAILURE MODES

Figure 13.1 shows the various types of available compressor designs. Within these nomenclatures there are specific compressor designs with there own failure modes. To obtain an accurate list of failure modes for an individual compressor, a detailed parts list is needed and a thorough analysis of the interaction of component parts is required. For example, several stages of compressor units may be included in the overall compressor system which will require the determination of the effects of failure of adjacent stages if there is a failure of one particular stage.

Failure modes for compressors and certain compressor parts are listed in Table 13-1. Some failure modes are more prevalent than others as a direct result of the variety of compressor types and differing environmental conditions of operation.

Table 13-1. Compressor Failure Modes
(References 2 and 86)

FAILURE MODE	FAILURE CAUSE	FAILURE EFFECT
General compressor failure modes (see specific compressor type failure modes below)		
Seal failure	See Chapter 3	- Reduced output
Bearing failure	See Chapter 7	- Low flow pulsation
Gear failure	See Chapter 8	- Compressor noise, vibration and complete failure
Belt failure	See Chapter 21	
Shaft failure	See Chapter 20	
Clogged filter assembly	- Contaminants - See Chapter 11 for additional failure causes	- Corrosion, excessive temperature causing winding damage
Temperature sensor failure	See Chapter 19	- Loss of overload protection
Loss of motor power source	- Compressor overload - Misalignment between motor and compressor - See Chapter 14 for additional failure causes	- Loss of compressor output - Contaminants from burnt motor windings
Corrosion, water hammer, freeze damage	- Moisture within the compressor - Discharge temperature <55 C higher than inlet air	- Acid gases mixed with internal moisture accelerates the corrosion process
Internal corrosion	- Faulty filtration - Acid gases from the environment	
Clogged suction strainer	- Mechanical wear - Erosion of close-clearance moving parts	- Loss of gas capacity
Pressure pulsations		
Loss of gas output	- Fractured compressor casing - Seal leakage - Connection failure - See individual compressor types below	- Compressor failure
Improper lubrication of mechanical parts	- Seal leakage	- Reduced compressor efficiency - Eventual loss of output

Table 13-1. Compressor Failure Modes (Continued)

FAILURE MODE	FAILURE CAUSE	FAILURE EFFECT
Increased friction and wear	- Contaminants - Poor lubrication - Excessive heat from running compressor outside limits	- Decreased performance, - Increased vibration
Enclosure damage caused by vibration	- High fluctuating stresses - Insufficient foundation	- Material fatigue
The following failure modes apply to reciprocating compressors and need to be considered in addition to the general compressor failure modes above		
Cylinder fails to move	- Spring loaded valve fails to open	- Loss of gas output
Cylinder leakage	- Mechanical wear - Damaged seal	- Reduced compressor efficiency
Liquid entering one or more compression cylinders	- Seal failure	- Permanent valve damage - Compressor failure
Damaged piston rings	- Low compressor oil - Wear	- Maintenance required
Inoperative suction on discharge valve	- Valve leakage - Discharge valve fails to open	- Reduced compressor efficiency
Damaged cylinder packing ring	- Moisture entering cylinder	- Permanent valve damage - Reduced compressor efficiency
Damaged piston – crankshaft connecting rod	- Mechanical binding - Loss of lubricant	- Noisy compressor - Reduced compressor efficiency
Unbalanced crankshaft	- See Chapter 20	- Noisy compressor - Reduced compressor efficiency
Mechanical overloading of piston	- Low compressor oil - Wear - Excessive duty cycle	- Compressor failure

Table 13-1. Compressor Failure Modes (Continued)

FAILURE MODE	FAILURE CAUSE	FAILURE EFFECT
The following failure modes apply to rotary compressors and need to be considered in addition to the general compressor failure modes above		
Variable clearance between helical rotors	- Bent shaft, shock, vibration	- Severe rotor wear - Compressor failure
Contaminants between helical rotors	- Compressor operated in contaminated environment - Acid in ambient air	- Severe rotor wear - Compressor failure
Loss of lube film between helical rotors	- Loss of lubricant - Worn rotors	- Loss of gas capacity
Loss of coolant capability	- Loss of injected lubricant	- Damaged rotors - Compressor failure
Clogged discharge port	- Clogged filter - Contaminants through rotor section	- Loss of compressor air output
Oil and gas not sufficiently separated in high pressure area	- Filter damage - Plugged filter	- Contaminated gas output
Leakage of output gas back to low pressure side (slippage)	- Mechanical wear	- Loss of gas capacity
Reduction of internal clearances	- Distortion of rotor due to cyclic loading;	- Rubbing, increased wear
Accumulation of water in the lubricant	- Cooler failure	- Early compressor failure
Valve sticking	- Over lubrication, moisture in oil	- Overheating, increased wear
The following failure modes apply to diaphragm compressors and need to be considered in addition to the general compressor failure modes above		
Accelerated curing, embrittlement of diaphragm	- Extreme high or low temperature	- Decreased performance
Corrosion or cracking of diaphragm	- Contaminants	- Decreased performance
Valve sticking	- Over lubrication, moisture in oil	- Overheating, increased wear

Table 13-1. Compressor Failure Modes (Continued)

FAILURE MODE	FAILURE CAUSE	FAILURE EFFECT
Diaphragm rupture or crack	- Inadequate strength characteristics - Insufficient material pasticity - Corrosion - Local stresses at seating surface - Material hardening	- Loss of air output

13.5 FAILURE RATE MODEL FOR COMPRESSOR ASSEMBLY

A compressor system is made up of one or more stages. For example, a single-stage compressor is comprised of a single element or group of elements in parallel. However, two and three-stage compression may be required where higher pressures are needed. The total compressor may therefore be comprised of elements or groups of elements in series to form a multistage compressor based on the change in temperature and pressure across each stage.

Every compressor to be analyzed for reliability will be a unique design and comprised of many different components. The following example equation will need to be modified for the particular compressor design.

$$\lambda_C = (\lambda_{FD} \bullet C_{SF}) + \lambda_{CA} + \lambda_{BE} + \lambda_{VA} + \lambda_{SE} + \lambda_{SH} \qquad \textbf{(13-1)}$$

Where: λ_C = Total failure rate of compressor, failures/million hours

λ_{FD} = Failure rate of fluid driver, failures/million hours (See Section 13.7)

C_{SF} = Compressor Service Multiplying Factor (See Table 13-5)

λ_{CA} = Failure rate of the compressor casing, failures/million hours (See Section 13.6)

λ_{BE} = Total failure rate of compressor shaft bearings, failures/million hours (See Chapter 7)

λ_{VA} = Total failure rate of control valve assemblies, failures/million hours (See Chapter 6)

λ_{SE} = Total failure rate of compressor seals, failures/million hours (See Chapter 3)

λ_{SE} = Failure rate of compressor shaft, failures/million hours (See Chapter 20)

Additional parameters in the equation that may be necessary include timing gears (Chapter 8), belt (Chapter 21), couplings (Chapter 17), sensors (Chapter 19), filter (Chapter 11) and clutch (Chapter 12).

Different compressor configurations such as piston, rotary screw and centrifugal have different parts within the total compressor and it is important to obtain a parts list for the compressor prior to estimating its reliability. For example, a reciprocating piston compressor will contain suction and discharge valves while a rotary screw compressor will not normally contain control valves but may contain some unique oil/gas separator filters. The fluid driver of the compressor may be only a small part of the total compressor failure rate. The failure rate for each part comprising the compressor must be determined before the entire compressor assembly failure rate, λ_C, can be determined. Section 13.7 provides some guidance as to the various compressor configurations. Failure rates for each part will depend on expected operational and environmental factors that exist during normal compressor operation. It is important to consider each compressor stage in a multi-stage compressor as a separate compressor with the total failure rate as the sum of the failure rates for the individual stages. For example, a total compressor system may be comprised of a centrifugal compressor followed by an oil-free reciprocating compressor. The total system failure rate is the sum of the failure rates for the two units.

13.6 FAILURE RATE MODEL FOR CASING

The compressor casing, normally a very reliable component, can have a large effect on the life of other components in the compressor assembly (especially seals and bearings). Casing construction design, connections, casing material and production testing are specified to meet the application, safety and casing-pressure requirements. The value of reliability of compressor casings, through the experience of many different manufacturers, can generally be equated to a λ_{CA} value of 0.010 failures/million hours.

13.7 FAILURE RATE MODEL FOR DESIGN CONFIGURATION

Various reliabilities are inherent in specific designs and types of compressors. For example, it is expected that the reliability due to wear will be different in a rotary screw compressor compared to a centrifugal compressor due to the nature of metal-to-metal contact and rotor speeds. The various chapters of this handbook can be used to estimate the failure rates of the individual component parts. The fluid driver parameter λ_{FD} can be approximated by data presented in Table 13-4 for various types of fluid drivers, developed from information collected by such sources as OREDA and the U.S. Navy.

Typical component parts of a compressor that need to be evaluated for reliability include the following:

- drive unit
- air end
- cooling system
- receiver tanks
- air dryers
- filters
- piping distribution system

A rotary compressor is typically comprised of the following components:

- air intake filter unit usually a two stage assembly required by rotary compressors
- rotor unit comprised of screws, lobes or vanes to compress the gas and a discharge port. A non-lubricated rotary compressor will contain timing gears to drive the rotors
- oil injection unit to supply lubrication between the rotating screws or vanes and prevent gas leakage
- aftercooler unit
- oil/gas separator unit to separate the oil from the gas
- oil filter unit
- oil cooler unit

A reciprocating compressor is typically comprised of the following components:

- compressing unit consisting of air cylinders, single or double acting piston for each cylinder, air inlet valves and discharge valves
- mechanical unit consisting of connecting rods, piston rods, crossheads, and a crankshaft and flywheel
- lubricating unit for bearings, gears, and cylinder walls consisting of a pump or force-fed lubricator to supply oil to the compressor cylinders
- regulation unit to maintain the pressure in the discharge line and gas storage tank at the required pressure

A centrifugal compressor is typically comprised of the following components:

- Rotating components mounted on a shaft that drives a central drum retained by bearings

13.7.1 Compressor Service Load Multiplying Factors

As mentioned previously in this Chapter, each type of compressor is uniquely designed for a particular application and requirement. Some designs are more sensitive to a particular operating environment and require particular attention as part of the reliability estimate. For example, inlet air condition is an important consideration in estimating compressor reliability. A proper inlet filtration prevents erosion of seals and close clearance moving parts. Some compressor designs are more sensitive than others to the entrance of contaminants into the compressor. Another factor is high ambient humidity that if not controlled can cause corrosion. Duty cycle, ambient operating temperature, lubrication quality, shock and vibration are all operating and environmental parameters that need to be considered with respect to the sensitivity of the design to that parameter. Table 13-5 provides multiplying factors that modify the base failure rate considering each potential operating or environmental condition.

13.8 DIAPHRAGM FAILURE RATE MODEL

The diaphragm compressor is a unique design employing a flexible diaphragm to establish compression of gas. The back and forth moving membrane is driven by a rod and a crankshaft mechanism as shown in Figure 13.3. Only the membrane and the compressor box come in contact with the pumped gas and thus the diaphragm compressor is often used for pumping explosive and toxic gases. Because of the unique design properties and applications of the diaphragm compressor this section is included to provide a more detailed analysis of the diaphragm itself. Table 13-4 can be used in lieu of this detailed procedure.

λ_{FD} in Equation (13-1) for the diaphragm compressor can be written as:

$$\lambda_{FD} = \lambda_{DI}$$

Where:

λ_{DI} = Total failure rate of the diaphragm assembly

And:

$$\lambda_{DI} = \lambda_{BFD} \bullet C_P \bullet C_{AC} \bullet C_{LC} \bullet C_T \qquad \textbf{(13-2)}$$

Where:

λ_{BFD} = Compressor diaphragm base failure rate, 0.58 failures/million hrs.

C_P = Factor for effects of axial loading (See Section 13.8.1)

C_{AC} = Factor for effects of atmospheric contaminants (See Section 13.8.2)

C_{LC} = Factor for effects of liquid contaminants (See Section 13.8.3)

C_T = Factor for effects of temperature (See Section 13.8.4)

13.8.1 Axial Load Multiplying Factor

Diaphragms, in general, are round flexible plates which undergo an elastic deflection when subjected to an axial loading. In the application of compressors, this axial loading and elastic deflection creates a reduction in volume of the space adjacent to the diaphragm. The gas is compressed and a pressure builds. The diaphragm can be designed in many different ways with variations in such parameters as materials, size and shape. The model developed for a compressor diaphragm is shown in Figure 13.2. It has a passive area in the center which is rigid. This area transmits a force from the push rod to the diaphragm. To be effective, the thickness of the rigid center should be at least 6 times the thickness of the diaphragm.

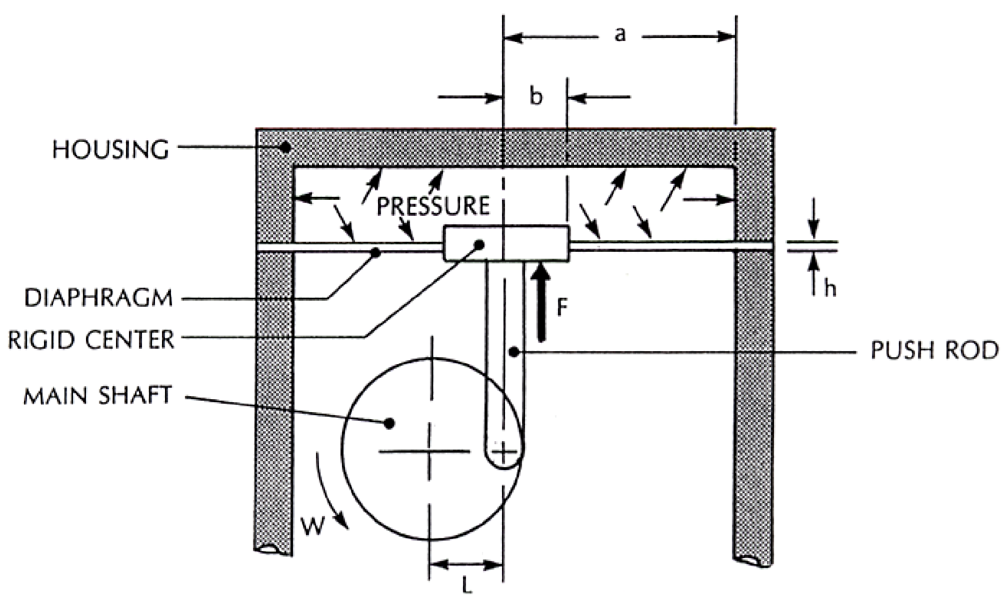


Figure 13.2 Compressor Diaphragm Model

The characteristic equations describing the compressor diaphragm are given in Equations (13-3) through (13-9) and are based on the following restrictive assumptions:

(1) Diaphragm is flat and of uniform thickness.

(2) Diaphragm material is isotropic and homogeneous.

(3) All forces, loads, and reactions are applied normally to the plane of the plate.

(4) Diaphragm thickness not greater than 20% of its diameter.

(5) The effects of shearing stresses and pressures on planes parallel to the surface of the diaphragm have not been taken into account. They are considered insignificant in diaphragms with thickness to radius ratios (*h/a*) of less than 0.15.

(6) The stresses created in a diaphragm due to bending and tensile loading may be combined by summing their values (method of superposition).

The characteristic equation of a rigid center diaphragm loaded by a force for any magnitude of deflection is given by Equation (13-3). It is applicable for (b/a) ratios greater than 0.05.

$$F = \frac{\pi E}{a^2}\left[\frac{h^3 y_o}{K_F} + h\, y_o^{\,3} B\right] \qquad \textbf{(13-3)}$$

Where: F = Force applied to rigid disk of diaphragm, lb

E = Modulus of elasticity, lbs/in²

a = Radius of diaphragm, in

h = Diaphragm thickness, in

y_o = Vertical deflection at center of diaphragm, in

K_F = Modified Stiffness Coefficient based on diaphragm

bending loads, $= \dfrac{3\left(1-\eta^2\right)}{\pi}\left[\dfrac{c^2-1}{4c^2} - \dfrac{\ln c^2}{c^2-1}\right]$ **(13-4)**

B = Stiffness coefficient based on diaphragm tensile loading, as follows:

$$= \frac{\dfrac{7-\eta}{3}\left(\dfrac{1+b^2}{a^2} + \dfrac{b^4}{a^4}\right) + \dfrac{\left(3-\eta\right)^2 b^2}{\left(1+\eta\right) a^2}}{\left(1-\eta\right)\left(1-\dfrac{b^4}{a^4}\right)\left(1-\dfrac{b^2}{a^2}\right)^2} \qquad \textbf{(13-5)}$$

η = Poisson's ratio

c = Ratio of radii (diaphragm-to-disk), a/b, in/in

b = Radius of rigid center plate of diaphragm, in

The maximum radial stress for a force-loaded diaphragm with rigid center occurs at the inner perimeter of the diaphragm (b):

$$\sigma_r = \frac{F\,K_F\,B_F}{2\pi h^2} \tag{13-6}$$

Where: σ_r = Maximum radial stress, lbs/in^2

B_F = Modified stiffness coefficient, based on diaphragm tensile loading

$$= \frac{2}{1-\eta^2}\;\frac{c^2\left(2c^2\ \ln c - c^2 + 1\right)}{\left(c^2-1\right)^2 - 4c^2\ \ln^2 c} \tag{13-7}$$

At equilibrium, where the force transmitted by the push rod in Figure 13.2 generates a maximum pressure in the chamber above the diaphragm (i.e., the rod has completed its stroke), a balance of forces in the vertical direction is established.

If the increased performance of a compressor is to be evaluated and the change in shaft power requirements are known, the following equation, in combination with Equation (13-6), can be used to evaluate the maximum induced stress in the diaphragm:

$$\sigma_r = \frac{396{,}000\,hp\,K_F\,B_F}{2\pi L\,\omega\,h^2} \tag{13-8}$$

Where: hp = Shaft output horsepower

L = Offset of eccentric shaft, in

ω = Output shaft speed, rpm

The maximum stress is calculated from Equation (13-8) for the compressor rated condition. Then the maximum stress for the actual operating condition is calculated in the same manner.

Empirical studies show that for moderate to high strains, a mechanical tearing of rubber, referred to as "mechanical-oxidative cut growth", can be the mechanism of failure for rubber diaphragms. The cut growth may greatly increase in the presence of oxygen. For this mode of failure, the fatigue life is inversely proportional to a power of the strain energy of the rubber. The strain energy is a characteristic of each type of rubber, and in turn, inversely proportional to the strain experienced by rubber under cyclic stressing. Figure 13.3 shows the stress-strain relationship for natural rubber compounds. Unlike many other engineering materials, rubber can be manufactured with a wide range of elastic moduli. Stiffness variations can be attained with no dimensional changes by varying the incorporation of fillers (reinforcing carbon blacks). This "hardness" variable is essentially a measurement of reversible elastic penetration (International Rubber Hardness Degrees or IRHD).

The stress developed in a rubber diaphragm can be calculated from Equation (13-6). Although rubber is flexible, (i.e., has low elastic and shear moduli), it is highly incompressible in bulk and its Poisson's ratio, η can be approximated as 0.5. This will facilitate the use of these equations. From the stress calculated, Figures 13.3 and 13.6 provide a corresponding load multiplying factor, C_P.

The value for strain obtained from Figure 13.3 must exceed 75%. Below this strain, the mechanical-oxidative cut growth mode of failure does not apply, and the C_P factor becomes 1.0.

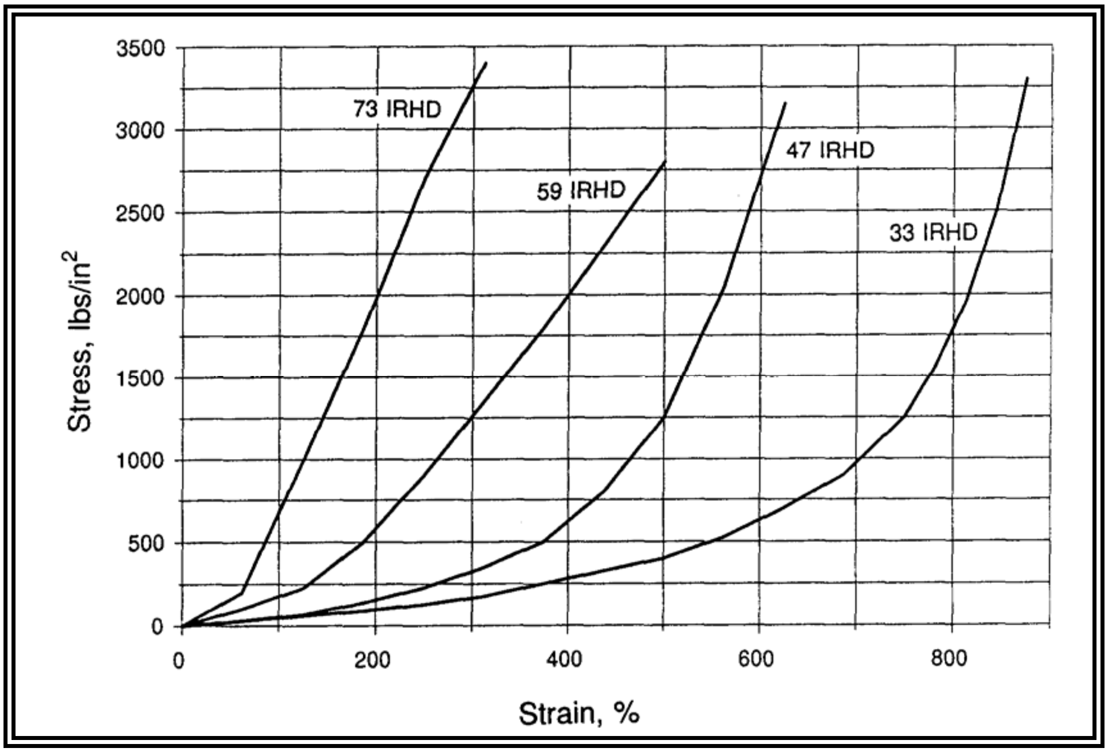


Figure 13.3 Tensile Stress-Strain Curves for Four Natural Rubber Compounds of Different Hardness (Ref 31)

13.8.2 Atmospheric Contaminant Multiplying Factor

The very small concentration of ozone in the atmosphere, normally a few parts per hundred million at ground level, may cause cracking in strained rubber components. Under cyclic conditions of strain below about 75%, ozone cut growth is the major factor in determining fatigue life.

Experimental data presented in Figure 13.7 illustrates that fatigue life is proportional to the concentration of ozone. The stress developed in a rubber diaphragm can be calculated from Equation (13-6). Poisson's ratio, η, can be equated to 0.5.

Table 13-2 can be used to determine the strain by dividing the value of stress obtained from Equation (13-6) by Young's modulus. Figure 13.7 and this strain value are then used to determine the contaminant air performance multiplying factor, C_{AC}.

13.8.2.1 Adjustment to Atmospheric Contaminant Multiplying Factor

In ozone-dominant failure potentials, the use of chemical anti-ozonant (coating) on the surface of the rubber diaphragm can reduce crack growth by a factor of 3. If a coating is used, the multiplying factor, C_{AC}, obtained from Figure 13.7 should be multiplied by 1/3.

13.8.3 Liquid Contaminant Multiplying Factor

Water absorption does not usually cause any significant deterioration of rubber, but the absorption of oil and solvents cause rubber to swell with a consequent deterioration in certain properties. Thin components can be expected to fail rapidly if the major surfaces are exposed to oil. Thick components are effectively protected by their bulk. Such components can last many years in an oily environment. Diffusion theory predicts that the mass of liquid absorbed per unit area of rubber (in the early stages of swelling) is proportional to the square root of the time taken for the absorption.

The rate of movement of the boundary between swollen and unswollen rubber is calculated from:

$$PR = \frac{L}{\sqrt{t}} \qquad \textbf{(13-9)}$$

Where: PR = Penetration rate, in/sec$^{0.5}$

L = Depth of the swollen layer, in

t = Time that a given mass of liquid is absorbed by a given surface, sec

The failure rate for a rubber diaphragm is dependent on the presence of liquid contaminants and the viscosity of the liquid in contact with it. Typical penetration rates are shown in Figure 13.4. Figure 13.4 reveals that the penetration rate into natural rubber decreases as the viscosity of the swelling liquid increases.

An adjustment for various types of diaphragm materials can be made using the multiplying factors presented in Table 13-3. These factors should be multiplied by the penetration rate obtained from Figure 13.4 prior to using the nomograph in Figure 13.5.

Table 13-2. Hardness and Elastic Moduli

HARDNESS, IRHD	YOUNG'S MODULUS, E, lb/in^2
30	130
35	168
40	213
45	256
50	310
55	460
60	630
65	830
70	1040
75	1340

13.8.4 <u>Temperature Multiplying Factor</u>

The variations in ambient temperature commonly occurring in practice are unlikely to greatly affect fatigue behavior. Experiments over a range of -32 to 212 F indicate only a slight effect of temperature on the fatigue life of crystallizing natural rubber. In general, rubbers become weaker as the temperature is raised. There is a steady fall in strength up to a critical temperature at which an abrupt drop occurs. For natural rubber, this temperature is about 212 ^oF.

A temperature multiplying factor, C_T, can be developed as follows:

For: -32F < T $\leq$ 212 F, C_T = 1.0

and for: T > 212 F, C_T = 6.7

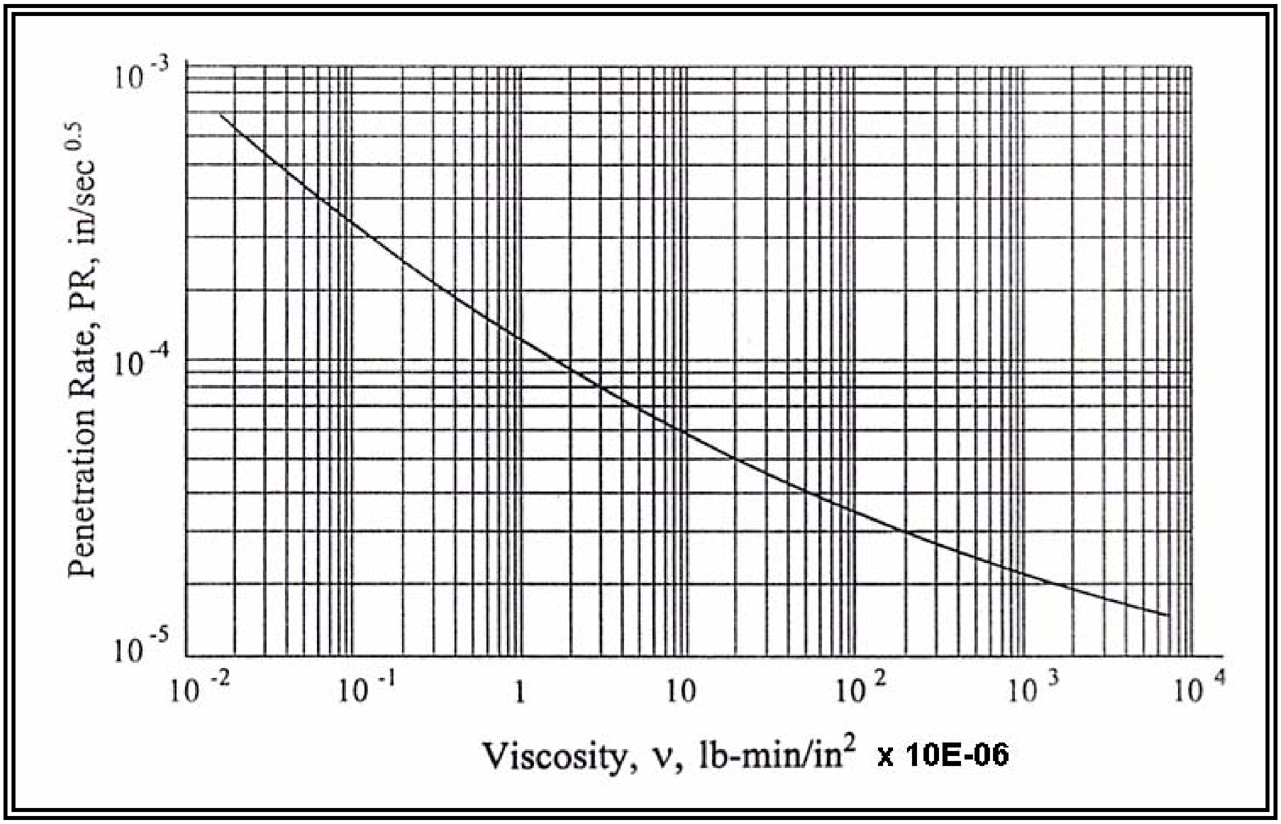


Figure 13.4 Effect of Liquid Viscosity on the Penetration Rate of Liquids into Natural Rubber

Table 13-3. Contaminant Adjustment Factor For Various Diaphragm Materials

RUBBER	X
Natural	1.0
Cis polybutadiene	1.3
Butyl	0.7
SBR	0.7
Neoprene WRT	0.4
Nitrile (38% acrylonitrile)	0.1
Metal	0.001

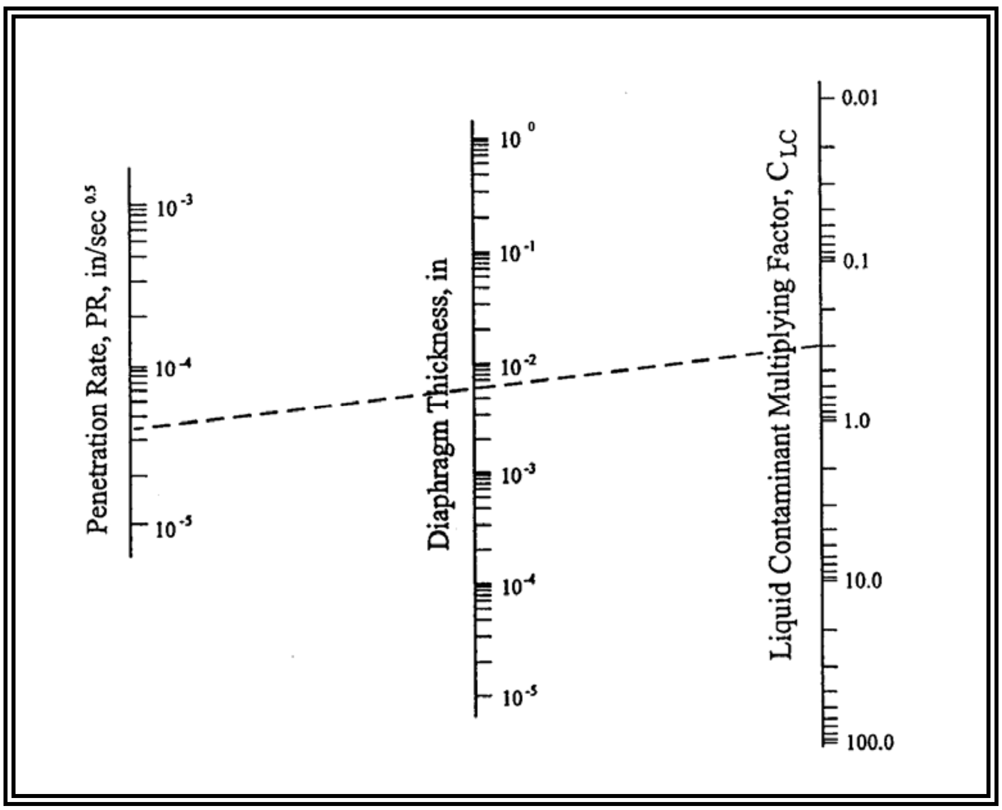
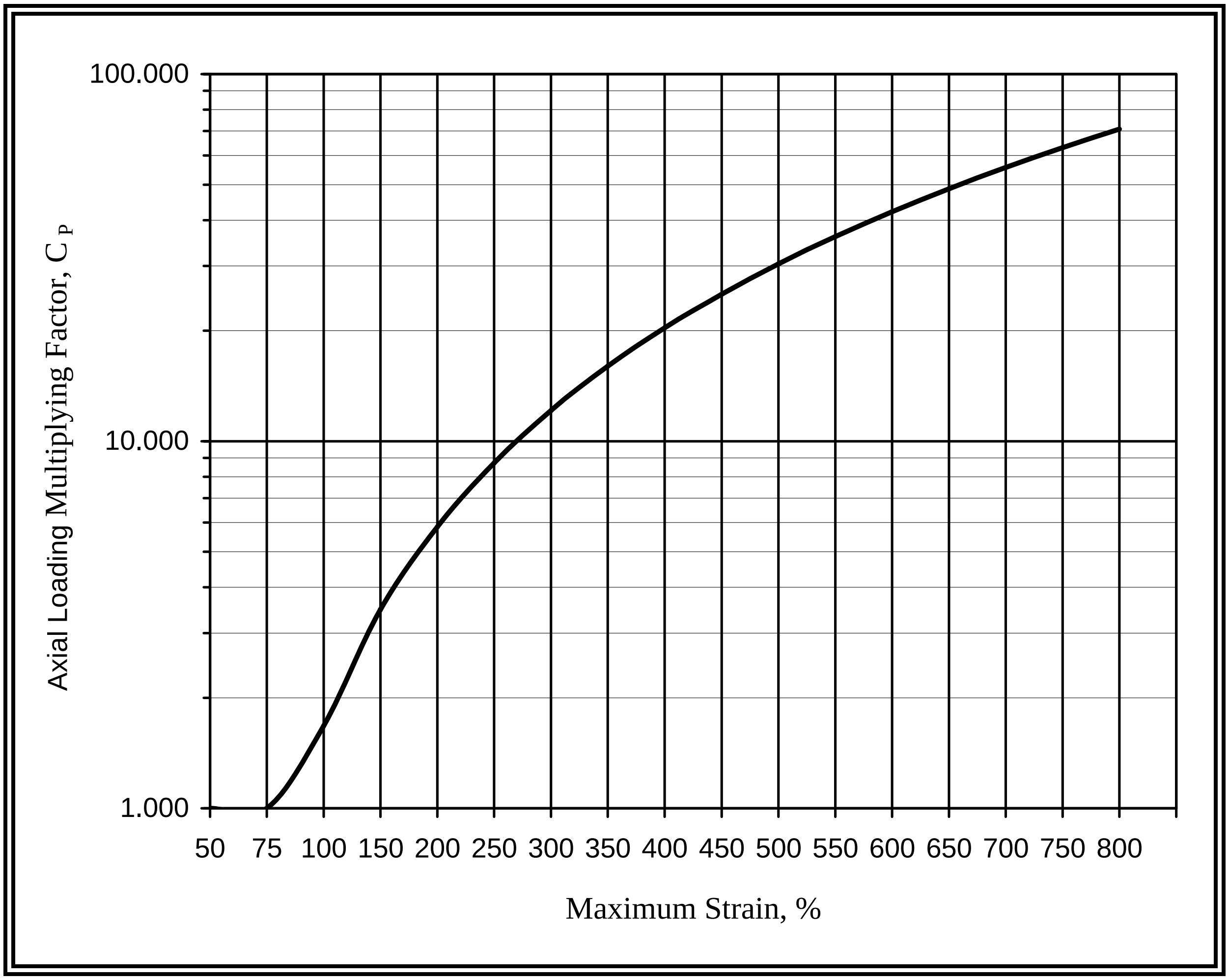


Figure 13.5 Nomograph for the Determination of Liquid Contaminant Multiplying Factor, C_{LC}

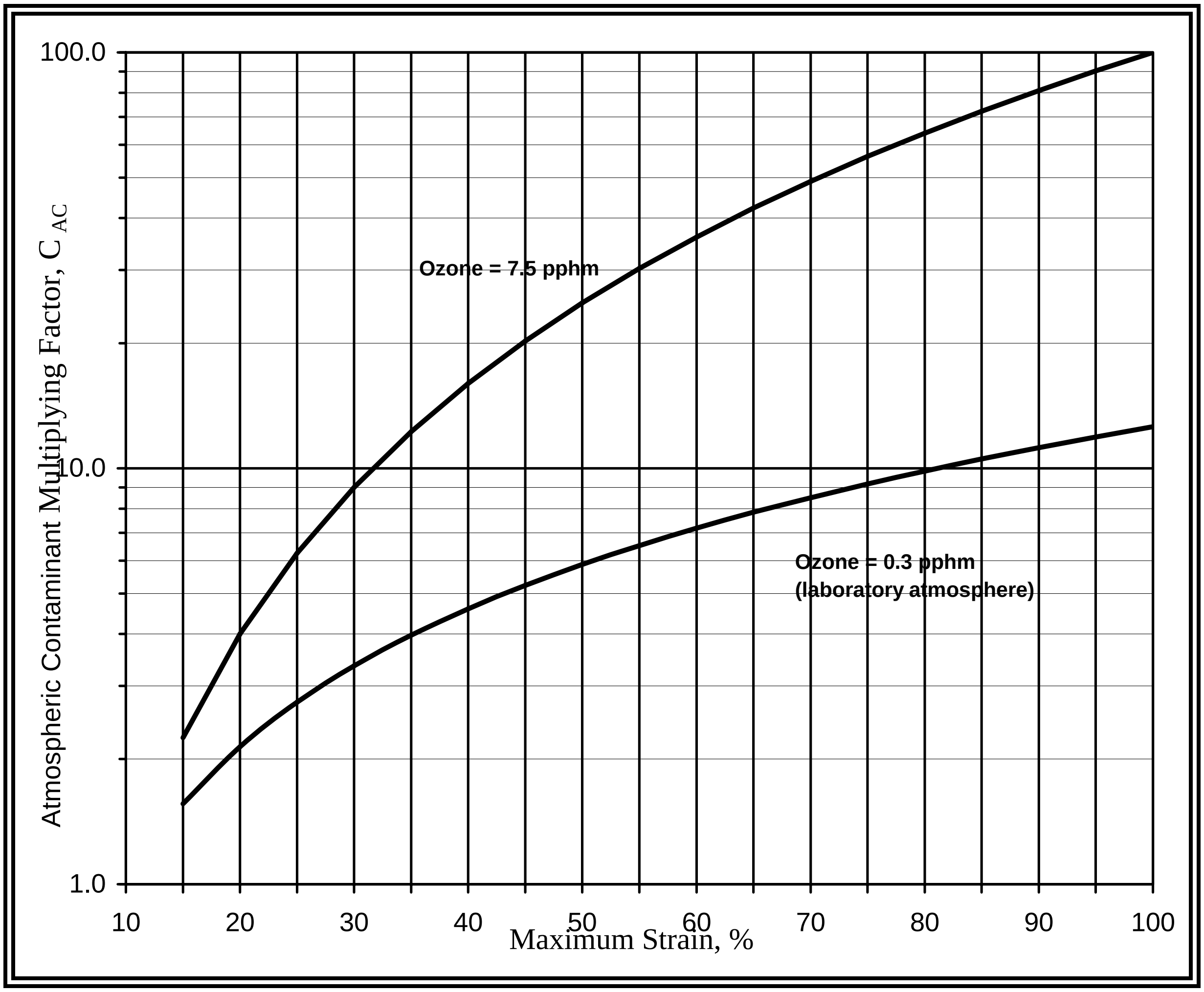


For $S \leq$ 75%: $C_P = 1.0$

For $S >$ 75%: $C_P = \left(\frac{S}{75}\right)^{1.8}$

Where: S = Strain, %

Figure 13.6 Axial Loading Multiplying Factor as a Function of Strain



For ozone 0.3 pphm: $C_{AC} = \left(\frac{S}{10}\right)^{1.1}$

For ozone 7.5 pphm: $C_{AC} = \left(\frac{S}{10}\right)^{2}$

Where S = Strain, %

Figure 13.7 Atmospheric Contaminant Multiplying Factor

Table 13-4. Failure Rate for Fluid Drivers (λ_{FD})

(*See note following table*)

FLUID DRIVER MODE	MODEL TYPE	BASE RATE* λ_{FD}
Radial flow	------	12.0
Axial flow	------	12.0
Reciprocating	Single piston	14.0
Reciprocating	Double acting piston	16.5
Reciprocating	Labyrinth	16.5
Reciprocating	Rubber Diaphragm **	22.8
Reciprocating	Metal Diaphragm	28.5
Rotary	Vane	12.0
Rotary	Screw	12.0
Rotary	Lobe	12.0
Rotary	Liquid Ring	12.0

* Failures/million hours of operation

** See Section 13.8 for specific failure rate calculations

Note: If the complete compressor has multiple stages determine the failure rate for each stage as an independent compressor and total the failure rates.

Table 13-5 Compressor Service Load Multiplying Factors

Multiplying Factor	Centrifugal	Rotary	Reciprocating	Diaphragm
Normal duty cycle, operating temperature and humidity, air cleanliness with proper filtration, lubrication quality, vibration and shock loading	1.0	1.0	1.0	1.0
High duty cycle (> 5 cycles per /hour)	1.2	1.2	1.4	1.2
Extreme operating temperatures	1.1	1.1	1.4	1.4
Non-scheduled lubrication check	1.1	1.2	1.1	1.2
High vibration level and/or heavy shock loading	1.2	1.4	1.3	1.5
Poor inlet air quality	1.1	1.4	1.1	1.3

13.9 REFERENCES

In addition to specific references cited throughout Chapter 13, other references included below are recommended in support of performing a reliability analysis of compressors.

2. "A Practical Guide to Compressor Technology", Second Edition, Heinz P. Bloch, John Wiley & Sons, 2006

26. Krutzsch, W.C., Pump Handbook, McGraw-Hill Book Company, New York (1968).

31. Nagel, W.B., "Designing with Rubber," Machine Design (June 23, July 7, July 21, Aug 11, 1977).

62. Baumeister, T, et al, Mark's Standard Handbook for Mechanical Engineers, McGraw-Hill Book Company

78. CDNSWC, "Interim Reliability Report on the MC-2A Compressor Unit", January, 1992

86. "Performance Prediction of Centrifugal Pumps and Compressors", 25th Annual Gas Turbine conference and 22nd Annual Fluids Engineering Conference Proceedings ASME 1980

97. “ The Chemical Engineering Guide to Compressors”, Richard Greene, McGraw Hill Publications Company, 1984

108. Daryl Beatty, Dow Chemical Company, “Oil analysis Boosts Compressor Reliability”. *Practicing Oil Analysis Magazine,* November 2004

109. Robert Moffatt, Gast Manufacturing Corp., “Prolonging Compressor Life”, Machine Design, May 11, 1978

128. “Rotary Screw or Reciprocating Air Compressor: Which One is Right?”, Bryan Fasano and Randy Davis, Gardner Denver, Plant Engineering, September 1998

CHAPTER **14**

ELECTRIC MOTORS

14.0 TABLE OF CONTENTS

14.1 INTRODUCTION

Motors convert electrical energy into mechanical energy and play a very important part in supplying power for all types of mechanical equipment such as pumps, compressors and machine tools. They are sometimes classified according to the type of electricity they require including direct current (DC) or alternating current (AC). If AC, the motor may be of a single phase or polyphase design. Electric motors are usually sized in horsepower. The most common sizes are fractional horsepower motors, i.e. 1/2 horsepower or 1/4 horsepower. Larger motors range in size to hundreds of horsepower.

There are many different types of motors to be analyzed for reliability such as the split phase motor, capacitor start motor and squirrel cage motor. The split phase motor is mostly used for "medium starting" applications. It has start and run windings, both are energized when the motor is started. When the motor reaches about 25% of its full load speed, a centrifugal switch disconnects the starter winding. The split phase motor is used where stops and starts are somewhat frequent such as in a refrigerator or air conditioner compressor.

The capacitor start motor has a capacitor in series with a starting winding and provides more than double the starting torque with one third less starting current than the split phase motor. Because of this improved starting ability, the capacitor start motor is used for loads which are hard to start such in a conveyor. The capacitor and starting windings are disconnected from the circuit by an automatic switch when the motor reaches about 75% of its rated full load speed.

A squirrel cage motor includes a rotor with a cylindrical shape containing longitudinal conductive bars usually made of aluminum or copper set into grooves and connected together at both ends by shorting rings forming a cage-like shape. The field windings in the stator set up a rotating magnetic field around the rotor. The relative motion between this field and the rotation of the rotor induces electric current in the conductive bars. In turn these currents lengthwise in the conductors react with the magnetic field of the motor, resulting in torque to turn the shaft. Washing machines and dishwashers are typical applications for the squirrel cage motor.

A gear motor is an assembly composed of an electric motor and a reduction gear in a single unit. A stepper motor converts electrical pulses into specific, rotational movements. Gear motors consisting of an electric motor and a reduction gear train are used in those applications requiring high torque at relatively low shaft speed such as in lifts, winches and garage door openers.

Electric motors are rated as to such variables as voltage, torque, temperature, and speed. Speed is usually specified as RPM at no load condition. As the motor is loaded down, the speed will slow down. If the electric motor is loaded too heavily, the motor shaft will stop. This stall speed should be avoided in any application. This chapter contains failure rate models that apply to all types of electric motors that can be used to support the development of mechanical equipment and provide a reliability estimate for a new motor application, proposed design modification, or an application that is different than specified parameters. The models are intended to focus attention on further design, testing or reliability analysis which should be accomplished to assure the allocated reliability of the motor in its intended operating environment.

14.2 CHARACTERISTICS OF ELECTRIC MOTORS

14.2.1 Types of DC Motors

DC motors consist of armature windings inside another set of field windings called the stator. Applying a voltage to the windings produces a torque in the armature, resulting in motion. DC motors are classified as either series-wound, shunt-wound, or compound-wound. In the series-wound motor the field windings and armature windings are connected in series so that all current that passes through the field windings also passes through the armature windings. The result is a powerful and efficient motor at high speed generating high torque for a given current. Speed will vary with load and can run away under no-load conditions. In the shunt-wound motor, the armature and field are both connected across the main power supply in parallel so that the armature and field currents are separate. Shunt wound motors generate the least torque for a given current but speed is quite constant with load and will not run away under no-load conditions. The compound-wound motor has a combination of series and shunt field windings resulting in a motor that is quite efficient and powerful, yet will not run away under no-load conditions.

14.2.2 <u>Types of Single-Phase AC Motors</u>

Most of the single-phase AC motors are induction motors distinguished by different arrangements for starting. Single-phase motors are used in sizes up to about eight horsepower for heavy starting duty, chiefly in home and commercial appliances for which polyphase power is not available.

The series wound single-phase motor has a rotor winding in series with the stator winding as in the series-wound DC motor. Since this motor may also be operated on direct-current, it is called a "universal motor". The series wound motor has a high starting torque and is used in vacuum cleaners, sewing machines, and portable tools. In the capacitor-start single-phase motor, an auxiliary winding in the stator is connected in series with a capacitor and a centrifugal switch. During the starting and accelerating period the motor operates as a two-phase induction motor. At about two-thirds full-load speed, the auxiliary circuit is disconnected by the switch and the motor then runs as a single phase induction motor.

14.2.3 <u>Types of Polyphase AC Motors</u>

The most extensively used polyphase motors are the induction type such as the squirrel cage induction motor introduced earlier. The wound-rotor type of induction motor has a squirrel cage and a series of coils set into the rotor, which are connected through slip-rings to external variable resistors. By varying the resistance of the wound-rotor circuits, the amount of current flowing in the circuits, and therefore the speed of the motor, can be controlled. Induction motors are manufactured with a wide range of speed and torque characteristics.

The synchronous motor is another type of polyphase AC motor. Unlike the induction motor, the rotor of the synchronous motor is connected to a DC supply which provides a field that rotates in step with the AC field in the stator. The synchronous motor operates at a constant speed throughout its entire load range, after having been brought up to this synchronous speed. This speed is governed by the frequency of the power supply and the number of poles in the rotor.

14.3 ELECTRIC MOTOR FAILURE MODES

A motor is one part of a system that includes its source of voltage, mounting assembly, shaft coupling and driven equipment. The operating environment includes ambient temperature, airborne contamination, shock and vibration. Motor reliability is usually associated with bearing life and the life of the windings before they require rewinding. Bearing failures are normally caused by poor maintenance practices such as allowing contaminates to enter the motor during lubrication, using the wrong grease or oil, and applying too much grease and not allowing the bearing to relieve the excess grease through the drain plug. Loading of the shaft is also a common bearing problem such as incorrect belt tension, dynamic overloading or misalignment. Chapter 7 provides the procedures for evaluating the bearing for reliability.

Another prominent failure mode for a motor is the shorting of the motor winding. Temperature rise is a function of the amount of heat generated in the rotor and stator per unit of time and the efficiency of the heat transfer system. Temperature rise of the windings is critical to motor reliability since insulation materials age over a period of time and this aging process is directly related to temperature. Eventually the materials lose their insulating properties causing a short circuit of the motor. Winding temperatures are related to the power losses of the motor (copper and iron). Copper losses occur as a result of the resistance of the winding and current flow. Iron losses (eddy current losses) are formed in the core of the motor. As these losses increase as a function of time and operation, the winding temperature increases. The permitted temperature rise of the windings is dependent on the class of insulation and temperature limits. Motor overheating is also caused by motor overloading, too high an ambient temperature and incorrect applied voltage. Table 14-2 provides temperature ratings for the various classes of motors.

Excessive frequency of starts and stops of the motor can also contribute to winding temperature rise without allowing the motor to cool between starts. The manufacturer will provide the maximum number of starts and stops per unit time as a function of load and speed. Limiting the frequency of startups per manufacturer's specifications provides adherence to predicted failure rate.

Other modes of motor failure are due to mechanical causes including unbalance, resonance and rotor deflection. The motor shaft is the most common mechanical problem caused by a worn or deformed shaft, incorrect coupling, shaft alignment, and overhung loads. Typical failure modes and their failure causes and effects are listed in Table 14-1. For additional information on individual parts of the motor, the particular chapter for that part should be reviewed as shown below:

1. Bearings (See Chapter 7, Section 7.5)
2. Windings (See Table 14.1 below and Section 14.5)
3. Brushes (See Table 14.1 below)

4. Armature (shaft) (See Chapter 20, Section 20.2)
5. Stator Housing (casing) (See Table 14.1 below)
6. Gears (See Chapter 8, Section 8.3)

Table 14-1. Electric Motor Failure Modes

FAILURE MODE	FAILURE CAUSE	FAILURE EFFECT
- Open winding - Shorted winding	- Insulation breakdown - High ambient temperature - High altitude - Mechanical overload - Frequent stops and starts - Dirt buildup on cooling fins - Vibration - Mechanical shock	- Motor won't start - Motor failure - Sparking at brushes
- Worn bearing: -- spalling --creeping or spin	- Excessive static load - Belt misalignment - Frequent starts and stops under heavy loads - Lubrication problem - Contamination - Overloading or high temperature	- Noisy - Heat build-up - Armature rubbing stator - Motor seized
- Cracked housing	- Fatigue - External shock - Vibration	- Leakage of dust into motor - Shorted or seized
- Sheared armature shaft - Cracked rotor laminations	- Fatigue - Misalignment - Bearing failure	- Seized - Armature rubbing stator
- Worn brushes -- brushes fail open	- Improper maintenance - Contamination - High temperature - Improper contact pressure	- Excessive sparking - Chatter or hissing noise - Motor runs too fast or too slow under load - Motor won't run

Table 14-1. Electric Motor Failure Modes (continued)

FAILURE MODE	FAILURE CAUSE	FAILURE EFFECT
- Noisy operation	- Worn or bent shaft - Shaft alignment - Mechanical vibration - Base plate distortion - Broken motor mounts	- Heat buildup
- Motor overheating	- Frequent starts - Incorrect supply voltage - High ambient temperature - Polyphase voltage unbalance > 1% - Motor overload - Blocked ventilation	- Short motor life - Motor failure
- Overload tripping	- Incorrect supply voltage - Excessive load speed	- Motor won't start
- Bearing failure	- Shaft misalignment - Incorrect coupling - Belt misalignment - Incorrect belt tension - Worn bearing	- Noisy operation - Motor failure

Additional details of failure modes for those components of a motor such as bearings and shafts are included in the applicable chapters of this Handbook.

14.4 MODEL DEVELOPMENT

The failure rate of a motor is affected by such factors as insulation deterioration, wear of sliding parts, bearing deterioration, torque, load size and type, overhung loads, thrust loads and rotational speed. The failure rate model included in this section is based upon identified failure modes of individual parts. The model developed is based on a fractional or integral horsepower AC type motor, although it will be general enough to be applied to most DC and AC motors.

The reliability of an electric motor is dependent upon the reliability of its parts, which may include: bearings, electrical windings, armature/shaft, housing, gears and brushes.

Failure mechanisms resulting in part degradation and failure rate distribution (as a function of time) are considered to be independent in each failure rate model. The total motor system failure rate is the sum of the failure rates of each of the parts in the motor:

$$\lambda_M = (\lambda_{M,B} \bullet C_{SF}) + \lambda_{WI} + \lambda_{BS} + \lambda_{ST} + \lambda_{AS} + \lambda_{BE} + \lambda_{GR} + \lambda_C \qquad \textbf{(14-1)}$$

Where:

λ_M = Total failure rate for the motor system, failures/million hours

$\lambda_{M,B}$ = Base failure rate of motor, failures/million hours (See Table 14-3)

C_{SF} = Motor load service factor (See Table 14-4)

λ_{WI} = Failure rate of electric motor windings, failures/million hours (See Section 14.5)

λ_{BS} = Failure rate of brushes, 3.2 failures/million hours/brush (Reference 68)

λ_{ST} = Failure rate of the stator housing, 0.001 failures/million hours (Reference 68)

λ_{AS} = Failure rate of the armature shaft, failures/million hours (See Chapter 20, Section 20.4)

λ_{BE} = Failure rate of bearings, failures/million hours (See Chapter 7)

λ_{GR} = Failure rate of gears, failures/million hours (See Chapter 8)

λ_C = Failure rate of capacitor (if applicable) See MIL-HDBK-217, (Reference 28)

14.5 FAILURE RATE MODEL FOR MOTOR WINDINGS

The life expectancy of a motor winding is primarily dependant on its operating temperature with respect to the permitted temperature rise of the winding. The temperature rise of the winding is a function of the motor design, insulation materials and the operating environment including shaft loading, duty cycle, altitude, and operating temperature. The insulation materials age over time and this aging process is directly related to temperature. Eventually, the materials lose their insulating properties and break down causing one or more short circuits.

Temperature rise occurs in a motor due to the losses that occur in the motor, normally copper and iron losses. The temperature inside the motor will depend on how effectively this heat can be removed by the cooling system of the motor. The difference between the internal and external temperatures is dependent on the thermal gradient and this difference is normally quite low.

The electric motor windings failure rate, λ_{WI}, is derived by Equation (14-2):

$$\lambda_{WI} = \lambda_{WI,B} \bullet C_T \bullet C_V \bullet C_{alt} \qquad \textbf{(14-2)}$$

Where: $\lambda_{WI,B}$ = Base failure rate of the electric motor windings, failures/million hours (See Section 14.5.1)

C_T = Multiplying factor which considers the effects of ambient temperature on the base failure rate (See Section 14.5.2 and Figure 14.1)

C_V = Multiplying factor which considers the effects of electrical source voltage variations (See Section 14.5.3 and Figure 14.2 for single phase motors or Figure 14.3 for three phase motors)

C_{alt} = Multiplying factor which considers the effects of operation at high altitudes (See Section 14.5.4 and Figure 14.4)

14.5.1 Base Failure Rate

$\lambda_{WI,B}$ is the base failure rate of the specific motor winding as supplied by the motor manufacturer. The winding will usually be specified in terms of expected life. The base failure rate is then:

$$\lambda_{WI,B} = \frac{1.0 \, x \, 10^6}{L_I} \qquad \textbf{(14-3)}$$

Where: $\lambda_{WI,B}$ = Failure rate, failures/million hours

and: L_I = Expected winding life, hours

If a manufacturer's winding life is not available, a winding life of 25,000 hours (failure rate $\lambda_{WI,B}$ = 40.0 failures/million hours) can be expected from most manufacturers (References 28 and 105). The multiplying factors for Equation (14-2) are described in the following paragraphs.

14.5.2 Temperature Multiplying Factor

Temperature is the primary factor that limits the life of motor windings. Heat causes the windings to age and deteriorate, so after time they break down and lose their insulation quality. When this happens, the related electrical components "short" and the motor burns out.

The manufacturer's rating of a motor based on insulation and expected life is provided in 25°C increments. The temperature rating for each class of insulation is defined as the maximum temperature at which the insulation can be operated to yield the rated winding life. The temperature rating for the various classes of insulation is shown in Table 14-2.

Table 14-2. Motor Insulation Ratings

Insulation Class	Temperature Rating *	Assumed Ambient Temperature **	Allowable Temperature Rise	Hot Spot Allowance
A	105° C	40° C	60° C	5° C
B	130° C	40° C	85° C	5° C
F	155° C	40° C	110° C	5° C
H	180° C	40° C	135° C	5° C

* Maximum operating temperature allowed which includes ambient temperature, allowable temperature rise and a hot spot allowance. Manufacturers may use a mixture of materials in their motors providing a higher allowable temperature than the listed temperature rating for the insulation which permits a higher allowable ambient temperature. Refer to manufacturer's specification if in doubt.

** After adding the ambient temperature + temperature rise + hot spot temperature, any difference between this sum and temperature rating can be applied to ambient temperature

Allowable temperature rise in Table 14-2 is the change in winding temperature from when the motor starts to the final elevated temperature under full load conditions. This temperature is above the ambient temperature around the motor prior to starting the motor. Temperature rise means that the heat produced in the motor windings, friction of the bearings, rotor and stator losses will continue to increase until the heat dissipation equals the heat being generated. A motor is designed so the temperature rise produced within the motor, when delivering its rated horsepower, and added to the industry standard 40^{o}C ambient temperature rating and an allowance for hot spots will not exceed the winding insulation temperature limit for that particular insulation class. There will be hot spots in the winding which are not measured by the normal resistance measurement that measures the mean temperature rise of the winding. Thus, a hot spot allowance is introduced into the difference between the temperature rating and the temperature limit (ambient temperature plus the temperature rise).

Ambient temperature in Table 14-2 above is the temperature of the air surrounding the motor or the room temperature in the vicinity of the motor. The temperature will obviously be higher than room temperature if the motor is operating in an enclosed cabinet. Ambient temperatures in Table 14-2 are assumed values established sufficiently high for most applications so that the following relationship between the temperatures can be used to prevent motor overheating.

Temperature Rating = Ambient Temperature + Temperature Rise + Hot Spot Allowance.

The winding temperature is determined by measuring both the ambient and the hot temperature resistances of the windings. The resistance measurement gives an average temperature which is more representative than spot measurements with a thermometer. This method has become standard because of the dimensional restrictions of so many motor designs, which prevent the use of thermometers. Since temperature measurements provide an average temperature, a standard hot spot temperature is provided in the table to consider hot spots in the winding not detected by the resistance measurements.

The equation for determining the motor winding temperature from resistance readings is as follows:

$$T_{rise} = \frac{R_H - R_C}{R_C}(K + T_C) \tag{14-4}$$

Where:

T_{rise} = Temperature Rise, ^{o}C

R_H = Hot winding resistance, ohms

R_C = Cold winding resistance, ohms

T_C = Cold temperature, ^{o}C

K = 234.5 (a copper temperature coefficient)

Allowing a motor to reach and operate at a temperature 10^{o}C above its maximum temperature rating shown in Table 14-2 will reduce the motor's expected life by 50%

Operating at 10°C above this, the motor's life will be reduced again by 50%. The same relationship exists in reverse if the motor is operated at a temperature 10°C below the rated temperature. For example, if a manufacturer provides a motor with an insulation Class F for a B Class environment, the motor can be expected to last four times as long.

As a result of this relationship between temperature and winding life, a failure rate multiplying factor considering the motor operating temperature is given by:

$$C_T = 2^{(T_O - 40)/10} \qquad \textbf{(14-5)}$$

Where: T_o = Ambient temperature surrounding motor with motor running at expected full load conditions, °C (See notes following)

Note 1 – If the temperature rise of the motor is known as a result of hot and cold resistance measurements T_o can be adjusted accordingly. For example, if the temperature rise of a Class F motor is found to be 100°C as opposed to 110 °C as shown in Table 14-2, this 10 °C differential can be applied to T_o reducing its value by 10 °C.

Note 2 – In determining T_o the installation location of the motor must be considered such as inside an equipment cabinet, next to other operating equipment, etc.

Figure 14.1 shows the effect of temperature on the base failure rate.

14.5.3 Voltage Multiplying Factor

To drive an existing mechanical load connected to the shaft, a motor must draw a fixed amount of power from the source. When the motor is subjected to voltages below the rated value, current must be increased to provide the same amount of power. This increase in current is a problem only if that current exceeds the motor's current rating. When the amperage is above the rated value, heat begins to build up in the motor increasing the probability of motor winding failure.

If the existing mechanical load is light, a decreased supply voltage will create an increase in current in approximately the same proportion as the voltage decrease. This change does not create a problem if the current remains within rated value. However, for a heavy mechanical load, the winding current is already high, possibly causing a voltage that may be lower than that without the load. Thus, any further decrease in voltage increases the probability of a current above rated value and overheating of the motor. Also, the current required to start a motor is much higher than for steady-state operation. Frequent starting as a result of a high operational duty cycle can contribute to an increase in voltage drop and must also be considered in determining the difference between actual and rated source voltage.

When the motor is subjected to a source voltage higher than rated value, the magnetic field of the motor can approach flux density saturation causing the motor to draw more current in an effort to magnetize the iron beyond the saturation point. This increase in amperage creates a corresponding increase in the probability of motor overheating. A failure rate multiplying factor can be established for those situations when the actual voltage is expected to differ from rated voltage:

For single phase motors:

$$C_V = 2^{10\left(V_D/V_R\right)} \tag{14-6}$$

Where V_D = Difference between rated and actual voltage

V_R = Rated voltage

Figure 14.2 shows the effect of voltage differential on the base failure rate.

For three phase motors, the voltage for all three phases must be equaled (balanced) so that the current values will be the same in each phase winding. When the voltages between the three phases are not equaled (unbalanced), the current increases dramatically in the motor windings causing premature motor failure. To determine the voltage unbalance, the average voltage of the three phases (AB, AC, BC) is measured and the average voltage determined. The average voltage is then subtracted from one of the voltages that will indicate the greatest voltage difference. This (greatest voltage difference divided by the average voltage) x 100 provides the percent voltage unbalance.

The industry standard recommends that the maximum voltage unbalance be limited to 1%. A motor should never be operated with a voltage unbalance exceeding 3%. A failure rate multiplying factor for voltage unbalance can be established as follows:

For three phase motors:

$$C_V = 1 + \left(0.40\, V_U\right)^{2.5} \tag{14-7}$$

Where: $V_U = \%\ \text{voltage unbalance} = 100 \text{ x } \dfrac{\text{greatest voltage difference}}{\text{average phase voltage}}$

And: V_U = 0% to 3%

Figure 14.3 shows the effect of voltage unbalance of the base failure rate.

14.5.4 Altitude Multiplying Factor

Motors operating up to and including 3,300 feet can be considered as operating at sea level. Above 3,300 the low density air does not allow a motor to cool as well as the air at sea-level. However, the decrease in ambient temperature characteristic of high altitudes somewhat compensates for the increase in temperature rise due to low air density. The altitude multiplying factor considers the difference between the effect of lower ambient temperature and loss of cooling capability. The following equation provides an altitude multiplying factor.

For operating altitudes > 3300 feet:

$$C_{alt} = 1.00 + 8x10^{-5} \; (a - 3300 \text{ ft}) \qquad \textbf{(14-8)}$$

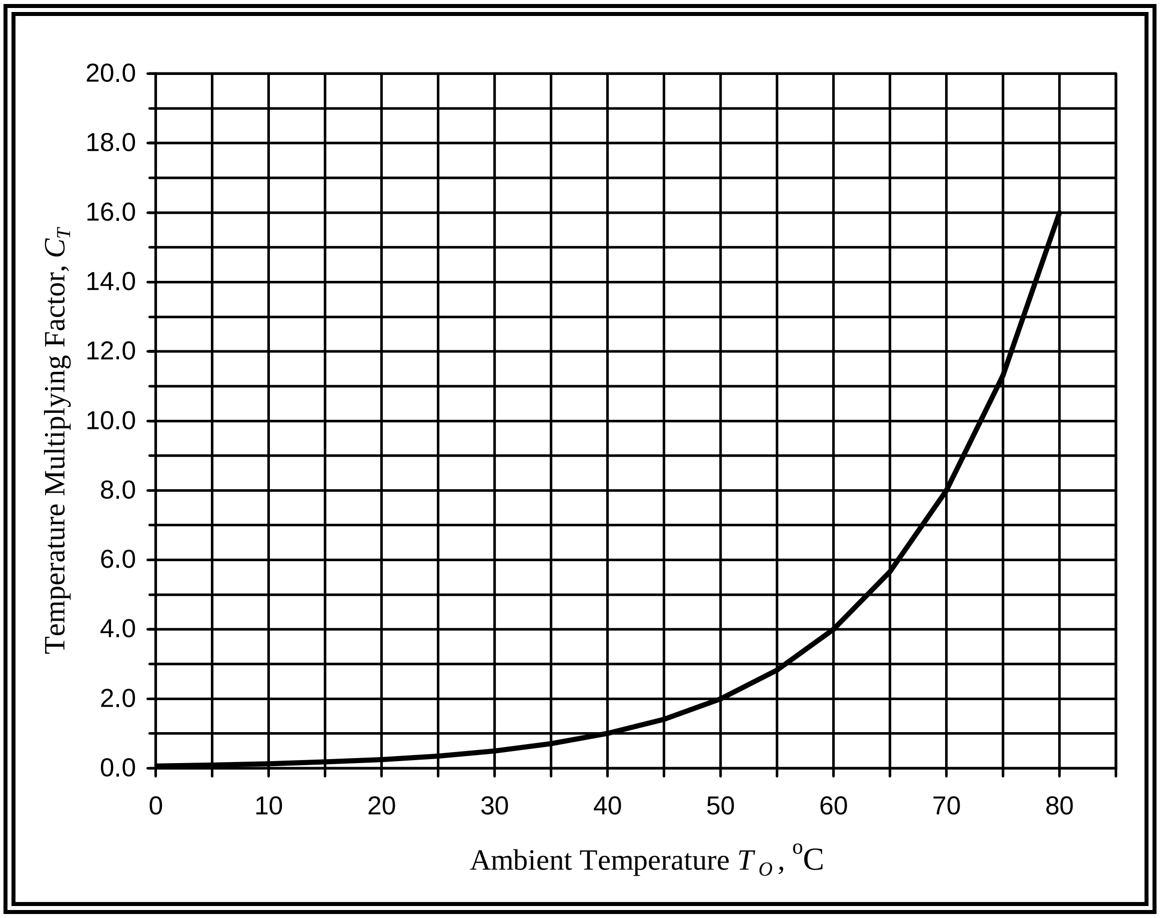
Where a = Operating altitude in feet

For altitudes $\leq$ 3300 ft, $C_{alt} = 1.0$

Figure 14.4 shows the effect of operating altitude on the base failure rate.

14.5.5 Motor Load

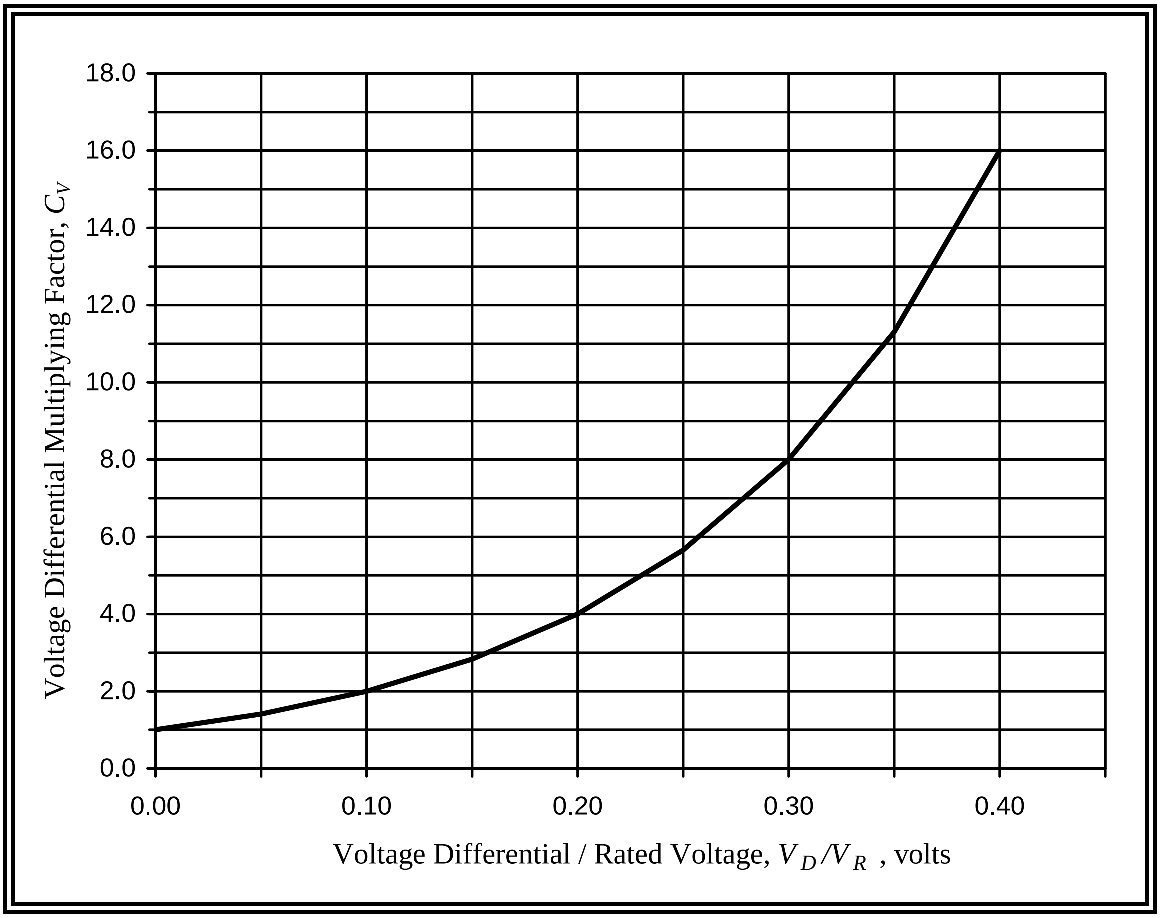
When a motor is operated at full load it has a given temperature rise. Operating the motor at loads above the manufacturer's specifications increases the motor temperature rise and increases the probability of early failure. It is assumed in the motor base failure rate equation that the torque and loading limits of the motor are not exceeded. Table 14-4 provides failure rate multiplying factors for severe loading.



$$C_T = 2^{(T_O - 40)/10}$$

Where: T_o = Ambient temperature surrounding motor with motor running at expected full load conditions (See notes following Equation 14-5)

Figure 14.1 Ambient Temperature Multiplying Factor, C_T

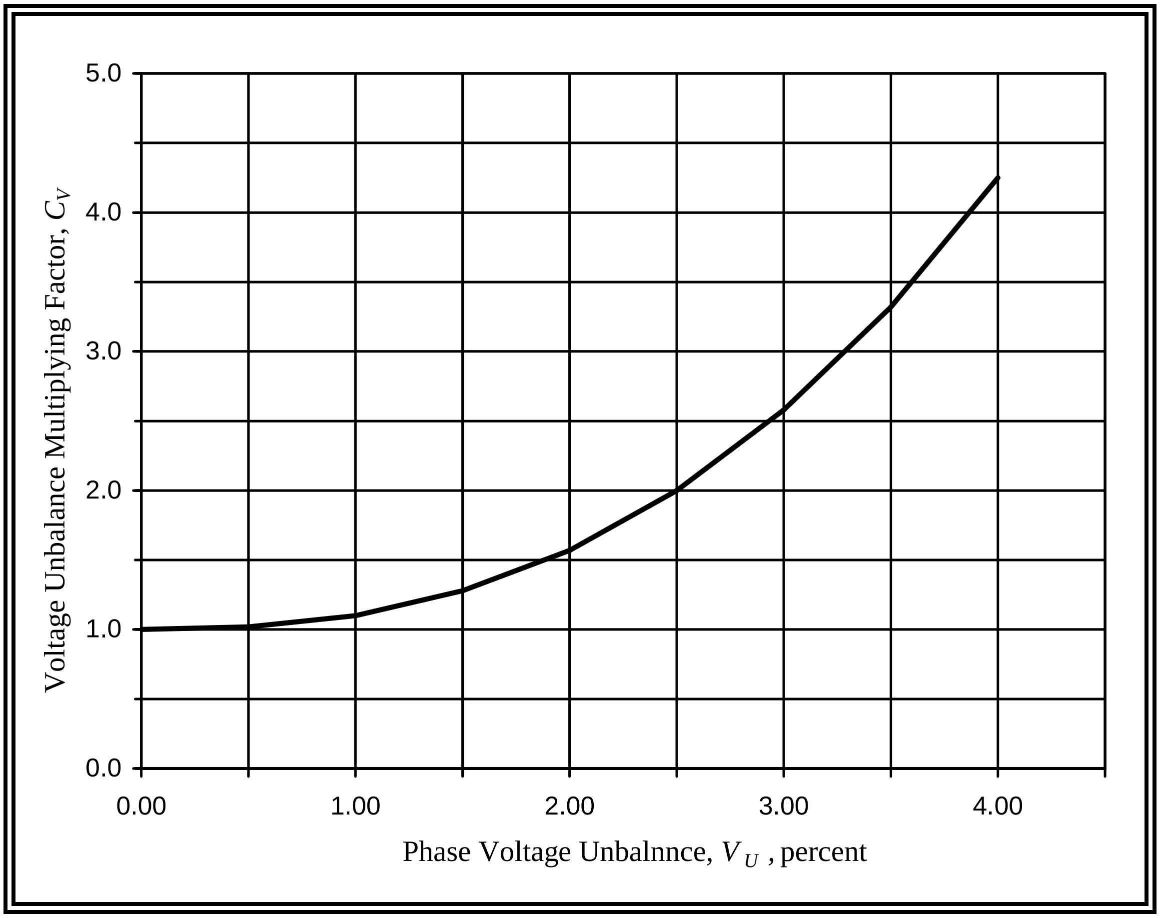


$$C_V = 2^{10\,(V_D/V_R)}$$

Where V_D = Difference between rated and actual voltage

V_R = Rated voltage

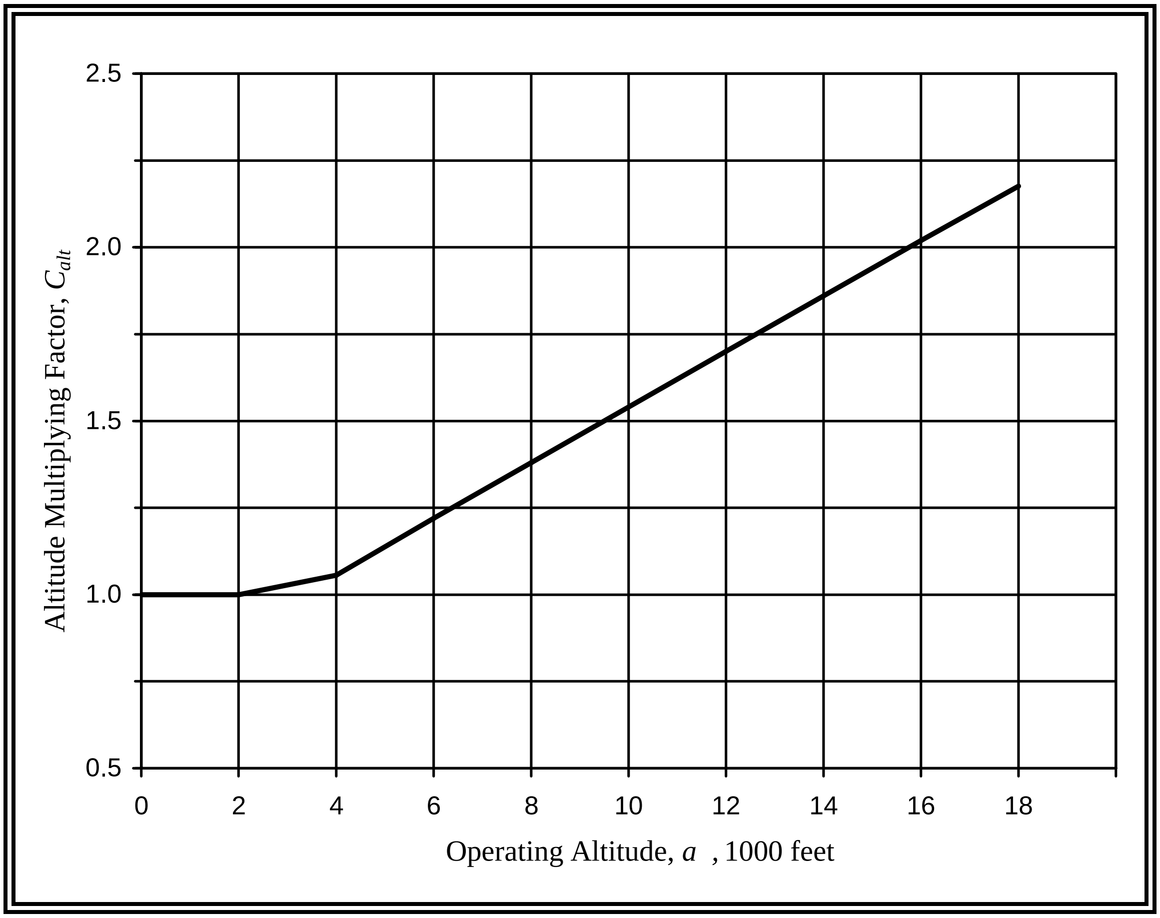
Figure 14.2 Supply Voltage Differential Multiplying Factor



$$C_V = 1 + \left(0.40\ V_U\right)^{2.5}$$

Where: V_U = % voltage unbalance = 100 x $\dfrac{\text{greatest voltage difference}}{\text{average phase voltage}}$

Figure 14.3 Voltage Unbalance Multiplying Factor



$$C_{alt} = 1.00 + 8x10^{-5} \ (a - 3300 \text{ ft})$$

Where a = Operating altitude in feet

Figure 14.4 Altitude Multiplying Factor

Table 14-3 Base Failure Rate of Motor, $\lambda_{M,B}$

Type of Motor	$\lambda_{M,B}$ (failures/million hours)
DC	2.17
DC brushless	1.75
AC single phase	6.90
AC polyphase	10.00

Table 14-4 Motor Load Service Factor, C_{SF}

Load Type	Load Description	C_{SF}
Uniform Load	One way continuous operation, minimal load fluctuation, no shock or vibration	1.00
Light Impact	Frequent starting and stopping, stepping motor operation, minimal shock and vibration	1.50
Medium Impact	Frequent bidirectional, reversible motor operation, moderate load impact ,shock and vibration	2.00
Heavy Impact	Subject to heavy vibration, shock loads, heavy load fluctuations	3.00

14.6 REFERENCES

In addition to specific references cited throughout Chapter 14, other references included below are recommended in support of performing a reliability analysis of electric motors.

28. MIL-HDBK-217, "Reliability Prediction of Electronic Equipment"

68. H. Wayne Beaty and James L. Kirtley, Jr., Electric Motor Handbook, McGraw-Hill Book Company 1998

105. OREDA Offshore Reliability Data, 5th Edition Det Norske Veritas, N-1363 Hovik, Norway 2009 ISBN 978-82-14-04830-8

127. Explaining Motor Failure, Austin Bonnett and Chuck Young EASA, EC&M Magazine, October 1, 2004

132. “Temperature Monitoring Is Key to Motor Reliability”, Thomas H. Bishop, Electrical Apparatus Service Association, Maintenance Technology Magazine, July 2004

This Page Intentionally Left Blank

CHAPTER 15

ACCUMULATORS

15.0 TABLE OF CONTENTS

15.1 INTRODUCTION

A hydraulic accumulator is a device designed to store energy. In a hydraulic system, fluid is pumped into the accumulator through a one-way valve. The accumulator holds the energy under pressure using a spring, compressed gas or raised weight which is used to exert a force against an incompressible fluid. The accumulator also has an output valve that is opened upon demand to supply fluid to other components of the system. An accumulator is used to store energy in such applications as:

- Fluid supply
- System pressure surge damping and shock suppression
- Stabilization of pressure fluctuations
- Leakage and thermal expansion compensation
- Emergency and standby power source

An accumulator is also used to smooth out the delivery flow of pumps. The pump may deliver the required power to the system in a pulsating flow which can produce pulsations detrimental to a high pressure system. An accumulator properly located in the hydraulic system will substantially cushion these pressure pulsations. In many hydraulic power applications the valve, actuator or other driven component stops or closes suddenly, creating a pressure wave that travels back through the system. This shock wave can develop peak pressures several times greater than normal working pressures causing objectionable noise or pump failure. An accumulator properly located in the hydraulic system will minimize this shock wave.

An accumulator design may include input and output valves, bladder, bleed plug, a high strength shell, piston, and fluid port. Typical accumulator designs are shown in Figure 15.1.

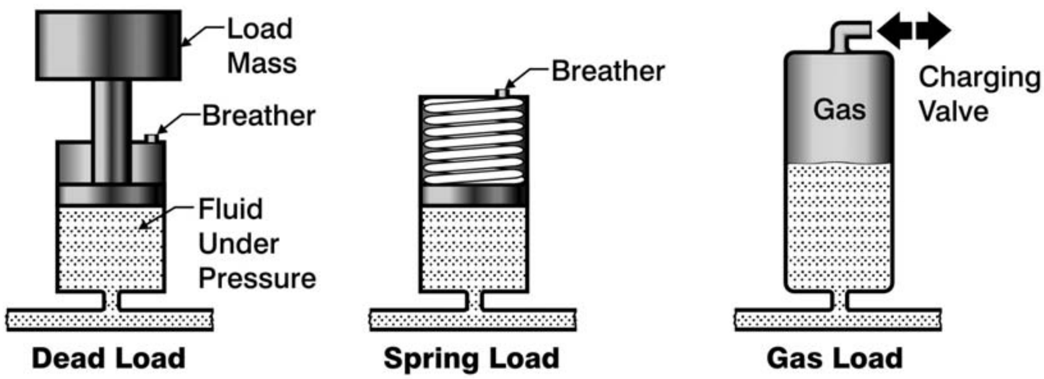


Figure 15.1 Typical Accumulator Designs

A dead load accumulator is comprised of a single acting vertical cylinder which raises a heavy load mass. It can be designed for large volumes but correspondingly heavy weights are needed resulting in a large physical size. The advantage of this type of accumulator design is the constant discharge pressure, whereas all other types exhibit a variation in pressure with respect to volume of fluid stored.

A spring loaded accumulator contains a single spring or several springs which move within a cylinder. As the volume of fluid in the accumulator is increased, the spring is compressed and the spring force is increased. The minimum pressure in the accumulator depends on the designed spring preload. The piston stroke and, therefore, the volume of fluid which can be stored is limited by the physical characteristics of the spring.

A gas loaded accumulator is designed to utilize a compressed gas such as nitrogen or air to pressurize the stored fluid. The accumulator may be a piston type, diaphragm type or bladder type to separate the noncompressable fluid and gas. Gas loaded accumulators can be very large. As discussed in the next section, accumulators are usually designed to be operated in the vertical position. The fluid pressure as a function of fluid volume in a gas loaded accumulator depends upon many factors such as the gas being used, the temperature of the gas and its pressure-volume characteristics.

15.2 FAILURE MODES

Accumulator failure is often defined as the inability to accept and exhaust a specified amount of fluid when operating over a specific system pressure range. Failure often results from unwanted loss or gain of precharge pressure. Correct precharge pressure is probably the single most important fact in prolonging the life of the accumulator.

In any type of accumulator utilizing a piston, the cylinder bore has to be machined, and wear will occur between the piston and cylinder body. Seals are built into the piston and these are subject to wear and leakage. Depending on the accumulator application, response time may be a factor. The response of the dead load accumulator will be somewhat slow due to the high inertia of the load and piston. The response of spring loaded accumulators will depend on the age of the spring and its modulus of rigidity. A response of a piston type accumulator will be adversely affected by the inertia of the piston and the effect of seal stiction. Typical accumulator failure modes are listed in Table 15-1.

If a piston type accumulator is inclined to a vertical position, the rate of wear will be increased due to the additional side load. Failure of a piston type accumulator tends to be gradual caused by deterioration of piston seals and wear in the cylinder bore. Failure of a diaphragm type accumulator will be more sudden, caused by the rupturing of the diaphragm. The failure rate of a diaphragm type accumulator may also depend on its physical characteristics.

A spring loaded accumulator must be evaluated closely for reliability to verify compatibility between the spring material and the surrounding medium. Any leakage past the seal could have a deteriorating effect on the spring material and its compression properties or fatigue life.

One of the main applications of an accumulator is the damping of fluid system pulsations or surges. The system effects of these pulsations must be evaluated as part of any reliability analysis. In some applications the pulsations are unimportant as they are partially smoothed by pipes upstream of the pump. A critical element of the reliability analysis is the probability of failure of other system components caused by accumulator failure. For example, a failed valve assembly within the accumulator which prevents fluid discharge may not be immediately detected and damage to other components may occur due to pressure transients. Shock waves produced as a result of the sudden closing of a downstream valve, for example, travels through the system fluid to the far end of the system and a decompression wave is formed which travels back to the valve. These waves travel back and forth until the energy is expended. The more rapid the valve closure, the more severe the pressure transient generated. Without detection of an accumulator failure, severe degradation and damage to system components could be occurring without operator or maintainer knowledge.

Table 15-1. Failure Modes for an Accumulator

FAILURE MODE	FAILURE CAUSE	FAILURE EFFECT
Seal Leakage	- Embrittlement - Wear - Distortion - Incompatibility with medium	- Leakage past piston - Internal leakage at valve - External leakage
Worn cylinder bore or piston surface	- Contaminants - Interaction with fluid medium	- Poor system response - Leakage - Loss of pressure
Loss of spring tension	- Corrosion - Fracture - Spring misalignment	- Poor system response - Inoperative accumulator
Piston stiction	- Surface wear - Corrosion	- Poor system response
Loss of pressure	- Ruptured gas bag - External leakage	- Poor system response - External component damage
Leakage of charge gas into fluid system	- Leakage past piston or bag - Sudden discharge of fluid from accumulator	- Poor system response - Contaminated fluid system
Inoperative accumulator	- Jammed output valve	- External component damage

15.3 FAILURE RATE MODEL

The failure rate of an accumulator will depend on several factors:

- Volumetric capacity
- Operating pressure
- Maximum flow rate

The failure rate of an accumulator is dependent on the sum of the failure rates of its component parts:

$$\lambda_A = \lambda_{SE} + \lambda_{SP} + \lambda_{PC} + \lambda_{VA} + \lambda_{CW} \qquad \textbf{(15-1)}$$

Where:

λ_A = Total failure rate of accumulator, failures/million hours

λ_{SE} = Failure rate of seals, failures/million hours (See Chapter 3)

λ_{SP} = Failure rate of springs, failures/million hours (See Chapter 4)

λ_{PC} = Failure rate of piston/cylinder interface, failures/million hours (See Chapter 9)

λ_{VA} = Failure rate of control valve, failures/million hours (See Chapter 6)

λ_{CW} = Failure rate of cylinder wall, failures/million hours (See Section 15.4 for thin walled cylinders and Section 15.5 for thick walled cylinders)

15.3.1 Seals

Specific failure modes of seals and procedures to determine their failure rates under different operating environments are discussed in Chapter 3. Of particular interest in the design evaluation of accumulators and other pressure vessels is the compatibility of the fluid medium and the seal material. The position of the accumulator in the fluid system must also be known to determine the side load on the piston and corresponding stress on the seal.

15.3.2 Springs

Specific failure modes of springs and procedures to determine their failure rates under different operating environments are discussed in Chapter 4. For most accumulators the failure rate equations in Chapter 4 for static springs can be used. The reliability of a spring is very sensitive to corrosion and the compatibility of the fluid and spring material must be considered.

15.3.3 Piston and Cylinder

The wear rate of the piston surface and cylinder bore will be sensitive to the position of the accumulator in its operating environment. Tilting of the accumulator from its vertical position will alter the side load of the piston. This parameter and others affecting the reliability of the piston/cylinder are included in the reliability equations contained in Chapter 9.

15.3.4 Valves

The reliability of valve assemblies which may be contained within the accumulator is determined using the equations contained in Chapter 6. Rate of flow into and out of the accumulator will affect the overall compressor reliability. Another failure mode to be considered in the design evaluation is the possibility of a sudden discharge of fluid from the output valve causing it to operate without fluid and creating an air lock.

15.3.5 Structural Considerations

The fluid contained within an accumulator under pressure creates stresses in the walls as shown in Figure 15.2. The state of stress is triaxial. A longitudinal or meridional stress acts parallel to the meridian; a circumferential, or hoop stress acts parallel to the circumference; and a radial stress acts outward at the surface. If the walls of the accumulator are relatively thin (thickness *t* is less than one-tenth the radius r_i) and of uniform shape, longitudinal and circumferential stresses will be uniform throughout the thickness of the wall. The radial stress, although varying from zero at the outside surface to a value equal to the internal pressure at the inside surface can be considered negligible. Section 15.4 provides equations for determining the stress levels of thin walled pressure vessels ($t/r_i < 0.1$) and Section 15.5 provides equations for determining the stress levels of thick walled pressure vessels ($t/r_i > 0.1$).

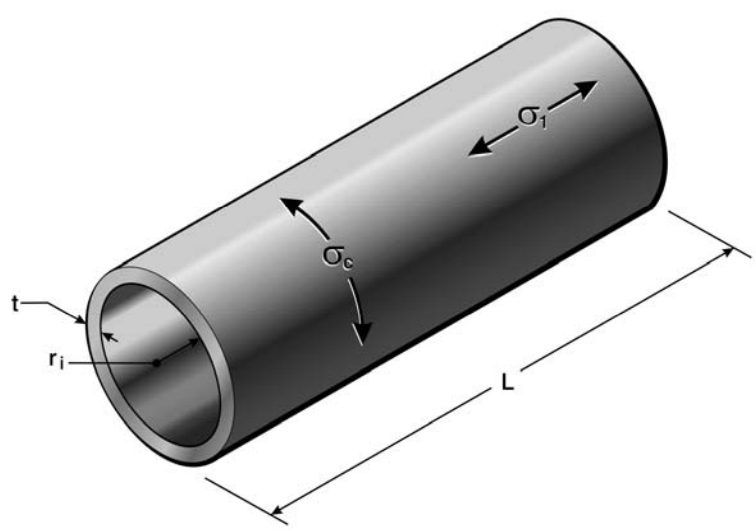


Figure 15.2 Stresses created in walls of accumulator

15.4 THIN WALL CYLINDERS

The shell thickness is designed to keep the maximum stresses below the yield strength of the material. The design thickness is the minimum required thickness computed by code formula plus an allowance for corrosion.

The walls of the accumulator will tend to expand in the radial direction when pressurized, causing the walls to stretch circumferentially. As a result of this radial expansion, stresses will occur acting in the circumferential direction. For thin walled cylinders, the circumferential strains are approximately the same at the inside and outside of the cylinder. Consequently, the circumferential stresses will be very nearly uniform throughout the wall thickness. The circumferential stresses can be related to the internal pressure of the accumulator by considering the equilibrium of a half cylinder shown in Figure 15.2 above. For equilibrium to occur, the resistive force due to circumferential stress acting on the cylinder wall must equal the force acting on accumulator wall as a result of applied fluid pressure. This equilibrium is shown with the following equation (Reference 38):

$$F_C = 2\pi \sigma_c t L \ - \ 2\pi P r_i L = 0 \qquad \textbf{(15-2)}$$

Where:

F_c = Circumferential force, lbs

P = Internal pressure, lbs/in^2

r_i = Internal radius, in

L = Cylinder length, in

σ_c = Circumferential stress, lbs/in^2

t = Wall thickness, in

Solving for σ_c, the circumferential stress, in the cylinder results in the following equation:

$$\sigma_C = \frac{P r_i}{t} \qquad \textbf{(15-3)}$$

Similarly, the equilibrium of forces in the longitudinal direction provides the following equation:

$$F_1 = \pi \ P \ r_i^2 \ - \ 2 \ \pi \ r_i \ t \ \sigma_l \qquad \textbf{(15-4)}$$

Where: F_1 = Longitudinal force, lbs

σ_l = Longitudinal stress, lbs/in2

and the corresponding longitudinal stress is:

$$\sigma_l = \frac{P\, r_i}{2\, t} \tag{15-5}$$

The effects of end plates and joints on the accumulator are a reduction in strength of the accumulator due to riveted joints, welding and other fabrication techniques. This reduction is accounted for by including a joint efficiency parameter in the circumferential and longitudinal stress equations:

$$\eta = \frac{\text{minimum strength of joint}}{\text{strength of solid material}}$$

Where: η = Joint efficiency parameter

The addition of η_c as a circumferential joint efficiency parameter provides the following equation:

$$\sigma_c = \frac{P\, r_i}{t\, \eta_c} \tag{15-6}$$

and the addition of η_l as a longitudinal joint efficiency factor provides the following equation:

$$\sigma_l = \frac{P\, r_i}{2\, t\, \eta_l} \tag{15-7}$$

The relative strength (efficiency) of a joint depends upon its design and type of joint. Table 15-2 provides values of η_c and η_l based on joint efficiencies that may be expected in the various types of joints if they are well designed. The actual value for the efficiency parameters, η_c and η_l, from Table 15-2 below can be adjusted depending on the confidence level in manufacturing techniques and quality control.

Table 15-2. Approximate Efficiencies of Joints (Ref. 57, 58)

TYPE OF JOINT	DESIGN (Number of Rows)	η_c, η_l
Riveted Lap Joint	Single	0.55
	Double	0.65
	Triple	0.75
Riveted Butt Joint	Single	0.65
	Double	0.75
	Triple	0.85
	Quadruple	0.90
Welded Butt Joint	Single	0.75
	Double	0.85

The ends of the accumulator are often hemispheres. The internal pressure in a thin spherical shell will create two mutually perpendicular circumferential stresses of equal magnitude and a radial stress. Again, a thickness/radius ratio of less than 1/10 provides a minimal value of radial stress. The resistive force due to circumferential stress, σ_c, acting on the accumulator wall to achieve equilibrium must equal the force on the hemisphere due to internal pressure, P:

$$F_{hc} = \pi P r_h^2 - 2 \pi r_h t_h \sigma_{hc} = 0 \qquad \textbf{(15-8)}$$

Where: F_{hc} = Circumferential force of the end section, lbs
P = Internal fluid pressure, lbs/in^2
r_h = Radius of the end section, in
t_h = Thickness of the end section material, in
σ_{hc} = Circumferential stress in the end section, lbs/in^2

Solving for σ_{hc}, with the addition of a joint efficiency parameter, provides an equation for maximum stress at the hemispherical ends.

$$\sigma_{hc} = \frac{P r_h}{2 t_h \eta_c} \qquad \textbf{(15-9)}$$

It will be noted that for the same wall thickness, the spherical ends of the accumulator provide twice the strength. The hemispherical ends, therefore, are sometimes thinner than the cylindrical section. Equations for various shapes of accumulators can be found in standard textbooks for s more detailed structural analysis.

The failure rate of the cylinder is determined by a base failure rate for the cylinder multiplied by stress level factors:

$$\lambda_{CW} = \lambda_{CW,B} + \lambda_c + \lambda_l \qquad \textbf{(15-10)}$$

Where: λ_{CW} = Failure rate of cylinder for use in Equation (15-1), failures/million hours

$\lambda_{CW,B}$ = Base failure rate of cylinder, 0.001 failures/million hours

λ_c = Failure rate considering compressive stress (See Section 15.6)

λ_l = Failure rate considering longitudinal stress (See Section 15.6)

15.5 THICK WALL CYLINDERS

If the wall thickness of the pressure vessel is more than one-tenth the radius, the circumferential and longitudinal stresses cannot be considered uniform throughout the thickness of the wall and the radial stress cannot be considered negligible. Reference 38 provides the equations for different shapes of thick walled containers:

$$\sigma_l = \frac{P r_i^2}{r_o^2 - r_i^2} \qquad \textbf{(15-11)}$$

Where: σ_l = Longitudinal stress, lbs/in^2

P = Internal pressure, lbs/in^2

r_i = Internal radius of the cylinder, in

r_o = External radius of the cylinder, in

$$\sigma_c = \frac{P r_i^2 \left(r_o^2 + r^2 \right)}{r^2 \left(r_o^2 - r_i^2 \right)} \tag{15-12}$$

Where: σ_c = Circumferential stress, lbs/in^2

r = Average radius of the cylinder, in

$$\sigma_r = \frac{P r_i^2 \left(r_o^2 - r^2 \right)}{r^2 \left(r_o^2 - r_i^2 \right)} \tag{15-13}$$

Where: σ_r = Radial stress, lbs/in^2

The failure rate of the cylinder is determined by a base failure rate for the cylinder multiplied by stress level factors:

$$\lambda_{CW} = \lambda_{CW,B} + \lambda_c + \lambda_l + \lambda_r \tag{15-14}$$

Where: λ_{CW} = Failure rate of cylinder wall for use in Equation (15-1), failures/million hours

$\lambda_{CW,B}$ = Base failure rate of cylinder, 0.001 failures/million hours

λ_c = Failure rate considering circumferential stress (See Section 15.6)

λ_l = Failure rate considering longitudinal stress (See Section 15.6)

λ_r = Failure rate considering radial stress (See Section 15.6)

15.6 FAILURE RATE CALCULATIONS

The structural aspects of the accumulator failure rate depend on the stress/strength relationships of the materials. The standard definition of reliability includes the probability that the strength random variable will exceed the stress random variable as shown in Figure 15.3.

$$R = P\left(S > s\right) = P\left(S - s\right) > 0 \tag{15-15}$$

Where: R = Reliability

P = Probability

S = Strength random variable

s = Stress random variable

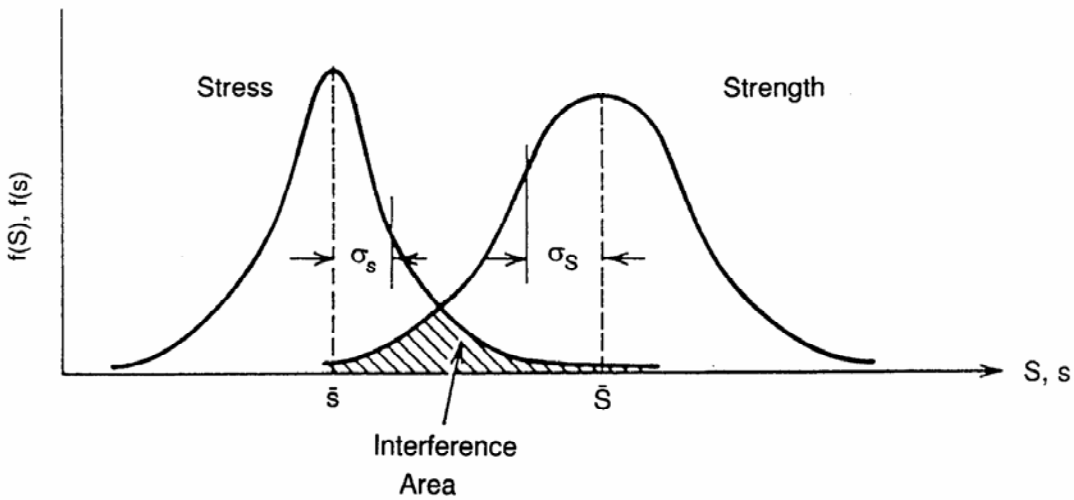


Figure 15.3 Stress Strength Relationship

The stress variable includes any parameter that tends to introduce a failure of the accumulator while the strength variable indicates any parameter resisting failure. Failure is defined to have occurred when actual stress exceeds actual strength for the first time. The ratio of strength to stress provides a safety factor:

$$n = \frac{F_y}{\sigma_x} \tag{15-16}$$

Where: n = Factor of safety

F_y = Material yield strength, lbs/in^2

σ_x = Mean value for the stress, σ_c, σ_l, or σ_r, lbs/in^2

A reliability analysis must include an estimate of the tail probabilities for stress and strength variables based on previous experience and intimate knowledge of the design and operating environment. The lower and upper limits on these probabilities quantify the uncertainty of the estimates. The probability distributions of yield strengths for steels are found to be normally distributed.

The standard normal variable of (S - s) will be equal to:

$$z = \frac{\mu_S - \mu_s}{\sqrt{\sigma_S{}^2 + \sigma_s{}^2}} \quad \textbf{(15-17)}$$

Where: z = Probability density function

μ_S = Mean value of F_y, lbs/in^2

μ_s = Mean value of σ_x, lbs/in^2

σ_S = Standard deviation of strength

σ_s = Standard deviation of stress

The probability density factor can be converted to reliability and failure rate using cumulative standard normal distribution tables and assuming $R = e^{-\lambda t}$.

15.7 REFERENCES

In addition to specific references cited throughout Chapter 15, other references included below are recommended in support of performing a reliability analysis of accumulators.

22. Howell, Glen W. and Terry M. Weathers, Aerospace Fluid Component Designers' Handbook, Volumes I and II, TRW Systems Group, Redondo Beach, CA prepared for Air Force Rocket Propulsion Laboratory, Edwards, CA, Report AD 874 542 and Report AD 874 543 (February 1970)

38. Roack and Young, Formulas for Stress and Strain, McGraw-Hill Book Company, New York (1975)

46. Fox, R.W., and A.T. McDonald, Introduction to Fluid Mechanics, John Wiley and Sons, New York (1978)

57. Deutschman, A.D., et al, Machine Design, Theory and Practice, MacMillan Publishing Co., NY (1985)

58. Parmley, R.O., Mechanical Components Handbook, McGraw-Hill Book Co., NY (1985)

This Page Intentionally Left Blank

CHAPTER 16

THREADED FASTENERS

16.0 TABLE OF CONTENTS

16.1 INTRODUCTION

Methods of fastening or joining components include bolting, brazing, soldering, and bonding with adhesives. One advantage of threaded fasteners is that they permit disassembly of the equipment for maintenance and repair. Threaded fasteners also allow the use of automated as well as standard manual tools for assembly and installation procedures enhancing operational reliability. The reliability of threaded fasteners in a given operating environment depends on the strength of materials, methods of fabrication and assembly, and the stress levels created by fatigue loads and environmental conditions. This chapter considers the reliability performance of externally and internally threaded fasteners. Other fasteners such as rivets and pin fasteners are discussed in Chapter 23.

16.1.1 Externally Threaded Fasteners

Bolts and screws are fasteners with a formed head on one end and an external thread on the other end. Studs are fasteners that incorporate external threads at each end. Structural bolts and screws are installed through prepared holes in the material to be joined. Various lengths of bolts and screws are produced to accommodate the thickness of the material to be fastened and the additional length of thread needed for proper engagement with the nut or with the internal tapped thread. These fasteners are subjected to tensile, shear, bending, and fatigue loads sensed by the joint. Failure rate is also subject to the environment imposed on the joint, which may include temperature extremes or exposure to various corrosive conditions.

16.1.2 Internally Threaded Fasteners

Specific fastening devices which incorporate internal (female) threads include nuts and inserts. They are intended to engage with the external threads of bolts, screws, and studs, and should be compatible to develop the full rated strength of the external thread.

16.1.3 Locking Mechanisms

One aspect of threaded fastener performance that is of concern is the fastener's susceptibility to loosening as a result of severe vibration or dynamic loadings acting on the joint. Since vibratory stresses cannot be totally eliminated, several methods have been developed which have proven effective in maintaining fastener integrity.

Self-locking nuts are integral fasteners which incorporate in the nut element a controlled high-torque feature which is designed to prevent rotation of the external threads, even if the initial tightening torque is completely relaxed. The same principle of an inherent self-locking feature is often extended to screws used in tapped holes.

Chemical thread-locking systems include adhesives which are applied to the fastener threads before installation, and which cure and effect a permanent bond after assembly.

Cotter pins are a supplemental locking device used with a slotted or castellated nut. They are installed through a drilled hole in the bolt threads to prevent rotation or movement of the nut after installation.

Safety lock wiring is also a supplemental locking system. Usually two or more fasteners in series must be wired to prevent rotation in the "off" direction. The nut end can be wired using slotted or castellated nuts and bolts with drilled holes. However, the predominant use of safety wiring is to secure screw heads which have been drilled to accommodate the wire where the screws have been installed in tapped or blind holes.

Lockwashers are used to resist the loosening effects of vibration, and come in a wide variety of designs for various applications. Conical spring washers are typically made of hardened and tempered steel that is slightly concaved. These washers deform when the bolt is tightened, acting as a spring that compensates for small losses in bolt tension due to thermal expansion or compression set of the gaskets. Helical spring washers employ a single-coil helical spring that flattens under load. The spring action assists in maintaining the bolt load, while the split edges provide a locking action by biting into the bearing surfaces. Toothed lockwashers provide a gripping action resulting from the teeth biting into the material of the bolt head or nut, and being deformed axially, with the application of tension from bolt torquing.

All locking systems should be compatible. For example, using an elastic stop nut (a nut with a deformable plastic insert) on a male thread drilled for use with a cotter pin, may destroy the holding capability of the nut because of damage to the insert.

16.1.4 Threads

The generally accepted standard for screw threads is the Unified thread form identified as "UNC" for the coarse-thread series and "UNF" for the fine-thread series. In addition to establishing diameter-pitch combinations, the Unified screw-thread system also defines the distinct profile and identification requirements for several screw-thread forms.

The relationship between nominal diameter and the number of threads per inch is referred to as the "diameter-pitch combination". There are several prominent thread forms which cover the majority of standards intended for general engineering use.

Coarse-thread series - This is perhaps the most widely used series of commercial and industrial fasteners. The thread form is particularly advantageous for applications requiring rapid assembly or disassembly, or for threading into lower-strength materials, such as castings, soft metals, and plastics.

Fine-thread series - For the same nominal diameter, this series incorporates more threads per inch. The result is a larger tensile stress area than that of the same size coarse thread, contributing to the greater strength capability of fine-thread fasteners. These fasteners are normally used where the length of thread engagement is short, or where a smaller lead angle is desired. Fine-thread-series fasteners are used extensively in aerospace and in applications where coarse threads would not be suitable. Other series include the extra-fine-thread, 8-thread, 12-thread, and 16-thread series.

Selection of the fastener to be used in a particular application depends on several factors:

- Function of the fastener
- Configuration of the joint being fastened
- Thickness and type of materials to be joined
- Type of loading on the fastener
- Operating environment of the fastener

16.2 FAILURE MODES

A fastener may experience either static loading or fatigue loading. Types of static loading include tension, shear, bending, torsion or a combination of these loading types. Types of fatigue loading include shock and vibration. Other common causes of fastener failure include environmental problems, manufacturing errors, incorrect installation and improper use of the equipment. The following sections provide a more detailed description of fastener failure modes. A summary of failure modes, causes and effects is included in Table 16-1.

16.2.1 Hydrogen Embrittlement

The phenomenon of hydrogen embrittlement in threaded fasteners has been mostly associated with high-strength (over 160 ksi) steel parts which have been furnished with either zinc or cadmium electroplating. During the plating process, atomic hydrogen can be trapped in or under the plating. Other sources of hydrogen can be traced to material pickling or alkaline or acid cleaning. Unless the free hydrogen is removed, when the fastener is used and stressed, as in a structural application, the hydrogen can attack the grain boundaries. The result is rapid crack propagation and often-catastrophic failure of the steel fastener.

Two major ways to avoid the problems of hydrogen embrittlement are to (1) bake the fasteners in a subsequent operation after plating to remove the excess hydrogen or (2) use mechanical plating processes in lieu of electroplating. Mechanical plating is the process of using glass beads to cold-weld a ductile metal (e.g., cadmium, zinc, etc.) onto a metal substrate by mechanical energy. Coating thicknesses are more uniform than when the hot-dip process is used. However, hot-dip fasteners have more corrosion protection built-in, due to the greater coating thickness.

16.2.2 Fatigue

When a failure of a threaded fastener occurs it is invariably a catastrophic failure occurring without warning. It is therefore important to consider the fatigue strength properties of the threaded fastener during reliability estimates. Research has established that rolled threads exhibit higher fatigue life than machined or ground threads. Further, threads rolled after heat treatment show better fatigue performance than comparable threads rolled before heat treatment. In addition, factors such as proper bolt head design, cold work of the head-to-shank fillet, quality control of the basic material used, and minimization of possible metallurgical defects all contribute significantly to improved fatigue life.

With respect to fatigue performance, it has been observed that failures normally develop at stress levels well below the static strength of the fastener. The two main types of joint fatigue loading are shear fatigue and tension fatigue. For shear-loaded joints, fatigue failure normally occurs in the plate or sheet material. Some of the factors which influence shear joint fatigue life include the applied fatigue or dynamic stresses, hole preparation, hole clearance, amount of induced bending, and fastener preload.

16.2.3 Temperature

Threaded fasteners will encounter both high-temperature and low-temperature service exposures in practice. Characteristically, materials used at very low temperatures will show an increase in tensile strength, but may sacrifice ductility. Conversely, at elevated temperatures, tensile strength properties are usually reduced, and above critical service temperature limits they may drop off dramatically.

16.2.4 Load and Torque

For every fastener system, there is an optimum torque range to develop the design clamp load. This is normally referred to as the "torque-tension relationship". Over torquing can result in excessive bolt yielding and possible subsequent relaxation, or even thread stripping and failure on installation. Too low an initial torque can contribute to potential fatigue and/or joint loosening with extended service life and vibration.

There are several factors which affect and influence the nominal torque-tension relationship, including condition of the threads, condition and squareness of the joint, method and equipment for torquing, installation from the nut or bolt head end, and lubrication. Possibly the most influential factor is the lubrication (plating and/or supplemental lubricant) on the fastener system, since the effective coefficient of friction can alter the installation torque requirements by as much as 50 to 100 percent.

Table 16-1 Threaded Fastener Failure Modes

Failure Mode	Failure Cause	Failure Effect
Male thread fracture	- Poor quality material - Corrosion - Fatigue	Eventual failure from shock, vibration
Broken shank	- Poor quality material - Over-torquing	Immediate failure, fastener replacement
Self loosening	- Under-torquing - Vibration	Loose joint; joint flexure under fatigue
Fractured bolt head, cracked nut	- Excessive preload	Immediate failure, fastener replacement
Thread stripping of external or internal thread	- Production variations in threaded assembly (bellmouthing) - Material tensile/shear strength variations - Radial displacement of nut - Excessive tensile force	Reduced thread strength, eventual failure
Loosening joint over time	- Overloading that exceeds preload - Insufficient preload - Poor quality washers (brinelling)	Severity of loose joint depends on application
Reduced fatigue life	- Bolt too tight - Bolt too loose	Eventual assembly failure
Galling during assembly or removal	- Too fine a thread for application - Incorrect material combination	Inability to remove fastener
Surface bearing deformation	- Bolt too tight	Fluid leakage
Crushed gasket	- Bolt too tight	Fluid leakage

16.2.5 Bolt and Nut Compatibility

Particularly where high-strength bolts are used, critical attention must be given to specifying the correct mating nut. Inadvertent specification of a lower-strength (grade) nut invites the possibility of nut thread stripping under high tensile loading. But more significantly, a weaker nut will not adequately develop the full clamp load capability of a high-strength bolt when subjected to the necessary installation torque. As a rule of thumb, the thickness (height) of a nut should approximate the diameter of the equivalent mating bolt to develop the full tensile strength properties of the bolt, if the bolt and nut materials have the same strength.

16.2.6 Vibration

Whereas fatigue loading is presumed to be relatively high with respect to the strength of the threaded fastener or the joint, vibration loads are relatively low, but may be associated with various ranges of cyclic frequencies. Critical combinations of frequency, loading, and amplitude can force a structure into resonance, often with catastrophic results. While the overwhelming majority of operating structures are not subjected to conditions of resonance, the vibration forces present (including random and steady-state vibration, shock, and impact) are sometimes serious enough to drastically affect the threaded fastener system.

Under repeated or extensive vibration, there is a tendency for the nut to rotate or loosen off the bolt threads. Continued vibration can actually result in the nut completely disengaging from the bolt, with subsequent loss of the bolt from the joint. Not as severe, but just as important, vibration loosening can reduce or completely relax the original preload in the bolt, causing the bolt to sense increased fatigue loads with continued exposure. What may have first started as vibration loosening may actually end as a fatigue failure because of the complex stress mechanisms involved.

16.3 STRESS-STRENGTH MODEL DEVELOPMENT

16.3.1 Static Preload

The most important factor that determines the preload induced in a bolt is the torque applied to tighten the bolt. As torque is applied to the nut, the bolt stretches creating a preload. The common method used to apply a predetermined torque is by means of a wrench which has a dial attachment that indicates the magnitude of torque being applied. The following empirical equation indicates the relationship between induced preload and applied torque.

$$T = d\,F_i\,c \tag{16-1}$$

Where:

T = Applied torque, in-lb

d = Major bolt diameter, in.

F_i = Initial preload, lb

c = Torque coefficient, See Table 16-3

For most applications, the value for the torque coefficient can be approximated by the values listed in Table 16-3. The values presented are typical because of the wide range of values reported from coating manufacturers. The user is cautioned to consult the manufacturer in critical applications.

A simple bolted joint can be dangerous unless it is properly designed for the loading and is assembled by a trained mechanic. In any fastening situation, the basic aim is to determine as accurately as possible the least expensive fastener that, when properly tightened, will secure a joint during product life. Properly applied assembly torque produces the wedging action of the fastener threads that elongate the bolt to produce tension. Tension (or preload) induced in a fastener at assembly should always be greater than any external load the joint will experience in service. A preload ensures optimum performance if it prevents the clamped parts from separating in service. However, too much preload can cause damage to the bolt, stripped threads and fracture of the material to be joined.

16.3.2 Temperature Effects of Fasteners in Clamped Joints

If the threaded fastener is to encounter temperatures above 400 degrees F, temperature changes must be considered in fastener joint design, in that they can act to change the joint clamping force and/or the fastener preload. If at any time during operation, the external loading exceeds the preload of the fastener at operating temperature, the fastener is considered to have failed. Therefore, the preload at ambient will need to be adjusted to compensate for the anticipated operating temperature. Correction factors for temperature effects can be modeled as follows:

$$F_{i,ambient} = \left[F_{i,operating} \bullet \frac{E_1}{E_2} \bullet \frac{1}{K_{SR}} \right] - \frac{A_S E_1}{L_E} \left(\Delta L_J - \Delta L_B \right) \tag{16-2}$$

Where:

$F_{i,ambient}$ = Preload at ambient temperature, lb

$F_{i,operating}$ = Fastener design preload at elevated operating temperature, lb

E_1 = Modulus of elasticity at room temperature, lbs/in^2 (See Table 16-2)

E_2 = Modulus of elasticity at operating temperature, lbs/in^2 (See Table 16-2)

K_{SR} = Correction factor accounting for stress relaxation in the fastener at elevated operating temperatures (See Figure 16.2)

A_s = Fastener tensile stress area, in^2

$$= 0.7854\left(d - \frac{0.9743}{N}\right)^2$$ (Reference 59) **(16-3)**

d = Major bolt diameter, in

N = Number of threads per inch

L_E = Effective length (as defined in Figure 16.1) of fastener, in

ΔL_J = Change in length or thickness of the joint, in

$$= L_G \bullet \alpha_J \bullet T_O \quad \textbf{(16-4)}$$

L_G = Thickness of clamped joint, in

α_J = Thermal coefficient of linear expansion for joint material (See Table 16-4)

ΔL_B = Change in grip length of the fastener, in

$$= L_G \bullet \alpha_B \bullet T_O \quad \textbf{(16-5)}$$

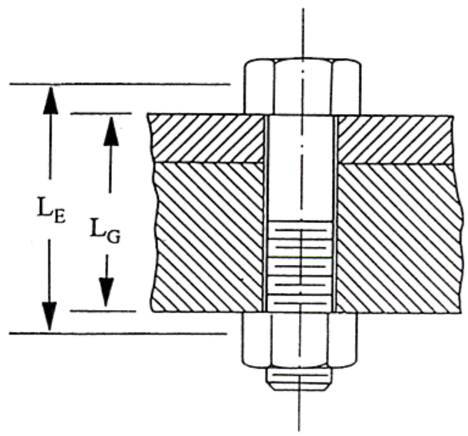
α_B = Thermal coefficient of linear expansion for bolt (See Table 16-4)

T_O = Elevated operating temperature, ^{o}F

Table 16-2 presents values of material moduli of elasticity vs. various temperatures. It should be noted that Equation (16-2) above will have lower preloads at room temperature assembly if the fastener joint is designed to be operated at a lower temperature (lower temperatures increase the stiffness of a bolted assembly). A reliability review must also consider the fact that the strength of most bolts decreases with rising temperatures.

Table 16-4 contains the values of coefficients of linear expansion for typical bolting material, α_B, and joint material, α_J. These values are used in computing ΔL_J and ΔL_B of Equation (16-2). It is necessary to note that the coefficients themselves are temperature dependent. The values in the table are based on room temperature. At elevated temperatures, with dissimilar joint/bolt materials, if the joint material expands more than the bolt material, the bolt will develop more stress or preload ($F_{i,operating}$) than it was designed to experience. To the contrary, if the bolt material expands more than the joint material at elevated operating temperatures, this will act to lessen the design bolt preload. The correction term in Equation (16-2) is therefore subtracted from the preload at assembly ($F_{i,ambient}$) to account for these situations.

At greatly elevated temperatures, many materials experience a slow increase in length under a heavy, constant load. This phenomenon is known as creep. A bolted joint assembly may experience a slightly different phenomenon, under which a steady loss of stress in a heavily loaded part whose dimensions are fixed or constrained, called stress relaxation. K_{SR} in Equation (16-2) accounts for this time/temperature dependent "stress relieving" factor when the bolted assembly is to be operated at extreme temperatures. Figure 16.2 contains several plots of correction factors for various bolt materials, derived from stress relaxation data over an elevated temperature range after 1000 hours exposure at each temperature. From the fairly constant initial ranges of each curve in Figure 16.2, it is evident that the stress relaxation effect is of concern only at the higher temperatures (i.e., 600°F and above).



$L_E = L_G$ + 1/2 (fastener head height + nut thickness), in

Figure 16.1 Determining Effective Fastener Length (L_E)

Stress relaxation losses are not repetitive in temperature cycling situations. The material stress value stabilizes at some lesser value after some period of time. This is because the tendency to relax decreases as the tensile stress (the driving force) in the bolt decreases.

16.3.3 Corrosion Considerations

Corrosion is another potential problem that needs to be addressed in evaluating threaded fasteners for reliability. Excessive corrosion can eventually lead to a reduction in preload or total loss of clamping force from material deterioration. A reliability review should verify the following:

(1) Materials in the joint assembly (bolts, nuts, structure) are identical, or as close together as possible in the galvanic series, minimizing electrical potential differences.

(2) In a situation of dissimilar metals, the larger amount of material should act as an anode, while the smaller amount of material behaves as the cathode.

(3) The installation of a sacrificial anode has been considered that can be replaced from time to time such as a block of material placed in the vicinity of the bolted joint, where material is sacrificed in a galvanic reaction.

(4) Stresses and/or stress concentrations in the fastener and joint are minimized by providing generous fillets, polishing surfaces, preloading bolts uniformly, etc. Stress concentrations at the root of a crack will tend to increase the galvanic reaction between the bolt and the adjacent material, aiding to the growth of fatigue cracks.

(5) Various coatings have been considered that can inhibit the process of corrosion.

(6) Periodic replacement of the bolts, prior to failure, has been considered based on prediction of life expectancy.

(7) Consideration that susceptibility to stress corrosion cracking can increase significantly with elevated temperatures.

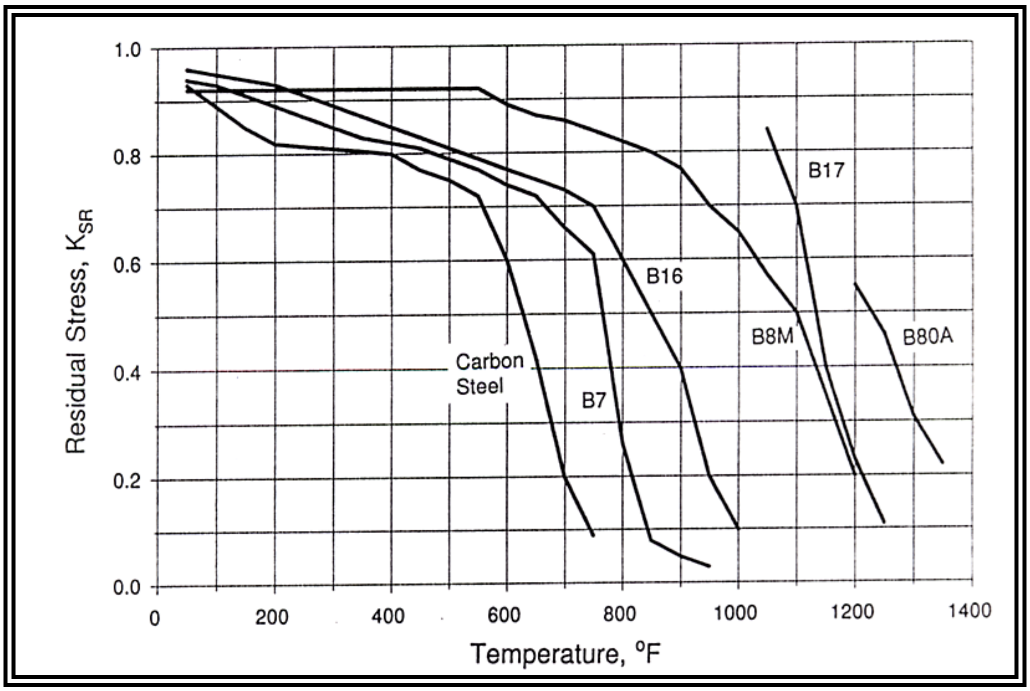


Figure 16.2 Stress Relaxation Factor, K_{SR}, for Various Operating Temperatures & Materials (After 1000 Hours) (Ref. 60)

There is a relationship between the depth of the threads on a fastener and its sensitivity to stress corrosion cracking. Generally, the larger diameter fasteners of a given material will have a lower threshold stress level than small bolts. For the same reason, fasteners with fine pitch threads are less sensitive than those with coarse threads.

16.3.4 Dynamic Loading

Machinery in operation is a dynamic situation. Threaded fasteners used in many applications have a dynamic load superimposed on a much larger static preload. These dynamic or fluctuating loads are augmented by stress concentrations and bending. In general, the higher the dynamic loads seen by the threaded fastener, the sooner it will fail. Whether a failure will occur or when depends on the extent and variation in dynamic loading. The usual method of determining fatigue life of a threaded fastener is to perform testing and establishing an S-N diagram, where S is the stress level and N is the number of cycles of applied load. Figure 16.3 is a typical S-N diagram showing the mean life of the fasteners under test. The curve shows that the cycle life will be very short when applied alternating stress levels are very high. As alternating stress levels are reduced, cycle life increases and below some stress level, the curve becomes flat and fatigue life becomes very large. This stress level is called the endurance limit of the fastener, and is defined as the complete reversing stress level below which fatigue life will be infinite.

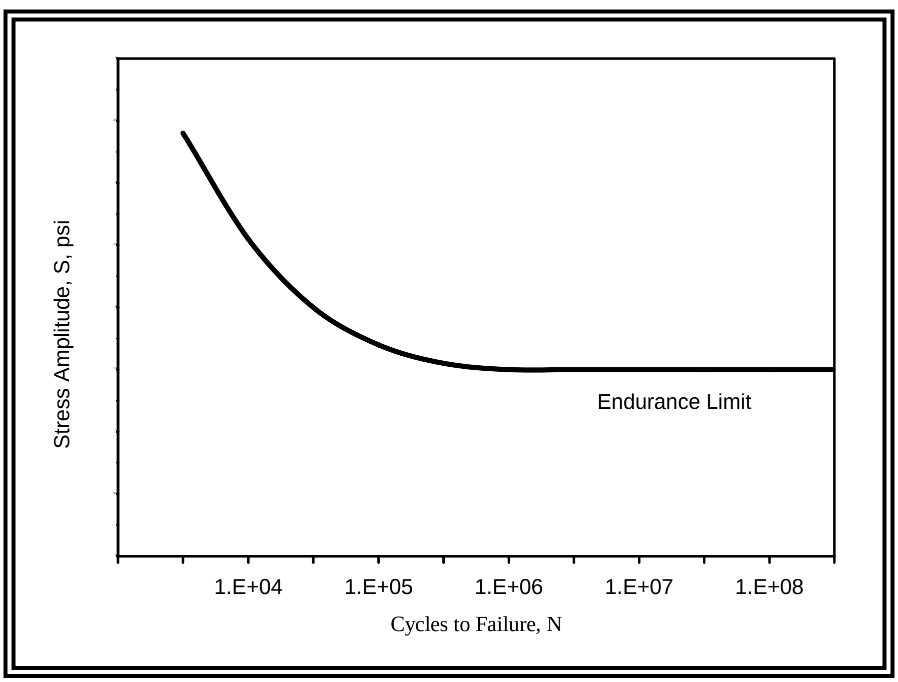


Figure 16.3 Typical S-N Diagram for Threaded Fastener

Unfortunately, endurance stress levels are usually only a small fraction of the static yield strength of the material and the S-N curves depend on the mean tension of the fastener in addition to the reversing stress level. The type of material used, heat treatment used and shape of the fastener all have an effect on the endurance limit. As a result it is not possible to use the S-N curve of the material being used as an estimate of the fatigue life of the specific threaded fastener.

16.3.5 Determination of Base Failure Rate

Data available from fastener manufacturers that have performed extensive testing can be used to determine the base failure rate, $\lambda_{F,B}$. However, since this data is somewhat specific to the conditions or environment of the test, it is often difficult to locate data that will yield the base failure rate of a specific type of fastener, under a particular set of loading conditions. This necessitates the development of a procedure to estimate the base failure rate of a generic fastener, with various characteristics and loading conditions.

During fatigue testing of a threaded fastener, stress values become less as the data is plotted against an increasing number of stress cycles as shown in Figure 16.3. While the graph becomes horizontal in the case of ferrous metals and alloys after the material has been stressed for a certain number of cycles, aluminum and other nonferrous materials do not have a horizontal asymptote, hence limit. To develop a model to predict fastener failures under dynamic loading, it is necessary to correlate the results of standard material fatigue tests to the geometry and loading conditions of the fastener to be used. It must be noted that even when the material of the test specimen and that of the mechanical fastener are identical, there will be significant differences between the fatigue curves for the two. Therefore, although correction factors will be presented in an effort to compensate for this, the user is cautioned that the analytical model developed will not yield absolutely precise results.

16.4 FAILURE RATE MODEL

Equation (16-6) provides a determination of an expected failure rate of a threaded fastener in hours of operation. Correction factors have been established to account for the individual contributions by surface finish, size differential, loading, temperature, etc.

$$\lambda_F = \lambda_{F,B} \bullet C_{SZ} \bullet C_L \bullet C_T \bullet C_I \bullet C_K \qquad \textbf{(16-6)}$$

Where: λ_F = Failure rate of fastener, failures/million hours

$\lambda_{F,B}$ = Base failure rate, failures/failures/million hours. See Table 16-5.

C_{SZ} = Multiplying factor considering the effects of size deviation from the S-N test specimen (See Section 16.4.1)

C_L = Multiplying factor considering the effects of different loading applications (See Section 16.4.2 and Table 16-6)

C_T = Elevated temperature multiplying factor (See Section 16.4.3)

C_I = Multiplying factor considering for the severity of in-service cyclic shock (impact) loading (See Section 16.4.4 and Table 16-7)

C_K = Stress concentration multiplying factor for fastener threads (See Section 16.4.5 and Table 16-8)

16.4.1 Size Multiplying Factor

Smaller machine parts tend to exhibit greater fatigue strength than larger ones, all other configurations and material properties being equal. Since larger surfaces have more defects overall, the probability of failure is greater in larger parts. The size multiplying factor is applicable to those situations where experimental fatigue life data is available and an adjustment is to be made reflecting a different size. A correction factor is established to account for a size differential, as well as the bending of a solid circular material, without constant rotation (Reference 39):

For bending or torsional loading:

$$C_{SZ} = \left(\frac{0.370\ d}{0.3} \right)^{-0.1133} \qquad \textbf{(16-7)}$$

Where: d = Major diameter of fastener, 2 inches or less

For axial loading:

$$C_{SZ} = 1.0 \qquad \textbf{(16-8)}$$

If Table 16-5 is used reflecting a standard grade of fastener, $C_{SZ} = 1.0$

16.4.2 Alternate Loading Multiplying Factor

S-N data for a specific threaded fastener may be based on loading in tension, compression, bending torsion or shear. The base failure rate shown in Table 16-5 assumes axial loading and Table 16-6 can be used to adjust the base failure rate for the appropriate loading.

16.4.3 Temperature Multiplying Factor

Typical rotating beam data are acquired at room temperature. However, fasteners are often called upon to clamp equipment at higher temperatures. Since a decline in static and dynamic strengths, creep, and thermal expansion must all be taken into account at higher temperatures, the following factor should be applied to the S-N test specimen data to achieve correction (Reference 39):

For steel operating above 160°F:

$$\text{For } T_O > 160^{o}\text{F:} \qquad C_T = \frac{460 + T_O}{620} \tag{16-9}$$

and:

$$\text{For } T_O \leq 160^{o}\text{F:} \qquad C_T = 1.0$$

where: T_O = Operating temperature of fastener, °F

16.4.4 Cyclic Shock/Impact Loading Multiplying Factor

A correction factor, C_I, must be applied when shock loads are present. In general, cyclic loads are less severe than applied shock loading. The factors are presented in Table 16-7.

16.4.5 Thread Correction Multiplying Factor

Observations of typical bolt failure pattern data from Reference 39 have revealed that only about 15% of failures occur under the head, due to the stress riser caused by the fillet. The risers found in the thread area accounted for the other 85%. Therefore, Table 16-8 presents the multiplying factor, C_K, for stress risers caused by the introduction of threads. The effects of notch sensitivity and surface finish have been incorporated.

Table 16-2. Elasticity Modulus (10^6 lbs/in^2) as a Function of Temperature
(Reference 60)

SPEC	GRADE	TEMPERATURE, degrees F					
		-325	-200	70	400	600	800
ASTM A193	B5	32.9	32.3	30.9	29.0	28.0	26.1
	B6	31.2	30.7	29.2	27.3	26.1	24.7
	B7	31.6	31.0	29.7	27.9	26.9	25.6
	B8-CL 1	30.3	29.7	28.3	26.5	25.3	24.1
	B16	31.6	31.0	29.7	27.9	26.9	25.5
ASTM A307		31.4	30.8	29.5	27.7	26.7	24.2
ASTM A320	L7	31.6	31.0	29.7	27.9	26.9	25.5
	L43	31.6	31.0	29.7	27.9	26.9	25.5
	B8	30.3	29.7	28.3	26.5	25.3	24.1
ASTM A325	Type 1,2,3	31.4	30.8	29.5	27.7	26.7	24.2
ASTM A354		31.2	30.8	29.3	27.5	26.5	24.0
ASTM A449		31.2	30.8	29.3	27.5	26.5	24.0
ASTM A453		30.3	29.7	28.3	26.5	25.3	24.1
ASTM A490		31.2	30.8	29.3	27.5	26.5	24.0
ASTM A540	B21,B22	31.6	31.0	29.7	27.9	26.9	25.5
	B23,B24	29.6	29.1	27.6	26.1	25.2	23.0
SAE J429	GR 1,2,4	31.4	30.8	29.5	27.7	26.7	24.2
	GR 5,7,8	31.2	30.6	29.3	27.5	26.5	24.0

Table 16-3. Typical Torque Coefficients
(Reference 60)

FASTENER MATERIAL/COATING	TORQUE COEFFICIENT, *c*
Aluminum on AISI 8740 alloy steel	0.52
Mild or alloy steel on steel	0.20
Stainless steel on mild/alloy steel	0.30
1" dia. A490	0.18
1" dia. A490 (rusty)*	0.39
Black Oxide	0.18
Cadmium plate (dry)	0.20
Cadmium plate (waxed)	0.19
Galvanized A325	0.46
Galvanized, hot-dip A325	0.09 - 0.37
Gold on stainless steel or beryllium copper	0.40
Graphitic coatings	0.09 - 0.28
Machine Oil	0.21
Moly paste or grease	0.13
Solid film PTFE	0.12
Zinc plate (waxed)	0.29
Zinc plate (dry)	0.30

* Exposed outdoors for two weeks

Table 16-4. Thermal Coefficients, α , of Linear Expansion (10^{-6} in/in/^{o}F) Evaluated at 70^{o}F (Reference 60)

SPEC	GRADE	TEMPERATURE, degrees F			
		70	400	600	800
ASTM A193	B5	6.5	7.0	7.2	7.3
	B6	5.9	6.4	6.5	6.7
	B7	5.6	6.7	7.3	7.7
	B8	8.5	9.2	9.5	9.8
	B16	5.4	6.6	7.2	7.6
ASTM A307		6.4	7.1	7.4	7.8
ASTM A320	L7	5.6	6.7	7.3	7.7
	L43	6.2	7.0	7.3	7.6
	L7M	6.2	7.0	7.3	7.6
	B8 CL 1	8.5	9.2	9.5	9.8
ASTN A325		6.2	7.0	7.3	7.6
ASTM A354		6.2	7.0	7.3	7.6
ASTM A449		6.2	7.0	7.3	7.6
ASTM A453		9.1	9.7	10.0	10.2
ASTM A490	651	6.2	7.0	7.3	7.6

Table 16-5 Fastener Base Failure Rate

Grade / Class	Diameter Inches	Proof Load kpsi	Base Failure Rate Failures/million hours
SAE 1	1/4 – 1 1/2	33	0.260
SAE 2	1/4 – 3/4	55	0.156
SAE 2	3/4 – 1 1/2	33	0.260
SAE 4	1/4 – 1 1/2	65	0.132
SAE 5	1/4 - 1	85	0.101
SAE 5	1 – 1 1/2	74	0.116
SAE 5	1 1/2 - 3	55	0.156
SAE 7	1/4 – 1 1/2	105	0.082
SAE 8	1/4 – 1 1/2	120	0.072
ISO 4.6	-- to 1 1/2	33	0.260
ISO 5.8	-- to 1 1/2	55	0.156
ISO 8.8	-- to 1 1/2	85	0.101
ISO 10.9	-- to 1 1/2	120	0.072
ASTM A325	1/2 - 1	85	0.101
ASTM A325	1 1/8 - 1 1/2	74	0.116
ASTM A354, BB	1/4 – 2 1/2	80	0.107
ASTM A354, BB	2 3/4 - 4	75	0.114
ASTM A354, BC	1/4 – 2 1/2	105	0.082
ASTM A354, BC	2 3/4 - 4	95	0.090
ASTM A354, BD	1/4 – 1 1/2	120	0.072
ASTM A490	1/2 – 1 1/2	120	0.072

Table 16-6. Load Multiplying Factors
(Reference 39)

TYPE OF LOAD APPLIED	C_L
Axial ($\sigma_{T,ult} \leq$ 220 kpsi)	1.09
Axial ($\sigma_{T,ult}$ > 220 kpsi)	1.00
Bending	1.00
Torsion and Shear	1.72

Table 16-7 Multiplying Factor for Impact Loading
(Reference 19)

IMPACT CATEGORY	C_I	C_I
	Normal Vibration	Continuous High Vibration
LIGHT (rotating machinery - motors, turbines, centrifugal pumps)	1.00	1.50
MEDIUM (rotary & reciprocating motion machines - compressors, pumps)	1.25	1.88
HEAVY (presses for tools & dies, shears)	1.67	2.50
VERY HEAVY (hammers, rolling mills, crushers)	2.50	3.75

Table 16-8. Multiplying Factor for Threaded Elements, C_K
(Reference 39)

SAE GRADE BOLT	ROLLED THREADS	CUT THREADS
	C_K	C_K
0-2	2.2	2.8
4-8	3.0	3.8

16.5 REFERENCES

In addition to specific references cited throughout Chapter 16, other references included below are recommended in support of performing a reliability analysis of threaded fasteners.

19. Hindhede, U., et al, Machine Design Fundamentals, John Wiley & Sons, NY, 1983.

39. Shigley, J.E., Mischke, C.R., Mechanical Engineering Design, McGraw-Hill Book Co., NY, 1989.

56. Haviland, G.S., "Designing with Threaded Fasteners", Mechanical Engineering, Vol 105, No. 10, Oct 83.

59. Handbook H28, Nat'l Bureau of Stds, Govt Printing Office, Washington, DC, 1957.

60. Bickford, J.H., An Introduction to the Design and Behavior of Bolted Joints, Marcel Dekker, Inc., NY, 1990.

61. Handbook of Corrosion Data, ASM International, Metals Park, OH, 1990

62. Baumeister, T, et al, Mark's Standard Handbook for Mechanical Engineers

63. Thomas Couplings Applications Manual

64. Bolam, J.R., "Coupling Alignment: The Reverse Indicator Method Simplified", P/PM Technology, July/Aug 90

65. Dvorak, P., "Sorting Out Flexible Couplings", Machine Design, 11 Aug 88

66. Robertson, R., and Smith, B., "Why Flexible Couplings Fail", Plant Engineering and Maintenance, Jun 89

94. Mechanical Designers’ Workbook, “Fastening, Joining & Connecting”, J. Shigley and C. Mischke, McGraw-Hill 1986

CHAPTER 17

MECHANICAL COUPLINGS

17.0 TABLE OF CONTENTS

17.1 INTRODUCTION

A mechanical system often requires an axial connection between two components such as a motor to a pump. The connection is accomplished between the two component shafts with use of a mechanical coupling. The coupling is designed to transmit power (torque) from one shaft to the other, causing both to rotate in unison and at the same RPM. Perfect alignment between the two shafts is almost impossible and the coupling will drift from its initial position to a degree depending on the application. If not compensated, mechanical wear of attached components will increase. Therefore, another purpose of the mechanical coupling is to compensate for minor amounts of misalignment and random movement between the two shafts.

Still another common purpose of a coupling is the provision of a break-point between the driving and driven shafts acting as a fuse if a severe torque overload should occur. This inclusion of a safety coupling can protect a serious high-cost equipment failure. Each of these coupling applications results in many unique designs of mechanical couplings depending on torque requirement, rotating speed, expected shaft misalignment, backlash limitations, vibration and other factors unique to the application and operating environment. A typical mechanical coupling is shown in Figure 17.1

There is no one type of coupling that can provide the universal solution to all coupling problems so there are many designs available for specific applications. Each coupling design has strengths and weaknesses that must be taken into consideration for the particular application to determine its reliability for that application. The initial selection of the coupling is related to the torque capability of the coupling in relation to operating stress levels. Startup and shutdown torque, momentary overloads and shaft resonance need to be evaluated to determine the maximum torque requirements. It is important to check the overall system design to be sure that the peak torque capacity of the power source being absorbed by the system does not exceed the normal torque capacity of the coupling.

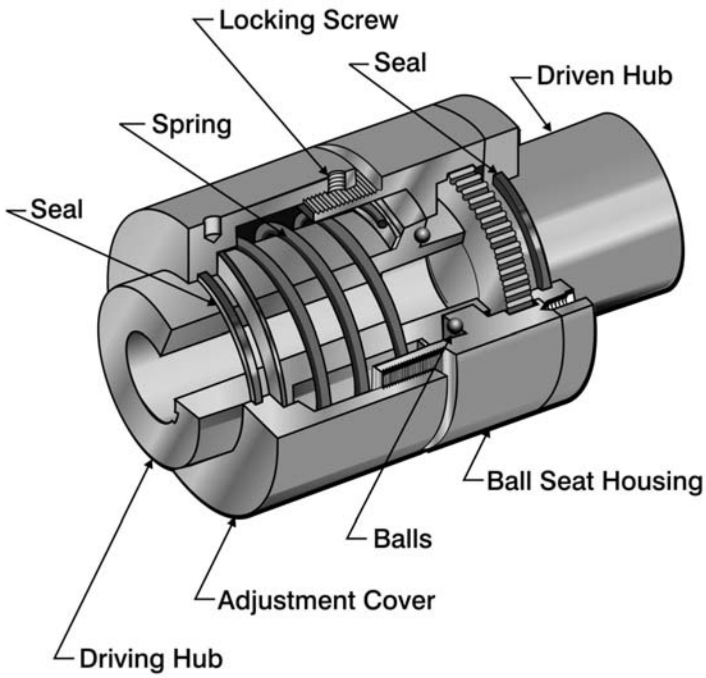


Figure 17.1 Typical mechanical Coupling

Misalignment is an important consideration in determining the failure rate of the total system containing a coupling. The maximum allowable misalignment is a function of the coupling material, the percentage of torque capacity being used, and the transmitted vibration. Misalignment may cause radial forces to be exerted on the driver and driven components. Excessive radial forces can cause stresses on bearings, seals and other parts to the point of premature failure.

A coupling may be rigid or flexible; rigid couplings being used in applications where misalignment is not a factor or where flexibility is not required. Thus, flexible couplings are more common. Flexible couplings are designed to accommodate various types of load conditions including rotation reversals, shock loading, load vibration, and misalignment.

17.1.1 Rigid Shaft Couplings

Rigid couplings are used when shafts have good collinear alignment. Although simple in design, rigid couplings are usually restricted to relatively low speed applications where good shaft alignment and minimum shaft flexibility can be expected. Various types of rigid shaft couplings include the following:

(1) The clamp/compression type coupling relies on the clamping force developed from the fasteners to connect the two shafts. Torsional forces are normally transmitted via shaft keys.

(2) The sleeve type coupling is generally a single piece housing that transmits torque via shaft keys or tapered bushings. Axial positioning of the coupling is maintained by retaining rings or threaded shaft collars.

(3) The flange coupling mates two coupling halves together in a plane that is perpendicular to the shaft centerline. Torque is normally transmitted between the shafts via the bolted fasteners in the flange.

17.1.2 Flexible Shaft Couplings

In many mechanical drive systems potential misalignment of a drive shaft, system vibration or variable drive torque requires a flexible drive. Flexible couplings are used to connect collinear shafts subject to one or more kinds of misalignment, while reducing the effect of shock and impact loads that may be transferred between shafts. Flexible couplings may employ material flexibility, mechanical flexibility or both types of flexibility.

Material flexibility refers to those flexible couplings containing resilient components that can accommodate shaft misalignment, as well as dampen shock and impact loads. This type of coupling possesses torsional flexibility, often acting as a "detuning" device by altering the vibration properties of the connected system. The flexible insert type coupling transmits torque through an oil resistant rubber spider assembled between two pairs of axially overlapping rigid jaws.

Mechanical flexibility refers to flexible couplings employing rigid parts that transmit torque without backlash or angular play (other than that due to manufacturing tolerances and wear). These types of couplings are generally incapable of dampening the transmittal of shock and impact loads. The bellows type coupling is used in applications involving large amounts of shaft misalignment, combined with low radial loading. The disc-type coupling can accommodate a smaller amount of angular misalignment than the bellows type, but by adding additional metallic "disks", radial load (torsional) carrying capacity can be greatly increased.

Due to the varying ranges of performance, caution must be taken in determining performance characteristics for the operating conditions to be encountered. Diaphragm couplings are rated for high speed operation, but accept only a small amount of misalignment. Although elastomeric and gear type couplings can accommodate a fair amount of angular misalignment, in general, the gear coupling is rated to accept heavier duty (horsepower). The reliability analysis needs to consider the fact that gear couplings require periodic maintenance in the replenishment of lubrication and seals.

17.2 FAILURE MODES OF COUPLINGS

Table 17-2 lists various failure modes encountered when using rigid or flexible couplings. A thorough reliability analysis can result in the elimination of many of these failure modes by properly selecting the correct type, size and torque rating of the coupling for the intended operational environment.

Table 17-2. Failure Modes of Couplings
(Reference 66)

FAILURE MODE	FAILURE CAUSE
Worn flexing element or shaft bushings	- Excessive shaft misalignment
Ruptured elastomeric flexing element, sheared hub pins or teeth	- Torsional shock overload
Fatigue of flexing element, hub pins, or discs	- Excessive starts and stops - Torsional vibration
Shaft bearing failure	- Lubricant failure - Excessive shaft misalignment - Operational temperature extremes
Loose hubs on shaft	- Torsional shock overload - High peak-to-peak torsional overload
Worn gear teeth	- Lubrication failure - High peak-to-peak torsional overload
High pitched hammering or clacking noise	- Excessive shaft misalignment - Loose hubs or bolt connections - Lubricant failure
Swollen, distorted, or cracked elastomeric flexing member, severe hub corrosion	- Chemical attack - Excessive heat

The connection of the coupling to the shafts can be keyed or keyless. Also, the fit of the coupling to the shafts can be categorized as interference or clearance. An interference fit, also known as a press fit, is achieved by friction after the coupling and the shaft are joined. With the shaft ground slightly oversize and the hole in the coupling undersized, when the two are joined, each is deformed upon being compressed so that they are locked together by friction. Thus a keyless coupling arrangement will always have an interference fit.

The design principle of the clearance fit is that the torque is transmitted through the key minimizing any sliding of the coupling on the shaft. Keyed clearance fit couplings are most frequently used on lower-power applications. Use of a clearance fit depends on the torque to be transmitted and operating speed. If the torque or misalignment is excessive, the coupling hub may toggle and become loose, causing material fretting. The interference fit with a keyed shaft is commonly used to axially locate the coupling hub and resist forces associated with unbalance and misalignment.

Coupling reliability is affected by the method used to mount the coupling hub on the shafts of connected equipment. Although clearance fit connections work satisfactorily for certain types of machinery, this practice should be avoided on any critical piece of equipment, because the reliability of the system can be affected. The most common cause of failure in this instance would be fretting of the coupling bore and of the shaft, and rolling of the key within the keyway due to looseness in the connection.

Initial alignment of the components connected by the shafts is one of the most critical factors affecting coupling performance and reliability. This is true regardless of the type of coupling employed. Flexible couplings are basically in-line devices which are intended to compensate for small amounts of shaft misalignment caused by bearing wear, foundation settling, thermal growth, etc. The more attention paid to initial alignment, the larger the reserve margin that will exist for accomplishing the intended purpose of the coupling. There are definite advantages to be gained from aligning equipment to more precise values than those recommended by the manufacturer. The primary advantage is that the reserve margin for accepting misalignment during the life of the equipment is increased accordingly.

Another factor to be considered, and one which is most important to satisfactory performance, is adherence to the manufacturer's bolt torque recommendations. Loose bolts can induce fretting corrosion, as well as hammering and pounding which will eventually destroy the bolts and coupling discs. Chapter 16 provides the procedures to be used in evaluating threaded fasteners for reliability.

Proper equipment maintenance is another factor affecting the operating life of couplings. For example, failure rate data of gear couplings indicates that at least 75% of all coupling failures are due to lack of lubrication. A well aligned gear type coupling requires periodic replenishment of the lubricant due to heat, oxidation, etc. There are other failure modes that occur with the accumulation of operating time. Specifically, shaft misalignment can develop after many cycles of operation as a result of thermal growth caused by differences between component material thermal expansion coefficients. At elevated operating temperatures, shaft centerlines may "grow" to be farther apart than at room temperature. In situations where a system will normally operate at elevated temperatures, the zero misalignment condition should be set at these elevated temperatures. In addition, total operational scenarios must be considered. If the system has a substantial cool-down or warm-up period of operation, then consideration must be given as to whether or not operations can be sustained during these periods of thermal adjustment and changes in misalignment.

Other failure modes are caused by vibration. Excessive vibration can act to bring about material fatigue, fastener loosening, or stress corrosion cracking. After extended operation, component wear can open clearances and augment vibration amplitudes. Increased vibration can act to worsen shaft misalignment. Another failure mode is caused by bearing wear. Lack of lubrication, contamination of the bearings, and wear can deteriorate performance of a bearing over a period of time. Bearing failure can increase vibration, which can in turn, adds to shaft operational misalignment.

17.3 FAILURE RATE MODEL FOR MECHANICAL COUPLING

An example of the type of data that coupling manufacturer's develop when testing their coupling designs is the endurance limit as shown in Figure 17.3. As shown by the curve as the stress level is reduced, cycle life is increased and below some stress level the curve becomes flat and the cycles to failure becomes very large. The data is normally design specific with speed and load limitations. The endurance limit in cycles to failure can be converted to a base failure rate as follows:

$$\lambda_{CP} = 1 / N \tag{17-1}$$

Where: λ_{CP} = Failure rate of coupling, failures/million cycles

N = Number of cycles to failure

In the event that manufacturer's data is not readily available, a base failure rate for a coupling can be derived from the sum of its component parts. A typical gear coupling unit is shown in Figure 17.4. In this example of a gear coupling unit, the failure rate is given by:

$$\lambda_{CP} = (\lambda_{CP,B} \bullet C_{SF}) + \lambda_{GR} + \lambda_{SE} + \lambda_{H} \tag{17-2}$$

Where: λ_{CP} = Failure rate of coupling, failures/million cycles

C_{SF} = Service factor multiplying factor (See Table 17-3)

λ_{GR} = Failure rate of gears, failures/million cycles (See Chapter 8)

λ_{SE} = Failure rate of seals, failures/million cycles (See Chapter 3)

λ_H = Failure rate of coupling housing including hubs, failures/million cycles (See Section 17.6)

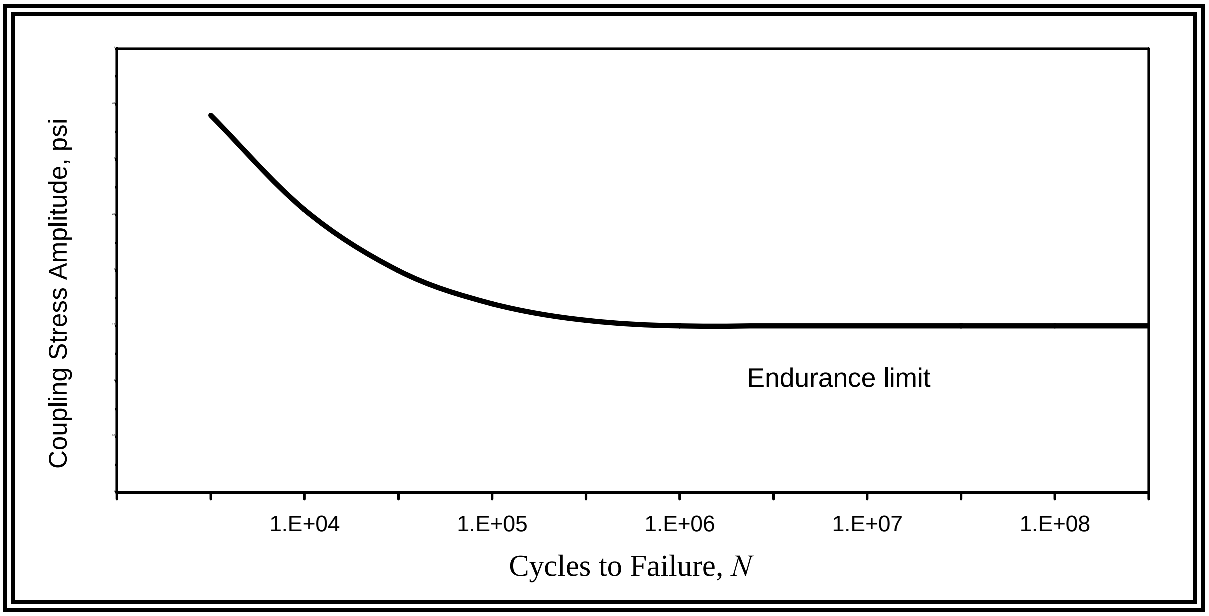


Figure 17.3 Stress as a Function of Cycles to Failure for a Typical Coupling

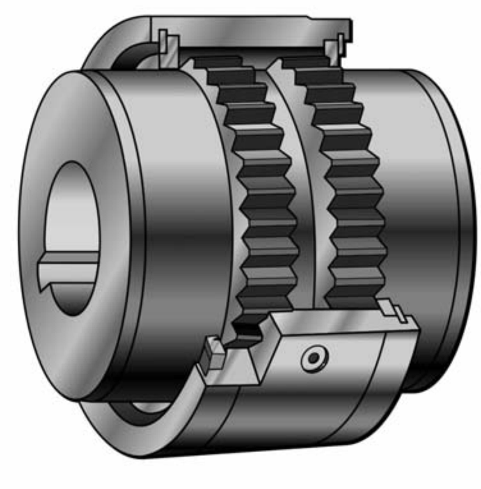


Figure 17.4 Gear Coupling (Reference 62)

A typical failure rate (λ_{CP}) of a coupling from data sources is 5.0 failures per million cycles. However, the failure rate of a coupling depends more on application than design or the manufacturing process. Rapid wear of the flexible member is typical of excessive shaft misalignment. Torn rubber insert, sheared rings or ruptured elastomer caused by high torsion impact loads or vibration are other failure considerations.

The torque capacity of a coupling is defined as its ability to transmit a required torque load. However, because of other factors, such as backlash, a coupling is selected with rated torque capacity many times greater than needed. Couplings are frequently specified in horsepower capacity (a function of torque and speed) at various speeds. Table 17.3 below contains some service factors to modify the expected failure rate of a coupling for the expected operating environment.

Table 17-3 Service Factors for a Mechanical Coupling

Driven Machinery	Normal Torque Characteristic	High or Non-uniform Torque
Uniform	1.1	1.2
Light shock	1.2	1.3
Medium shock	1.3	1.4
Heavy shock	1.4	1.5

17.4 UNIVERSAL JOINT

A universal joint is a joint in a rigid rod that allows the rod to 'bend' in any direction. It consists of a pair of hinges located close together, oriented at 90 degrees relative to each other connected by a cross shaft. Figure 17.5 shows a typical universal joint.

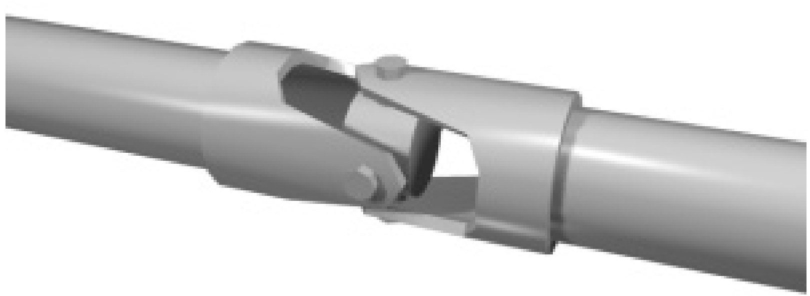


Figure 17.5 Typical Universal Joint

Universal joints are commonly used in shafts that transmit rotary motion. In evaluating a universal joint for reliability it is important to consider not only the driver and load or the size of the shaft. Potential misalignment and vibration, operating range, maximum torque and operating speed must also be analyzed.

17.5 FAILURE RATE MODEL FOR UNIVERSAL JOINT

The failure rate of a universal joint can be presented as a sum of the failure rates of its individual component parts:

$$\lambda_{UJ} = \lambda_{BE} + \lambda_{SE} + \lambda_{H} + \lambda_{F} \qquad \textbf{(17-3)}$$

Where:
λ_{UJ} = Failure rate of universal joint, failures/million cycles
λ_{BE} = Failure rate of bearings, failures/million cycles (See Chapter 7)
λ_{SE} = Failure rate of seals, failures/million cycles (See Chapter 3)
λ_{H} = Failure rate of coupling housing, failures/million cycles (See Section 17.6)
λ_{F} = Failure rate of coupling fasteners, failures/million cycles (See Chapter 16)

The life expectancy of a joint is a function of the application requirements such as torque, speed, and joint angle, as well as other factors. Therefore, the basic load-speed-life-stress relationships applicable to rolling element bearings are useful in life computations for universal joints employing similar rolling elements. These relationships have been established in the bearing chapter of this handbook.

17.6 FAILURE RATE OF THE COUPLING HOUSING

The housing for a mechanical coupling is a very reliable component. Defined as λ_H, the housing failure rate will have a greater effect on the reliability of the coupling unit from the standpoint of how it affects other less reliable components. For instance, the housing may have an average life expectancy of 10 years compared to the life of a seal or bearing of only one or two years. However, the type of housing used in the coupling assembly can have a large effect on the lifetime of the bearings and seals. This is due to differing loads placed on the shaft by the lubricant. The failure rate of the coupling housing (λ_H) itself can be estimated at 0.001 failures/million cycles.

17.7 REFERENCES

In addition to specific references cited throughout Chapter 17, other references included below are recommended in support of performing a reliability analysis of mechanical couplings.

19. Hindhede, U., et al, Machine Design Fundamentals, John Wiley & Sons, NY, 1983.

57. Deutschman, A.D., et al, Machine Design; Theory and Practice, MacMillan Publishing Co, NY, 1975

58. Parmley, R.O., Mechanical Components Handbook, McGraw-Hill Book Co., NY, 1985.

62. Baumeister, T, et al, Mark's Standard Handbook for Mechanical Engineers

63. Thomas Couplings Applications Manual

64. Bolam, J.R., "Coupling Alignment: The Reverse Indicator Method Simplified", P/PM Technology, July/Aug 90

65. Dvorak, P., "Sorting Out Flexible Couplings", Machine Design, 11 Aug 88

66. Robertson, R., and Smith, B., "Why Flexible Couplings Fail", Plant Engineering and Maintenance, Jun 89

67. Universal Joint & Driveshaft Design Manual, Series No. 7, Society of Automotive Engineers, Inc, Warrendale, Pa.

71. Dennis N. Pratt, "Investigation of Spline Coupling Wear", Report No. SY-51R-87, December 1987, Naval Air Warfare Center, Patuxent River, Maryland

72. Randall F. Barron, "Engineering Model for Mechanical Wear", Report No. CMLD-CR-09-88, June 1988, Louisiana Tech University

CHAPTER 18

SLIDER CRANK MECHANISMS

18.0 TABLE OF CONTENTS

18.1 INTRODUCTION

The slider crank mechanism is usually not thought of as an independent mechanical component but rather as an integral part of a more complex piece of equipment such as the piston rod and related components of an internal combustion engine. Figure 18.1 shows a typical slider crank mechanism. The normal function of this particular device is the conversion of rotational force into a linear force or vice versa.

The typical slider crank mechanism includes bearings, rods, linkages, seals and a sliding surface such as a cylinder wall. Wear of these parts becomes the primary failure mechanism, the failure modes and effects being dependent upon the application. The geometry of the design plays an important part of the reliability analysis since the mechanical advantage and the wear pattern are greatly influenced by the positioning of parts.

18.2 FAILURE MODES OF SLIDER CRANK MECHANISMS

The more predominant failure modes of a slider crank mechanism can be readily identified with frictional action on like or dissimilar materials. The component parts of a slider crank mechanism are subject to wear in varying degrees and the normal approach to reliability analysis is to establish the expected life of the individual parts in the projected operating environment.

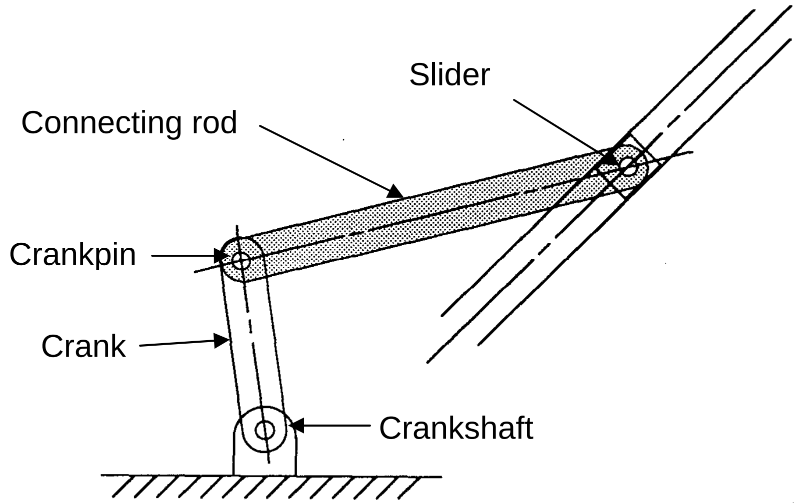


Figure 18.1 Typical Slider Crank Mechanism

Slider crank wear will manifest itself in several ways to cause the degradation of the slider crank to the point of failure. This threshold of failure must be defined in terms of jamming friction, side movement, limit of travel, alignment of parts, etc. Some of the failure modes to be considered are included in Table 18-1.

Table 18-1. Typical Failure Modes of Slider Crank Mechanisms

FAILURE MODE	FAILURE CAUSE
Linkage does not move in intended direction	Bearing parts separate causing jam
Restricted linkage travel	Excess load combined with bearing wear Mechanical seal leakage
Broken linkage	Fatigue of linkage member
Linkage alignment out of tolerance	Bearing deformed
Poor linkage response	Loss of lubricant film thickness Mechanical seal leakage

In the slider crank mechanism shown in Figure 18.1 bearings will be located at each of the rotating parts. Bearing wear will usually be influenced by the lubrication film thickness that is maintained, the side load on the bearing, the contamination level, and corrosion. Chapter 7 presents an approach to evaluating bearing life for these considerations.

A slider crank mechanism will normally contain a mechanical seal that provides a barrier between the moving or rotating surfaces and prevents the trapped fluid from migrating into undesired areas. For example, in the case of an engine cylinder the rings on the piston prevent the combustion gases from going into the lubricant and also prevent loss of energy due to combustion gas by-pass. The O-ring and flat seals are designed to prevent the lubricant from contaminating other part areas and also, the loss of lubricant. Seals and gaskets are described in Chapter 3 and the models contained therein should be used for the analysis of seal and gasket reliability.

Crankshafts and connecting rods within the slider crank mechanism are subject to fatigue and may crack although this failure mode is rare. Chapter 20 contains the procedures for evaluating these components.

18.3 MODEL DEVELOPMENT

The failure rate model for the slider crank mechanism can be expressed by the following equation:

$$\lambda_{SC} = \lambda_{BE} + \lambda_{RD} + \lambda_{SE} + \lambda_{RI} + \lambda_{SM} \tag{18-1}$$

Where: λ_{SC} = Total failure rate for slider crank, failures/million operations

λ_{BE} = Failure rate for bearings, failures/million operations (See Section 18.3.1 and Chapter 7, Section 7.5)

λ_{RD} = Failure rate for rods/shafts, failures/million operations (See Section 18.3.2 and Chapter 20, Section 20.4)

λ_{SE} = Failure rate for seals/gaskets, failures/million operations (See Section 18.3.3 and Chapter 3, Section 3.2)

λ_{RI} = Failure rate for rings/dynamic seals, failures/million operations (See Section 18.3.4 and Chapter 3, Section 3.3)

λ_{SM} = Failure rate for slider mechanism, failures/million operations (See Section 18.3.5 and Chapter 9, Section 9.3)

Failure rates are determined for the individual parts comprising the slider crank mechanism. Then, a total failure rate of the slider crank mechanism is determined using Equation (18-1).

18.3.1 Bearings

One of the predominant failure modes of a slider crank mechanism is caused by a malfunctioning bearing surface. Both roller and sliding bearings can be included in a slider crank design. Failure rate equations for roller/ball bearings are included in Chapter 7.

Typical sliding bearings are shown in Figure 18.2. The sliding bearing is usually comprised of three elements including a fixed member, moving member, and the lubricant film separating the fixed and moving members. An example sliding bearing is shown in Figure 18.3. The sliding bearing may be characterized as a shaft rotating within a sleeve. Sliding bearings can be classified by material, load direction, lubrication method, and configuration. Sliding bearings are well suited for large loads encountered in slider crank mechanisms. Although sliding bearings may have less running friction than rolling bearings, their starting friction is much higher. Rolling bearings are also easier to lubricate during service life. Sliding bearings are well suited to slider crank mechanisms in low speed applications where shock and vibration may occur.

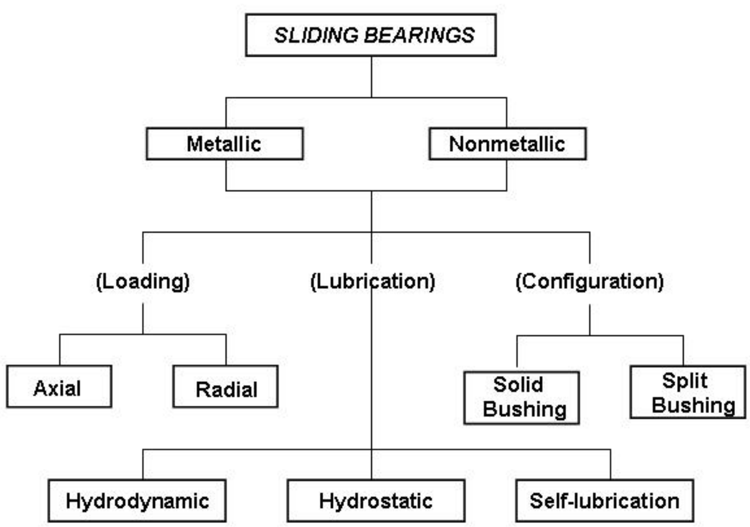


Figure 18.2 Sliding Bearing Classifications
(Reference 19)

In a hydrostatic bearing the pressure is always present at the predetermined level supplied by an external pump which forces lubricant into the system. Hydrodynamic sliding bearings are characterized by the load being carried by a lubrication film being generated by shaft rotation. Hydrodynamic bearings are more prone to initial wear than hydrostatic bearings because lubrication does not occur until there is rotation of the shaft. The friction at start-up is largely due to a direct contact between the journal and sleeve. The life of the bearing then is limited due to wear at start-up and stopping.

The reliability of the sliding crank mechanism will, of course, be affected by the lubricant being used. All liquids provide lubrication but some do better jobs in particular applications. Dry lubricants, for example, will adhere very well to the bearing surfaces but tend to wear quite rapidly and their capacity to minimize friction is only fair. The concept of lubricant viscosity is illustrated in Figure 18.3.

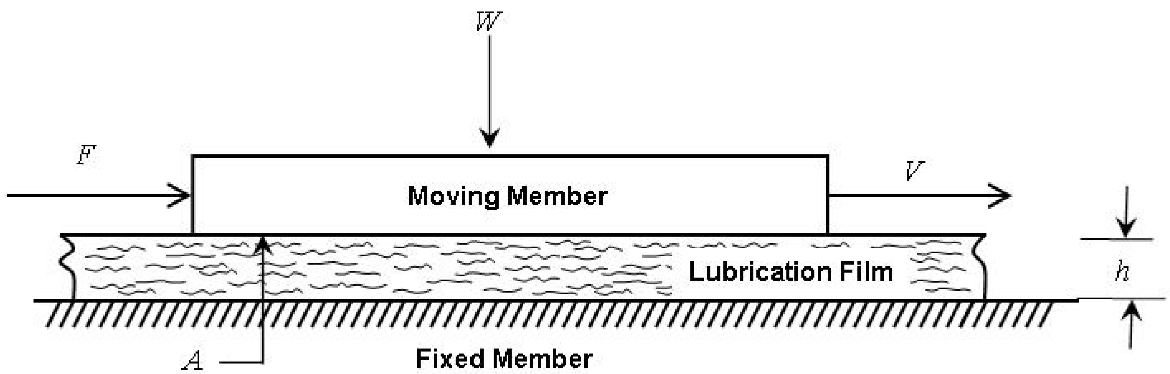


Figure 18.3 Sliding Bearing Lubricant Viscosity
(Reference 19)

In this figure, a film of lubricant adheres to the stationary plate and supports a moving plate. In order to move the upper plate to the right at a constant velocity V, it is necessary to exert some constant force F. Thus a shear stress is applied at the wetted surface of the moving plate equal to (Reference 19):

$$\tau = \frac{F}{A} \quad \textbf{(18-2)}$$

Where: τ = Shear stress, lbs/in^2

F = Applied Force, lb

A = Area of plate surface in contact with lubricant, in^2

The rate of shearing strain R' is defined by the following equation:

$$R' = \frac{V}{h} \tag{18-3}$$

Where: V = Velocity of moving member

h = Lubricant film thickness

The ratio of shearing stress to rate of shearing strain is called the dynamic viscosity, ν (Reference 19):

$$\nu = \frac{\tau}{R'} = \frac{F\,h}{A\,V} \tag{18-4}$$

Three types of lubrication occur in practice. They differ in the degree to which the lubrication is carrying the load. A full film lubrication physically separates the shaft and bearing surfaces by a relatively thick lubricant film of about 15 microns. This film prevents any metal-to-metal contact at the operating conditions. The coefficient of friction will be low, usually not above about 0.005. Full film operation implies minimum power losses and maximum life expectancy of the parts.

Complete boundary lubrication means that the bearing and shaft surfaces are being rubbed together with only a very thin lubricant film adhering to each surface and preventing direct contact. The coefficient of friction is high, in the range of 0.1.

Mixed film lubrication means that there is both boundary and full film lubrication. Part of the load is carried by small pools of self-pressurized lubricant. Other areas of the surfaces are rubbing with only a thin film of lubricant separating the peaks. A typical friction coefficient for this type of lubrication is 0.02.

Figure 18.4 shows the coefficient of friction plotted against a bearing characteristic number $\nu n/P$. The three operating variables in this bearing characteristic number are the lubricant's viscosity, shaft speed, and unit bearing load:

$$\alpha = \frac{\nu\,n}{P} \tag{18-5}$$

Where α = Bearing characteristic number

ν = Lubrication viscosity, lb-min/in^2

n = Shaft speed, revolutions/sec

P = Unit bearing load, lbs/in^2

The unit bearing load is defined as the ratio of the bearing's load to its projected area:

$$P = \frac{W}{D\,L} \tag{18-6}$$

Where: W = Bearing load, lb

D = Bearing diameter, in

L = Bearing length, in

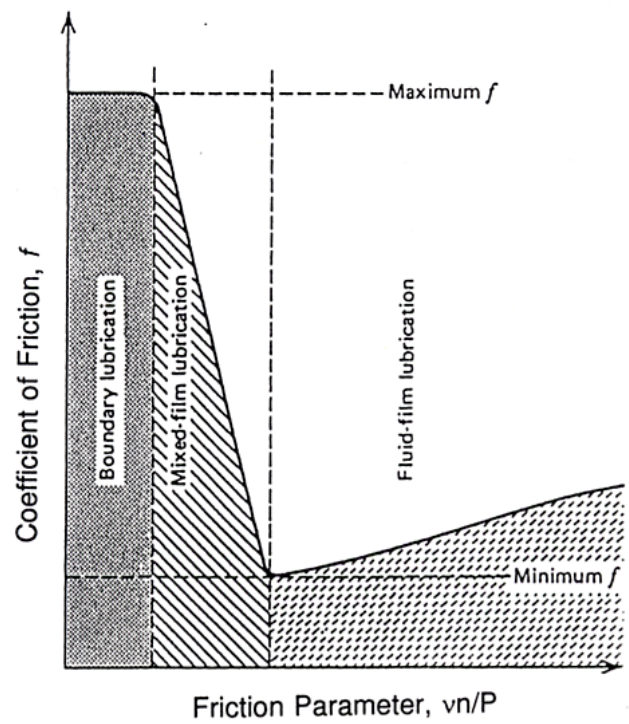


Figure 18.4 Variation of Friction Coefficient with the $\nu n/P$ factor
(Reference 19)

This bearing characteristic number provides a method of determining any potential problem with lubricant film. Any low viscosity, low shaft speed, or high unit bearing load implies a low value for $\nu n/P$. Conversely, the higher $\nu n/P$, the easier it is to establish a full-load-supporting film.

As shown in Figure 18.4, for the largest values of $\nu n/P$, there is fluid-film, or hydrodynamic lubrication regime. In this regime of operation the coefficient of friction attains a minimum of about 0.001. Operating with a slightly greater $\nu n/P$ value will assure an adequately thick film and a margin of safety with a somewhat greater power loss. The lowest values of $\nu n/P$ correspond to the type of complete boundary lubrication. The friction coefficient remains constant throughout this lubrication regime; its actual value depends on the character of the surfaces and the lubricant. The mid-regime is that of mixed-film lubrication. In this regime a decrease in $\nu n/P$ is accompanied by a sharp increase in friction coefficient.

Many bearings utilized in mechanical devices are considered light service and can operate successfully in the mixed-film or boundary lubrication regimes. Typical office equipment and appliances with latching mechanisms contain self-lubricated bearings. Manufacturers of bearings for light service usually base bearing selection on the *PV* factor, the product of unit bearing load P and rubbing velocity V. This factor indicates what bearing temperature will be reached and what rate of wear can be expected. Temperature rise and wear rate are maintained within reasonable limits by controlling the *PV* factor.

The unit bearing load P, already defined by Equation (18-6) is the ratio of the bearing's load to its projected area. The rubbing velocity V must be calculated differently for oscillating shaft motion than for continuous rotation. For continuous rotation the rubbing speed is (Reference 19):

$$V = 5\pi D n \tag{18-7}$$

Where: V = Rubbing velocity, ft/min

D = Bearing diameter, in

n = Shaft speed, revolutions/sec

However, if the shaft is oscillating relative to the bearing, the design value for V is based on the average rubbing speed.

$$V = \frac{\pi D \theta f}{72} \tag{18-8}$$

Where: θ = Total angle traveled per cycle, degrees

f = Frequency of oscillation, cycles/sec

The use of sleeve or journal bearings in severe service requires a full bearing or a thick lubrication film to support the load. While an external pump may be used to supply a lubricant under pressure to the bearing's feed hole, within the bearing itself it is the shaft that acts as a pump and pumps the oil adhering to it into the wedge-shaped oil film that supports the load. With the shaft stationary, the shaft simply rests on the bottom of the bearing. But at start-up the shaft begins to roll up the bearing wall. As it climbs, it also begins to pump oil between itself and the bearing. As this oil is pumped, the shaft lifts off the bearing surface and moves in the direction of rotation. At operating speed, the shaft has developed a wedge-shaped film between itself and the bearing that supports the shaft and its load. The radial displacement of the shaft's center from the bearing's center is the eccentricity. The pressure distribution in the oil film achieved depends on factors such as shaft speed, load, lubricant viscosity, bearing clearance, and length-to-diameter ratio.

The failure rate of a bearing is obtained using the equations contained in Chapter 7. The information obtained from this section can be used in conjunction with the procedures in Chapter 7 to estimate the failure rate of the bearing.

18.3.2 Rods/Shafts

The reliability of the rod or shaft is generally very high when compared to the components in the slider crank mechanism. Generally, the life expectancy will be at least three times that of the bearing. The possibility that the rod or shaft will fracture can best be determined using finite element techniques. The effects of the rod or shaft breakage on adjacent components are of greater importance than the reliability of the rod or shaft itself.

The failure rate of a rod, λ_{RD}, can be estimated at 0.001 failures/million hours for fracture. If deemed critical, procedures for determining the base failure rate and the multiplying factors for a rod or shaft can be found in Chapter 20.

18.3.3 Seals/Gaskets

The failure rate of a seal or gasket is determined by the ability of the seal to restrict the flow of fluid from one region to another for the intended life in the prescribed operating environment. Section 3.2 of Chapter 3 contains procedures and equations for determining λ_{SE} in Equation (18-1).

18.3.4 Rings, Dynamic Seals

The sealing surface of rings and other dynamic seals are perpendicular to the shaft with contact between primary and secondary rings to achieve a dynamic seal at various speeds, pressures and temperatures. The procedures contained in Chapter 3 can be used to determine λ_{RI} in equation (18-1), the failure rate for rings.

18.3.5 Slider Mechanism

The wear life of the sliding surface area depends on the correlation of wear of the two surfaces involved with the material strength and the stress imposed on the slider mechanism. From a time standpoint, wear of the two surfaces will occur in two phases. The first or constant wear phase is characterized by the shearing of asperities due to sliding action. During this period the wear rate is practically linear as a function of the number of mechanical cycles and the wear depth at the end of the constant wear phase is one half the original surface finish. During the second or severe wear phase, wear debris becomes trapped between the two sliding surfaces and gouging of the surfaces takes place. The wear rate begins to increase very rapidly and failure of the slider mechanism is imminent. Chapter 9, Section 9.3 contains the procedure and equations to determine the number of cycles at which point the slider crank mechanism is determined to have failed due to wear.

18.4 REFERENCES

In addition to specific references cited throughout Chapter 18, other references included below are recommended in support of performing a reliability analysis of a slider crank mechanism.

19. Hindhede, U., et al, Machine Design Fundamentals, John Wiley & Sons, NY, 1983.

57. Deutschman, A.D., et al, Machine Design; Theory and Practice, MacMillan Publishing Co, NY, 1975

58. Parmley, R.O., Mechanical Components Handbook, McGraw-Hill Book Co., NY, 1985.

62. Baumeister, T, et al, Mark's Standard Handbook for Mechanical Engineers

72. Randall F. Barron, "Engineering Model for Mechanical Wear", Report No. CMLD-CR-09-88, June 1988, Louisiana Tech University

CHAPTER 19

SENSORS AND TRANSDUCERS

19.0 TABLE OF CONTENTS

19.1 INTRODUCTION

A sensor is a hardware device that measures a physical quantity and produces a signal which can be read by an observer or by an instrument. For example, a thermocouple converts temperature to an output voltage, which can be read by a voltmeter. For accuracy, all sensors are calibrated against known standards. A transducer is a device that is actuated by power from one system and supplies power usually in another form to a second system. For example, a loudspeaker is a transducer that transforms electrical signals into sound energy. A standard definition for a sensor or transducer does not exist and the words sensor and transducer are used synonymously, specific names being given depending on the application. Also, suffixed derivatives ending in -meter such as accelerometer, flowmeter and tachometer are used. For convenience, the basic sensor will be used in this chapter to describe these units.

Analog sensors produce a continuous output signal or voltage, which is generally proportional to the quantity being measured. Physical quantities such as temperature, velocity, pressure and displacement are all analog quantities, as they tend to be continuous in nature. For example, the temperature of a liquid can be measured using a thermometer or thermocouple, which continuously responds to temperature changes as the liquid is heated up or cooled down. Digital sensors produce a discrete output signal or voltage that is a digital representation of the quantity being measured. Digital sensors produce a discrete (non-continuous) value, which may be outputted as a single bit or a combination of the bits to produce a single byte output.

One of the important newer applications of sensors to system reliability is prognostics and diagnostics. A prognostic/diagnostic system uses sensors to monitor operational and environmental conditions and translates any changes to component remaining life using the equations in this Handbook embedded in the prognostic/diagnostic processor. As preestablished threshold values are exceeded, the embedded reliability models calculate component overstress and cumulative damage and provide maintenance personnel of impending malfunctions and the next best maintenance action to be performed.

19.2 TYPES OF SENSORS

In almost every product for commercial and military application, the number of sensors and transducers continues to increase. The application of “smart sensors” with digital communication techniques and resulting improved accuracy and self-healing sensor networks has created an ever-increasing application of sensor technology. Table 19-1 is a partial listing of the applications for sensors and transducers.

Table 19-1. Typical Sensor Applications

Sensor Classification	Section Reference	Typical Sensor/Transducer
Thermal	19.2.1	Thermostat, thermister, thermocouple, thermopile
Force	19.2.2	Mechanical force, strain gauge, torque
Positional	19.2.3	Potentiometer, LVDT, rotary encoder
Fluid	19.2.4	Pressure, flow, viscometer
Optical	19.2.5	Photodiode, phototransistor, photodetector, infrared, fiberoptic
Motion	19.2.6	Displacement, velocity, acceleration, vibration, shock
Presence	19.2.7	Proximity
Environmental	19.2.8	Temperature, altitude, humidity, smoke

19.2.1 <u>Thermal Sensors</u>

Thermal sensors vary from simple ON/OFF thermostatic devices that control a domestic hot water system to highly sensitive semiconductor types that control complex processing facilities. Temperature sensors measure the amount of heat energy within an object and detect any physical change to that temperature. There are many different types of thermal sensors and all have different characteristics depending upon their actual application. Typical thermal sensors include the following:

- Resistance Temperature Detector: Same as a power resistor

- Thermistor: A thermistor is a passive resistive device that changes its physical resistance with temperature producing a measurable voltage depending on the current through the device. A thermistor is generally made from ceramic type semiconductor materials such as oxides of nickel, manganese or cobalt coated in glass. Thermistors are generally connected in series with a suitable biasing resistor to form a potential divider network and the choice of resistor gives a voltage output at some pre-determined temperature point or value.

- Thermocouple: A thermocouple converts thermal energy into electrical energy. It consists of two junctions of dissimilar metals, such as copper and constantan that are welded or crimped together. A thermocouple is created whenever two dissimilar metals touch and the contact point produces a small thermoelectric voltage as a function of temperature (Seebeck voltage). The thermocouple is a commonly used temperature sensor because of its simplicity, small size, ease of application and speed of response to changes. A thermopile is composed of thermocouples connected in series.

19.2.2 <u>Force Sensors</u>

Force sensors are used to obtain an accurate determination of pulling and/or pressing forces. The force sensor creates an electrical signal which corresponds to the force measurement to be used for further evaluation or process control. Force sensors are commonly used in automotive vehicle assemblies such as brakes, suspension units and air-bag systems.

A force sensor generally measures the applied force from the proportional deformation of a spring element; the larger the force, the more this element deforms. Many force sensors employ the piezoelectric principle exhibited by quartz. Under load, quartz crystals produce an electric charge proportional to the mechanical load applied; the higher the load, the higher the charge. Thus, in piezoelectric force sensors, quartz serves as both the spring element and the measurement transducer.

A strain gauge is a device used to measure the strain of an object. The strain gauge is the fundamental sensing element for many types of sensors, including pressure sensors, load cells, torque sensors and position sensors. The majority of strain gauges are foil types. They consist of a pattern of resistive foil which is mounted on a backing material and as the foil is subjected to stress, the resistance of the foil changes. This results in a signal output, related to the stress value.

Commonly, torque transducers use strain gauges applied to a rotating shaft or axle. The strain gauge bridge requires voltage and a torque transducer is required to receive the signal from the rotating shaft. This can be accomplished using slip rings, rotary transformers or wireless telemetry.

19.2.3 Positional Sensors

A positional sensor is one that permits a linear or angular position measurement. It can either be an absolute positional sensor or a relative one (displacement sensor).

A conventional potentiometer is an analog device that can be used as a positional sensor to vary, or control, the amount of current that flows through an electronic circuit. The potentiometer can be either angular (rotational) or linear (slider type) in its movement providing an electrical signal output that has a proportional relationship between the actual wiper position and its resistance change. A digital potentiometer consists of resistor arrays, switches, logic gates, multiplexers, and data converters. Digital potentiometers have excellent setability, better resolution, and lower noise levels than conventional potentiometers. They are more stable over time and their resistance drifts minimally. They are more reliable and exhibit a lower temperature coefficient of resistance than analog potentiometers.

The Linear Variable Differential Transformer (LVDT) is an inductive type positional device that works on the same principle as an AC transformer. It is a very accurate device for measuring linear distances with an output proportional to the position of its moveable core. It basically consists of three coils wound on a hollow tube, one forming the primary coil and the other two coils forming identical secondary coils 180^o out of phase from either side of the primary coil. An armature connected to the object being measured slides up and down inside the tube. The AC excitation voltage applied to the primary winding induces an EMF signal into the two secondary windings providing an amplitude that is a linear function of core displacement.

The LVDT must be calibrated for the particular application. Any mechanical change in the application such as a part that has been moved or replaced requires recalibration. It is important that the calibration of an LVDT take place in contact with the part it is to measure.

Rotary Encoders resemble potentiometers but are non-contact optical devices used for converting the angular position of a rotating shaft into an analog or digital data code. Rotary encoders utilize light from an LED or infrared light source that is passed through a rotating high-resolution encoded disk containing the required code patterns.

Photodetectors scan the disk as it rotates and an electronic circuit processes the information into a digital form as a stream of binary output pulses that are fed to counters or controllers which determine the actual angular position of the rotating shaft.

19.2.4 Fluid Sensors

A fluid pressure sensor detects a pressure difference between a detecting pressure and reference pressure and converts the difference into an electric signal. Pressure sensors are used to measure pressures of gases or liquids. Pressure measurements typically are made as absolute, gauge, or differential measurements. Absolute pressure sensors measure a pressure relative to a vacuum, gauge sensors measure a pressure relative to atmospheric pressure, and differential sensors measure a pressure difference between two inputs.

Generally, a pressure sensor for sensing a gas or liquid pressure has a diaphragm that acts as a pressure sensing element and is configured such that deflection (pressure deformation) of a diaphragm under a fluid pressure applied through a pressure port is converted to an electrical signal, thereby enabling the fluid pressure to be measured.

A fluid flow sensor is a device for sensing the rate of fluid flow. Typically a flow sensor is the sensing element used in a flowmeter or datalogging device to record the flow of fluids. There are various kinds of flow sensors and flowmeters including some that have a vane that is pushed by the fluid and can drive a rotary potentiometer or similar device. Other flow sensors are based on sensors which measure the transfer of heat caused by the moving medium.

The flow sensor can normally measure velocity, flow rate or totalized flow. Flow sensors are sometimes related to sensors called velocimeters that measure speed of fluids flowing through them. The flow sensor technology can be based on such things as light, heat, electromagnetic properties, ultrasonic and many other technologies in a wide spectrum. A flow sensor can work by direct measurement or inferential measurement. Several types of flow sensors are non-mechanical and normally work by the inferential method. As is true for all sensors, accuracy of a fluid sensor measurement requires a calibration applicable to the sensor function.

A viscometer is an instrument used to measure the viscosity of a fluid. Viscometers only measure one flow condition. In general, either the fluid remains stationary and an object moves through it, or the object is stationary and the fluid moves past it. The drag caused by relative motion of the fluid and a surface is a measure of the viscosity.

19.2.5 Optical Sensors

Optical sensors are passive devices that convert radiant light energy into an electrical signal output. The most common type of photoconductive device is the photoresistor which changes its electrical resistance in response to changes in the light intensity. Photojunction devices are PN-Junction light sensors or detectors made from silicon semiconductors that can detect both visible light and infrared light levels. This class of photoelectric light sensors includes the photodiode and the phototransistor. Phototransistor light sensors operate the same as photodiodes except that they can provide current gain and are much more sensitive than the photodiode.

Photodetectors, also known as proximity sensors, are used to determine if a moving object enters the range of a sensor. The most commonly found photodetector is the "electric eye". This type of sensor works by projecting a beam of light from a transmitter to a receiver across a specific distance. As long as the beam of light maintains a connection with the receiver, the circuit remains closed. If an object passes through the beam of light, the continuity of the circuit is lost, and the circuit opens. An example of this type of sensor is a garage door opener safety sensor that will halt the closing of the door if an object breaks the beam.

Active infrared sensors project a beam of light in the infrared spectrum and receive the returning reflection from objects in the sensor's range. Infrared sensors can be used as proximity sensors, such as in automatic doors. Passive infrared sensors are used to measure the radiation of heat within its range. Examples of passive infrared sensors include "heat-seeking" missile guidance systems and infrared thermography systems.

Fiberoptic sensors can be used to measure a wide range of physical phenomena, depending on the configuration of the sensor. Optical fibers can be coated with materials that respond to changes in strain, temperature, or humidity. Optical gratings can be etched into the fiber at specific intervals to reflect specific frequencies of light. As the fiber is strained, the distances between the gratings change, allowing the physical strain to be measured.

19.2.6 Motion Sensors

Motion sensors are designed to measure the rate of change of position, location, or displacement of an object that is occurring. If the position of an object is changing as a function of time the first derivative gives the speed of the object and if the speed of the object is also changing, then the first derivative of the speed gives the acceleration.

Displacement sensors measure the distance an object moves and they can also be used to measure object height and width. Optical and magnetic displacement sensors are used to detect the amount of a linear displacement. Linear and angular displacement sensors are used for high-precision machining and measuring, for manufacturing and testing components with very tight dimensional tolerances. Magnetic displacement sensors consist of one or more magnets producing an induction field and typically measure linear or rotational displacement providing an output proportional to absolute linear or rotary position displacement of the elements.

Velocity sensors measure the linear velocity of an object using either contact or non-contact techniques. In cable-extension linear velocity sensors, the moving object is attached to a cable, which is typically connected to a potentiometer. As the object moves, the potentiometer's resistance value changes. Magnetic induction sensors are non-contact linear velocity sensors that use an induced current from a magnetic field to measure the linear velocity. Microwave sensors use microwave technology to determine speed, whereas fiber optic sensors use fiber optics or laser technology to determine speed. Piezoelectric linear velocity sensors utilize a piezoelectric material that is compressed generating a charge that is measured by a charge amplifier. Often, piezoelectric sensors are used in vibration velocity measurement applications.

The simplest acceleration type measures mass motion by attaching the spring mass to the wiper arm of a potentiometer. In this manner, the mass position is conveyed as a changing resistance. The Linear Variable Differential Transformer (LVDT) takes advantage of the natural linear displacement to measure mass displacement. In these instruments, the LVDT core itself is the seismic mass. Displacements of the core are converted directly into a linearly proportional voltage. These accelerometers generally have a natural frequency less than 80 Hz and are commonly used for steady-state and low-frequency vibration.

Vibration sensors are sensors for measuring, displaying and analyzing linear velocity, displacement and proximity, or acceleration. They can be used on a stand-alone basis, or in conjunction with a data acquisition system. Vibration sensors are available in many forms. They can be raw sensing elements, packaged transducers, or as a sensor system or instrument, incorporating features such as local or remote display, and data recording. The primary elements of importance in shock measurements are that the device has a natural frequency that is greater than 1 kHz and a range typically greater than 500 g. The primary accelerometer that can satisfy these requirements is the piezoelectric type.

19.2.7 <u>Presence Sensors</u>

A proximity sensor is a sensor able to detect the presence of nearby objects without any physical contact. A proximity sensor often emits an electromagnetic or electrostatic field, or a beam of electromagnetic radiation (infrared, for instance), and looks for changes in the field or return signal. The object being sensed is often referred to as the proximity sensor's target. Different proximity sensor targets demand different sensors. For example, a capacitive or photoelectric sensor might be suitable for a plastic target while an inductive proximity sensor requires a metal target.

Proximity sensors can have a high reliability and long functional life because of the absence of mechanical parts and lack of physical contact between sensor and the sensed object. The maximum distance that this sensor can detect is defined as its "nominal range". Some sensors have adjustments of the nominal range or means to report a graduated detection distance. Proximity sensors are also used in machine vibration monitoring to measure the variation in distance between a shaft and its support bearing. This is common in large steam turbines, compressors, and motors that use sleeve-type bearings.

A proximity sensor is divided in two halves and if the two halves move away from each other, then a signal is activated. An example application of a proximity sensor is for window security. When the window opens an alarm is activated.

19.2.8 Environmental Sensors

Environmental sensors include those used to measure temperature, humidity, wind speed, barometric pressure, etc. They are often connected in a network. Temperature sensors are discussed in Section 19.2.1. An analog humidity sensor gauges the humidity of the air relatively using a capacitor-based system. The sensor is made out of a film usually made of either glass or ceramics. The insulator material which absorbs the water is made out of a polymer which takes in and releases water based on the relative humidity of the given area. This changes the level of charge in the capacitor of the electrical circuit.

A digital humidity sensor works via two micro sensors that are calibrated to the relative humidity of the given area. These are then converted into the digital format with an analog to digital conversion.

Most smoke detectors work either by optical detection (photoelectric) or by physical process (ionization). They may also use both detection methods to increase sensitivity to smoke. In a photoelectric device, smoke can block a light beam reducing the light reaching a photocell used to set off an alarm. Ionization detectors have an ionization chamber and a source of ionizing radiation. Released radiation particles ionize the oxygen and nitrogen atoms in the chamber. These positively-charged atoms are attracted to the negative plate in the chamber and the electrons are attracted to the positive plate, generating a small, continuous electric current. When smoke enters the ionization chamber, the smoke particles attach to the ions and neutralize them, so they do not reach the plate. The drop in current between the plates triggers the alarm.

19.3 FAILURE MODES OF SENSORS

Typical failure modes for individual sensors are shown in Table 19-2. These example failure modes must be modified for the particular application. A design analysis of equipment reliability (See Chapter 24) performed on the sensor assembly will help focus on the modifying factors to be used on published failure rates. Failure modes need to be addressed for the specific sensor network being analyzed. Typical subsystems of a sensor network include:

- Data processing subsystem – provide and process the measured quantities for further real-time use by the human and communication interfaces and/or at the electrical output subsystem.
- Sensor subsystem – convert the physical or chemical quantities into conditioned electrical signals for use in the data processing unit.
- Human interface – read the information on a display and enter request for data
- Communications interface – connect the instrument to external systems
- Electrical output subsystem – convert the digital information into one or more analog signals
- Power supply unit – supply power to the other smart sensor subsystems.

Reliability is defined differently in real time applications than for delay tolerant applications. In real time applications reliability is dependent upon the arrival time of sensor data and a delay in data processing may be a sensitive failure mode. In a delay tolerant configuration data loss is the factor determining reliability. There are other significant differences in failure modes when evaluating a sensor network for reliability. Additional failure modes to be considered include information overload, timing errors, node interference, power depletion and digitization errors. Table 19-3 lists some typical failure modes of sensor network applications.

Timing in conventional sensors is not normally an issue since the sensor continuously measures the parameter and presents a resulting 4-20mA signal. In a sensor network architecture based on interrupts, the sensor might receive many interrupts of high priority and might subsequently fail to attend to the tasks of lower priority. Another potential problem with a sensor network is data integrity. With conventional sensors, parameters are set using fixed components such as resistors, and are therefore highly insensitive to external influences. In the case of a sensor network, the sensor may lose the parameter settings stored in memory during a power failure. Failure modes of the communications link in sensor networks include information overload of the communications interface causing events happening simultaneously to be communicated sequentially, thus confusing the operator when deciding what caused a disturbance.

Table 19-2. Typical Failure Modes of Individual Sensors

FAILURE MODES	FAILURE CAUSES	FAILURE EFFECT
Incorrect signal from sensor element	- Reduced signal level - Impedance mismatch - A/D conversion error	Potential processing error
Loss of signal from sensor element	- Chip failure - Corroded sensor	Loss of signal to processor
Complete loss of signal in transmission line	- Broken wire - Fiber optic, RF interruption	Loss of signal to processor
Signal error in transmission line	- Power line interference - Contaminants in fluid system	Potential processing error
Incorrect signal in computation device	- Error in algorithm	Processing error
Power supply loss of voltage	- Power supply malfunction	Loss of signal to processor
Improper response to information recipient	- Incorrect interpretation - System malfunction - Error in algorithm	Potential system malfunction
Calibration error	- Software error - Error in algorithm	Potential system malfunction
Battery energy depletion	- Battery malfunction	Loss of signal to processor

Table 19-3. Typical Failure Modes of Sensor Networks

FAILURE MODES	FAILURE CAUSES	FAILURE EFFECT
Loss of voltage to all components	Power Supply malfunction	Loss of signal history, calibration and settings information
Loss of voltage to sensor subsystem/ADC	Wire failure	Loss or incorrect measurement of input signal
Loss of voltage to processor	Wire failure	Loss of signal history, calibration and settings information
Loss of voltage to transceiver	Wire failure	Loss of or incorrect output or settings
Loss of voltage to human interface	Wire failure	No display of data, setting of parameters not possible
Defect in D/A conversion	Hardware error	Incorrect output
Incorrect signal from sensor	Defect in A/D conversion, software error	Incorrect output
Loss of signal from sensor	Sensor failure	Node failure
Timing error from sensor	Software error	Incorrect output
Conditioning of data from sensor is incorrect	Software error	Incorrect output
Corrupted signals to data processing or external system	Information overload, software error	Loss of or incorrect output
Error in signal processing	Software error	Incorrect output
Error in calibration	Failure of communications interface, software error	Incorrect output
Error in data storage	Hardware/software error	Incorrect output, incorrect calibration
Incorrect display of data	Human error	Incorrect action by controller
Incorrect setting of parameters	Human/software error	Incorrect output
Loss of clock, wrong or changing frequency	Hardware error	Incorrect output

19.4 SENSOR FAILURE RATE MODEL DEVELOPMENT

A typical sensor arrangement is shown in Figure 19.1. Here a differential transformer is used in conjunction with a spring to measure weight. The sensing element is the spring which is compressed when the weight is applied against the sensor changing the position of the central core of the differential transformer. The transformer converts the position of the core into an electrical signal with the magnitude proportional to the applied weight.

The failure rate of the sensor in Figure 19.1 is essentially determined by the failure rates of the compression spring and the transformer. The failure rate of the spring can be determined by using the procedures in Chapter 4. MIL-HDBK-217 contains procedures for determining the failure rate of the transformer (Reference 28).

Other typical sensors for mechanical flow sensing are shown in Figure 19.2. In Figure 19.2(a), a spring loaded hinged vane is pushed open as fluid flows through the sensor; and in Figure 19.2(b), the fluid pushes against a spring restrained plug. Again the failure rates of the individual parts are determined using procedures in the applicable chapters of this handbook for cylindrical wear, springs, etc.

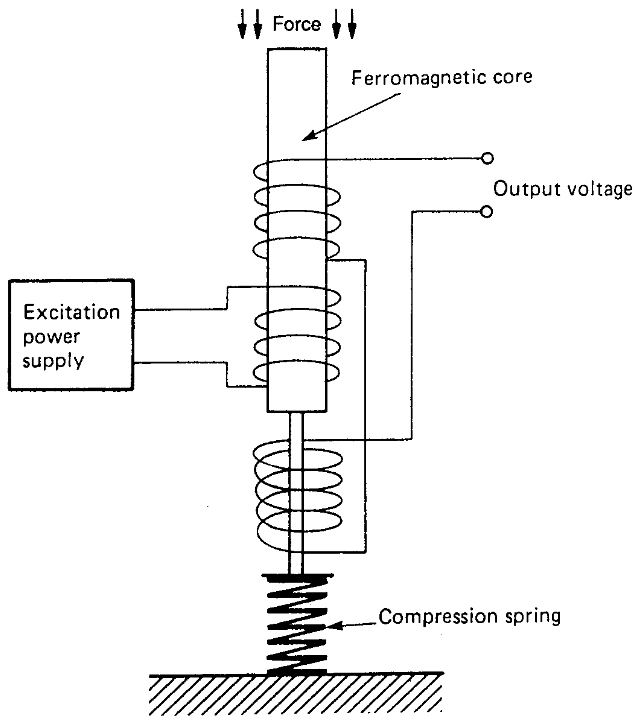


Figure 19.1 Typical Positioning Sensor Unit

It is important to realize that determining the failure rate of a sensor is more involved than combining failure rates of the individual parts. As shown in Figure 19.3, sensors are used as part of a larger sensing/monitoring/detection system and the reliability of the total system is sensitive to its defined requirements. Accuracy, repeatability, sensitivity, drift, and other performance requirements are an important part of the failure rate estimate. Redundant sensors are often used to back up other sensors that give no warning of imminent failure and which may fail catastrophically. Self test features and scheduled test inputs to check for sensor drift can avoid system failure. Response to sensor data by the operator and/or other parts of the system must also be considered in estimating the failure rate of a sensor system.

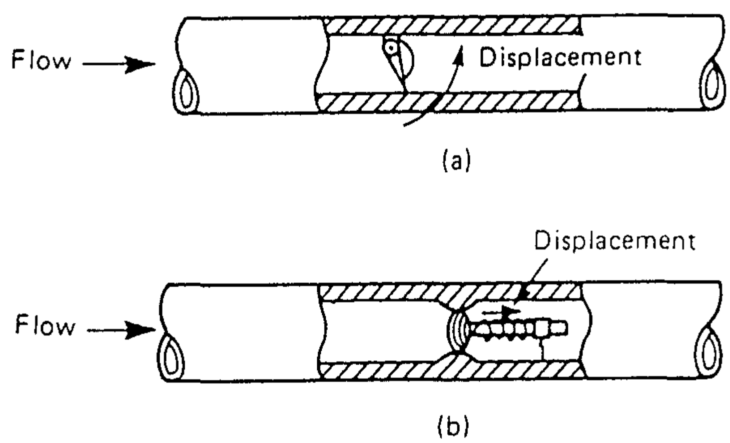


Figure 19.2 Typical Flow Sensor Units

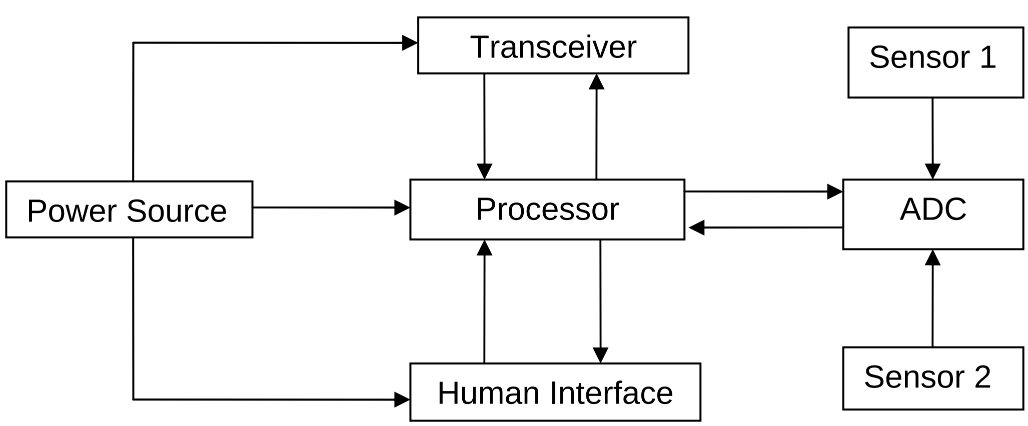


Figure 19.3 Typical Sensor Network

Though sensors provide vital environmental information and feedback to microcontrollers, they also increase processor workload because their signals must be linearized, filtered, temperature compensated, scaled, and converted. New integrated circuits, however, can convert temperature, pressure, position movement, and acceleration signals to computer compatible equivalents. One-chip functions include a/d converters, multiplexers, current-source, and a microcomputer that controls the converter and processor sensor data. Sensor-specific information such as calculated values and calibration factors can be stored in a local memory unit. The fact that each and every sensor system is different requires that the sensor be separated into its individual parts for analysis using the appropriate chapters of the Handbook to estimate each failure rate.

In redundant sensor configurations a possible design method is the use of diversity. Diversity is the use of different technologies to perform a required function. This practice may involve such an approach as sensors from various manufacturers with different sensor technology and software. Diversity has the advantage that it can reduce the probability that two or more sensors fail simultaneously, although this effect is limited by the fact that two or more sensors may still contain the same faults or experience a common cause failure mode. A common cause failure mode is the failure of multiple items from a single cause which is common to all of them such as a power supply failure. A disadvantage of diversity can be the increased complexity of maintenance, which in itself can lead to a higher probability of failure of the sensor configuration. Whether the use of diversity is advisable depends on the design of the sensor network and the application.

A reliability model of the overall sensor network can be written as follows:

$$\lambda_{TD} = \lambda_{TD,B} + \lambda_S + \lambda_T + \lambda_C + \lambda_P \ldots\ldots + \lambda_X \qquad \textbf{(19-1)}$$

Where: λ_{TD} = Failure rate of sensor, failures/million hours

$\lambda_{TD,B}$ = Base failure rate of sensor, failures/million hours

λ_S = Failure rate of sensing element (See Table 19-4)

λ_T = Failure rate of the transmission line (See Reference 28)

λ_C = Failure rate of the computational device (See Reference 28)

λ_P = Failure rate of the power source (See Reference 28)

λ_X = Failure rate of other components comprising the sensor or sensor network

The unique designs of sensors and special applications in terms of sensitivity, drift, etc. make it difficult to assign an accurate base failure rate for a sensor. Typical published failure rates for sensors are shown in Table 19-4 for commercial applications. Various publications contain historical failure rates for sensors. The data does not usually provide the design information, the environment/operating conditions under which the sensor was used, the fluid medium in contact or any of the characteristics listed in the section on failure modes. The failure rate can vary to a great extent mainly because the failure rate of a sensor is mainly dependent upon manufacturing practices and environmental conditions. Use of these failure rates must be adjusted based on experience with similar applications and operating environment. MIL-HDBK-217 may be helpful in applying multiplying factors for the intended operating environment. It is recommended that the considerations provided in Table 19-5 be applied in estimating sensor and sensor network failure rates.

Table 19-4. Typical Failure Rates for Sensing Elements
(Reference 88 and various data sources)

Sensor Transducer Classification	Sensor Transducer	Technology	Failure Rate λ_S
Thermal	RTD	Resistive	1.50
	Thermistor	Semiconductor	3.50
	Thermocouple *	Thermoelectric	5.00
	Infrared	Emissivity	7.50
	Thermostat	Bimetallic	20.00
Mechanical Force/Pressure	Force transducer	Semiconductor	4.00
	Strain gauge	Resistive	7.50
	Strain gauge	Semiconductor	23.00
	Torque transducer	Magnetic	4.50
	Load cell	Strain gauge	23.00
Fluid	Fluid pressure sensor	Piezoelectric	13.00
	Fluid flow sensor	Resistive	14.30
	Fluid level sensor	Capacitance	10.70
	Air flow sensor	Vane meter	8.50
Optical	Photodiode	Semiconductor	0.16
	Phototransistor	Semiconductor	0.65
	Photodetector	Semiconductor	0.03
	Infrared sensor	Semiconductor	7.50
	Solar cell	Photovoltaic	1.00
Position	Analog potentiometer	Resistive	1.50
	Digital potentiometer	Semiconductor	2.50
	LVDT	Transformer	7.50
	Rotary encoder	Optical	8.00
	Rotary encoder	Magnetic	5.00

Table 19-4. Typical Failure Rates for Sensing Elements (continued)
(Reference 88 and various data sources)

Sensor Transducer Classification	Sensor Transducer	Technology	Failure Rate λ_S
Motion	Linear displacement	Resistive	2.50
	Rotary displacement	Resistive	3.50
	Displacement sensor	Capacitive	10.70
	Displacement sensor	Inductive	8.50
	Velocity sensor	Hall effect	2.50
	Velocity sensor	Electro-magnetic	3.50
	Velocity sensor	Rotational	2.50
	Optical - Photosensor	Semiconductor	0.16
	Optical - Infrared	Semiconductor	7.50
Presence, Proximity	Proximity sensor	Electro-magnetic	15.38
	Infrared	Semiconductor	7.50
Environmental	Altitude sensor	Piezoelectric	3.39
	Humidity sensor	Resistive	20.44
	Humidity sensor	Capacitive	20.44
	Accelerometer	Piezoresistive	15.00
	Accelerometer	Piezoelectric	13.00
	Strain Gauge	Semiconductor	23.00
	Smoke detection	Ionization	8.00
	Smoke detection	Photoelectric	6.50

* A thermocouple exhibits a very small output voltage and will normally require an amplifier unless inputted to a measuring device. Failure rate does not include amplifier, controller or power supply.

Table 19-5. Sensor and Sensor Network Failure Rate Considerations

1. Battery life, battery backup, uninterruptible power supply
2. Sensor connectors, connections
3. Individual sensor failures/redundant sensor connection
4. Average reading of multiple sensors
5. Sensor accuracy requirements
6. Short term and long term drift
7. Sensor positioning in target area
8. Data routing efficiency
9. Number of sensor nodes in the network
10. Data generation rate of sensor nodes
11. Energy consumption rate
12. Response timing requirements
13. Corrosive environment, protective coating
14. Operating temperature
15. Shock and vibration requirements
16. All potential failure modes listed in Table 19-2 and/or Table 19-3

19.5 REFERENCES

In addition to specific references cited throughout Chapter 19, other references included below are recommended in support of performing a reliability analysis of a sensor, transducer or sensor network.

19. Hindhede, U., et al, Machine Design Fundamentals, John Wiley & Sons, NY, 1983.

28. MIL-HDBK-217, "Reliability Prediction of Electronic Equipment"

88. Reliability Analysis Center, "Nonelectronic Parts Reliability Data", NPRD-95

113. Meine van der Meulen, "On the Use of Smart Sensors, Common Cause Failure and the Need for Diversity", Center for Software Reliability, City University, London, 2008

114. Zhanyang Zhang and Miriam R. Tausner, "Using Markov Process to Model Wireless Sensor Network Life Expectancy with QoS Constraints, Department of Computer Science, College of Staten Island/City University of New York, 2008

CHAPTER 20

SHAFTS

20.0 TABLE OF CONTENTS

20.1 INTRODUCTION

A shaft is a rotating member, usually of a circular cross section, used to transmit power or motion. It provides the axis of rotation, or oscillation of other parts such as gears, flywheels and pulleys and controls the geometry of their motion. One of the functions of a shaft is transmitting torque from one element to another on the shaft. Power is transmitted by means of rotational motion and developed torque from one end of the shaft to the other. An example would be a compressor as a load driven by a motor with the compressor shaft connected to the motor shaft via a shaft coupling. Typical shaft loads include such components as fluid drivers, gears, splines and pulleys. Reliability issues for shafts include material strength, rotational speed, shear stress, temperature, and the operating environment.

Flexible shafts provide the capability of transmitting motion around corners. A flexible shaft is made by winding several layers of wire around a central core. Flexible shafts are rated by specifying the torque corresponding to various radii of curvature of casing. A wire rope may be used as a fexible shaft. The wire rope section in Chapter 23 provides some additional information on evaluating a wire rope for reliability.

A spindle is a short shaft. Failure modes to be considered in evaluating a spindle for reliability and the equations and procedures for determining the failure rate for a given application are the same as a typical shaft.

Shafts are usually designed for an infinite life. Beyond 1 million stress cycles the fatigue strength of most steels becomes constant. This value of material endurance strength divided by the shaft load provides a factor of safety for the shaft material. Materials used for the manufacture of shafts less than 3.5 inches in diameter are usually cold-rolled or cold-drawn steel. Larger shafts are usually hot-rolled and then machined to remove the decarburized surface. Shafts larger than 6 inches in diameter are usually forged and machined to size.

After the failure rate of the shaft is determined from the procedures in this chapter, it can be added to other failure rates of the component such as the pump, motor or compressor for a total component failure rate.

20.2 FAILURE MODES

Most shafts are subject to combined rotational and bending stresses. These in turn may be static, variable and dynamic. Sometimes flexible couplings are used as shaft connections to reduce bending loads. Shafts are generally designed for an infinite life and will therefore have a very low failure rate. The effect on other components is often a more serious question. For example, in the case of a shaft supporting a gear, excessive bending of the shaft will cause misalignment of gear teeth and uneven distribution of tooth load, in turn, causing excessive tooth wear.

Keyways cut into the shaft reduce its load carrying ability, particularly when impact loads or stress reversals are involved. When a completely reversing stress is applied to a stress concentration point, growth of a crack is accelerated at this point that can lead to shaft fracture. Keys and keyways should be used only to locate components on the shaft avoiding stress concentrations on the key. Changes in shaft radii contribute to stress concentrations. Large radii between shaft sections minimize stress concentrations at the shoulders.

Shafts in rotation can become very unstable at certain speeds and damaging vibrations and deflections can occur. The result is not only failure of the shaft but also damage to components of the machine of which the shaft is a part. The rotational speed at which this phenomenon occurs is called the critical speed of the shaft.

Typical failure modes of a shaft are listed in Table 20-1.

Table 20-1 Shaft Failure Modes

Failure Mode	Failure Cause	Failure Effect
Bent shaft	- Excessive load, torque - Impact loads - Bearing failure	- Assembly vibration - Damaged bearing, impeller, wear ring, mechanical seal, gear box
Excessive shaft deflection	- Dynamic loading on shaft - Reversing loads - Critical shaft speed exceeded - Unbalanced load	- Eventual shaft damage - Damaged bearing, impeller, wear ring, mechanical seal, gear box
Shaft misalignment	- Improper assembly - Worn bearings - Excessive load	- Assembly vibration - Damaged bearing, impeller, wear ring, mechanical seal, gear box
Damaged surface finish	- Corrosion - Contaminants - Manufacturing process - Thermal expansion at high temperatures	- Shaft bearing failure
Shaft fatigue / fracture	- Stress riser at fillet - Stress concentration at keyway - Shaft radii changes - Bending fatigue - Excessive velocity - High torque load	- Damaged bearing, impeller, wear ring, mechanical seal, gear box
Fretting corrosion	- Relative movement of tightly fitted parts	- Surface cracks, eventual shaft failure - Bearing, gear, coupling corrosion

20.3 MODEL DEVELOPMENT

Shafts are primarily designed on the basis of the torsional moment which they transmit. This torque can be calculated using the following expression:

$$T = \frac{3.96\,x\,10^{5}\,hp}{2\,\pi\,N} \tag{20-1}$$

where: T = torque, in-lbs

hp = transmitted horsepower

N = Shaft rpm

and the shear stress on the shaft can be calculated from the following:

$$S_S = \frac{16\,T}{\pi\,d^3} \tag{20-2}$$

where: d = shaft diameter, in

20.4 FAILURE RATE MODEL FOR SHAFTS

The reliability of the shaft itself is generally very high when compared to other components. Studies have shown (Reference 26) that the average failure rate for the shaft itself is about eight times less than mechanical seals and about three times less than that of the ball bearings. The possibility that the shaft itself will fracture, or become inoperable is very unlikely when compared to the more common component failure modes. Normally, it will be the seals or bearings of the component that will cause the initial problems. The effect of the shaft on reliability of other components is of greater importance than the reliability of the shaft itself.

The shaft reliability model is shown by the following equation:

$$\lambda_{SH} = \lambda_{SH,B} \bullet C_f \bullet C_T \bullet C_{DY} \bullet C_{SC} \tag{20-3}$$

Where: λ_{SH} = Shaft failure rate, failures/million cycles

$\lambda_{SH,B}$ = Shaft base failure rate, failures/million cycles (See Section 20.4.1)

C_f = Shaft surface finish multiplying factor (See Section 20.4.2)

C_T = Material temperature multiplying factor (See Section 20.4.3)

C_{DY} = Shaft displacement multiplying factor (See Section 20.4.4)

C_{SC} = Stress concentration factor for shaft discontinuities (See Section 20.4.5)

The above reliability equation for shafts assumes a constant rotational torque loading due to transmitted power and completely reversed bending loads on the shaft due to its rotation. In the case of large reciprocating loads such as those in piston engines and compressors, this assumption must be analyzed and the failure rate adjusted accordingly.

20.4.1 Base Failure Rate

In a typical fatigue test, the test specimen is stressed to some value and the number of stress fluctuations to fracture noted. The applied stress is then plotted against the number of cycles to failure. For many ferrous metals and thermosetting plastics the S-N (stress vs. cycles) curve approaches an asymptotic value at some stress level called the endurance or fatigue limit. Figure 20.1 shows an example of fatigue data for mild steel subjected to reversed stresses with an endurance limit of 27,000 lbs/in^2 (Curve A). For many materials there is no definite endurance limit and a value is determined at a selected number of cycles, usually 10^8 cycles (Curve B). The base failure rate, $\lambda_{SH,B}$, is estimated from the material endurance factor by the following relationship:

$$\lambda_{SH,B} = \frac{1}{N} \qquad \textbf{(20-4)}$$

Where: $\lambda_{SH,B}$ = Shaft base failure rate, failures/million cycles

N = Number of cycles to failure at application stress level, S_{ED}

S_{ED} = Material endurance limit, lbs/in^2

The endurance limit, S_{ED}, for some common steels and alloys is shown in Table 20-2.

The multiplying factors account for environmental conditions that vary from the normal operation. The base failure rate represents values that can be expected if all conditions during normal operation are those of the original design. The following discussion explains the various multiplying factors.

20.4.2 Shaft Surface Finish Multiplying Factor

C_f is the shaft surface finish factor that adjusts the base failure rate by an amount depending on the type of manufactured finish. If the design calls for a particular finish, then a variation from this finish during the manufacturing process will alter the reliability of the shaft. Table 20-3 shows the values and equations for the various finishes versus material tensile strength.

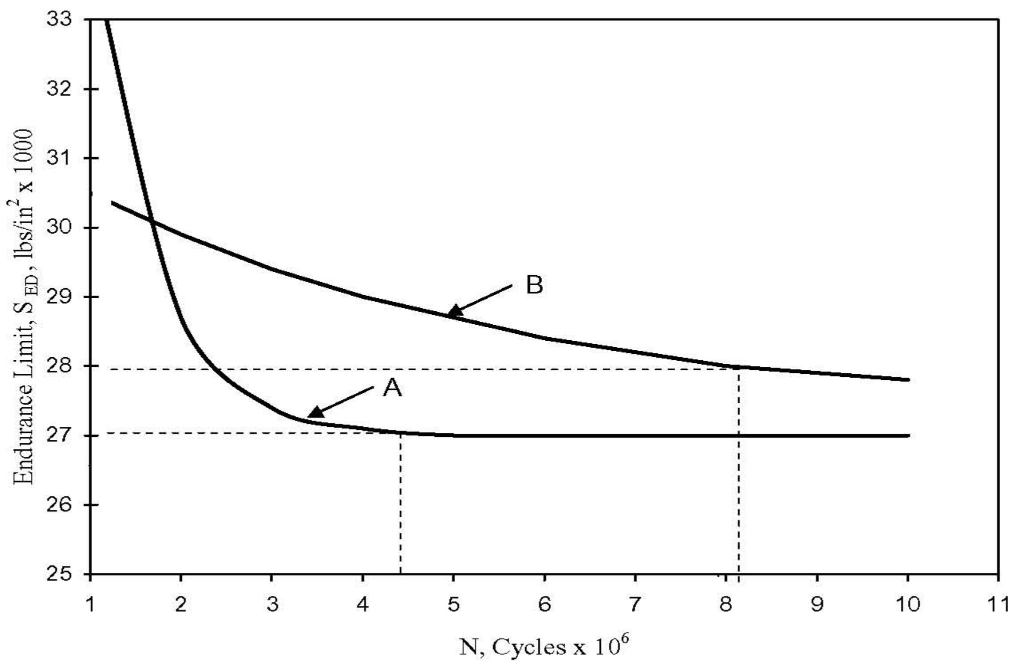


Figure 20.1 Typical S-N Curves and Endurance Limits

20.4.3 Material Temperature Multiplying Factor

Typical material fatigue data are acquired at 160 °F. To compensate for the decline in static and dynamic strengths, creep, and thermal expansion at high temperatures, the temperature factor, C_T, represented by Equation (20-5) is applicable for temperatures greater than 160 ^{0}F (Reference 19).

$$C_T = \frac{460 + T_{AT}}{620} \text{ for } T_{AT} > 160^O F \quad \textbf{(20-5)}$$

and: $$C_T = 1.0 \text{ for } T_{AT} \leq 160^O F$$

Where: T_{AT} = Operating temperature, ^oF

20.4.4 Shaft Displacement Multiplying Factor

The shaft displacement factor, C_{DY}, will vary with the amount of load the shaft will see. Shaft misalignment and excessive deflection have a significant influence on the reliability of shaft bearings and any seals mounted on the shaft. The estimate of shaft deflection depends on the weight of the shaft, its physical dimensions and any unbalance force caused by fluid flow. The basic equation for determining the shaft displacement multiplying factor is (References 123 and 129):

$$Y = \frac{F\,l^3}{3E\,I} \qquad \textbf{(20-7)}$$

Where: Y = Shaft deflection at load, in

F = Fluid radial unbalance force or load weight, lb

l = Shaft overhang from shaft bearing, in *

E = Modulus of elasticity of shaft material, lbs/in^2 (See Table 20-5)

I = Shaft moment of inertia ($\pi d^4/64$), in^4

d = Shaft diameter, in

* The length of the shaft from the center of the radial support bearing to the center of the impeller or other shaft load.

A typical shaft assembly is shown in Figure 20.2. Equation (20-8) can be used to establish the influence of shaft deflection on failure rate for this design. Similar equations will apply to the applicable shaft configurations.

$$C_{DY} = \frac{0.0043\,F}{Eb}\left[\frac{X^3}{I_X} + \frac{L^3}{I_L} + \frac{M^3}{I_M} + \frac{N^3}{I_N}\right] \qquad \textbf{(20-8)}$$

Where: b = Specified shaft deflection, in (See Table 20-6)

X,L,M,N = Length of shaft section, in

20.4.5 Stress Concentration Multiplying Factor

Most shaft failures originate at stress concentrations caused by discontinuities of shaft geometry and loads. The previous section contains those parameters affecting shaft loading. The total stress concentration factor is determined from the design of the fillet between shaft sections and the extent of any shaft grooves along each shaft section. The total shaft stress concentration factor is then determined as follows:

$$C_{SC} = C_{SC,R} + C_{SC.G} \tag{20-9}$$

where:

C_{SC} = Shaft stress concentration factor

$C_{SC,R}$ = Stress concentration factor due to transition between shaft sections

$C_{SC,G}$ = Stress concentration factor due to shaft grooves

A stress concentration factor for each shoulder radii can be found using the following equation:

$$C_{SC,R} = \left(\frac{0.3}{r/d}\right)^{0.2} \ x \ \left(\frac{D}{d}\right)^{(1-r/d)} \tag{20-10}$$

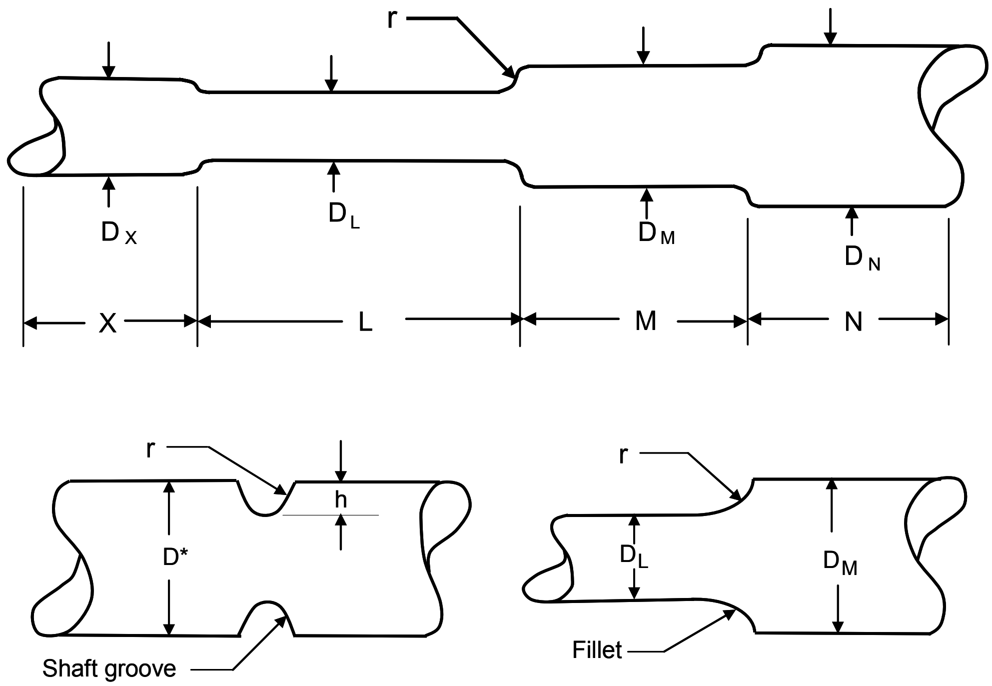
Where:

r = Radius of fillet, in

D = Initial shaft diameter, in (for example D_L in Figure 20.2)

d = Transitioned shaft diameter, in (for example D_M in Figure 20.2)

Table 20-4 provides typical stress concentration factors for shaft grooves, $C_{SC,G}$. If there are no grooves in the shaft, $C_{SC,G}$ will be equal to 1.0.



D* = Diameter of applicable shaft section, D_L, D_M, D_N, D_X
h = depth of shaft groove
r = radius of fillet or shaft groove

Figure 20.2 Typical Shaft Assembly

Table 20-2. Average Values of Endurance Limits
(Reference 39)

MATERIAL	ENDURANCE LIMIT S_{ED}
Steel, $\sigma_{T,ult} \leq$ 200 kpsi	0.50 $\sigma_{T,ult}$
Steel, $\sigma_{T,ult}$ > 200 kpsi	100 kpsi
Magnesium	0.35 $\sigma_{T,ult}$
Nonferrous Alloy	0.35 $\sigma_{T,ult}$
Aluminum Alloy (wrought)	0.40 $\sigma_{T,ult}$
Aluminum Alloy (cast)	0.30 $\sigma_{T,ult}$

Table 20-3. Shaft Surface Finish Factor

FINISH	C_f
Polished	1.0
Ground	0.89
Hot Rolled	$0.94 - 0.0046\, T_S + 8.37 \times 10^{-6} (T_S)^2$
Machined or Cold Drawn	$1.07 - 0.0051\, T_S + 2.21 \times 10^{-5} (T_S)^2 - 3.57 \times 10^{-8} (T_S)^3$
Forged	$0.75 - 4.06 \times 10^{-3}\, T_S + 7.58 \times 10^{-6} (T_S)^2$

Note: T_S = Tensile strength of material, kpsi

Table 20-4 Stress Concentration Factor $S_{SC,G}$ for Shaft Groves

h/D	h/r						
	0.1	**0.5**	**1.0**	**2.0**	**4.0**	**6.0**	**8.0**
0.05	1.10	1.45	1.60	2.00	2.05	---	---
0.10	1.00	1.27	1.40	1.70	2.00	2.25	---
0.20	1.00	1.10	1.20	1.31	1.60	1.75	2.00
0.30	1.00	1.10	1.10	1.20	1.35	1.48	1.55

Table 20-5 Shaft Material Strengths

Shaft Material	Tensile Strength (Ultimate) σ_u ksi	Endurance Strength σ_e ksi	σ_e / σ_u	Modulus of Elasticity E mpsi
Alloy steel	100 - 240	44 - 106	0.44	30
Stainless steel	80 - 230	24 - 69	0.30	29
High carbon steel	90-210	39 - 90	0.43	30
Cast steel, carbon	70 - 100	35 - 50	0.50	30
Low alloy cast steel	70 - 200	35 - 100	0.50	30
Cast aluminum	20 - 48	8 - 18	0.38	10.3
Wrought aluminum	22 - 83	8 - 29	0.35	10.0 – 10.6

Table 20-6 Allowable Shaft Bending

Application	Allowable Shaft Deflection b, inches *
Actuator	0.007
Compressor	0.025
Motor	0.010
Pump	0.007

* Note 1: Default value = 0.007 inches

Note 2: If the application requires or permits a different allowable shaft bending, b can be adjusted accordingly.

20.5 REFERENCES

In addition to specific references cited throughout Chapter 20, other references included below are recommended in support of performing a reliability analysis of a shaft assembly.

19. Hindhede, U., et al, “Machine Design Fundamentals”, John Wiley & Sons, NY, 1983

26. Igor J. Karassik et al, Pump Handbook, McGraw-Hill Book Company, New York (1986).

39. Shigley, J.E., Mischke, C.R. “Mechanical Engineering Design”, McGraw-Hill Book Company, NY, (1989)

123. Centrifugal Pump & Mechanical Seal Manual, William J. McNally, 2009

129. “Run Times”, August 2005, Dale B. Andrews, Lawrence Pumps, Inc.

131. “Tidal Current Turbine Reliability: Power Take-off Train Models and Evaluation, C. Iliev and D. Val, Third International Conference on Ocean Energy, October 2010

CHAPTER 21

BELT AND CHAIN DRIVES

21.0 TABLE OF CONTENTS

21.1 INTRODUCTION

There are a number of means to mechanically change speeds and transmit power in industrial applications. V-belts, synchronous belts, and chains are three methods of transmitting power between two shafts separated by a wide distance, and they are used over a wide range of speed ratios.

V-belt drives are a common means of transmitting horsepower and reducing speed. They are quiet and require very little maintenance. These drives transmit power through friction created by a wedging action of the belt in the sheave groove. V-belts work optimally with speed ratios of up to 6:1, but are available with drive ratios up to 10:1.

Synchronous belt drives offer positive engagement between mating teeth of a toothed belt and a toothed sprocket. No lubrication is required and they can operate at higher speeds than other positive engaging drives such as chain drives. Additionally, they offer less noise at slow speeds than chain drives.

Synchronous belts with a trapezoidal tooth profile are called timing belts. These belts are used in power transmission applications up to 50 horsepower. Another synchronous belt with a curvilinear tooth profile has a more rounded tooth and has come to be known as the high torque drive, or HTD. HTD sprockets can transmit up to 300 horsepower. With a modified curvilinear tooth profile, a higher tooth angle and slightly shallower tooth, synchronous belts can transmit up to 500 horsepower.

Applications requiring positioning, indexing, constant speeds, and no-slip are ideal for synchronous belt drives due to their positive tooth engagement. Additionally, wet environments, where chains can rust and V-belts might slip, are other ideal applications. The no-slip feature of synchronous belts leads to greater efficiency which reduces power consumption. However, alignment, noise and high shock loads are more critical with synchronous belt drives. Alignment needs to be held to within 1/4º to prevent rapid wear. High horsepower and high-speed synchronous belt drives can experience noise problems due to rapid tooth interaction. They are also more susceptible to damage from high shock loads. However, properly applied and installed, synchronous belt drives can last at least 12,000 hrs.

High torque, slow speed applications are ideal for chain drives. Chain drives are compact, economical, and easy to install. They are capable of operating in high temperature environments and provide no slip with no special tensioning. Additionally, they offer the same efficiency advantages as synchronous drives due to their no-slip feature.

Chains are also used in serpentine drive applications and specialty conveyors because a variety of chain attachments and types of chain are available. The versatility, availability, and compactness of chain drives allow them to be used in all industries. While synchronous belts and V-belts are offered only in fixed belt lengths, a chain can be set to any length, providing for unrestricted center distance. Improper lubrication is the primary cause for premature chain failure.

21.2 BELT DRIVES

A belt drive is a low cost means of transmitting rotary motion from one shaft to another. Belt drives are smooth running, operationally quiet, and resistant to start-up or momentary overloads. Recently improved reinforcing materials have made belt drives more practical where formally only chain drives would have been reliably employed. Belt drives are used when large distances between shafts make gears impractical or when operational speed is too high for chain drives. A belt drive is an assembly of belts and pulleys including a means of attaching sheaves or pulleys to their respective shafts to transmit power. Pulleys and belts can be used to increase or reduce speed or torque, or to transfer power from one shaft to another.

As shown in Figure 21.1, belts are either flat or exist in a V-shaped cross section running on grooved pulleys.

- Flat belts: Usually a composite construction with cord reinforcement, suited for high speeds and relatively low power.

- Round belts: Used in agricultural machinery drives and light duty or appliance drives such as vacuum cleaners. Round belts are similar to V-belts and they run in V sheaves.

- V-belts require less tension than do flat belts, because they have more surface area contacting the pulley and therefore more friction. V-belts are comprised of a load carrying cord tensile member located at the pitch line, embedded in a relatively soft matrix which is encased in a wear resistant cover. The wedging action of a V-belt in a pulley groove results in a drive which is more compact than a flat belt drive, but short center V-belt drives are sensitive to shock. The narrow (wedge) V-belt provides more tensile member support than the classical V-belt. They handle an equivalent load, but with narrower face width and smaller diameter.

- Timing or synchronous belts are a specific class of belts that contain toothed members similar to spur gears. Timing belts are positive rather than friction driven since they rely on gear-like teeth on the pulley and belt. The teeth on the belt mesh with the teeth on the pulley providing for a positive, non-slip rotational drive assembly. Because of the toothed nature of the components, specific reduction ratios can be obtained along with increased power torque ratings over non-toothed pulleys and belts.

There are several different belt designs. One is a fabric-wrapped construction. The wrap provides a protective envelope that is designed to prevent damage and prolong life. In a non-wrapped belt, none of the cross section is allocated to a wrap; thus, the total section consists of working tensile material. This allows a higher power rating for belts with non-wrapped construction.

V-belts are applied in a variety of applications but work best in applications greater than 500 RPM. They offer a great benefit in that they will slip upon overload, thus protecting other, more expensive equipment from load surges.

The service life of a properly designed V-drive is approximately 20,000 to 25,000 hours. This life can be dramatically less if the drive is not installed or applied properly. V-belt drives are limited to a maximum misalignment of 1/2º, and standard belts are limited to operating temperatures from -20ºF to 140ºF. Published belt ratings are often based on a temperature of 85ºF. In addition to ambient temperatures, excessive slip caused by worn sheave grooves, tight bends, and poor ventilation can lead to increased belt operating temperatures. Belt life will be drastically reduced above the baseline temperature as shown by the temperature multiplying factor curve in Figure 21.7.

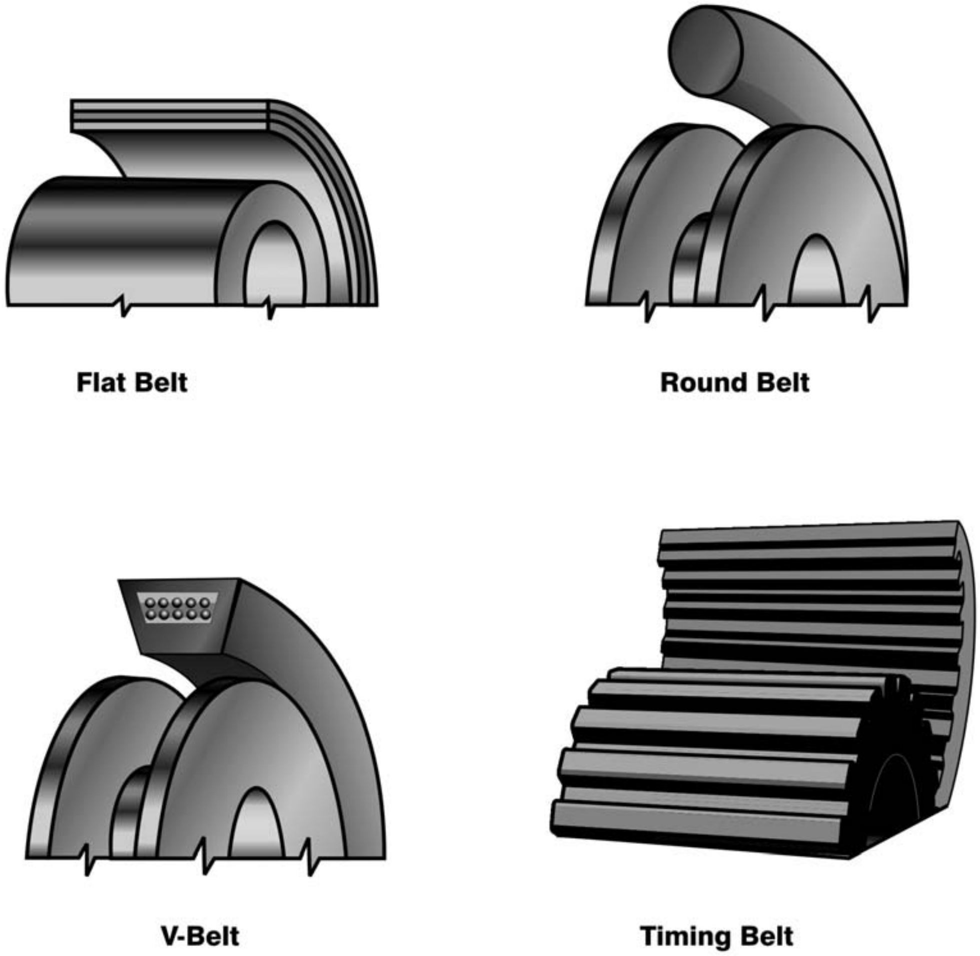


Figure 21.1 Belt Types

Reliable operation of a belt drive depends on substantial tension to create a friction that allows the belts to grip the pulleys allowing them to turn and keeping them from slipping as they turn. Because V-belts operate based on friction and mechanical advantage, proper tensioning of V-belts is the single most important factor necessary for long, satisfactory operation. Too little tension will result in slippage, causing rapid belt and sheave wear. Too much tension can result in excessive stress not only on the belts, but also bearings and shafts. For maximum life, the belt must operate within prescribed tension ranges and in environments compatible with material life.

21.3 FAILURE MODES OF BELT DRIVES

The most common failure mode of belts is a broken belt caused by improper tensioning. There is a tension at which the belt experiences optimum service life. Above this tension belt fatigue translates into a decrease in belt life; below this tension belt slip results in reduced belt life. Table 21-1 presents a summary of failure modes for belt drives.

Table 21-1. Typical Failure Modes of Belt Drives

FAILURE MODE	FAILURE CAUSE	FAILURE EFFECT
Improper operating tension	Installation error	Belt failure
Pulley/sheave misalignment	Installation error	Sidewall cracking and belt failure
Worn pulley/sheave	Incorrect tension	Belt slippage and rapid wear rate
Temperature extreme	Belt slippage, operating environment	Belt hardening and reduced life
Chemical contamination	Operating environment	Belt wear and eventual failure
Foreign objects in the belt drive assembly	Operating environment	Belt wear and eventual failure
Belt slip	Insufficient tension	Excessive heat and wear generated with reduced belt life
Belt fatigue	Excessive tension	Broken belt
Worn belt and pulley/sheave,	Large starting and stopping forces greater than 10% above operating conditions	Premature belt failure
Normal wear rate	Normal repetitive stressing	Eventual belt failure
Rapid belt deterioration	Heat build-up due to inadequate ventilation	Variation in drive ratio and reduced belt life
Improper belt drive operation	Loose pulley/sheave on shaft	Sidewall cracking and belt failure

Misalignment can be caused by non-parallel shafts, offset pulleys, or shafts or pulleys that have an angular skew at installation. Misalignment causes belts to experience sidewall stresses and uneven shock loads. As a result, the belt fails due to sidewall cracking or rollover inside the sheaves.

Belts running on worn pulleys are susceptible to slippage and excessive wear in localized sidewall regions causing a reduction in belt life. Belts run on worn sheaves can roll over in the sheaves, especially if there is any misalignment.

Belts are usually designed to operate within the temperature range of -20F to 140 F. Elevated temperatures cause hardening, increased dynamic stiffness and reduced ultimate elongation of the cushion compound, reducing the belts flexibility. The loss in flexibility results in increased stress on the cushion stock. Eventually these stresses become too high for the compound to handle and a cushion crack appears. Continued exposure to elevated temperatures will cause a continuous increase in stress level and continued cracking. Eventually, the cushion stock will not be able to support the cord line and the belt will fail to function.

21.4 RELIABILITY PREDICTION OF BELT DRIVES

21.4.1 Belt Misalignment

Parallel misalignment results from pulleys being mounted out of line from each other. Parallel misalignment is generally more of a concern with V-type belts than with synchronous belts because V-type belts run in grooves and are not as free to move sideways on the pulleys. Synchronous belts on the other hand are usually able to self align themselves as they run. However, if there is not enough clearance between the edge of the belt and the pulley flange, the belt can become pinched causing serious performance problems and early belt failure. Therefore, in order to maximize performance and reliability, synchronous drive assemblies should be aligned closely, a maximum allowable misalignment of ¼ degree. Parallel and angular misalignments are shown in Figure 21.2.

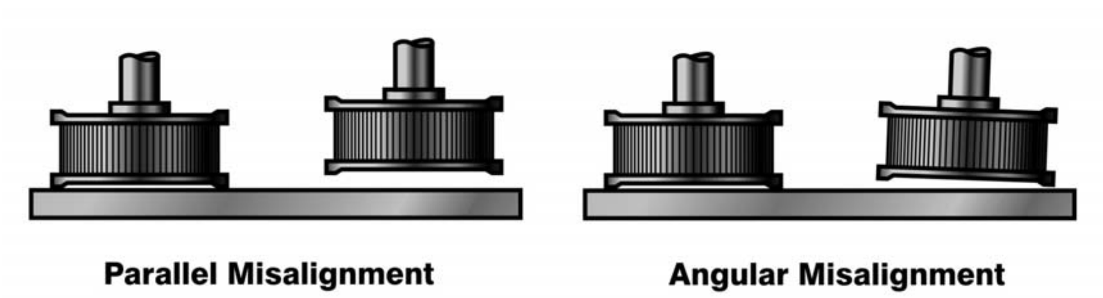


Figure 21.2. Belt Misalignment

21.4.2 Belt Speed

Belt speed V is defined as:

$$V = \frac{12DN}{\pi} \tag{21-1}$$

Where:
V = Belt speed, ft/min
D = pitch diameter, in
N = rotational speed, RPM

The ratio between the velocity in RPM of the faster shaft and that of the slower shaft is called the speed ratio:

$$Speed\ Ratio = \frac{RPM\ \ faster\,shaft}{RPM\ slower\,shaft} = \frac{Dia.larger\ pulley}{Dia.smaller\ pulley} \tag{21-2}$$

Belt speed is an important consideration in estimating the design life of a belt. Higher speeds require increased belt tension to compensate for higher centrifugal force. In general, increasing speed increases belt life because the tight side belt tension decreases with increasing speed. See Equation (21-3). However, if the belt speed is too high, the increase in centrifugal tension can more than offset the decrease in tight side belt tension and belt life can be reduced. See Equation (21-5). In timing belt drives (toothed belt) higher speeds generate dynamic forces resulting in increased tooth stresses and shorter belt life. Belt manufacturers normally provide a horsepower rating of the belt for specified belt speeds. The impact of belt speed on belt loading is explained in the next section.

21.4.3 Belt Loading

The belt on a belt drive assembly must be sufficiently tight to prevent slip causing a belt tension on both sides of the driven pulley or sheave. When the belt drive assembly is stationary with no power being transmitted, the tension on both sides of the pulley is equal. When the drive is rotating and transmitting power there is a tight side tension on the input side of the driven pulley and a slack side tension on the output side of the pulley. Figure 21.3 shows these tensions.

The ratio of the tight side to the slack side tension is called the tension ratio. The higher the tension ratio, the closer the belt is to slipping, i.e., the belt is too loose. The difference in these two tensions is a working tension. The working tension is calculated as follows:

$$T_W = \frac{33{,}000\,HP}{V} \qquad \textbf{(21-3)}$$

where:

T_W = Tight side tension (T_T) – slack side tension (T_S), lbs
HP = Horsepower, hp = 33,000 ft-lbs/min
V = Belt speed, ft/min

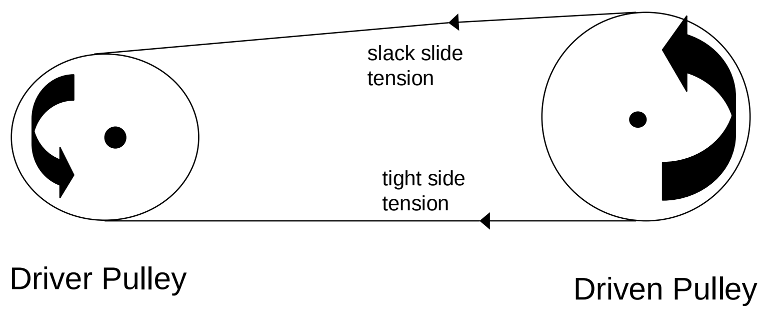


Figure 21.3 Belt Tensions

A belt experiences three types of tension as it rotates around a pulley:

- Working tension, T_W. Equal to tight side tension, T_T – slack side tension, T_S (See equation (21-3))

- Bending tension, T_B. As the belt bends around the pulley, one part of the pulley is in tension, the other in compression. The bending of the belt around a pulley introduces tension in the tensile member of the belt, the amount of tension dependent on the radius of the bend.

$$T_B = \frac{C_B}{d} \qquad \textbf{(21-4)}$$

where:

T_B = Bending tension, lbs
C_B = Constant depending on belt size and construction, in-lbs
d = Pulley diameter, in

- Centrifugal tension, T_C occurs in a belt drive because of the centrifugal force caused by the belt which has weight rotating around the pulley. Centrifugal tension depends on the belt speed.

$$T_C = MV^2 \tag{21-5}$$

where: T_C = Centrifugal tension, lbs
M = Constant related to belt weight, lbs/(ft/min)^2
V = Belt speed, ft/min

There are certain times as the belt moves on the pulley when the three types of tension are cumulative:

$$T_{PEAK} = T_T + T_B + T_C \tag{21-6}$$

These tensions on the belt are shown in Figure 21.4

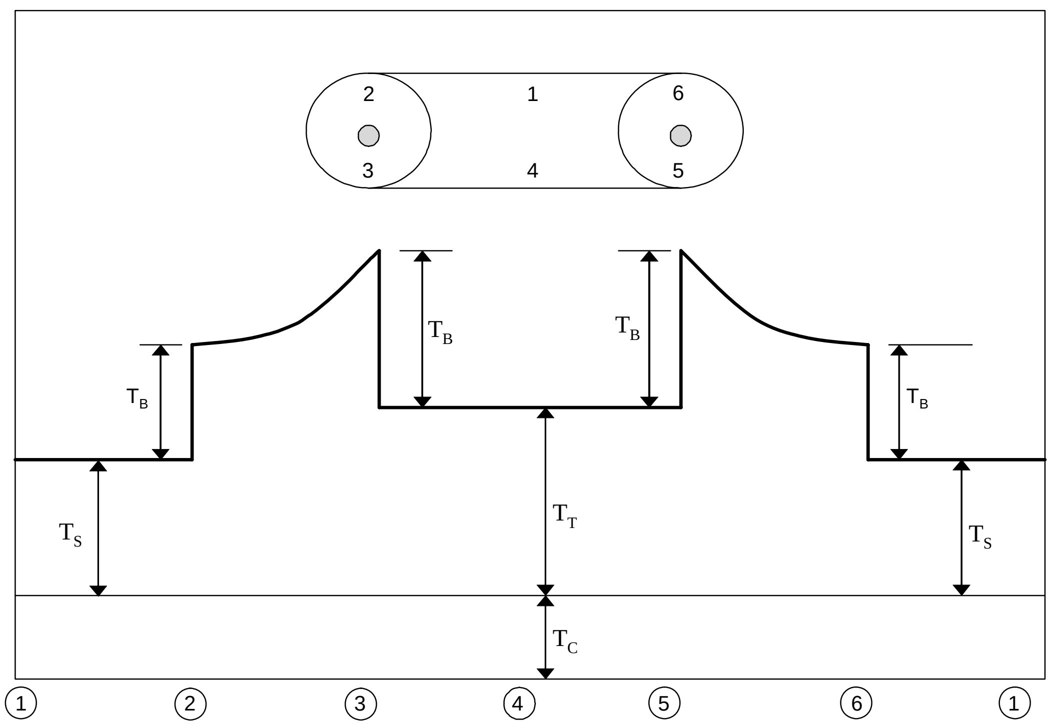
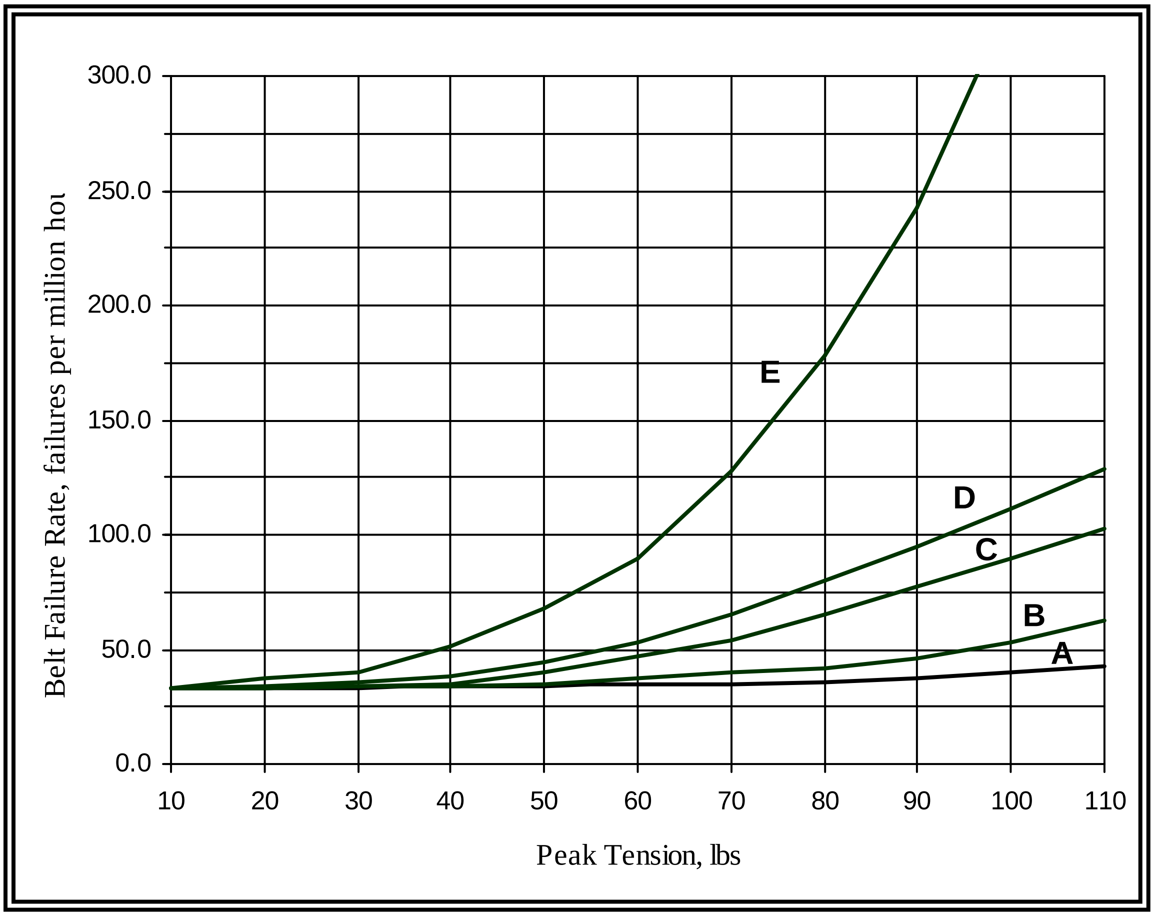


Figure 21.4 Belt Drive Tensions

Peak tension, T_{PEAK}, is directly related to belt life. The impact of peak tension (loading) on belt failure rate is shown in Figure 21.5.



A = 100 ft/min
B = 200 ft/min
C = 500 ft/min
D = 750 ft/min
E = 1000 ft/min

Figure 21.5 Impact of Belt Loading on Failure Rate

21.4.4 Belt Rating

The manufacturer of the belt will usually publish the horsepower rating of the particular belt being used or considered for a belt drive design. In some cases the torque rating is supplied which can be equated to horsepower using the equations in Section 21.4.3. It may also be possible to use the horsepower rating of the prime mover for the drive shaft. It must be remembered that the life of a flat or round belt is very much dependent on the tension of the belt and therefore dependent on maintenance procedures and the tension of the belt at the initial installation and any equipment repair. After the actual horsepower of the belt is computed using the equations in Section 21.4.3, it can be compared to the rated horsepower of the belt. The relationship of this horsepower ratio to estimated failure rate is shown in Figure 21.6

21.4.5 Belt Operating Temperature

Belts can operate successfully in temperatures from about -20 F to 140 F. Outside this temperature range, belt reliability depends on special belt materials. Belt life will normally decrease exponentially with an increase in temperature. Cold temperatures have a more dramatic effect on belt life. In extremely cold weather, the belt can fail when compounds within the belt structure reach their glass transition temperature causing a catastrophic belt failure.

Excessive slip caused by worn sheave grooves, tight bends and poor ventilation can lead to increased belt operating temperatures. These factors need to be considered in addition to the ambient operating temperature in determining belt temperature.

21.5 FAILURE RATE MODEL FOR BELT DRIVE

The failure rate of the total belt drive can be estimated with the following equation:

$$\lambda_{BD} = \lambda_{BD,B} \bullet C_{BL} \bullet C_t \bullet C_{PD} \bullet C_{BT} \bullet C_{BV} \bullet C_{SV} + \lambda_P \tag{21-7}$$

where:

λ_{BD} = Failure rate of belt drive under specific operating conditions, failures/million hours

$\lambda_{BD,B}$ = Base failure rate of belt, 40 failures/million hours

C_{BL} = Multiplying factor for belt loading (See Section 21.4.3, Section 21.4.4 and Figure 21.6)

C_t = Multiplying factor for belt operating temperature (See Section 21.4.5 and Figure 21.7)

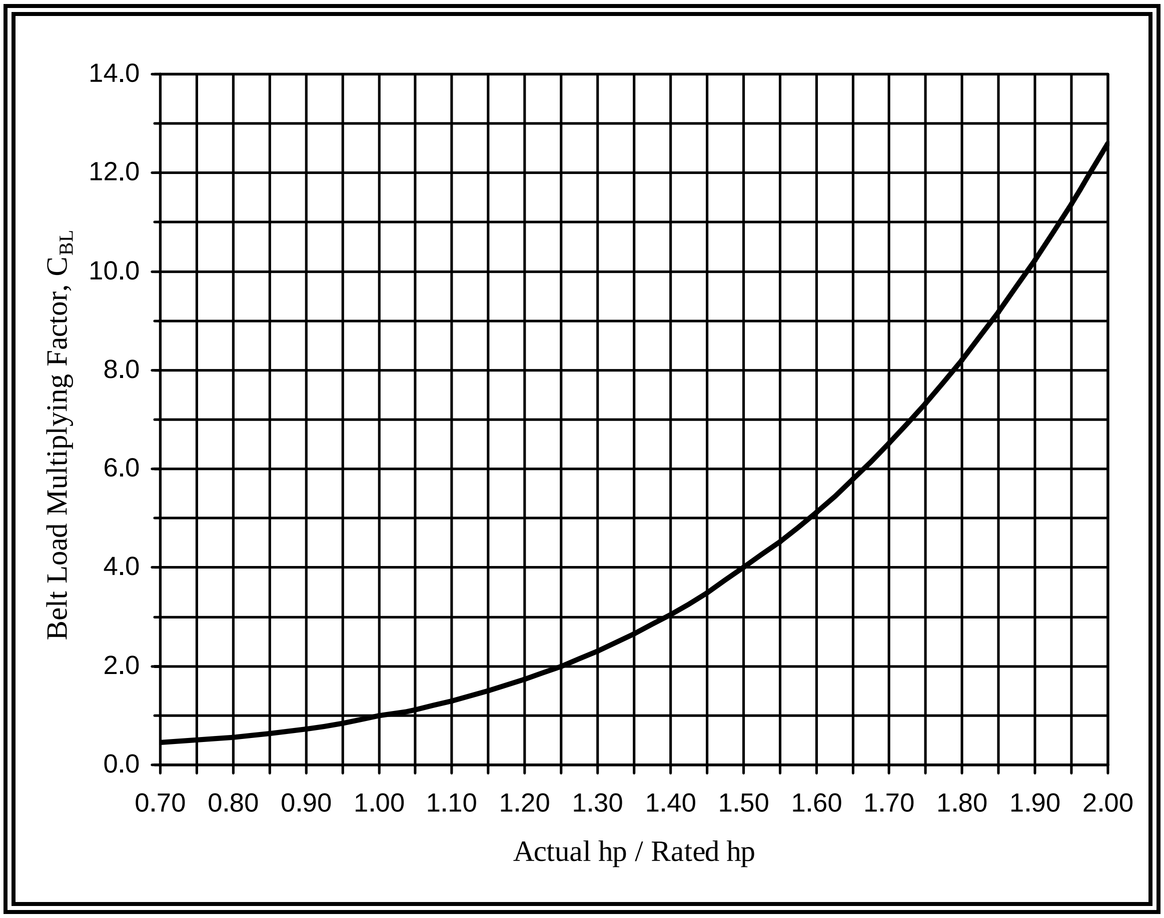
C_{PD} = Multiplying factor for pulley diameter (See Figure 21.8)

C_{BT} = Multiplying factor for belt type (See Table 21-2)

C_{BV} = Multiplying factor for belt drive operating service (See Table 21-3)

C_{SV} = Multiplying factor for belt drive shock environment (See Table 21-4)

λ_P = Failure rate for driver and driven pulleys, 0.8 failures/million hours for flat pulleys, 1.5 failures/million hours for grooved pulleys



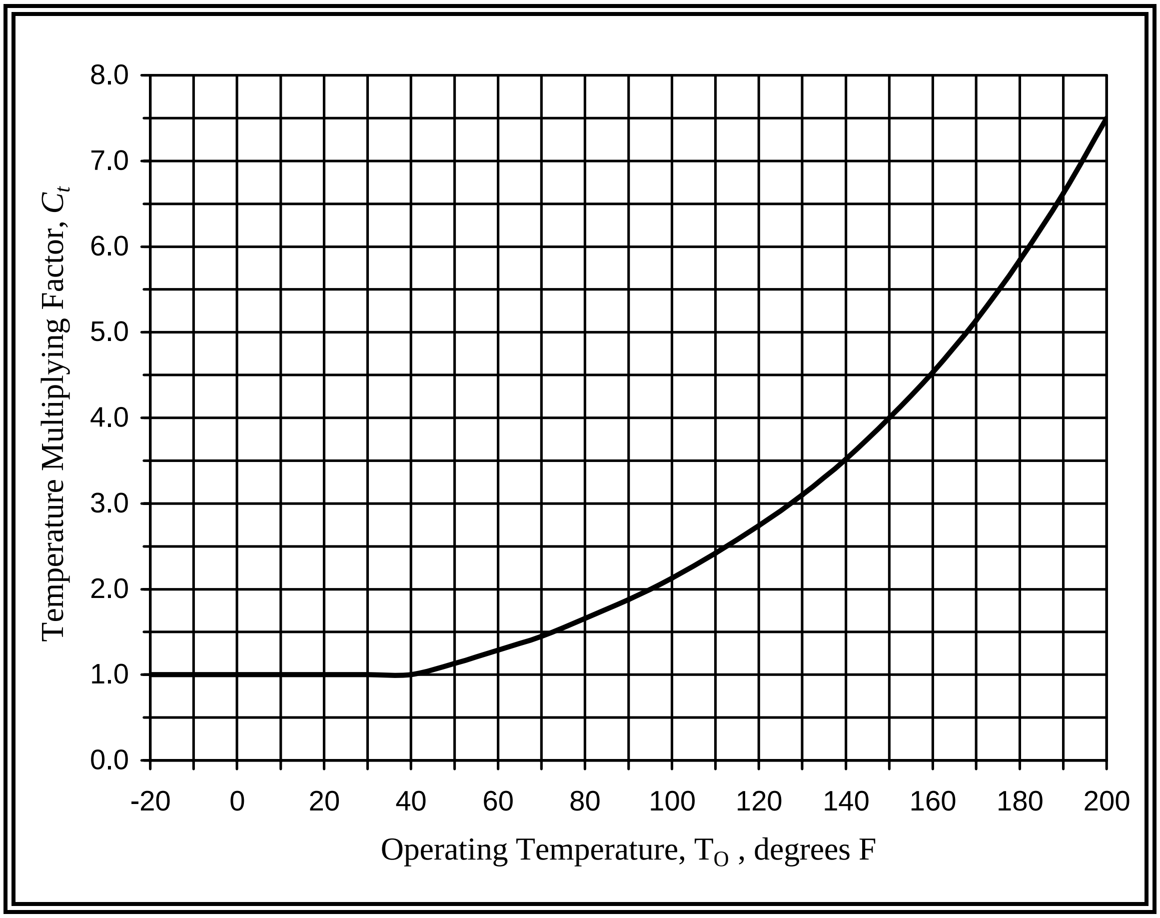
$$C_{BL} = 0.3 + \left(\frac{hp_O / hp_D}{1.1} \right)^{4.2}$$

where: C_{BL} = Belt load multiplying factor

hp_O = Operating load, ft-lbs/min

hp_D = Design or specified load, ft-lbs/min

Figure 21.6 Belt Load Multiplying Factor, C_{BL}

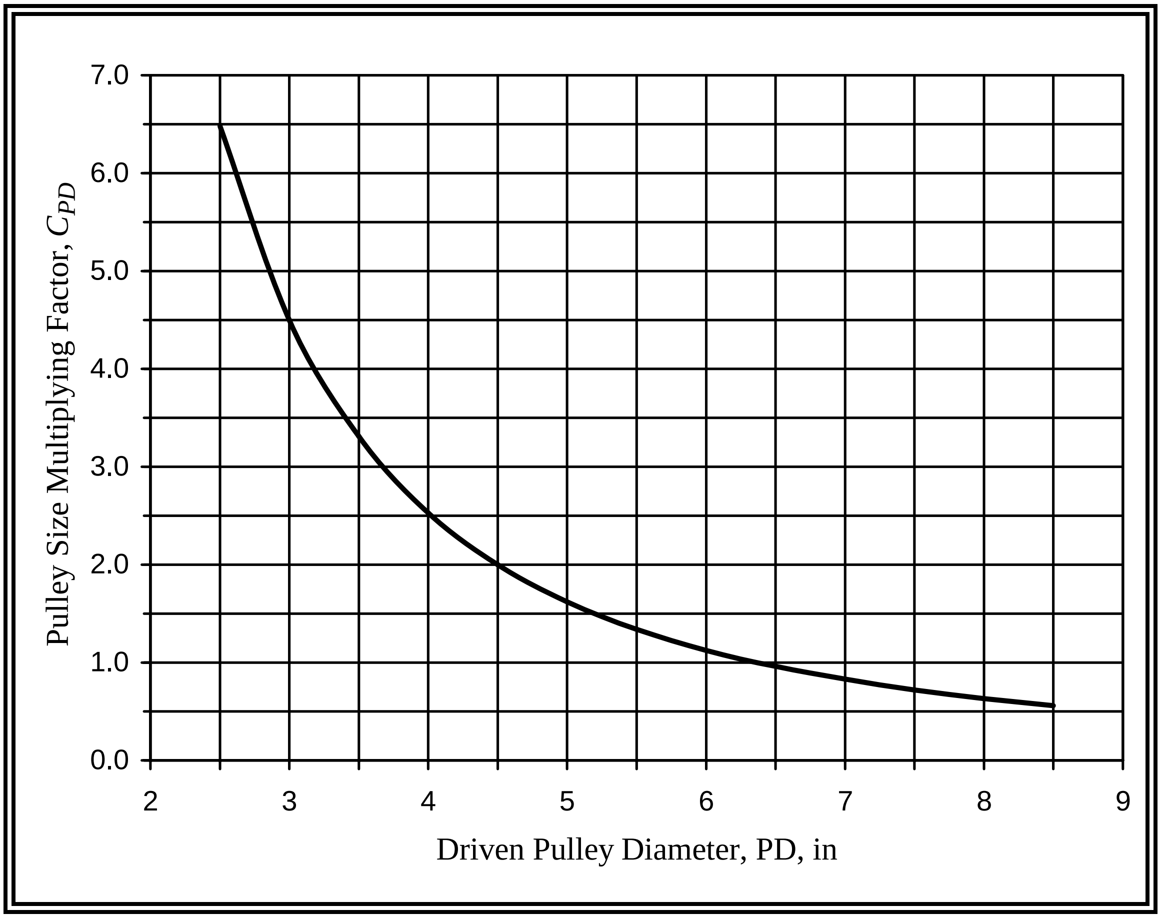


$$C_t = \frac{1}{2^t}$$

where: $t = 0$ for Operating Temperature , T_O = -20 F to 40 F

and: $t = \frac{40 - T_O}{55}$ for operating temperature , T_O > 40 F

Figure 21.7 Multiplying Factor for Belt Operating Temperature, C_t



$$C_{PD} = 2\,x \left(\frac{4.5}{PD}\right)^2$$

where: PD = Diameter of driven pulley, in

Figure 21.8 Multiplying Factor for Pulley Diameter, C_{PD}

Table 21-2 Belt Type Multiplying Factor, C_{BT}

Belt Type	Belt Type Multiplying Factor, C_{BT}
SPZ	0.8
SPA	0.48
SPB	0.33
SPC	0.18
Y	9.09
Z	4.16
A	0.93
B	0.54
C	0.30
D	0.14

Table 21-3 Belt Operating Service Factors, C_{BV}

Reference 39

<table>
<tr><th colspan="6">Driver Pulley</th><th>Driven Pulley</th></tr>
<tr><td colspan="3">Low or normal torque (small AC motors, shunt wound DC motors, small engines</td><td colspan="3">High or non-uniform torque (single phase AC motors, series wound DC motors, large internal combustion engines)</td><td rowspan="2">Typical loads</td></tr>
<tr><td>Intermittent service</td><td>Normal service</td><td>Continuous service</td><td>Intermittent service</td><td>Normal service</td><td>Continuous service</td></tr>
<tr><td>1.0</td><td>1.1</td><td>1.2</td><td>1.1</td><td>1.2</td><td>1.3</td><td>Blowers, small fans, centrifugal pumps, compressors</td></tr>
<tr><td>1.1</td><td>1.2</td><td>1.3</td><td>1.2</td><td>1.3</td><td>1.4</td><td>Generators, machine tools, rotary pumps</td></tr>
</table>

Table 21-4. Shock Environment Service Factors, C_{SV}

Driven Machinery	Source of Power	
	Low or Normal Torque	High or Non-uniform Torque
No shock	1.1	1.2
Light shock	1.2	1.3
Medium shock	1.3	1.5
Heavy shock	1.4	1.7

21.6 CHAIN DRIVES

A chain is a reliable machine component, which transmits power by means of tensile forces, and is used primarily for power transmission and conveyance systems. The function and uses of a chain drive are similar to those of a belt drive. The two most common types of chain include steel chain, especially the type called roller chain, which makes up the largest share of chains being produced, and plastic chain. Metal chain drives are normally used for applications below 3000 rpm where accuracy and reliability must be greater than that provided by rubber belts. Chain drives will usually maintain a constant speed under varying load conditions because the metal chain does not slip or stretch and will need only infrequent adjustment.

A chain drive is a combination of chains and sprockets and a means of shaft mounting. A chain can be defined as a series of links, connected and fitted into one another to form what is in effect a flexible rack with a series of integral journal bearings. In its simplest form a chain drive consists of two sprockets of arbitrary size and a chain loop. Sprockets are wheels with external teeth shaped so that they can fit into the links of the drive or driven chain. The center distance from one hinge or joint to the next is known as the pitch of the chain and is the primary identifying dimension.

Some of the advantages of chain drives over belt drives include:

- No slippage between chain and sprocket teeth
- Negligible stretch allowing chain drives to carry heavy loads
- Less sensitive to dust and humidity than belts and not adversely affected by sun, oil or grease
- Operation at higher temperatures than belt drives

- Longer operating life expectancy because flexure and friction contact occur between hardened bearing surfaces separated by an oil film
- Longer shelf life because of less susceptibility to storage conditions and non-deterioration of metal

Some of the potential reliability problems associated with chain drives include:

- Chain drives can elongate due to wearing of link and sprocket teeth contact surfaces
- Chain drives require frequent or, in some applications, continual lubrication
- Sprockets often need to be replaced when worn chains are replaced because of wear
- Chain drives are limited to slower speeds than belt drives

21.7 FAILURE MODES OF CHAIN DRIVES

There are several ways a chain can fail:

- In a tensile failure, the chain is overloaded in tension until it is stretched so badly it will not function properly, or it is literally pulled apart.
- If the chain is loaded repeatedly in tension, at a load below the yield strength (the chain is not stretched) microscopic cracks will develop in the link plates. These cracks continue to grow until the chain breaks.
- In a wear failure, material is removed by sliding, or sliding combined with abrasion or corrosion, until the chain will not fit the sprockets. Either the mechanism will not function properly or the remaining material is so thin that the chain breaks.
- Galling is rapid wear caused by metal seizure between the chain pin and bushing. This rapid wear is caused by the combination of excessively high speeds and loads.

Parts of a chain drive are shown in Figure 21.9. Chain drives require tensioning on a regular basis. When the chain has "stretched" by approximately 3 percent the chain typically requires replacement. All the components of the chain (rollers, pins, bushings) have lost their case hardened surface properties at this point and failure is imminent.

Design life of a chain depends on adequate lubrication. Proper lubrication reduces the wear on all moving surfaces of the chain, and helps cushion the chain drive from shock loading. The type of lubrication method varies dependent upon the speed and operating environment. Manual lubrication involves periodic application of oil from an oiler or brush and is used for slow speed and intermittent operation. Oil bath lubrication is best for most applications and involves submerging the lower strand of chain in a chain casing oil sump. Recirculating oil systems or forced lubrication can be used to provide continuous lubrication to the chain. This method is more common in high speed, high horsepower applications.

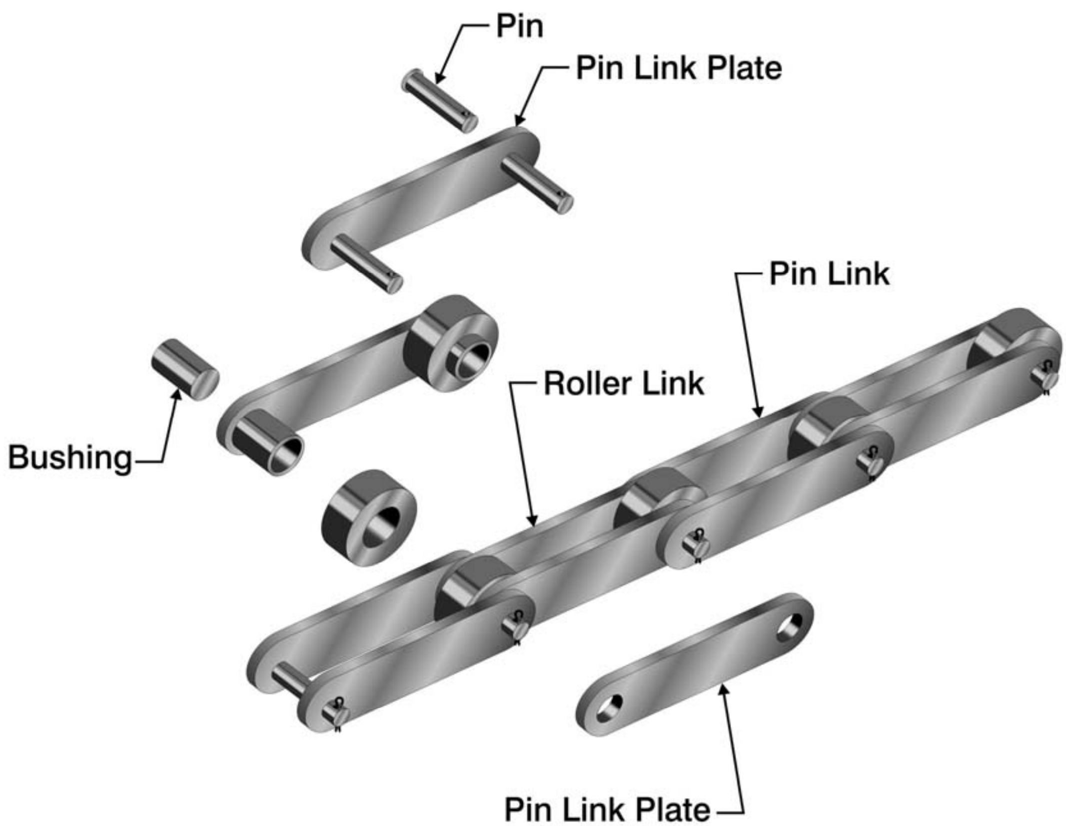


Figure 21.9 Parts of a Chain Drive

The key factor causing a chain to jump the sprocket teeth is chain wear elongation. Because of wear elongation, the chain creeps up on the sprocket teeth until it starts jumping sprocket teeth and can no longer engage with the sprocket. When a chain is operating, the outer surface of the pin and inner surface of the bushing rub against one another creating constant wear. Proper lubrication reduces the amount of wear but does not eliminate it.

As the surface of the pin is reduced by wear, the rigidity of the pin decreases and eventually fatigue failure may result. How much wear is acceptable and at what point operation is no longer acceptable depends on the application. When wear elongation is less than or equal to 1.5 percent for a transmission chain, or less than or equal to 2 percent for a conveyor chain, there is almost no risk of fatigue failure.

Chain sprockets can operate in a variety of environments. Dust and dirt can easily be separated from oil through proper circulation and filtering, allowing chain drives to be operated in rather harsh environments. Furthermore, chain drives can be used in high temperature environments, provided sufficient lubrication is available that won't break down at elevated temperatures.

In summary, the failure modes of a chain drive will depend on its application and the allowable performance in terms of noise and chain elongation. Table 21-5 provides a summary of possible chain drive failure modes.

Table 21-5. Typical Failure Modes of Chain Drives

FAILURE MODE	FAILURE CAUSE	FAILURE EFFECT
Turned/galled pins	Inadequate lubrication	Damaged chain
Enlarged holes	Overload	Damaged chain
Broken pins and/or link plates	Extreme overload	Damaged chain and sprockets
Worn link plate contours	Chain rubbing on chain guide	Damaged chain
Cracked link plates	Excessive loading; corrosive environment	- Chain fatigue failure - Stress corrosion failure
Broken, cracked or deformed rollers	Excessive speed, chain riding too high on sprocket teeth	Damaged chain
Chain climbs sprocket teeth	Excessive chain slack; overload	Chain and sprocket wear failure
Foreign objects in the chain drive assembly	Operating environment	Chain wear and eventual failure
Excessive noise	Chain interference; loose casing; excessive chain slack; inadequate lubrication	Eventual chain damage
Bushing/roller fatigue	Chain loading	Chain wear and eventual failure
Chain elongation	Link-pin joint wear	Chain begins to skip teeth followed by complete failure
Tensile fatigue loading	Sideplate loading	Sudden catastrophic chain failure

21.8 RELIABILITY PREDICTION OF CHAIN DRIVES

For a chain drive to transmit power, the driven sprocket must be rotated against a resisting torque. In contrast to belt drives, one strand of a chain drive must always be slack. Thus, power is transmitted solely by the tension side.

Chain drives are less sensitive to dust and humidity than belts and are not adversely affected by sun, oil or grease. They can also operate at much higher temperatures than belts and they do not slip. Chain drives do require frequent lubrication (continual in some applications), they require very close alignment, and since the do not slip they provide no overload protection.

In Figure 21.10, the left sprocket is the driving side (power input) and the right sprocket is the driven side (power output). As counterclockwise rotation power is applied to the driving sprocket while adding resistance to the driven sprocket, then the chain is loaded in tension mainly along the D-A span, and tension is smaller in the other parts. Figure 21.11 shows this relation.

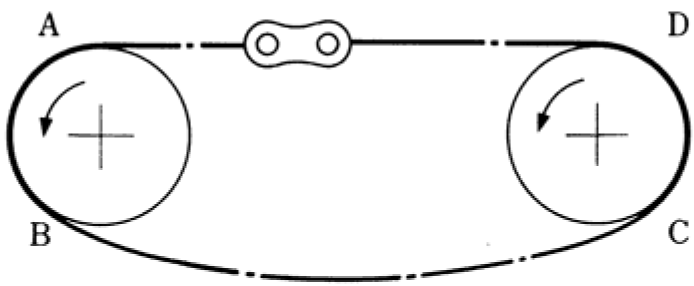


Figure 21.10 Typical Chain Drive Under Load

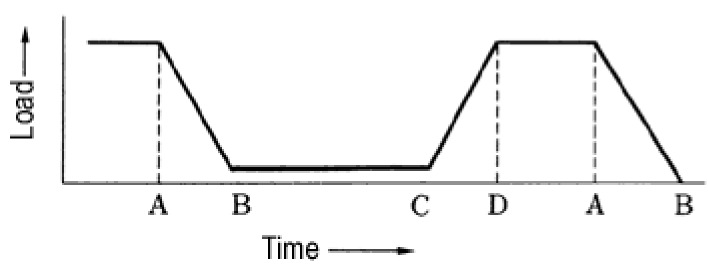


Figure 21.11 Chain Load with the Addition of Resistance

As a chain is subject to increasing stress or load, it becomes longer. This relationship can be graphed as shown in Figure 21.12. The vertical axis shows increasing stress or load, and the horizontal axis shows increasing strain or elongation. In this stress-strain graph, each point represents the following:

- O-A: elastic region
- A-C: plastic deformation
- B: maximum tension point
- C: actual breakage

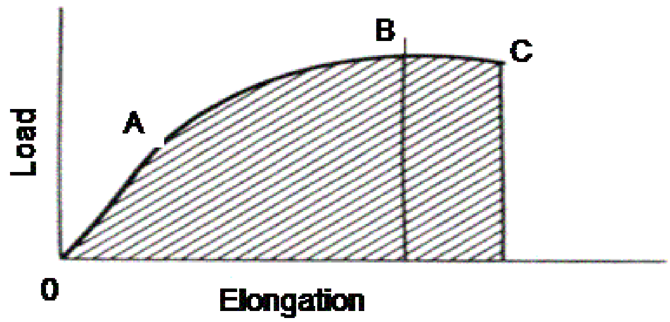


Figure 21.12 Effect of Increasing Stress on Chain Length

The chain speed for a given sprocket diameter and sprocket angular velocity is given by:

$$S = \frac{\pi n D}{12} = \frac{PNn}{12} \qquad \textbf{(21-8)}$$

where:

S = Chain speed, ft/min

n = Sprocket speed, rpm

D = Sprocket pitch diameter, in

P = Chain pitch, in (equal to the center-to-center distance between chain hinges or joints)

N = Number of teeth of sprocket

For a chain drive with two sprockets, the speed ratio is:

$$\frac{n_2}{n_1} = \frac{N_1}{N_2} \quad \textbf{(21-9)}$$

And the chain tension is:

$$T = \frac{33{,}000 \bullet HP}{S} \quad \textbf{(21-10)}$$

where:

T = chain tension, lbs

S = Chain speed, ft/min

HP = Horsepower to be transmitted, hp

21.9 FAILURE RATE MODEL FOR CHAIN DRIVE

A graph showing the area of reliable operation of a chain drive and the impact of chain speed on horsepower and resulting failure mechanisms is shown in Figure 21.13. The location of Line O-A in Figure 21.13 depends on the chain's allowable tension, including the fatigue strength of the connecting or offset links and the centrifugal force in high-speed rotation. Line B-C depends on the breakage limit of the bushing and roller. In this kind of breakage of the bushing and roller, there is no fatigue limit as there is with the link plates. The location of line D-E depends on the bearing function of the pin and the bushing. The range defined within these three lines (O-A, B-C, and D-E) is the usable range. When the chain is used at low speeds, it is limited by line O-A, the fatigue limit.

Horsepower ratings are based upon the number of teeth and the rotating speed of the smaller sprocket, either drive or driven. The pin-bushing area, as it affects allowable working load, is the important factor for medium and higher speeds. For very low speeds, the limiting factor is the ultimate tensile strength of the chain. Figures 21.14 and 21.15 show the effect of RPM on horsepower rating of the chain.

In normal operation, the position at which the chain and the sprockets engage will fluctuate and the chain will vibrate along with this fluctuation. Even with the same chain, if the number of teeth in the sprockets is increased (change to larger diameter), vibration will be reduced. A decrease in the number of teeth in the sprockets will cause an increase in vibration. This is because there is a pitch length in chains, and they can only bend at the pitch point.

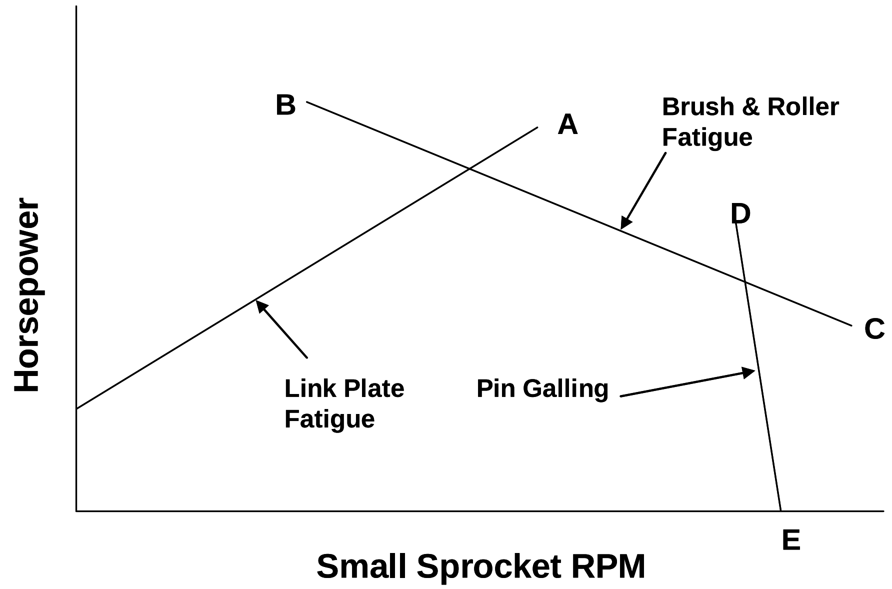


Figure 21.13 Typical Chain Drive Failure Modes

Considering the failure modes and stress considerations of chain drives, the following equation can be developed for estimating the failure rate of chain drives:

$$\lambda_{CD} = \lambda_{CD,B} \bullet C_{CV} \bullet C_{CS} \bullet C_{CT} \bullet C_{CI} \bullet C_{ST} + \lambda_{CS} \qquad \textbf{(21-11)}$$

where:

λ_{CD} = Failure rate of chain drive under specific operating conditions, failures/million hours

$\lambda_{CD,B}$ = Base failure rate of chain, 15 failures/million hours

C_{CV} = Multiplying factor for operating service (See Table 21-7)

C_{CS} = Multiplying factor for chain speed (See Table 21-8)

C_{CT} = Multiplying factor for chain operating temperature (See Table 21-9)

C_{CI} = Multiplying factor for lubrication method (See Table 21-10)

C_{ST} = Multiplying factor for sprocket design (See Figure 21-16)

λ_{CS} = Failure rate for driver and driven sprockets, 0.8 failures/million hours

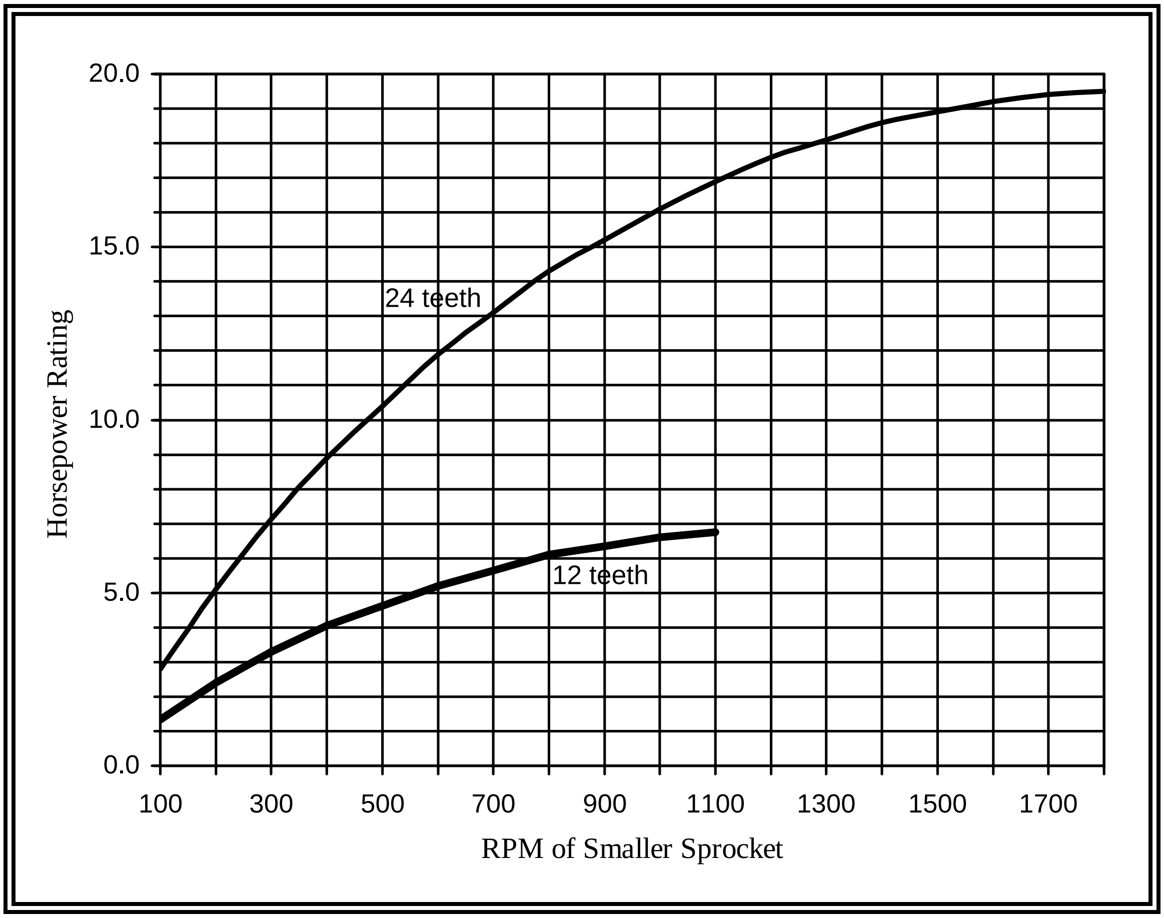


Figure 21.14 Horsepower Ratings for Typical Single-Strand, Roller-Chain Drive (ANSI No. 60, ¾ inch pitch)

It is important to remember that when determining the horsepower ratings from manufacture's catalogs that the total horsepower rating of a multiple strand is not a simple multiplication of a single chain horsepower rating times the number of strands. Use the applicable factor from Table 21-6 to determine the total horsepower rating of multiple strand chains.

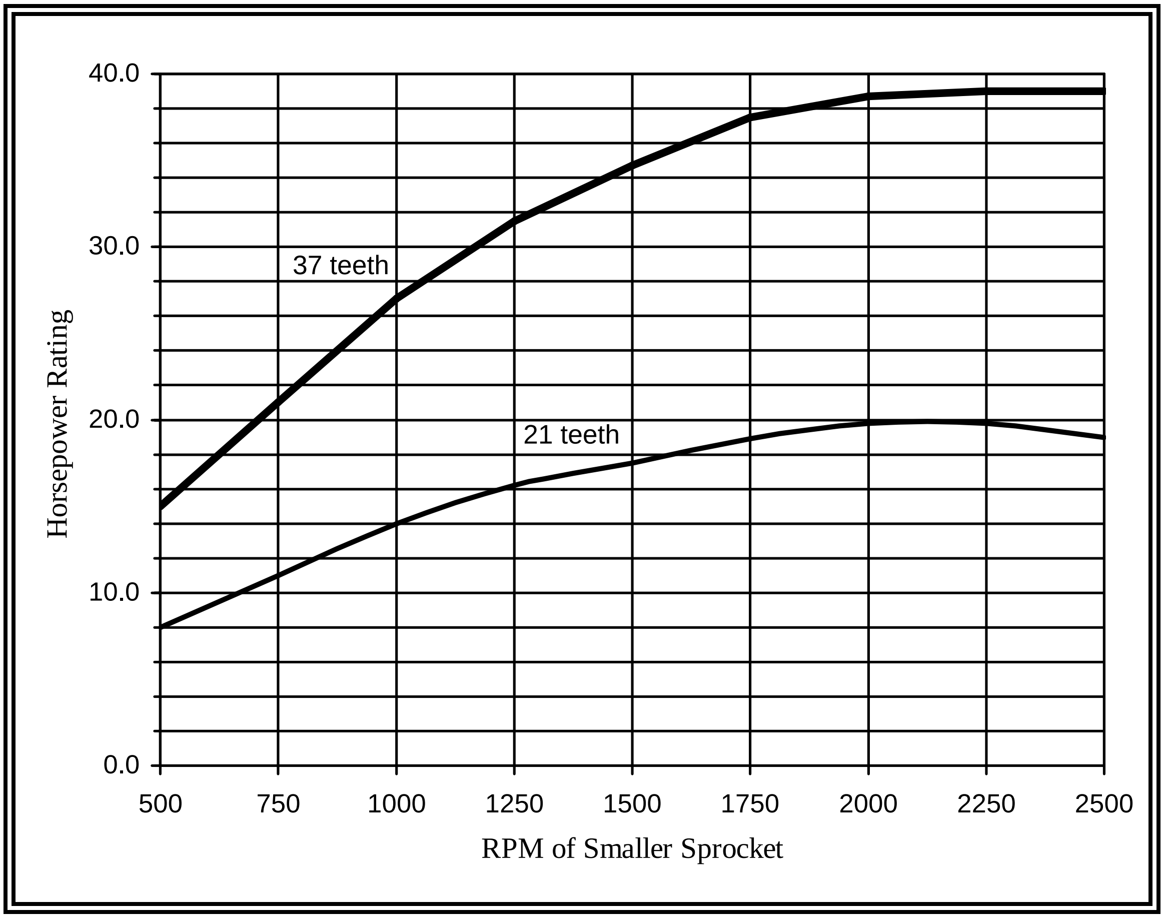


Figure 21.15 Horsepower Ratings Per Inch Width for Typical Silent-Chain Drive (¾ inch pitch)

Table 21-6. Multiple Strand Factor

Number of Roller Chain Strands	Multiple Strand Factor
1	1.0
2	1.7
3	2.5
4	3.3
5	3.9
6	4.6

Table 21-7. Chain Drive Operating Service Factors, C_{CV}

Driver Sprocket			Driven Sprocket
Smooth running (electric motors, turbines)	Slight shock (electric motors with frequent starts, internal combustion engine with hydraulic coupling)	Moderate shocks (internal combustion engines with mechanical coupling	Type of Driven Load
1.0	1.1	1.3	Smooth (conveyers with small load fluctuations, centrifugal blowers)
1.4	1.5	1.7	Some impact (conveyers with some load fluctuations, centrifugal compressors, marine engines)
1.8	1.9	2.1	Large impact (machines with reverse or large impact loads

Table 21-8 Chain Speed Multiplying Factor, C_{CS}

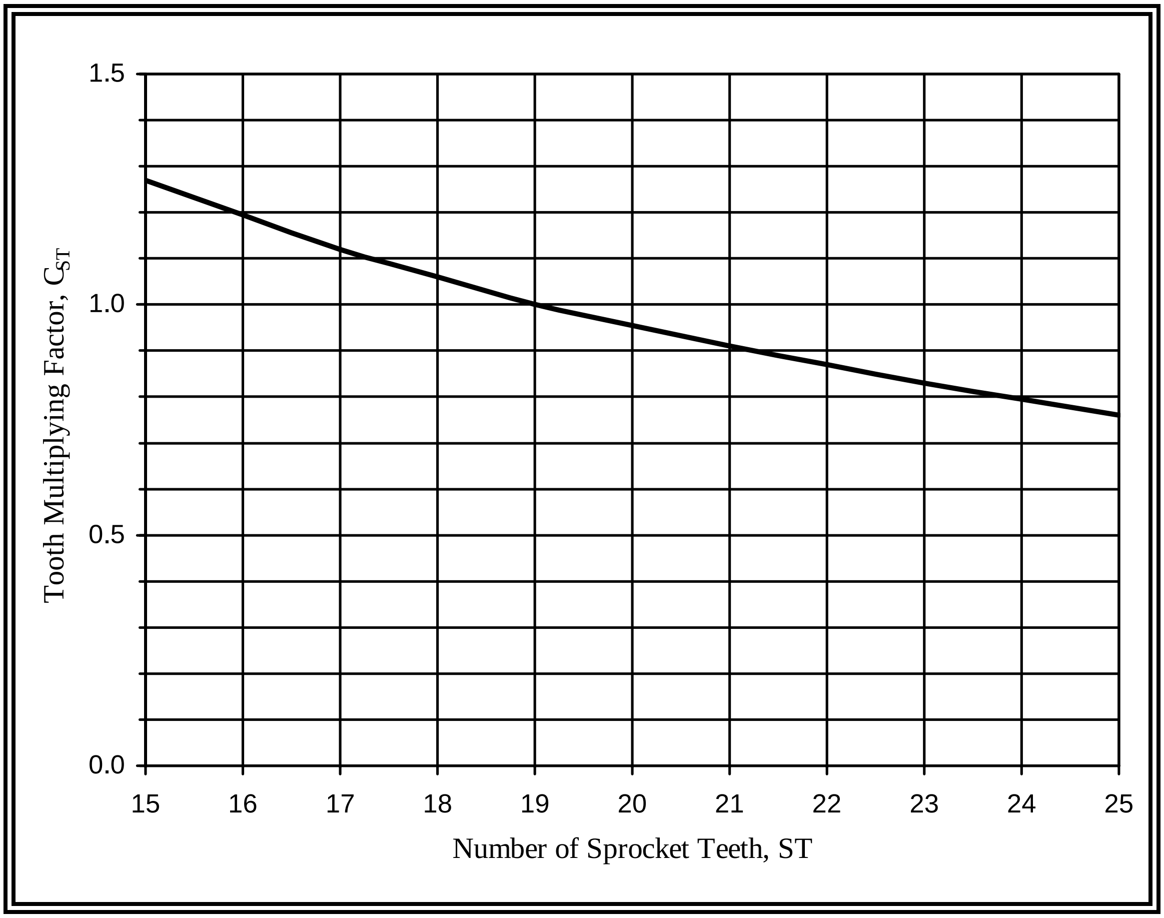
Chain Speed	C_{CS}
< 50 ft/min	1.0
50 – 100 ft/min	1.2
> 100 – 160 ft/min	1.4

Table 21-9 Chain Temperature Multiplying Factor, C_{CT}

Temperature	C_{CT}
$\leq$ 170 ^oC	1.0
170 – 200 ^oC	1.5
> 200 – 260 ^oC	2.0

Table 21-10 Chain Lubrication Multiplying Factor, C_{CI}

Type of Lubrication	C_{CI}
Manual Operation	1.5
Drip Lubrication	1.0
Bath Lubrication	0.8
Stream Lubrication	0.7



$$C_{ST} \;=\; 19/ST$$

where:

ST = Number of teeth on smaller sprocket

Figure 21.16 Multiplying Factor for Sprocket Design

21.10 REFERENCES

In addition to specific references cited throughout Chapter 21, other references included below are recommended in support of performing a reliability analysis of belt and chain drives.

19. Hindhede, U. et al, Machine Design Fundamentals, John Wiley & Sons, NY, 1983

39. Shigley, J.E., Mischke, C.R., Mechanical Engineering Design, McGraw-Hill Book Co., NY, 1989.

58. Parmley, R.O., Mechanical Components Handbook, McGraw-Hill Book Co., NY, 1985

84. Machinery's Handbook, 27th Edition, Industrial Press, 2004

87. Childs, T.H.C., Dalgarno, K.W., Day, A.J. and Moore, R.B., "Automotive Timing Belt Life and a User Design Guide", Proceedings of the Institute of Mechanical Engineers, 1998

88. Reliability Analysis Center, "Nonelectronic Parts Reliability Data", NPRD-95

89. SINTEF Industrial Management, "OREDA Offshore Reliability Data", 4th Edition, 2002

90. The Complete Guide to Chains, Tsubakimoto Chain Company, 1995

91. Renold Transmission Chain Selection Procedure

106. Design and Analysis of Machine Elements, Douglas Wright, Department of Mechanical and Materials Engineering, The University of Western Australia

107. Salzman, R.H. and Reaburn, S. M. "Probabilistic Modeling for Timing Belt Fatigue Life Predictions Using Accelerated Testing", Int. J. of Materials & Product Technology, 2001

110. John L. Wright, "Chains for Drives and Conveyors – Lube 'Em to Last", Machinery Lubrication Magazine, March 2002

This Page Intentionally Left Blank

CHAPTER 22

FLUID CONDUCTORS

22.0 TABLE OF CONTENTS

22.1 INTRODUCTION

Fluid conductors are the means by which the entire fluid system is connected. It is therefore very important to assure the reliability of the various connections in the system. Many times in evaluating a fluid system for reliability, the major components such as pumps, valves and actuators are analyzed in detail while overlooking the fluid-carrying link between the components. Individual fluid conductors are usually very reliable from a design standpoint but there may be a large number of these components in the overall system and their reliability can be very sensitive to the operating environment. Even with all the technological advancements in hydraulic systems, fluid leakage remains one of the most aggravating causes for unscheduled maintenance actions.

Conductors of a fluid power system are basically of three types: pipe, tubing and hose. A pipe is a rigid conductor not intended to be bent or shaped into a configuration. Tubing is a semi-rigid fluid conductor which is usually bent into a desired shape for the particular application. A hose is a flexible fluid conductor which can be adapted to components that move during operation. Newer designs are combining tubing and hoses as hybrid assemblies. These assemblies provide the strength and heat dissipation characteristics of metal tubing with the flexibility and vibration dampening characteristics of hose.

There are many factors to consider when evaluating the reliability of a fluid conductor system. The failure modes presented in this Chapter should be considered for the particular application of the fluid conductor because the failure rate of a fluid conductor is probably more sensitive to the operating environment of the system in which it is installed than to its design parameters. Each application must be evaluated individually because of the many installation, usage and maintenance variables that affect the failure rate.

22.2 PIPE

Piping components of a fluid system are designed for an internal pressure to compensate for the most severe conditions of fluid pressure and temperature expected in normal operation. In addition to normal fluid operating pressures, potential back pressures, pressure surges, temperature fluctuations and performance variations of pumps, valves and other components must also be considered in the evaluation of fluid conductor systems. To meet these requirements, the greatest required pipe thickness and highest flange rating must be utilized.

Older fluid conductor systems are comprised of threaded pipe and many hydraulic systems had NPT ports (American Standard Taper Pipe Threads for General Use). This type of connection is the least reliable for high-pressure fluids as the thread itself provides a leak path. Because pipe threads are deformed when tightened, any subsequent movement, either loosening or tightening, increases the potential for leaks. Threaded connections have more recently been replaced by more reliable soft seal connections.

Most failures of fluid conductor systems occur at or within the interconnection points such as fittings and flanges. These areas are sensitive to installation and maintenance procedures and the inclusion of seals. Reliability of the system therefore depends on the proper selection of interconnecting components that are compatible with each other and the operating environment. Considerations include strength, ductility, hardness, and corrosion resistance. System temperatures also need to be evaluated when estimating reliability. A single over-temperature event of sufficient magnitude can permanently damage all the seals in an entire high-pressure fluid system resulting in numerous leaks. Prolonged operation at above-normal temperatures can produce the same results.

Typical pipe system materials include high density polyethylene (HDPE), cross linked polyethylene (PEX), acrylonitrile butadiene styrene (ABS), polyvinyl chloride (PVC), stainless steel, carbon steel, copper and ductile iron. One material does not exhibit a significant difference in lifespan from another. When properly designed and installed, pipe systems of any of these materials can be sufficiently durable to exhibit extremely low failure rates. Plastic pipe can exhibit some long term failure modes due to chemical attack, deterioration from ultraviolet rays, change in dimension (creep) and environmental stress cracking. Failure data indicates no difference in the failure rate between pressure and gravity flow systems assuming proper pipe material selection and layout and proper tightness of pressure system connections to avoid leaks.

22.2.1 Failure Modes of Pipe Assembly

As mentioned previously, most failures of a pipe assembly occur at or within the interconnection points. Failure modes of a pipe assembly include material properties that are incompatible with the fluid being transported or with the surrounding operating environment. A common failure mode in pipe systems is caused by a sudden reduction in liquid flow in a pipe. When a valve is abruptly closed, dynamic energy is converted to elastic energy creating a pressure wave called water hammer that can cause pipe failure.

The following failure modes need to be considered when evaluating a pipe assembly for reliability:

- Burst failure caused by internal pressure
- Buckling caused by external pressure
- Bending failure
- Stress related failure from applied loads
- Excessive leakage at the interconnection points

Table 22-1 provides a summary of potential failure modes of a pipe assembly.

22.2.2 Reliability Prediction of Pipe Assembly

The failure modes presented above should be considered for the particular application of the fluid conductor. The failure rate of a fluid conductor is extremely sensitive to the operating environment of the system in which it is installed as compared to the design of the pipe. Each application must be evaluated individually because of the many installation, usage and maintenance variables that affect the failure rate.

The internal pressure in piping normally produces stresses in the pipe wall because the pressure forces are offset by pipe wall tension. The longitudinal stress from pressure is calculated by:

$$S_L = \frac{P_D\, D}{4t} \qquad \textbf{(22-1)}$$

where: S_L = Longitudinal stress, psi
P_D = Internal design pressure, psi
D = Outside pipe diameter, in
t = Pipe wall thickness, in

Table 22-1. Typical Failure Modes of Pipe Assemblies

FAILURE MODE	FAILURE CAUSE	FAILURE EFFECT
Damaged connector	- Corrosion - Improper torque on fitting - Gasket failure	Gradual increase in system leakage
Burst failure	- Rapidly applied load - Pressure transients	Catastrophic pipe assembly failure
Buckling failure	- Insufficient piping supports	Immediate leakage above system requirements
Bending failure	- Bend radius less than allowable	Immediate leakage above system requirements
Crack in rigid pipe	- External stress	System leakage
Leakage	- Chemical incompatibility with fluid - Chemical attack/improper thread sealant - Ultraviolet deterioration	Gradual increase in system leakage
Fatigue failure	- Water hammer from upstream component	System fluid leakage

The burst pressure of the pipe is determined as follows:

$$P = \frac{2\,t\,S}{D} \qquad \textbf{(22-2)}$$

where: P = Burst pressure, psi

t = Pipe wall thickness, in
S = Tensile strength of pipe material, psi
D = Outside pipe diameter, in

And the working pressure is equal to:

$$WP = \frac{P}{sf} \tag{22-3}$$

where: WP = Working pressure, psi
sf = Safety factor (normally equal to approximately 3.0)

The failure rate of a piping assembly depends primarily on the connection joints and the base failure rate of a piping assembly from various data sources can be estimated at 0.47 failures/million operating hours per connection and the total failure rate of the pipe assembly can be estimated with the following equation:

$$\lambda_P = \lambda_{P,B} \bullet C_E \tag{22-4}$$

where: λ_P = Failure rate of pipe assembly, failures per million hours
$\lambda_{P,B}$ = Base failure rate of pipe assembly, 0.47 failures/million hours
C_E = Environmental factor, see Table 22-2

Table 22-2. Pipe Assembly Environmental Factor, C_E

Operating Environment	Multiplying Factor, C_E
Normal duty, non-flex pipe, ambient conditions, no vibration or shock	1.0
Heavy duty, flexible pipe under random pulsations	1.2
Severe duty, vibration and shock environment	1.4

See Chapter 3 of this Handbook to determine the failure rate of any seals being used in the pipe connectors.

22.3 TUBING

Components on many types of hydraulic equipment are connected together by semi-rigid tubing. Tubing is light-weight compared to pipe and can be bent and assembled on-site to meet specific system configurations. Material types for tubing include aluminum, corrosion-resistant steel and titanium. These materials provide good heat dissipation for the system fluid.

There are many factors that can lead to tubing failure. Because it is semi-rigid, tubing can transmit vibration from one component to another throughout the equipment. Therefore, many hydraulic designs utilize bent tubes and hoses providing the weight and bend advantages of bent tube with the flexibility and vibration dampening characteristics of hose. Although tubing is generally a better heat dissipater than hose, in some applications the use of hose can actually result in less heat buildup because of improved laminar flow through the more gradual bends created between hose connections.

Connectors for hydraulic tubing are either permanent swaged fittings or AS (Aerospace Standard) type fittings. Swaged fittings require special repair equipment and procedures, but their reliability is extremely good and they are light-weight. AS fittings require tubes to be flared at 37 degrees on the ends which adds to the cost of the tubing. AS fittings need to be selected based on their pressure rating. A common cause of leakage from 37 degree flare fittings is incorrect torque. Insufficient torque results in inadequate seat contact, while excessive torque can result in damage to the tube and connector through cold-working. Newer AS fittings are not flared; instead, the sleeve bites into the tube. All metal-to-metal connections, such as compression and flared type, are sensitive to excessive torque. Thus, the failure rate of line connections is dependent on assembly methods and will significantly affect the infant failure region of the failure rate bathtub curve.

When the conductor assembly is used to provide actuator movement, it will be subject to vibration, which changes the torque on plumbing connections and causes metal fatigue. Reliability is thus affected by the routing of the fluid conductor assembly and methods of tubing support. Tube bending and fabrication require special equipment, proper training and experience to acquire reliable connections. Misalignment and excess bending cause strain on the tubing, which can lead to leakage or line failure once in service. Improper deburring and flaring can eventually lead to stress cracks after the lines have been installed. All of these considerations need to be part of the reliability analysis.

22.3.1 Failure Modes of Tubing Assembly

Leakage is the most common failure mode in fluid systems. Connections that incorporate an elastomeric seal such as a BSPP (British Standard Pipe Thread) or a SAE-4 bolt flange offer the highest seal reliability. NPT is the least reliable type of connector for high pressure hydraulic systems because the thread itself provides a leak path. The threads are deformed when tightened and as a result, any subsequent loosening or tightening of the connection increases the potential for leaks. Table 22-3 provides a summary of potential failure modes for tubing assemblies.

Table 22-3. Typical Failure Modes of Tubing Conductor Assemblies

FAILURE MODE	FAILURE CAUSE	FAILURE EFFECT
Stress cracks	- Excessive strain from misalignment - Improper deburring and flaring	System leakage
Metal fatigue	- Vibration	System leakage
Compressed air line leak	- Line misalignment - Vibration - Excessive torque loading on compression fittings	Reduced compressor life
Damaged connector	- Improper torque on fitting - External impact	System leakage
Tubing burst	- Pressure transients - Suddenly applied load	Immediate tubing assembly failure
Fluid leakage	- Chemical incompatibility with fluid - Improper thread sealant	Gradual increase in system leakage
Tube buckling failure	- Tubing support failure	Immediate leakage above system requirements

Plastic tubing will exhibit different failure modes than that for metal tubing. One common failure mode is caused by the change of properties of the plastic over time and/or temperature. The strength and stiffness of many common plastics change dramatically over a relatively small change in temperature. Some plastics such as PVC can become brittle and will shatter when exposed to lower temperatures. The stiffness of many plastic materials changes when exposed to heat. Likewise, the amount of load or stress that some plastics can withstand will change over time (creep). Plastics can also fail when combining stress and chemicals (environmental stress cracking).

22.3.2 Reliability Prediction of Tubing Assembly

The equations for determining the burst and working pressures of a tubing assembly are the same as those derived for pipes in Section 22.2.2. The failure rate of a tubing assembly depends primarily on the connection joints. Accordingly, the base failure rate can be derived from various data sources as 1.33 failures/million operating hours per connection and the total failure rate of the tubing assembly can be estimated with the following equation:

$$\lambda_T = \lambda_{T,B} \bullet C_E \tag{22-5}$$

where: λ_T = Failure rate of tube assembly, failures per million hours

$\lambda_{T,B}$ = Base failure rate of tube assembly, 1.33 failures/million hours

C_E = Environmental factor, see Table 22-4

Table 22-4. Tubing Assembly Environmental Factor, C_E

Operating Environment	Multiplying Factor, C_E
Normal duty, ambient conditions, no vibration or shock	1.0
Heavy duty, random fluid pulsations	1.2
Severe duty, vibration and shock environment	1.4

See Chapter 3 of this Handbook to determine the failure rate of any seals being used in the tubing connections.

22.4 HOSE

A hose assembly is comprised of three basic elements including the inner tube to convey the fluid, reinforcement to withstand the fluid pressure, and outer cover to protect the hose from abrasion and other environmental conditions. The failure rate of a hose assembly depends on its size, temperature, application, media, and pressure. However, even knowing all the operating parameters of a system, it is still difficult to predict the expected service life of a hose because of unknown service and operating conditions. The predicted service life in this section assumes an adequate maintenance program that includes fluid cleanliness and visual inspections for abrasion, heat damage, etc.

22.4.1 Failure Modes of Hose Assembly

Hose assemblies have a finite life, the main factor contributing to failure being service conditions. The following factors will affect the service life of metal hose:

- Pitting corrosion
- High fluid velocity combined with chemical abrasives
- Stress corrosion
- Vibration
- Torsion fatigue
- Tight radius bending and constant motion

Table 22-5 provides a summary of failure modes for hose assemblies.

Operating pressure – The maximum operating pressure within the hose should not exceed the recommended working pressure as specified by the manufacturer. Specified burst pressure of the hose should never be used as the operating pressure. Exposing the hose to surge pressures and pressure spikes above the working pressure of the hose can cause the hose to expand and contract and shorten hose life.

Operating temperatures – Continuous use of a hose at or above the maximum allowable operating temperature will cause deterioration of the hose material and loss of fitting retention. Fluid and ambient temperatures, both static and transient must not exceed the limitations of the hose. A single over-temperature event of sufficient magnitude or prolonged operation at above-normal temperature can permanently damage the seals in the hose assembly resulting in leakage. At higher temperatures, plasticizers tend to leach out of elastomers at a rate depending on the temperature and duration at the high temperature. Proximity of the hose to other components producing high temperatures can contribute to this failure mode. Excessively low temperatures can cause the hose to harden, take on a permanent set, and initiate hose cracking.

Application – An individual hose assembly is designed for a specific range of pressures, flows and temperatures and deviating from these design parameters in actual usage will obviously shorten the design life. Excessive flow velocity, for example, will damage the inner tube, especially at hose bends, and cause premature failure. Improper hose length can also cause failures from hose stretching in a hose assembly that is too short or excessive bending in a hose that is too long for the application. In general, mobile applications face harsher conditions than non-mobile permanent installations due to abrasion.

Table 22-5. Typical Failure Modes of Hose Assemblies

FAILURE MODE	FAILURE CAUSE	FAILURE EFFECT
Metal hose fatigue failure	- High flow velocity - Flexing of corrugations	- Continual increase in size of crack until complete fracture
Excessive shear stress in metal hose	- Twisting the hose during installation	- Circumferential cracks
Irregular cracks in metal hose	- Vibration	- Leakage and eventual failure
Inner tube deterioration	- Elevated temperature	- Eventual hose failure - Contaminants entering system
Excessive fluid temperature	- High fluid temperature caused by excessive flow velocity	- Premature hose failure
Fractured hose	- Excessively small bend radius - Hose bend immediately behind the coupling	- Reduced ability to withstand internal pressure
Hose leakage	- Continuous exposure to high temperature - Chemical deterioration	- Loss of hose flexibility
Inner tube failure	- Inadequate compatibility with fluid	- Eventual hose failure

External forces – Flexing the hose to less than its minimum bend radius, twisting or kinking the hose will reduce hose life. Evaluating a hose for reliability must therefore include an examination of fittings and adapters designed to prevent the impact of external forces. Excessive abrasion can damage the hose cover, accelerating hose failure.

Fluid and environmental compatibility – The expected life of a hose assembly depends on the chemical resistance of the tube, cover, 0-ring fitting and other hose components and compatibility with the fluid being used and the environment. A fluid incompatible with a hose will shorten the design life of the hose. Consideration must be given to the chemical composition of the environment surrounding the fluid conductor assembly as well as the media being conducted including abrasive particles and corrosive properties when estimating the failure rate. Ultraviolet light, ozone, salt water and various chemicals can shorten hose life

Hose configuration – The size of the hose assembly components must be adequate to keep pressure losses to a minimum and avoid damage to the hose due to heat generation or excessive turbulence.

22.4.1.1 Metallic Hoses

There are several factors associated with flexing that affect the service life of corrugated metal hose. Service life may be affected by factors external to the metal hose assembly such as the chemical composition of the environment surrounding the hose assembly as well as the media being transferred.

Turbulent flow of abrasive chemical media over the alloy surface of the hose may cause accelerated corrosion or erosion-corrosion. Liquids or gases that have suspended solid particles will wear or remove the oxide protective film of the hose and leave the alloy exposed and more susceptible to corrosion.

Fatigue is another failure mode to be considered. The flexing of the corrugations during hose operation can cause a failure of a progressive nature. Stress generated by flexure, pulsation, torsion, vibration and flow induced vibration are some other causes for fatigue failure. Applications where the flow of a liquid or gas is above manufacturer specifications and a liner is not incorporated into the hose assembly can result in premature fatigue failure. The high flow velocity causes the corrugations to vibrate at a high frequency and, if the vibration is near the natural frequency of the hose, failure can occur very quickly.

22.4.1.2 Non-metallic Hoses

The most common causes of a non-metallic hose assembly failure include:

- flexing the hose to less than the specified minimum bend radius
- twisting, pulling, kinking, crushing, or abraiding the hose
- operating the fluid system above maximum or below minimum temperature
- exposing the hose to surges in pressure above the maximum operating pressure
- intermixing hose, fittings and other parts that are not compatible

All hoses are rated with a maximum working temperature. Exposure to continuous high temperatures can lead to the hose losing its flexibility. Exceeding these temperatures can reduce hose life by as much as 80%. When hoses are exposed to high external and internal temperatures simultaneously, there will be a significant reduction in hose service life.

Bending a hydraulic hose in more than one plane results in the twisting of its wire reinforcement. A twist of five degrees can reduce the service life of a high-pressure hydraulic hose by as much as 70% and a seven degree twist can result in a 90% reduction in service life. Multi-plane bending is usually the result of inadequate clamping where the hose is subjected to mechanical motion.

Operating conditions have a direct effect on the service life of a hose assembly. Temperature extremes can accelerate the aging of the hose's rubber tube and cover. Frequent and extreme pressure fluctuations will accelerate the fatigue of hose material.

22.4.2 Reliability Prediction of Hose Assembly

The failure rate of a hose is very hard to determine because of the varied operating conditions. For example, failure rate data indicates that approximately 80% of hose failures are attributable to external physical damage through pulling, kinking, crushing, or abrasion of the hose. Abrasion caused by hoses rubbing against each other or surrounding surfaces is a very common type of hose damage. In operation, hoses on diesel engines will have higher failure rates than those used on gasoline engines since diesel engines run hotter than gasoline engines, temperature accelerates chemical reaction and chemical deterioration causes hose failure.

The failure rate of a hose in its operating environment can be determined with the following equation:

$$\lambda_H = \lambda_{H,B} \bullet C_T \bullet C_E \tag{22-6}$$

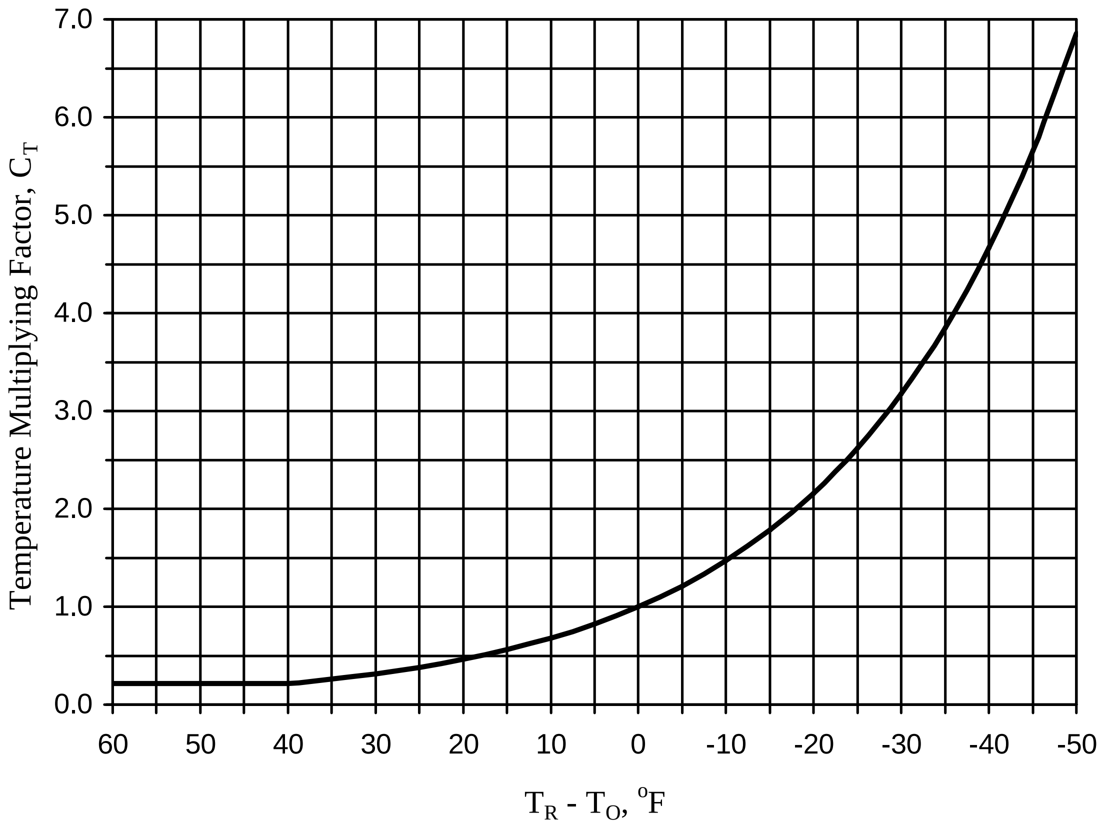
where:

λ_H = Failure rate of hose assembly, failures per million hours

$\lambda_{T,B}$ = Base failure rate of hose assembly, 1.85 failures/million hours

C_T = Temperature factor, see Figure 22.1

C_E = Environmental factor, see Table 22-6



$$C_T = \frac{1}{2^t}$$

Where: $t = \frac{(T_R - T_O)}{18}$ for $(T_R - T_O) \leq 40\ ^oF$

and: $C_T = 0.21$ for $(T_R - T_O) > 40\ ^oF$

T_R = Rated Temperature of Hose, ^oF

T_O = Operating Temperature of Hose, ^oF

Figure 22.1 Temperature Multiplying Factor, C_T

Table 22-6. Hose Assembly Multiplying Factor, C_E

Environment	Environmental Factor, C_E
Normal duty, ambient conditions, no vibration or shock	1.0
Heavy duty, random fluid pulsations	1.2
Severe duty, vibration and shock environment	1.5

22.5 REFERENCES

In addition to specific references cited throughout Chapter 22, other references included below are recommended in support of performing a reliability analysis of fluid conductors.

88. Reliability Analysis Center, “Nonelectronic Parts Reliability Data”, NPRD-95

92. PST =>Solutions! ,Volume 3, October 1996

CHAPTER 23

MISCELLANEOUS PARTS

23.0 TABLE OF CONTENTS

23.1 INTRODUCTION

The failure rate of a total mechanical or hydraulic system depends upon the active components of the system such as pumps, valves, gears and springs. The system will normally contain passive or non-moving parts as well, such as pins, couplings, nuts and bolts. The reliability of these passive or non-moving parts can be very hard to predict partly because their individual failure rates are small and little failure rate data have been compiled. A structural analysis of each part is often not warranted; yet, their cumulative failure rate data must be considered in the total reliability analysis. This chapter provides some guidance on estimating miscellaneous part failure rates. It also provides the appropriate section in this Handbook for specific mechanical parts where additional guidance on evaluating the part for reliability can be obtained.

The design evaluation procedures in other chapters of this Handbook provide a failure rate based on a detailed understanding of the design and operating environment of the component being considered. In contrast, the items in this Chapter are normally passive parts with limited availability of published failure rate data. The provided failure rate may require a modification based on personal experience with the part in the system being analyzed and its operating environment.

23.2 AXLE

An axle is a non-rotating member which carries no torque and is used to support rotating wheels and pulleys. The failure rate of an axle is dependent on material properties, weight of the wheel or pulley and its operating environment including shock and vibration. It is difficult to estimate a failure rate for an axle without performing a detailed stress analysis, since an axle is normally designed for infinite life and the failure rate almost totally dependent on operating conditions.

In the process of evaluating an axle for reliability, it is important to consider the total axle system. A bent axle, for example, will cause subsequent failure of bearings, bushings, wheels and other components that are mounted on the axle.

23.2.1 Axle Failure Modes

The following table includes some typical failure modes to consider when evaluating an axle for reliability.

Table 23-1. Typical Failure Modes for an Axle

FAILURE MODE	FAILURE CAUSE	FAILURE EFFECT
Fatigue failure at stress raiser	Shock, vibration	Crack growth, eventual breakage
Bent axle	Bending stresses	Axle mounted component failure
Unstable rotation	Vibration	Axle fatigue and breakage
Uneven rotation	Loose mounting assembly	Axle mounted component failure

23.2.2 Axle Failure Rate

A stress-strength analysis is required to accurately estimate the failure rate of an axle for the intended operating environment. For estimating purposes, a typical failure rate for a shaft will be 0.01 failures/million hours. Consideration of this failure rate must include the evaluation of the parts mounted on the shaft that may be sensitive to axle failure mechanisms. The procedures in Chapter 20 can be used to determine the failure rate for small axles and shafts that are a part of pump, motor, compressor or other component.

23.3 BOLT

Structurally, a bolt acts as a pin to keep two or more parts from slipping relative to each other, or to clamp two or more pieces together. In either case the bolt must be tightened properly if it is to perform its intended function. Thus, the failure rate of a bolted joint is very much dependent on the assembly or rework process.

23.3.1 Bolt Failure Modes

The possible failure modes of a bolted joint depend on the application. In fluid systems, for example, a failed bolted joint may result in leakage, the definition of leakage, in turn, dependent on the type of fluid being contained and the criticality of the leak. For machinery that is subject to shock and vibration, self-loosening, corrosion and fatigue are failure modes of concern. Table 23-2 lists some of the potential failure modes of a bolted joint.

Table 23-2 Typical Failure Modes for a Bolt

FAILURE MODE	FAILURE CAUSE	FAILURE EFFECT
Loose bolt	Insufficient clamp force or preload, excessive shock and/or vibration	Joint movement, fluid leakage, eventual fatigue failure
Plastic deformation	Applied force overload, excessive bearing pressure	Tensile failure
Sheared bolt	Repeated loading, vibration	Fatigue failure
Stripped threads	Excessive preload	Eventual bolt failure

23.3.2 Bolt Failure Rate

Approximately 75 different factors affect the tension created in a single bolt when a torque is applied. Still more variables must be considered when a group of bolts in an assembly is tightened. Fortunately, the design process of bolted joints is well established for normal machine designs and the failure rate quite low. A typical failure rate of a bolt based on published failure rate data is 0.12 failures per million hours. This failure rate can usually be assumed for the bolted system including the washers, nuts and threadlocking devices. Table 23.3 provides recommended multiplying factors to adjust the base failure to consider the operating environment.

Table 23-3 Service Factors for a Bolt

LOADING	NORMAL VIBRATION	CONTINUOUSLY HIGH VIBRATION
Light loading (office type equipment)	1.00	1.50
Medium loading (compressors, large motors)	1.25	1.88
Heavy loading (Production equipment, trucks, aircraft, shipboard)	1.67	2.50
Very heavy loading (construction equipment, tanks	2.50	3.75

Just as failure modes are dependent on the application of the equipment, so must the failure rate be adjusted per the individual machine design and the number of bolts being used. For a complex machine design where weight or potential deformation becomes a factor, a finite element or stress analysis is usually performed. Detailed procedures for evaluating a bolt or other threaded fastener for reliability are contained in Chapter 16.

23.4 BUSHING

A mechanical bushing is a type of vibration isolator providing an interface between two parts, dampening transmitted energy from one part to the other. A bushing is also used as a bearing that is inserted into a housing to provide a bearing surface for rotary applications. This is the simplest type of bearing, comprising just a bearing surface and no rolling elements. The shaft in contact with the bearing slides over the bearing surface.

There are many different bushing designs for particular applications. Sleeve bushings or journal bearings are used to protect steel shafts from wear due to a combination of pressure and rotating motion. Slide bushings are used where accurate linear movement of a large object is required. A ball bushing permits rotation, sliding motion or both. A bushing blank is used as an insert in a hole to reduce the inside diameter of the hole and protect the surrounding body structure from damage resulting from vibrational and loading stress. Many bushings are press fit into structures and can be replaced after wear without requiring replacement of the entire structure.

Chapter 7 contains the procedures for evaluating a plain bearing for reliability including the failure modes to be considered. These same procedures can be used for determining the failure rate of a bushing.

23.4.1 Bushing Failure Modes

Failure modes of a bushing depend on the application of the bushing. Mechanical bushings may be a fixed or removable cylinder metal lining used to constrain, guide, or reduce friction, or they may be a threaded adapter to permit joining of pipes with different diameters. In some applications leakage past the bushing can be a potential failure mode. Some leakage can be expected since by design there is a gap between the bushing and shaft. Bushings are also used in chain drives providing separation between lubrication and contaminants. Chapter 21 contains failure mode information on these types of bushings. Table 23-4 provides typical failure modes for bushings to be considered in a reliability analysis.

Table 23-4. Typical Failure Modes for Bushing

FAILURE MODE	FAILURE CAUSE	FAILURE EFFECT
Loose on shaft	Vibration	Loose coupling
Corrosion	Dissimilar materials/fluid	Shaft failure
Structural failure	Misalignment, excessive load	Bushing failure
Damaged set screw	Over tightening	Bushing failure
Loss of radial or axial movement	Loss of lubrication	Damaged grease seal

23.4.2 <u>Bushing Failure Rate</u>

The failure rate of a bushing depends on many design factors including the yield strength of the material, hardness, elastic modulus, thermal conductivity, and galling resistance. Application variables include load, shaft speed, surface contaminants, lubrication and clearance. In highly loaded, slowly rotating systems, the maximum bearing pressure is considered a key design criterion. Maximum pressure is the highest compressive stress a bushing can sustain without failure due to cracking or spalling. Normally, this maximum pressure should be no greater than twice the yield strength of the bushing material. For less highly loaded but more rapidly rotating systems, the pressure-velocity (PV) limit is a factor determining failure rate. In this case pressure is the total weight supported by the shaft converted to a compressive load, and velocity is the rotational speed converted to a velocity.

Typical published failure data for a bushing would indicate a failure rate of 0.72 failures/million hours. Obviously, this rate needs to be adjusted depending on personal experience with the application. In some cases the bushing is an integral part of a shaft. Chapter 20 provides guidance on determining a failure rate for this application. Chapter 21 contains information on the use of bushings in a chain linkage.

23.5 CAM MECHANISM

A cam is a projecting part of a rotating wheel or shaft that strikes a lever at one or more points on its circular path. The cam can be a simple tooth to deliver pulses of power or an eccentric disc or other shape to produce a smooth oscillating motion in the follower. A cam-follower is the lever or output link making contact with the cam. The follower can be in the form of a roller, it may be a flat face or spherical in shape.

Cam-followers are fitted between the camshaft lobe and the driven mechanism such as the end of a valve stem or push rod which it operates. Cam-followers reduce wear on the camshaft and ensure that the driven mechanism operates smoothly. Follower size is determined by the total useful force requirement.

After the cam-follower displacement curve is established, the unit is sized for long service life with minimum weight and size. Cam size with respect to pressure angle and radius of curvature is an important design consideration. The pressure angle at any point on the profile of a cam is the angle between the direction where the follower wants to go at that point and where the cam wants to push it. It is the angle between the tangent to the path of follower motion and the line perpendicular to the tangent of the cam profile at the point of cam-roller contact. If the pressure angle exceeds 30 degrees for a radial cam for example, the force exerted on the cam may too great resulting in reduced life and accuracy.

23.5.1 Cam Mechanism Failure Modes

The primary failure mode of a cam-follower is wear. A cam does not necessarily operate at a constant drive speed and high speed cam-follower mechanisms often exhibit substantial drive speed fluctuations. These fluctuations are caused by changes in the angular velocity of the driving mechanism due to load variations, and backlash between the driving mechanism and the cam. The following table provides failure modes that need to be considered in a reliability analysis of cam mechanisms.

Table 23-5. Typical Failure Modes for a Cam Mechanism

FAILURE MODE	FAILURE CAUSE	FAILURE EFFECT
Cam follower error	Excessive pressure angle, high torque during follower acceleration	Cam mechanical error
Drive speed fluctuation	Load variation and/or backlash	Rapid wear and mechanical error
Loose shaft/cam contact	Loose shaft pin	Loss of mechanical motion
Worn cam- follower contact	Excessive load, binding	Cam mechanical error and loss of follower contact
Cam/follower material fracture	Excessive load	Loss of follower contact

Chapter 18 of this Handbook contains some additional guidance for evaluating the design of slider-crank mechanisms and the failure modes to be considered.

23.5.2 Cam Mechanism Failure Rate

The failure rate of a cam mechanism from various data sources is 6.1 failures per million hours. This failure rate is dependent on the operating speed of the cam mechanism; friction and wear rate between the driving mechanism, cam, and follower; and the material properties of the individual parts. Equations contained in Chapter 9 can be used to estimate the wear rate of a cam mechanism. Chapter 18 of this Handbook contains some additional guidance for evaluating the design of slider-crank mechanisms.

23.6 FITTING

Fittings are designed to seal fluid within a fluid system. A multitude of options exist for fittings in fluid systems. One type of fitting is an all-metal fitting which relies on metal-to-metal contact. Another design employs O-rings to contain pressurized fluid. In either design, tightening threads between the mating halves of the fitting or between the fitting and component port, forces the two mating surfaces together to form a seal. Quick disconnect couplings can significantly improve the serviceability of fluid systems by saving maintenance time. Couplings incorporating valves can automatically shut off fluid flow and maintain system pressure.

It is important to consider chemical compatibility of the coupling materials with respect to fluid media and the strength of coupling materials with respect to functional requirements of the application. Plastic couplings are lightweight and used for many low pressure applications. Metal couplings are typically chosen when high fluid pressure, vibration and other more severe operating conditions call for greater strength and durability.

23.6.1 Fitting Failure Modes

Leakage is the primary failure mode of fluid systems. Compression fittings that incorporate an elastomeric seal offer the best seal reliability. NPT is the least reliable type of connector for high pressure hydraulic systems because the thread itself provides a leak path. The threads become deformed when tightened and subsequent loosening or tightening of the connection increases the probability of leakage. Flared fittings are inexpensive and simple in design. The metal-to-metal seal of the flare is subject to seepage at high fluid pressures. Some flare joint fittings employ a conical washer between the nose and flare to reduce the probability of leakage. A common cause of leakage from flare joints is incorrect torque. Insufficient torque results in inadequate seat contact and excessive torque can cause cold working damage to the tube and connector. Torque can also be affected by vibration, vibration also causing metal fatigue of the connector.

The reliability of a fitting employing an elastomeric seal depends on fluid compatibility and fluid temperature. Fluid operating temperatures above 85^{o}C can cause damage to many seals. Table 23-6 provides some typical failure modes to be considered in evaluating a fitting for reliability. See Chapter 3 for a listing of failure modes for the seal itself.

Table 23-6. Typical Failure Modes of a Fitting

FAILURE MODE	FAILURE CAUSE	FAILURE EFFECT
Fluid seepage	- Damaged sleeve or ferrule - Damaged O-ring - Incorrect assembly	Eventual fitting failure
Fractured plastic housing	- Shock, vibration	Fitting failure
Leakage	- Fluid incompatibility	- Generation of contaminants in system - Eventual fitting and component failure
Damaged flare joint, fluid leakage	- Incorrect tightening during assembly	Fitting failure
Damaged seal	- Fluid temperature above acceptable limit	System leakage

23.6.2 <u>Fitting Failure Rate</u>

The failure rate of a fitting can be estimated as 1.3 failures per million hours for a threaded fitting and 2.4 failures per million hours for a quick disconnect fitting based on published failure rate data. However, failure modes as discussed in the previous section need to be addressed in estimating the failure rate. NPT ports, for example, will have a higher failure rate for high-pressure hydraulic systems because the thread itself provides a leakage path. The evaluation procedures in Chapter 3 can be used to estimate the reliability of an individual O-ring seal. If the fitting contains a shut-off valve, refer to Chapter 6 to estimate its reliability.

23.7 FLYWHEEL

Flywheels store kinetic energy as a mechanical battery and are used to smooth the variations in shaft speed that are caused by loads or power sources that vary in a cyclic fashion.

Overall performance of the flywheel depends on a sufficient moment of inertia, matching the power source to the load, and resulting performance requirements. One of the main considerations in flywheel design is balancing. By design, flywheels are devices with large inertia and they must, therefore, be balanced to remove eccentric loading and reduce the loading on bearings and other components.

The spinning of a flywheel creates stress at the inner hub connection which can lead to fracture Estimating the failure rate of a flywheel must therefore consider the flywheel velocity. Rotating parts such as a flywheel can be simplified to a rotating ring to determine stress levels.

23.7.1 Flywheel Failure Modes

Flywheels develop large stresses at their inter hub connection due to dynamic forces caused by spinning. These stresses can lead to failure. Table 23-7 includes some failure modes to consider when evaluating flywheel reliability.

Table 23-7. Typical Failure Modes for a Flywheel

FAILURE MODE	FAILURE CAUSE	FAILURE EFFECT
Flywheel loosened from shaft	Broken shaft/wheel connection	Uncontrolled release of energy
Flywheel fracture	Centrifugal forces	Complete loss of output energy
System vibration	Unbalanced flywheel	Shaft/bearing damage

23.7.2 Flywheel Failure Rate

The failure rate of a flywheel depends on a design balance between rotational speed, material density and tensile strength. Flywheel performance depends on an optimum energy-to-mass ratio and the flywheel must therefore spin at the maximum possible speed since kinetic energy increases only linearly with mass but increases as the square of rotational speed. However, a rapidly rotating object is subject to centrifugal force that can create sudden fatigue failure. While dense material can store more energy than low density material, the flywheel is also subject to higher centrifugal force and thus fails at lower rotational speed than one comprised of low density material. Therefore, the tensile strength of the flywheel material becomes an important reliability consideration.

A base failure rate of 0.2 failures per million hours can be assumed for a flywheel based on published failure rate data. Table 23-8 provides service factors to modify the expected failure rate of a flywheel for the expected operating environment.

Table 23-8. Service Factors for a Flywheel

Driven Machinery	Normal Torque, Low Velocity	Normal Torque, High Velocity	High Torque, Low Velocity	High Torque, High Velocity
Uniform	1.1	1.3	1.2	1.4
Light shock	1.2	1.4	1.3	1.5
Medium shock	1.3	1.5	1.4	1.6
Heavy shock	1.4	1.6	1.5	1.7

23.8 HINGE

A hinge is a joint that holds two parts together so that one can swing relative to the other. Most hinges are separate parts which are added to a completed assembly for further installation. Living hinges, on the other hand, are molded into a part during the forming process. A Butt Hinge is composed of two plates attached to abutting surfaces of a door and door jamb and joined by a pin. A Gravity Hinge closes automatically as a result of the weight of a door to which it is attached. A Strap Hinge is a surface mounted hinge with long flaps of metal on each side, by which it is secured to a door and adjacent post or wall. A Pintle Hinge pivots about an upright pin or bolt. A Spring Loaded Hinge can automatically open or close a door, eliminating the need for a latch to secure the door in its resting position.

Rotation of a hinge is permitted along only one axis, and during bending the neutral axis does not change length. Therefore, in the elastic case, the hinge can be considered as a beam undergoing pure bending, and the neutral axis forms a circular arc. Built-in plastic hinges generally undergo some plastic deformation. The hinge must stretch to compensate for plastic deformation, the hinge undergoing tension as well as bending.

23.8.1 Hinge Failure Modes

Most hinge failures are caused by excessive loading of the part they are supporting. Typical failure modes for a hinge are listed in Table 23-9. In the event of a spring loaded hinge, see Chapter 4, Section 4.2 for specific failure modes of a mechanical spring.

Table 23-9. Typical Failure Modes for a Hinge

FAILURE MODE	FAILURE CAUSE	FAILURE EFFECT
Bent pin	Excessive loading	Hinge Failure
Weak spring	Shock loads	Hinge failure
Damaged plating	Corrosion	Eventual hinge failure
Noisy hinge	Contaminants, bent pin, loss of lubrication	Eventual hinge failure
Jammed hinge	Excessive loading, damaged pin	Hinge failure

23.8.2 Hinge Failure Rate

The failure rate of a living hinge depends on whether strain in the hinge is purely elastic or whether plastic strain is also present. Plastic strain is further divided into pure bending strain and bending plus tensile strain. The design reliability of a hinge depends on the type of stress it is expected to carry in service.

The failure rate of both a metal hinge and a living hinge can be expected to be 0.5 failures per million cycles for normal applications. Chapter 18 provides equations and procedures to evaluate a slider crank mechanism for reliability considering the operating environment in determining the failure rate. Procedures for determining the failure rate of a hinge spring are contained in Chapter 4.

23.9 KEYS AND PINS

Keys and pins are used on shafts to secure rotating elements such as gears and pulleys. Keys are used to enable the transmission of torque from shaft to the gear, pulley or other shaft supported element. Pins are used for axial positioning and for transfer of torque and/or thrust. Typical types of pins include cotter pins, dowel pins, hitch pins, taper pins and roll pins. The Woodruff key is often used to position a wheel against a shaft shoulder.

23.9.1 Key and Pin Failure Modes

Typical failure modes to be considered in the evaluation of reliability for keys and pins are included in the following table.

Table 23-10. Typical Failure Modes for Keys and Pins

FAILURE MODE	FAILURE CAUSE	FAILURE EFFECT
Sheared pin or key	Loose part on shaft	Damage to higher indenture level component
Loose pin or key	Enlarged holes	Loose part on shaft
Damaged pin	Shock/ vibration	Misalignment of part
Damaged keyway	Excessive axial loading	Axial movement of mounted part

23.9.2 Key and Pin Failure Rate

Pins are subject to loading in shear. Fatigue failure begins with a small crack. Stress concentration factors for keyways depend on the fillet radius at the bottom and ends of the keyway. Published failure rates for keys and pins indicate a failure rate for keys and pins of 0.35 failures/million hours.

23.10 PILLOW BLOCK

Pillow blocks are comprised of a bearing mounted in a housing and typically used to provide load support for a rotating shaft. In many belt drive systems the drive mechanism that couples the motor to a fan or blower is retained by pillow block bearings. Chapter 7 provides guidance on evaluating the bearing for reliability. As with most bearings, pillow block bearings are specified in terms of L_{10} life and the procedures in Chapter 7 can be used to modify the L_{10} life for the intended pillow block application.

23.10.1 Pillow Block Failure Modes

Pillow blocks are used to support a shaft, the mounting surface being on a parallel plane with the axis of the shaft. Elongated bolt holes in the base of the unit allow for some adjustment of the pillow block. Therefore correct assembly of the pillow block is critical to achieving the expected life expectancy of the pillow block. A pillow block mounted at an angle with respect to the shaft can cause scoring of the shaft, heat build up and early seal leakage. The housing material for a pillow block bearing is typically made of cast iron or pressed steel and failures of the housing itself are almost nonexistent. Typical failure modes are listed in the following table.

Table 23.11. Typical Failure Modes for a Pillow Block

FAILURE MODE	FAILURE CAUSE	FAILURE EFFECT
Worn bearing race	Excessive side loading, misalignment,	Shaft vibration
Bearing seizure	Loss of lubricant, excessive rotational velocity, high temperature	Shaft seizure
Loss of lubricant	Seal leakage	Bearing wear
Damaged housing	Shock, vibration	Damaged pillow block
Loose shaft coupling	Loose set screw	Early shaft wear

See Chapter 7, Section 7.4 for specific bearing failure modes. See Chapter 3, Section 3.2 for specific seal failure modes. See Section 23.13 later in this Chapter on set screws for specific set screw failure modes.

23.10.2 <u>Pillow Block Failure Rate</u>

Although the pillow block is comprised of a standard bearing and a housing, the effect of a misaligned pillow block on the shaft must be considered in evaluating a pillow block for reliability. Typically, a pillow block will experience a failure rate of 5.0 failures per million revolutions. The L_{10} life of the pillow block may need to be adjusted after considering the potential failure modes.

23.11 POWER SCREWS

A power screw is a mechanical device for translating rotational motion to linear motion. Typical applications include the screw for jacks, presses and vises.

The wear life of a power screw is difficult to predict because of the number of variables to consider including load, speed, screw material, surface finish, lubrication, duty cycle, operating temperature, and environmental factors such as vibration and the existence of abrasive and corrosive contaminants. Power screws are subject to both adhesive and abrasive wear. These wear mechanisms are described in Chapter 9. Also, Chapter 16, Section 16.3 provides the equations and procedures for evaluating the stress-strength aspects of a power screw.

Ball screws are a type of power screw. Ball screws allow balls to roll between the screw shaft and nut to achieve high efficiency. Manufacturers usually report the rated load that a given ball screw can exert for 1 million inches of cumulative travel. This is a similar rating method as for the L_{10} life of ball bearings and the same relationship can be used to estimate the life of the particular power screw:

$$\frac{\lambda_{PS}}{\lambda_{PS,B}} = \left(\frac{L_A}{L_S}\right)^3 \qquad \textbf{(23-1)}$$

where:

λ_{PS} = Failure rate of power screw, failures/million hours

$\lambda_{PS,B}$ = Base failure rate of power screw from published life, failures/million hours

L_A = Equivalent radial load, lbs

L_S = Basic dynamic load rating, lbs

This concept of estimating the life of a part with respect to loading is explained in Chapter 7.

23.11.1 Power Screw Failure Modes

The nuts of a power screw system used for moving loads such as in the lead screw of a lathe are usually made of a softer material than that of the screw. This process eliminates the potential problem of material bonding and galling under load pressure if the same materials were to be used. Also, the nut is cheaper than the drive screw thus reducing maintenance costs. The failure rate of the nut can therefore be expected to have a higher failure rate than the screw. For applications that position and support heavy loads such as scaffolding, like materials are sometimes used.

Another failure mode to consider is wear. Plating of a power screw is an important design feature because of the stiction involved under load. The first level of wear on a plated part will be the plating itself. Once the plating has worn away, bare metal is exposed to the environment. For moving load applications, proper lubrication is important to achieve maximum service life. Typical failure modes for a lead screw are listed in Table 23.12.

Table 23.12. Typical Failure Modes for a Power Screw

FAILURE MODE	FAILURE CAUSE	FAILURE EFFECT
Screw bonded to nut	Loss of lubrication	Load can not be moved
	Excessive load pressure	
Scored screw shaft	Misalignment	Eventual power screw failure
Fractured nut	Excessive Pressure-Velocity	Failed power screw operation
Worn screw threads	Foreign contaminants, loading	Unstable power screw operation

23.11.2 Power Screw Failure Rate

The number of variables as described previously affecting the failure rate of a drive screw make it extremely difficult to predict. Without some personal experience with the design of the power screw in its intended operating environment, it is normally best to evaluate the individual parts of the power screw using the procedures in Chapters 9 and 16.

23.12 RIVET

There are two common types of rivets, lap-joint and butt-joint. In a lap-joint rivet the plates to be joined overlap each other and are held together by one or more rows of rivets. In the butt-joint, the plates to be joined are in the same plane and are joined together with a cover plate which is riveted to both plates by one or more rows of rivets. Factors to be considered in evaluating a riveted joint for reliability include type of joint, spacing of rivets, type and size of rivets, hole-size and rivet material.

Both lap-joint and butt-joint rivet assemblies are subject to shear, tension, and crushing.

The shearing stress in the rivet will be:

$$S_S = \frac{P}{A_S} \tag{23-2}$$

where: S_S = Shearing stress, lb/in^2

P = Tensile load on the joint, lbs

A_S = Shear area, in^2

The controlling tensile stress in the plate occurs near the rivet hole:

$$S_T = \frac{P}{(w-d)t} \tag{23-3}$$

where: S_T = Tensile stress, lb/in^2

w = Width of the plate, in

d = Hole diameter, in

t = Plate thickness, in

The crushing stress due to load transfer at the contact between plate and rivet is:

$$S_C = \frac{P}{td} \tag{23-4}$$

where: S_C = Crushing stress level at the area of the rivet, lb/in^2

The total design will include a sufficient number of rivets to ensure that the failure of any one rivet because of improper installation or damage will not result in immediate failure of the structure.

23.12.1 Rivet Failure Modes

The primary failure modes in a conventional riveted joint assembly are rivet shear and plate cracking due to load bearing in the sheet. In lower stress aluminum structures, failure is more frequent in the plate, the failure mode being tension cracks starting at stress concentrations on hole-surfaces. In higher stress applications, the failure mode may be either shear of the rivets or plate cracking.

Rivets may fail by:

- Shearing through a cross section of the rivet

- Crushing

Plates may fail by:

- Tearing along a single line from center of rivet hole to edge of plate
- Shearing along two parallel lines extending from opposite sides of the rivet hole to the edge of the plate
- Tearing between adjacent rivets
- Tensile failure of the plate
- Crushing

23.12.2 Rivet Failure Rate

Riveted structures designed in accordance with manufacturer's specifications and assembled without additional stress raisers can be expected to have a fatigue failure rate of 0.08 failures per million hours. Both the method of making the hole for the rivet and the presence of a countersink affect the fatigue strength. For example, for single-row joints, fatigue strength increases with decreasing pitch; for three-row joints, fatigue strength increases with increasing spacing; and for multi-row joints, fatigue strength increases with increasing numbers of rows. In evaluating rivet joint designs for reliability it is important to recognize the fact that slightly out-of-round holes can increase the failure rate by a factor of 10. Assurance of rivet assembly reliability is normally assured by trial and error experimentation on sample products.

23.13 SETSCREW

Setscrews are special fasteners designed to hold gears and pulleys on shafts. Unlike bolts and tap screws that depend on tension to develop a clamping force, the setscrew depends on compression to develop a clamping force. The resistance to axial or rotary motion of the collar or hub relative to the shaft is called the holding power. This holding power is a frictional resistance between the contact surfaces of the collar and shaft plus the penetration of the setscrew into the shaft. The primary failure mode of the setscrew-shaft combination is a loose setscrew and loss of holding power. Of particular importance to the failure rate of the setscrew is the size of the setscrew and the holding power provided by the clamping action. Holding power is generally specified as the tangential force in pounds. The size of the shaft to be used with a particular size of setscrew will determine the holding power required. Cup-point and cone-point setscrews for example, penetrate the shaft deeper than oval-point or flat-point setscrews.

Setscrew selection is usually based on the setscrew diameter being equal to approximately one-half the shaft diameter. Manufacturer's data provide more exact design criteria. The shape of the screw head affects the seating torque that can be attained because it determines how much torque can be transmitted to the screw. For example, less torque can be transmitted through a slotted setscrew than a socket head setscrew. The holding power of the slotted setscrew is about 45% less than that of the socket head setscrew.

When a setscrew is used in combination with a key, the screw diameter should be equal to the width of the key. In this combination, the setscrew holds the parts in an axial direction only. Torsional load on the parts is then carried by the key. The key should be tight fitting so that no motion is transmitted to the screw. Under high reversing or alternating loads, a poorly fitted key will cause the screw to back out and lose its clamping force.

23.13.1 Setscrew Failure Modes

An important consideration in evaluating a setscrew-shaft combination is the holding power provided by the clamping action. Holding power is generally specified as the tangential force in pounds, the resistance to axial or rotary motion of the collar or hub relative to the shaft. Total reliability of the setscrew-shaft combination is also dependent on the number of setscrews being used. Two setscrews give more holding power than one, but not necessarily twice as much. Holding power is approximately doubled when the second setscrew is installed in an axial line with the first but is only about 30% greater when the screws are diametrically opposed. Normally a displacement of 60 degrees is used when two setscrews need to be installed on the same circumferential line providing a holding power of 1.75 times the holding power of one setscrew.

Another failure mode to be considered in the setscrew-shaft combination is corrosion created by dissimilar materials. A thin film of rust-preventive oil is normally used to protect unplated setscrews, or cadmium or zinc plated setscrews can be used. A summary of failure modes for setscrews is shown in the following table.

Table 23.13. Typical Failure Modes for a Setscrew

FAILURE MODE	FAILURE CAUSE	FAILURE EFFECT
Axial motion of collar or hub	- Shock and vibration resulting in loose setscrew	Loss of shaft motion
Rotary motion of collar or hub	- Shock and vibration resulting in loose setscrew	
Damaged setscrew	- Incorrect torque at installation - Severe operating environment resulting in corroded setscrew	

23.13.2 <u>Setscrew Failure Rate</u>

Setscrews are subject to axial and rotational loading in shear. Published failure rates for setscrews are in the range of 0.35 failures/million hours. Table 23.14 provides some recommended multiplying factors to adjust the base failure rate for the intended operating environment.

Table 23.14. Service Factor Multiplying Factors for a Setscrew

OPERATING ENVIRONMENT	LOW HUMIDITY	HIGH HUMIDITY
Normal duty	1.0	1.2
Random shock and vibration	1.2	1.4
Continuous shock and vibration	1.5	1.8

23.14 WIRE ROPE

A typical wire rope contains wire twisted in one direction to form the strands, and the strands twisted in the opposite direction to form the rope. In the completed rope the visible wires are approximately parallel to the axis of the rope. In another wire rope design the wires in the strand and the strands in the rope are twisted in the same direction so that the outer wires run diagonally across the axis of the rope. This wire rope design provides more resistance to abrasive wear and failure to fatigue but is harder to handle and more likely to kink and untwist.

When a loaded rope is bent over a sheave, the rope stretches like a string, rubs against the sheave, and causes wear of both the rope and the sheave. The amount of wear that occurs depends upon the pressure of the rope in the sheave groove. This pressure is called the bearing pressure given by:

$$p = \frac{2F}{dD} \qquad \textbf{(23-5)}$$

where:

p = Bearing pressure, pounds

F = Tensile force on the rope, lbs/in^2

d = Rope diameter, inches

D = Sheave diameter, inches

The optimum wire rope for a specific application is a difficult challenge because a high rating in one design parameter generally means a lower rating in another parameter. These design parameters include:

- Strength-resistance to breaking - the total load to be considered includes those loads created by abrupt starts and stops and shock loads in addition to the dead weight.

- Resistance to bending fatigue - the sharper the bend, the higher is the probability of fatigue failure. Accelerating the rate of travel also increases the fatigue factor. Close-coupled reverse bending will further increase the probability of fatigue failure. The greater the number of wires in each strand, the greater the resistance of rope to bending fatigue.

- Resistance to vibrational fatigue - vibration in the system is transmitted through the wire rope in the form of shock waves. This energy is absorbed by the various components and can eventually lead to wire failure in the rope.

- Resistance to abrasion - abrasion can occur on drums, sheaves or whenever rope rubs against itself or other material. Abrasion occurs internally whenever rope rubs against itself or other material. Abrasion also occurs internally whenever wire rope is loaded or bent and it weakens the rope by wearing away metal from inside and outside wires. Excessive wear can be caused by faulty sheave alignment, incorrect groove diameters, or improper drum winding.

- Resistance to crushing - crushing is the effect of external pressure on a rope, which damages it by distorting the cross-section shape of the rope, its strands and/or the core. When a rope is damaged by crushing, the wire, strands and core are prevented from moving and adjusting as required during operation. Crushing resistance is the ability to resist and absorb external forces.

23.14.1 <u>Wire Rope Failure Modes</u>

The main issue with the reliability of a wire rope is safety, since a failure of a wire rope can result in catastrophic consequences. The rated capacity of a wire rope is load which a new wire rope may handle under given operating conditions and at an assumed design factor. A design factor of 5 is typical for a wire rope (operating loads not to exceed 20% of the catalog breaking strength). Operating loads need to be reduced when life or valuable property are at risk. A design factor of 10 is usually chosen when wire rope is used to carry personnel (operating loads not to exceed 10 % of catalog breaking strength). The wire rope design must also be reviewed to assure that wire rope attachments have the same working load limit as the wire rope being used. Typical failure modes of a wire rope assembly are listed in Table 23.15.

Table 23.15. Typical Failure Modes for a Wire Rope

FAILURE MODE	FAILURE CAUSE	FAILURE EFFECT
Breakage	Abrupt starts, sudden stops exceeding dead weight load	Loss of load
Fatigue failure	Sharp bending around sheave, vibrational fatigue, installation error, loss of lubricant	Loss of load
Wire crushing	External forces	Strands and core do not adjust resulting in erratic movement
Damaged rope	Faulty sheave alignment	Excessive wear, early failure

23.14.2 Wire Rope Failure Rate

Wire rope manufacturers provide a fatigue diagram for their products similar to an S-N diagram. Assuming the ultimate tensile strength of the wire material is selected for the application including published safety factors, the wire rope can be expected to have a failure rate of approximately 17.5 failures per million operations.

23.15 REFERENCES

In addition to specific references cited throughout Chapter 23, other references included below are recommended in support of performing a reliability analysis of miscellaneous parts.

19. Hindhede, U., et al, Machine Design Fundamentals, John Wiley & Sons, NY, 1983.

39. Shigley, J.E., Mischke, C.R., Mechanical Engineering Design, McGraw-Hill Book Co., NY, 1989.

88. Reliability Analysis Center, “Nonelectronic Parts Reliability Data”, NPRD-95

93. Mechanical Engineers Handbook, Myer Kutz, et al, John Wiley & Sons, 1986

94. Mechanical Designers' Workbook, "Fastening, Joining & Connecting", J. Shigley and C. Mischke, McGraw-Hill 1986

95. Mechanical Designers' Workbook, "Mechanisms", J. Shigley and C. Mischke, McGraw-Hill 1986

96. Mechanical Designers' Workbook, "Power Transmission Elements", J. Shigley and C. Mischke, McGraw-Hill 1986

97. B. Demeulenaere and J. Schutter , "Input Torque Balancing Using an Inverted Cam Mechanism", ASME, 2005

100. Fittings and Flanges, Hydraulics & Pneumatics magazine,

101. John H. Bickford, "An Introduction to the Design and Behavior of Bolted Joints", Marcel Dekker, Inc., 1990

105. OREDA Offshore Reliability Data, 5th Edition Det Norske Veritas, N-1363 Hovik, Norway 2009 ISBN 978-82-14-04830-8

This Page Intentionally Left Blank

CHAPTER 24

DESIGN ANALYSIS OF EQUIPMENT AVAILABILITY

25.0 TABLE OF CONTENTS

24.1 INTRODUCTION

The normal procedure for evaluating equipment reliability and maintainability is the Failure Mode, Effects, and Criticality Analysis (FMECA). The FMECA identifies the failure modes for each component of the system and the effects of these failure modes are then identified through the next higher assembly up to the system level. The severity and the occurrence probability of the failure mode are then determined. The maintenance philosophy of the equipment and procedures for maintaining the equipment are normally established using a Reliability Centered Maintenance (RCM) analysis. Fault paths of equipment performance are identified using a Fault Tree Analysis (FTA) so that any potential hazards or system level faults can be corrected through equipment redesign early in the design phase.

The FMECA, FTA and RCM are normally performed as individual analysis efforts and the total effort is an extremely time consuming process. The contribution of these analysis procedures is questionable. They are normally a contractual requirement rather than being applied as a design tool. As a result, the process is not begun during the early stages of design development and by the time all of the analyses are completed, the effect on design improvement is negligible. Other common shortcomings include the failure to consider the effect of system operating conditions at the component level, failure to properly investigate the internal features of mechanical components, and failure to consider the interaction of mechanical assemblies. The isolated performance of the analysis efforts results in inadequate inputs to the design process with ineffective results in reliability enhancement.

The Reliability, Maintainability and Availability (RMA) analysis procedure contained in this Chapter is designed to streamline the analysis process permitting effective contribution to product reliability. The RMA evaluates a design as to its projected performance considering Mean Time Between Failure (MTBF), Mean Time To Repair (MTTR) and Operational Availability (A_O). With this RMA procedure, the process can be completed using one single worksheet rather than completing the FMECA, FTA and RCM procedures individually. The RMA procedure outlined in this chapter is designed to be used in conjunction with other Handbook chapters that identify failure modes and effects and determine their probabilities of occurrence.

24.2 RMA PLANNING

The usefulness of the RMA as a design tool and in the decision making process is dependent upon the effectiveness with which problem information is communicated for early design attention. To be effective, the RMA must be iterative to correspond with the nature of the design process itself. The extent of effort and sophistication of approach used in the RMA will be dependent upon the nature and requirements of the individual program. This makes it necessary to tailor the RMA to each individual development program. Tailoring requires extensive planning so that regardless of the degree of sophistication, the RMA will contribute meaningfully to program decisions regarding the feasibility and adequacy of a design approach.

Planning the RCM task involves system description development, worksheet design, procedure implementation, worksheet modifications to reflect design changes, and results analysis to provide design guidance. Establishing ground rules for conducting the analyses, identifying the lowest indenture level of analysis, identifying a coding system, and defining system failures all need to be considered in RMA analysis planning. Planning the RMA analysis effort will prevent the duplication of efforts within the same development program.

Timeliness is another important factor in assuring effective implementation of the RMA analysis. While the objective of the RMA is to identify all modes of failure within a system design, its first purpose is the early identification of all catastrophic and critical failure possibilities so they can be eliminated or minimized through design correction at the earliest possible time. Therefore, the RMA should be initiated as soon as preliminary design information is available at the higher system levels and extended to the lower levels as more information becomes available on the items in question.

Although the RMA is an essential reliability task, it also provides information for other purposes. The RMA is used for assurance of system maintainability, safety, survivability, and efficient logistics support.

24.2.1 Worksheet Formats

RMA worksheets that organize and document the analysis need to be designed to optimize understanding of failure modes and effects at all indenture levels of the system. A description of worksheet entries is included in Section 24.4.3 and worksheet examples are included in Section 24.6. It is important that these examples be modified for the particular system being analyzed.

24.2.2 Ground Rules and Assumptions

Ground rules identify the RMA analysis approach to be used including hardware analysis, functional analysis or combination analysis (see Section 24.4.1), the lowest indenture level to be analyzed and general statements of what constitutes a failure of the equipment in terms of performance criteria and allowable limits. Every effort should be made to identify and record all ground rules and analysis assumptions prior to initiation of the analysis. These ground rules provide a consistent approach when completing the worksheets. It must be recognized that ground rules and analysis assumptions may need to be altered for any item if design requirements are modified.

24.2.3 Indenture Level

The indenture level applies to the system hardware or functional level at which failures are postulated. The lowest indenture level of analysis is usually established at the lowest replaceable assembly of the system such as an actuator, valve or pump. The objective is to identify the failure modes of all parts that can contribute to an equipment failure such as an O-ring or spring. Failure modes at this level are normally included at the replaceable part level. Examples are included in the worksheets in Section 24.6.

24.2.4 Coding System

A coding system is established for consistent identification of system functions and equipment and for tracking failure modes. If a functional block diagram is available for the system, the coding system of that diagram should be used for the worksheets. Any coding system based upon the hardware breakdown structure can be used that provides complete visibility of each failure mode. The same coding system must be used for all worksheets to permit tracking of the failure modes.

24.2.5 Failure Definition

General statements of what constitutes an equipment failure of the item in terms of performance parameters and allowable limits for each specified output needs to be established. Failure definitions need to consider the equipment operating environment as well as design parameter limits.

24.3 GENERAL PROCEDURE

The RMA systematically examines the equipment to the lowest established indenture level and identifies all potential failure modes up to the total system level. In the early stages of design when system definitions and functional descriptions are not available, the initial analysis is performed to the lowest possible indenture level. When system definitions and functional definitions become available, the analysis is extended to the lowest established indenture level.

24.3.1 System Definition

System definition requires a review of all descriptive information available on the system to be analyzed. The following is representative of the information and data required for system definition and analysis.

- **Technical Specifications and Development Plans** - Technical specifications and development plans generally describe what constitutes and contributes to the various types of system failure. These will state the system objectives and specify the design and test requirements for operation, reliability, and maintainability. Detailed information in the plans will provide operational and functional block diagrams showing the overall functions the system must perform for successful operation. Time diagrams and charts used to describe system functional sequence will aid in determining the time-stress as well as feasibility of various means of failure detection and correction in the operating system. Acceptable performance limits under specified operating and environmental conditions are developed for the system and equipments.

Mission and environmental profiles describe equipment performance requirements in terms of functions and tasks to be performed related to the anticipated environments for each mission phase and operating mode. Function-time relationships from the time-stress relationships of the environmental conditions can then be established. A definition of the operational and environmental stresses the system is expected to undergo, as well as failure definitions, must be developed for an effective RMA analysis.

- **Trade-off Study Reports** - These reports, if available, help to identify areas of marginal and state-of-the-art design and explain any design compromises and operating restraints. This information will aid in determining the possible and most probable failure modes and related causes in the system.

- **Design Data and Drawings** - Design data and drawings identify each item and the item configuration that perform each of the system functions. System design data and drawings will usually describe internal and interface functions of the system beginning at system level and progressing to the lowest indenture level of the system. Design data will usually include either functional block diagrams or schematics that will facilitate construction of reliability block diagrams.

- **Reliability Data** - The determination of the possible and probable failure modes requires an analysis of reliability data on the item selected to perform each of the system internal functions. It is always desirable to use reliability data resulting from reliability tests run on the specific equipment to be used with the tests performed under the identical conditions of use. When such test data are not available, procedures in this Handbook can be used to estimate the reliability of the part for the intended operating environment. Reliability data from MIL-HDBK-217 for electronic parts or from operational experience and tests performed under similar use conditions on items similar to those in the systems can be used.

24.3.2 RMA Process

The RMA is most effective if initiated as an integral part of the early design process and updated to reflect design changes. This process will prove beneficial at design review meetings. The RMA will also be useful in defining special test considerations, quality inspection points, preventive maintenance actions, operational constraints, useful life, and other pertinent information and activities necessary to minimize failure risk. The following discrete steps are used in performing the RMA:

a. Define the system to be analyzed. Complete system definition includes identification of internal and interface functions, expected performance at all indenture levels, system restraints, and failure definitions. Functional narratives of the system should include descriptions of each mission in terms of functions which identify tasks to be performed for each mission phase and operational mode. Narratives should describe the environmental profiles, expected mission times and equipment utilization, and the functions and outputs of each item.

b. Construct block diagrams. Functional and reliability block diagrams which illustrate the operation, interrelationships, and interdependencies of functional entities should be obtained or constructed for each item configuration involved in the system's use. All potential item and interface failure modes are identified and their effect on the immediate function or item and on the system defined. Each failure mode in terms of the worst potential consequences which may result is identified and a severity classification category assigned.

c. Identify failure detection methods and compensating provisions for each failure mode. Corrective design or other actions required to eliminate the failure or control the risk can be identified for design review and corrective action.

d. Identify effects of corrective actions or other system attributes, such as requirements for logistics support. Document the analysis and summarize the problems which could not be corrected by design and identify the special controls which are necessary to reduce failure risk.

24.3.3 Severity Classification

Severity classifications are assigned to provide a qualitative measure of the worst potential consequences resulting from item failure. A severity classification is assigned to each identified failure mode and each item analyzed in accordance with the severity classification categories defined as follows:

Category I - Catastrophic - A failure which may cause death or total system loss

Category II - Critical - A failure which may cause severe injury, extensive property damage, or major system damage that results in loss of mission *

Category III - Marginal - A failure which may cause minor injury, minor property damage, or minor system damage that results in mission delay or degradation.

Category IV - Minor - A failure not serious enough to cause injury, property damage or mission delay or degradation, but which will necessitate repairs at a later time.

* In some cases the loss of mission may be sufficiently severe so that it may be classified as a Category I failure.

24.4 RMA ANALYSIS PROCEDURE

24.4.1 Analysis Purpose and Approach

The purpose of the RMA is to study the effects of item failure on system operation and to classify each potential failure according to its severity. Variations in design complexity and available data will generally dictate the analysis approach to be used. There are two primary approaches for accomplishing an RMA. One is the hardware approach which lists individual hardware items and analyzes their possible failure modes. The other is the functional approach which recognizes that every item is designed to perform a number of functions that can be classified as outputs. The outputs are listed and the failure modes that can contribute to an unsuccessful output are analyzed. For complex systems, a combination of the functional and hardware approaches may be considered. The RMA may be performed as a hardware analysis, a functional analysis, or a combination analysis and may be initiated at either the highest indenture level and proceed through decreasing indenture levels (top-down approach) or at the part or assembly level and proceed through increasing indenture levels (bottom-up approach) until the RMA for the system is complete.

The functional approach is normally used when hardware items cannot be uniquely identified or when system complexity requires analysis from the initial indenture level downward through succeeding indenture levels. The functional approach is normally utilized in an initial indenture level down fashion (top-down approach); however, it can be initiated at any level of indenture and progress in either direction. Each identified failure mode is assigned a severity classification that is utilized during design to establish priorities for corrective actions.

The hardware (bottom-up) approach is normally used when hardware items can be uniquely identified from schematics, drawings, and other engineering and design data. The hardware approach is normally utilized in a part level up fashion (bottom-up approach); however, it can be initiated at any level of indenture and progress in either direction. Each identified failure mode is assigned a severity classification which will be utilized during design to establish priorities for corrective actions.

The procedures contained in this Handbook assume a bottom-up hardware approach. This approach combined with the procedures in the Handbook for estimating the failure rates of hardware items in their intended operating environments provides the optimum method of performing an RMA analysis. After the failure modes are identified for each hardware item, the probability of occurrence is determined and then severity classifications are assigned to each failure mode and each item to provide a basis for establishing corrective action priorities. First priority is given to the elimination of identified Category I (catastrophic) and Category II (critical) failure modes. The combination of failure mode probability of occurrence and severity category permits a criticality rating of failure modes requiring additional investigation as to corrective action requirements.

24.4.2 System Definition

The first step in performing the RMA analysis is to define the system to be analyzed. Functional narratives are developed for each mission, mission phase, and operational mode and include statements of primary and secondary mission objectives. The narratives include system and part descriptions, operational modes, expected mission times and equipment utilization, and conditions which constitute system and part failure. Developing a system definition is extremely beneficial when completing the RMA analysis worksheets providing continuity at the lower indenture levels. The system definition is also important during design reviews when failure mode criticality is evaluated with respect to equipment performance and mission requirements. The following paragraphs describe those subjects that comprise a system definition.

- **Mission Functions and Operational Modes** - The system definition includes descriptions of each mission in terms of functions which identify the task to be performed and the functional mode of operation for performing the specific function. Mission functions and operational modes are identified starting at the highest system level and progressing to the lowest indenture level to be analyzed.

- **Environmental Profiles** - Environmental profiles include the anticipated environmental conditions for each mission and mission phase. When a system is to be utilized in more than one environment each different environmental profile needs to be described. The intended use of the system and its equipments are developed from mission time statements for each environmental profile. These profiles will be used in determining the time-stress relationships and the feasibility of failure detection methods and compensating provisions in the operating system.

- **Mission Time** - A quantitative statement of system function time requirements is developed and included in the system definition. Function-time requirements are developed for items which operate in different operational modes during different mission phases and for items which function only if required.

- **Block Diagrams** - Block diagrams which illustrate the operation, interrelationships, and interdependencies of functional entities of a system are advisable to provide the ability for tracing failure mode effects through all levels of indenture. Functional and reliability block diagrams show the functional flow sequence and the series dependence or independence of functions and operations. Block diagrams can be constructed in conjunction with or after defining the system and display the system as a breakdown of its major functions. More than one block diagram will usually be required to display alternative modes of operation, depending upon the definition established for the system. All inputs and outputs of the item as a whole are shown on the diagram.

Each block on the diagram should be designated by a consistent and logical item number that reflects the functional system breakdown order. A uniform numbering system developed in functional system breakdown order is required to provide traceability and tracking through all levels of indenture.

A functional block diagram illustrates the operation and interrelationships between functional entities of a system as defined in engineering data and schematics. A functional system block diagram provides a functional flow sequence for each indenture level of analysis. A reliability block diagram defines the series dependence or independence of all functions of a system or functional group for each life-cycle event. The reliability block diagram provides identification of function interdependencies for the system.

24.4.3 RMA Analysis worksheet

The documentation of the RMA analysis is the next step and is accomplished by completing the columns of the established RMA worksheet. Guidelines for designing the worksheets are included in Section 24.2. Worksheet examples are included in

Section 24.6. The following paragraphs contain a description of RMA worksheet entries.

- **Equipment Part Number** – A part number for the component being analyzed is assigned. The schematic diagram symbol or drawing number can be used to properly identify the component.

- **Equipment Functional Description** - The name or nomenclature of the item or system function being analyzed for failure mode and effects is listed. A uniform identification code with general requirements is used to provide consistent identification of system functions of the equipment and provide complete visibility of each failure mode.

- **Function ID Number** - A sequential code number used to track worksheet entries, provides a reference between worksheets and summary reports.

- **Function** - A concise statement of the function performed by the hardware item is listed which includes both the inherent function of the part and its relationship to interfacing items.

- **Failure Mode ID Number** – A sequential code number is assigned that will be used to track individual failure modes. Only one failure mode should be referenced by the Failure Mode ID number. The sequence of numbers resumes for each Function ID Number.

- **Failure Modes and Causes** - All predictable failure modes for each indenture level analyzed are identified and described. Potential failure modes are determined by examination of item outputs and functional outputs identified in applicable block diagrams and schematics. Failure modes of individual item functions are postulated on the basis of narratives and failure definitions included in the ground rules. The most probable causes associated with the postulated failure mode are identified and described. Failure Mode Tables in the various chapters of this Handbook provide listings of possible failure modes for the various mechanical components.

Since the failure mode may have more than one cause, all probable independent causes for each failure mode are identified in the indenture levels per the indenture level analysis. To assist in assuring that a complete analysis is performed, each failure mode and output function is examined in relation to the following typical failure conditions:

a. Premature operation.

b. Failure to operate at a prescribed time.

c. Intermittent operation.

d. Failure to cease operation at a prescribed time.

e. Loss of output or failure during operation.

f. Degraded output or operational capability.
g. Other unique failure conditions, as applicable, based upon given characteristics and operational requirements or constraints.

- **Mission Phase or Operational Mode** - A concise statement of the mission phase and operational mode in which the failure occurs is entered on the worksheet. Where a specific event or time can be defined from the system definition and mission profiles, the most definitive timing information should also be entered for the assumed time of failure occurrence.

- **Failure Effects** - The consequences of each assumed failure mode on item operation, function, or status is identified, evaluated, and recorded on the worksheet. Failure effects focus on the specific block diagram element which is affected by the failure under consideration. The failure under consideration may impact several indenture levels in addition to the indenture level under analysis; therefore local, next higher level, and end effects need to be evaluated. Failure effects also consider the mission objectives, maintenance requirements and personnel and system safety.

Local effects concentrate specifically on the impact an assumed failure mode has on the operation and function of the item in the indenture level under consideration. The consequences of each postulated failure affecting the item are described along with any second-order effects which result. The purpose of defining local effects is to provide a basis for evaluating compensating provisions and for recommending corrective action. It is possible for the local effect to be the failure mode itself.

Next higher level effects concentrate on the impact an assumed failure has on the operation and function of the items in the next higher indenture level above the indenture level under consideration. The consequences of each postulated failure affecting the next higher indenture level are described.

End effects evaluate and define the total effect an assumed failure has on the operation, function, or status of the uppermost system. The end effect described may be the result of a double failure. For example, failure of a safety device may result in a catastrophic end effect only in the event that both the prime function goes beyond limit for which the safety device is set and the safety device fails. Those end effects resulting from a double failure need to be indicated on the worksheets and can be detailed in the Remarks column.

- **Failure Detection Method** - A description of the methods by which occurrence of the failure mode is detected by the operator is recorded on the worksheet. The failure detection method, such as a visual or audible warning device, automatic sensing device, sensing instrumentation or other unique indication is identified.

A description of indications which are evident to an operator that a system has malfunctioned or failed, other than the identified warning devices, are recorded. Proper correlation of a system malfunction or failure may require identification of normal indications as well as abnormal indications. If no indication exists, identify if the undetected failure will jeopardize the mission objectives or personnel safety. If the undetected failure allows the system to remain in a safe state, a second failure situation should be explored to determine whether or not an indication will be evident to the operator. Indications to the operator should be described as follows:

a. Normal. An indication that is evident to an operator when the system or equipment is operating normally.

b. Abnormal. An indication that is evident to an operator when the system has malfunctioned or failed.

c. Incorrect. An erroneous indication to an operator due to the malfunction or failure of an indicator such as an instrument, sensing device or visual or audible warning device

A description of the most direct procedure that allows an operator to isolate the malfunction or failure is recorded. An operator will know only the initial symptoms until further specific action is taken such as performing a more detailed built-in-test (BIT) or operational procedure. The failure being considered in the analysis may be of lesser importance or likelihood than another failure that could produce the same symptoms and this must be considered. Fault isolation procedures require a specific action or series of actions by an operator, followed by a check or cross reference either to instruments, control devices, circuit breakers, or combinations thereof. This procedure is followed until a satisfactory course of action is determined.

- **Failure Restoration Method** - The method by which the failure is repaired or other action to be taken in event the failure mode does occur is entered on the worksheet. Typical repairs include on-site adjustment or lubrication, recalibration or part replacement with a spare. Off-site repairs requiring extensive down-time need to be considered.

- **Time to Repair** - The time to repair the identified failure mode is determined utilizing the maintenance strategy as described for the failure restoration method. Administrative delay time may be included either with the time to repair or as noted in the remarks column.

- **Remarks** - Any pertinent remarks pertaining to and clarifying any other column in the worksheet line are noted in the Remarks column. This column is used to amplify and clarify the entries on this particular worksheet. Typical remarks include safety provisions in the design, special maintenance requirements, logistics support of repaired items and spare parts requirements. This entry also may include a notation of unusual conditions, failure effects of redundant items, recognition of particularly critical design features or any other remarks that amplify the line entry.

The purpose of determining the criticality of a failure mode is to rank each potential failure mode identified according to the combined influence of severity classification and its probability of occurrence based upon the best available data. As parts configuration data and failure rate data become available, criticality numbers should be calculated and incorporated in the analysis. The failure probability levels, when used, should be modified as the system is better defined. There are failure rate data sources other than this Handbook such as MIL-HDBK-217 for electronic parts, OREDA and NPRD. Section 24.5, References, provides information on these sources. Calculation of a criticality number for each failure mode and component part is accomplished by completing the applicable columns of the worksheet.

- **Failure Rate Data Source** - The source of the failure rate data such as this Handbook, reliability testing data, MIL-HDBK-217, OREDA or NPRD is recorded on the worksheet. See Section 24.5 for identification of failure rate data sources.

- **Base Failure Rate** - The Base Failure Rate from the failure rate data source is recorded on the worksheet. The base failure rate is typically the failure rate found in the failure rate data source that must then be adjusted for the intended operating environment.

- **Adjustment Factors** - The base failure rate from the failure rate data source may need to be adjusted for the operating and environmental conditions of the equipment being analyzed. Adjustment factors may be the multiplying factors as contained in the various chapters of this Handbook, they may be the analyst's best judgment or it may be that no adjustment factor is needed. For example, if testing data for the particular part is being used, adjustment factors may not be applicable. Explanations can be included in the Remarks column.

- **Part Failure Rate** - The part failure rate (λ_p) as calculated using the procedures in this Handbook or from the appropriate failure rate data source is entered on the worksheet. Where appropriate, application factors, environmental factors, and other factors as may be required are applied to the base failure rates (λ_b) obtained from this Handbooks or other reference material to adjust for differences in operating stresses.

- **Failure Effect Probability** - β values are the conditional probability that the failure effect will result in the identified criticality classification, given that the failure mode occurs. The β values represent the analyst's judgment as to the conditional probability the loss will occur. Typical β values are as follows:
  Actual loss = 1.00
  Probable loss = >0.10 to <1.00
  Possible loss = >0.0 to 0.10
  No effect = 0.0

- **Failure Mode Ratio** - The fraction of the part failure rate (α) related to the particular failure mode under consideration is evaluated by the analyst and recorded. The failure mode ratio is the probability expressed as a decimal fraction that the part or item will fail in the identified mode. After all potential failure modes of a particular part or item are listed, the sum of the α values for that part or item will equal one. Individual failure rate multipliers may be derived from failure rate source data or from test and operational data. If failure mode data are not available, the α values will represent the analyst's judgment based upon analysis of the item's functions.

- **Duty Cycle** – In many cases the component being analyzed is only used part of the time with respect to the total equipment operating time. Examples include a valve or an actuator that is used only for a particular mode of equipment operation.

- **Detectability** - The probability that the failure will be detected by the operator in event of component failure is determined based on the analyst's knowledge of the system.

- **Operating Time** - The operating time in hours or the number of operating cycles of the component per mission is derived from the system definition and listed on the worksheet.

- **Failure Mode Criticality Number** - The value of the failure mode criticality number (C_m) is calculated and listed on the worksheet. C_m is the portion of the criticality number for the item due to one of its failure modes under a particular severity classification. For a particular severity classification and operational phase, the C_m for a failure mode may be calculated with the following equation:

$$C_m = \lambda_p \ \beta \ \alpha \ d \ t$$

where:

C_m = Criticality number for the failure mode
λ_p = Part Failure Rate
β = Conditional probability that failure mode results in listed end effect
α = Failure Mode Ratio
d = Probability that the failure mode will be detected by the operator or other means such as an alarm or monitor
t = Duration of applicable operating hours or mission phase (hours or cycles)

- **Item Criticality Number** - The second criticality number calculation is for the item under analysis. Criticality numbers (C_r) for the items of the system are calculated and listed on the worksheet. A criticality number for an item is the number of system failures of a specific type expected due to the item's failure modes. The specific type of system failure is expressed by the severity classification for the item's failure modes.

For a particular severity classification and mission phase, the C_r for an item is the sum of the failure mode criticality numbers.

$$C_r = \sum_{n=1}^{j} (\lambda_P \, \beta \, \alpha \, d \, t) n$$

where:

C_r = Criticality number for the item

n = The failure modes in the item that fall under a particular criticality classification

j = Last failure mode in the item under the criticality classification

- **Compensating Provisions** - Compensating provisions, either design provisions or operator actions, which circumvent or mitigate the effect of the failure are identified and evaluated. This step is required to record the true behavior of the item in the presence of an internal malfunction or failure.

Compensating provisions which are features of the design at any indenture level that will nullify the effects of a malfunction or failure are described on the worksheet. These include control items that will halt generation or propagation of failure effects, or activate backup or standby items or systems.

Design compensating provisions include:

a. Redundant items that allow continued and safe operation.

b. Safety or relief devices such as monitoring or alarm provisions which permit effective operation or limits damage.

c. Alternative modes of operation such as backup or standby items or systems.

Compensating provisions which require operator action to circumvent or mitigate the effect of the postulated failure need to be described. The compensating provision that best satisfies the indication(s) observed by an operator when the failure occurs is determined. This may require an investigation to determine the most correct operator action(s). The consequences of any probable incorrect action(s) by the operator in response to an abnormal indication should be considered and the effects recorded.

- **Severity Classification** - A severity classification category is assigned to each failure mode and item according to the failure effect. The effect on the functional condition of the item under analysis caused by the loss or degradation of output is identified so the failure mode effects will be properly categorized. For lower levels of indenture where effects on higher indenture levels are unknown, a failure's effect on the indenture level under analysis is described by the severity classification categories as follows:

Category I - Catastrophic - A failure which may cause death or total system loss

Category II - Critical - A failure which may cause severe injury, extensive property damage, or major system damage that results in loss of mission *

Category III - Marginal - A failure which may cause minor injury, minor property damage, or minor system damage that results in mission delay or degradation.

Category IV - Minor - A failure not serious enough to cause injury, property damage or mission delay or degradation, but which will necessitate repairs at a later time.

* In some cases the loss of mission may be sufficiently severe so that it may be classified as a Category I failure.

- **Remarks** - Any pertinent remarks pertaining to and clarifying any other column in the worksheet line are noted in the Remarks column. Examples include a description of the failure rate data source, adjustment factors and α or β factors.

The Fault Tree Analysis is designed to assure the safety of equipment operation. From previous worksheets, the end effects at the total system level are reviewed. There will be many similar end effects that can be consolidated into the most critical system level failure effects. These are listed on the worksheet (Sheet 5 of the example worksheets) along with the previous worksheet reference numbers and occurrence probabilities. If a fault tree diagram has been developed, the fault path reference is listed. The method by which the system level failure effect is detected and repaired is then entered. Completion of this worksheet provides assurance that all system level failure modes have been considered with respect to reliability and logistics support.

The criticality matrix provides a means of identifying and comparing each failure mode to all other failure modes with respect to severity. The matrix is constructed by inserting item or failure mode identification numbers in matrix locations representing the severity classification category and the probability of occurrence. The resulting matrix display shows the distribution of criticality of item failure modes and provides a tool for assigning corrective action priorities. A sample Criticality Matrix is shown in Figure 24.1. The frequency of occurrence can be displayed as a probability of occurrence, as a severity level or failure rate. The numbers within the blocks of Figure 24.1 represent the identification numbers of individual failure modes. The resulting display indicates the relative criticality of the failure modes.

24.5 REFERENCES

28. MIL-HDBK-217, "Reliability Prediction of Electronic Equipment".

88. Reliability Analysis Center, “Nonelectronic Parts Reliability Data”, NPRD-95

105. OREDA Offshore Reliability Data, 4th Edition Det Norske Veritas, NO-1322 Hovik, Norway 2002, ISBN 8251500877

118. MIL-STD-1629, “Procedures for Performing a Failure Mode, Effects and Criticality Analysis”, August 1998

24.6 WORKSHEET EXAMPLES

The following pages contain examples worksheets used for the RMA analysis. It is important to note that these examples need to be modified for the system being analyzed. The preceding Sections of this Chapter need to be reviewed thoroughly before deciding on the Worksheet formats.

MechRel

Data Collection Worksheet

Equipment Assembly Name							Project: ____________ Date: ____________ Compiled By: ____________	
Equipment Part Number	**Equipment Functional Description**	**Function ID Number**	**Function**	**Failure Mode ID No.**	**Failure Modes**	**Cause ID No.**	**Failure Causes**	**Mission Phase or Operational Mode**
Part Number	Part Name	1	Functional description of the part	1	Only one failure mode should be entered for each row because of the individual effects, α and β values and compensating provisions	1	Description of all probable independent causes of this specific failure mode	The mode in which the system operating when the failure mode occurs or effects equipment operation
						2		
						3		
				2		1		
						2		
		2		1		1		
				2		1		
						2		
						3		
				3		1		
		3		1		1		
						2		
						3		
						4		
				2		1		
Part Number	Part Name							

Sheet 1

MechRel

Data Collection Worksheet

Equipment Assembly Name							Project: ______________ Date: ______________ Compiled By: __________
Failure Mode ID No.	**Failure Effects**			**Failure Detection Method**	**Failure Restoration Method**	**Time to Repair**	**Remarks**
	Local Effects	**Next Higher Level**	**End Effects**				
Failure Mode Reference Number from Sheet 1	Consequences of the failure mode on operation or function of the failure mode at the part level	Impact of the failure mode at the next higher indenture level	Total effect of the failure mode on the operation, function or status of the system	Failure detection methods by which occurrence of the failure mode are detected by the operator, warning devices, alarms, sensing devices	Method by which failure is repaired such as part replacement, lubrication, adjustment, local repair	Time to repair for the failure mode based on restoration method considering admin delay time	Remarks clarifying or amplifying worksheet entries such as design features, safety provisions, maintenance requirements, logistics support of repairs and spare parts

Sheet 2

MechRel

Data Collection Worksheet

Equipment Assembly Name							Project: ____ Date: ____ Compiled By: ____	
Failure Mode ID No.	**Failure Rate Data Source**	**Base Failure Rate (λ_b x 10^6)**	**Adjustment Factors (π Factors)**	**Part Failure Rate (λ_p x 10^6)**	**Failure Effect Probability (β)**	**Failure Mode Ratio (α)**	**Duty Cycle (DC)**	**Detectability (d)**
Failure Mode Reference Number from Sheet 1	RAC, OREDA, NPRD, NSWC Handbook	Failure rate from source	Any correction factors required to convert acquired failure rate to application failure rate	Base failure rate multiplied by π factor	Conditional probability that identified failure mode will result in end effect (0.0 to 1.0)	Ratio of part failure rate related to identified failure mode	A value between 0.0 and 1.0 indicating the time ratio the component is functioning when the equipment is in operation	Probability that failure mode will be detected prior to failure

Sheet 3

MechRel

Data Collection Worksheet

Equipment Assembly Name						Project: ______ Date: ______ Compiled By: ______
Failure Mode ID No.	**Operating Time (t)**	**Failure Mode Criticality Number Cn**	**Item Criticality Number (ΣCn)**	**Compensating Provisions**	**Severity Classification**	**Remarks**
Failure Mode Reference Number from Sheet 1	The operating time in hours or operating cycles between overhauls	λ_p x β x α x d x t	Criticality number for the part	Provisions in the design or operator actions which circumvent or mitigate the effect of failure such as redundancy, alarm provisions or alternate modes of operation	Severity Category (See Remarks)	Remarks to clarify or amplify entries on the worksheet such as explanations of failure rate sources, adjustment factors or calculations,
						Category I is catastrophic (a failure which may cause death or total property damage)
						Category II is a critical failure, one that may cause severe injury, extensive property damage or loss of mission
						Category III is a marginal failure effect, one that may cause minor injury, minor property damage or mission degradation
						Category IV is a minor failure effect, a failure mode not serious enough to cause injury, property damage, but may necessitate repairs

Sheet 4

MechRel

Data Collection Worksheet

Equipment Assembly Name						Project: ____ Date: ____ Compiled By: ____
Single Fault Event	**End Effects**	**Reference Numbers**	**Occurrence Probability**	**Fault Path Reference**	**Maintenance Philosophy**	**Remarks**
System Level Fault Identification	End effects from Sheet 2 applicable to system level fault	Reference Failure Mode numbers	Combined probability of occurrence for system level fault	Reference from fault path if applicable	Methods by which system level fault is detected and repaired	Remarks concerning fault detection, reliability, severity and repair

Sheet 5

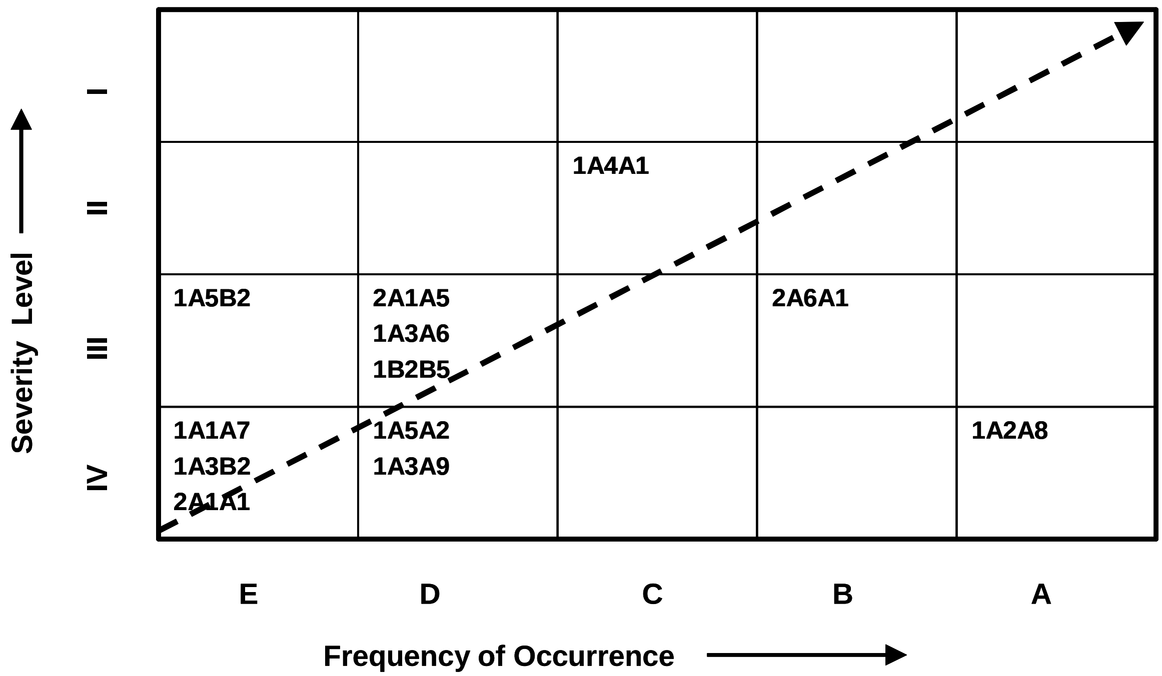


Figure 24.1 Sample Criticality Matrix

CHAPTER 25

REFERENCES

1. Abadzheva, R.N. et al., "Effects of Brake Fluid Components on Rubber," Khimiya i Tekhnologiya Topliv i Masel, No. 8, pp. 18-19 (Aug 1982).

2. “A Practical Guide to Compressor Technology”, Second Edition, Heinz P. Bloch, John Wiley & Sons, 2006

3. Anderson, A.E., "Wear of Brake Materials," in: Wear Control Handbook, M.B. Peterson and W.O. Winer, Eds., pp. 843-857, Am. Soc. Mech. Eng., New York (1980).

4. Armstrong, E.L., W.R. Murphy and P.S. Wooding, "Evaluation of Water Accelerated Bearing Fatigue in Oil-Lubricated Ball Bearings," Lubrication Engineering, Vol. 34, No. 1, pp. 15-21 (1 Nov 1977).

5. Bauer, P., M. Glickmon, and F. Iwatsuki, "Analytical Techniques for the Design of Seals for Use in Rocket Propulsion Systems," Volume 1, ITT Research Institute, Technical Report AFRPL-TR-65-61 (May 1965).

6. Bayer R.G., A.T. Shalhey and A.R. Watson, "Designing for Zero Wear," Machine Design (9 January 1970).

7. Bishop, F.E. and William M. Needleman, "The Effects of Fluid Contamination on Component Wear," Pall Corporation.

8. Block, H. and D. Johnson, "Downtime Prompts Upgrading of Centrifugal Pumps," Chemical Engineering Magazine, pp. 35-38 (25 Nov 1985).

9. Boone, Tony D., "Reliability Prediction Analysis for Mechanical Brake Systems," NAVAIR-SYSCOM Report (Aug 1981).

10. "Boston Gear Catalog" Catalog 100, INCOM International Inc., Quincy, Massachusetts 02171.

11. Canterbury, Jack, and James D. Lowther, "Application of Dimensional Analysis to the Prediction of Mechanical Reliability," Naval Weapons Support Activity, Washington Navy Yard, Wash., D.C., Report ADAD35295 (September 1976).

12. Carson, Harold, Springs: Troubleshooting and Failure Analysis, Marcel Dekker, Inc., New York. (1983).

13. Cormier, K.R., "Helicopter Drive System R&M Design Guide," Division of United Technologies Corp., Stanford, CT 06602, Report ADAD69835 (April 1979).

14. "Engineering Guide to Spring Design" Associated Spring, Barnes Group INc., Form No. 515 (1981).

15. "Fabrication and Testing of Lightweight Hydraulic System Simulator Hardware - Phase II," Report NADC-79024-60, prepared by Rockwell International, Columbus, Ohio, for Naval Air Systems Command, Washington, D.C.

16. Ferodo Limited, Friction Materials for Engineers, Stockport, England (1969).

17. Field, G.J., "Seals That Survive Heat," Machine Design (1 May 1975).

18. Hauser, D.L. et al., "Hardness Tester for Polyur," NASA Tech Briefs, Vol. 11, No. 6, p. 57 (1987).

19. Hindhede, U., et al, Machine Design Fundamentals, John Wiley & Sons, NY, 1983.

20. Ho, T.L., F.E. Kennedy and M.B. Peterson, "Evaluation of Materials and Design Modifications for Aircraft Brakes," NASA Report CR134896 (Jan 1975).

22. Howell, Glen W. and Terry M. Weathers, Aerospace Fluid Component Designers' Handbook, Volumes I and II, TRW Systems Group, Redondo Beach, CA prepared for Air Force Rocket Propulsion Laboratory, Edwards, CA, Report AD 874 542 and Report AD 874 543 (February 1970).

23. Hubert, Christopher J., John W. Beck and John H. Johnson, "A Model and the Methodology for Determining Wear Particle Generation Rate and Filter Efficiency in a Diesel Engine Using Ferrography," Society of Automotive Engineers Paper No. 821195 (1982).

24. Hudgens, R.D. and L.B. Feldhaus, "Diesel Engine Lube Filter Life Related to Oil Chemistry," Society of Automotive Engineers Paper No. 780974 (1978).

25. Johnson, R.L. and K. Schoenherr, "Seal Wear," in: Wear Control Handbook, M.B. Peterson and W.O. Winer, Eds., Sect. 5, pp 727-754, American Society of Mechanical Engineers, New York, (1980).

26. Igor J. Karassik et al, Pump Handbook, McGraw-Hill Book Company, New York (1986).

27. May, K.D., "Advanced Valve Technology," National Aeronautics and Space Administration, NASA Report SP-5019 (February 1965).

28. MIL-HDBK-217, "Reliability Prediction of Electronic Equipment".

29. Minegishi, H. et al., "Prediction of Brake Pad Wear/Life by Means of Brake Severity Factor as Measured on a Data Logging System," SAE Paper 840358 (1984).

30. Mordkowitz, A., "Predicting Service Life for Zero Wear," Machine Design (10 January 1974).

31. Nagel, W.B., "Designing with Rubber," Machine Design (June 23, July 7, July 21, Aug 11, 1977).

32. Neale, M.J., Tribology Handbook, Butterworths, London.

33. Needleman, William M., "Filtration for Wear Control," in: Wear Control Handbook, M.B. Peterson and W.O. Winer, Eds., Sect. 4, pp 507-582, Americna Society of Mechanical Engineers, New York, (1980).

35. "Optimum Design of Helical Springs," Machine Design, (6 November 1980).

36. Orthwein, William C., Clutches and Brakes: Design and Selection, Marcel Dekker, Inc., New York (1986).

37. Rhee, S.K. and P.A. Thesier, "Effects of Surface Roughness of Brake Drums on Coefficient of Friction and Lining Wear," SAE Paper 720449 (1972).

38. Roack and Young, Formulas for Stress and Strain, McGraw-Hill Book Company, New York (1975).

39. Shigley, J.E., Mischke, C.R., Mechanical Engineering Design, McGraw-Hill Book Co., NY, 1989.

40. Spokas, R.B., "Clutch Friction Material Evaluation Procedures," SAE Paper 841066 (1984).

41. "Standard Product Catalog," Catalog SPC 82, The Falk Corporation, Milwaukee, Wisconsin.

44. Sibley, L.B., "Rolling Bearings," in: Wear Control Handbook, M.B. Peterson and W.O. Winer, Eds., Sect. 5, pp 699-726, American Society of Mechanical Engineers, New York (1980).

45. Barron, Randall F., Revision of Wear Model for Stock Actuators, Engineering Model for Mechanical Wear (July 1987).

46. Fox, R.W., and A.T. McDonald, Introduction to Fluid Mechanics, John Wiley and Sons, New York (1978).

47. Machine Design, 1985 Fluid Power Reference Issue, Penton/IPC, Inc., (Sept. 19, 1985).

48. Kragelsky, I.V. and V.V. Alisin, Friction, Wear, and Lubrication, Vol. 2, pg. 30, Pergamon Press, London (1981).

49. Kuhlmann-Wildorf, D., "Parametric Theory of Adhesive Wear in Uni-Directional Sliding," Wear of Materials, pp. 402-413, American Society of Mechanical Engineers, New York (1983).

50. Bentley, R.M. and D.J. Duquette, "Environmental Considerations in Wear Processes," Fundamentals of Friction and Wear of Materials, pp. 291-329, American Society For Metals, Metals Park, Ohio (1981).

51. Sarkar, A.D., Wear of Metals, pp. 62-68, Pergamon Press, London (1976).

52. Lundberg, G. and A. Palmgren, "Dynamic Capacity of Rolling Bearings," Acta Polytechnica, No. 7 (1974).

53. Rumbarger, John H., "A Fatigue Life and Reliability Model for Gears," American Gear Manufacturers Association Report 229.16 (January 1972).

54. AGMA Standard for Surface Durability Formulas for Spiral Bevel Gear Teeth, American Gear Manufacturers Association Report 216.01 (January 1964).

55. AGMA Standard Nomenclature of Gear Tooth Failure Modes, American Gear Manufacturers Association Report 110.04 (August 1980).

56. Haviland, G.S., "Designing with Threaded Fasteners", MECHANICAL ENGINEERING, Vol 105, No. 10, Oct 83.

57. Deutschman, A.D., et al, Machine Design; Theory and Practice, MacMillan Publishing Co, NY, 1975

58. Parmley, R.O., Mechanical Components Handbook, McGraw-Hill Book Co., NY, 1985.

59. Handbook H28, Nat'l Bureau of Stds, Govt Printing Office, Washington, DC, 1957.

60. Bickford, J.H., An Introduction to the Design and Behavior of Bolted Joints, Marcel Dekker, Inc., NY, 1990.

61. Handbook of Corrosion Data, ASM International, Metals Park, OH, 1990

62. Baumeister, T, et al, Mark's Standard Handbook for Mechanical Engineers, McGraw-Hill Book Company

63. Thomas Couplings Applications Manual

64. Bolam, J.R., "Coupling Alignment: The Reverse Indicator Method Simplified", P/PM Technology, July/Aug 90

65. Dvorak, P., "Sorting Out Flexible Couplings", Machine Design, 11 Aug 88

66. Robertson, R., and Smith, B., "Why Flexible Couplings Fail", Plant Engineering and Maintenance, Jun 89

67. Universal Joint & Driveshaft Design Manual, Series No. 7, Society of Automotive Engineers, Inc, Warrendale, Pa.

68. H. Wayne Beaty and James L. Kirtley, Jr., Electric Motor Handbook, McGraw-Hill Book Company 1998

69. Norman Yudewitz, "Predict Relay Life Reliably with Simple Empirical Equations", Electronic Design, February 1, 1979

70. "Validation of Gearbox Reliability Models from Test Data", Report No. 87-D-0075, October, 1987, Eagle Technology, Inc.

71. Dennis N. Pratt, "Investigation of Spline Coupling Wear", Report No. SY-51R-87, December 1987, Naval Air Warfare Center, Patuxent River, Maryland

72. Randall F. Barron, "Engineering Model for Mechanical Wear", Report No. CMLD-CR-09-88, June 1988, Louisiana Tech University

73. Dennis Pratt, "Results of Air Compressor Reliability Investigation", Report No. TM 88-38 SY, January 1989, Naval Air Warfare Center, Patuxent River, Maryland

74. D. Pratt, "Results of Gear Pump Reliability Investigation", Report No. TM 89-24 SY, February 1990, Naval Air Warfare Center, Patuxent River, Maryland

75. D. Pratt, "Results of Centrifugal Pump Reliability Investigation", Report No. TM 89-69 SY, February 1990, Naval Air Warfare Center, Patuxent River, Maryland

76. D. Pratt, "Results of Pneumatic Impact Wrench Reliability Investigation", Report No. TM 90-88 SY, December 1990, Naval Air Warfare Center, Patuxent River, Maryland

77. Randall F. Barron and Herbert G. Tull, III, "Failure Rate Model for Aircraft Brakes and Clutches", Report No. DTRC-CMLD-CR-01-90, August 1990, Louisiana Tech University

78. Randall F. Barron and Herbert G. Tull, III, "Failure Rate Model for Aircraft Brakes and Clutches", Report No. NSWC-92/LO2, August 1992, Louisiana Tech University

78. CDNSWC, "Interim Reliability Report on the MC-2A Compressor Unit", January, 1992

79. D. Pratt, "Results of Air Force MC-2A Air Compressor Unit Reliability Investigation", Report No. TM 92-89 SY, March 1993, Naval Air Warfare Center, Patuxent River, Maryland

80. D. Pratt, "Results of Dayton 5A701 Linear Actuator Reliability Investigation", Report No. TM 93-89 SY, February, 1994, Naval Air Warfare Center, Patuxent River, Maryland

81. TD 84-3 R&M Data for Industrial Plants, A.P. Harris and Associates, 1984

82. Metals Handbook, American Society for Metals, 1985, ISBN 0-87170-188-X

83. Handbook of Chemistry and Physics, 86th Edition, CRC Press, 2005

84. Machinery's Handbook, 27th Edition, Industrial Press, 2004

85. Lev Nelik, “What Happens When a Pump No Longer Operates At Optimum Conditions”, Pumps and Systems, February 2005

86. “Performance Prediction of Centrifugal Pumps and Compressors”, 25th Annual Gas Turbine conference and 22nd annual Fluids Engineering Conference Proceedings ASME 1980

87. Childs, T.H.C., Dalgarno, K.W., Day, A.J. and Moore, R.B., "Automotive Timing Belt Life and a User Design Guide", Proceedings of the Institute of Mechanical Engineers, 1998

88. Reliability Analysis Center, "Nonelectronic Parts Reliability Data", NPRD-95

89. SINTEF Industrial Management, "OREDA Offshore Reliability Data", 4th Edition, 2002

90. The Complete Guide to Chains, Tsubakimoto Chain Company, 1995

91. Renold Transmission Chain Selection Procedure

92. PST =>Solutions! ,Volume 3, October 1996

93. Mechanical Engineers Handbook, Myer Kutz, et al, John Wiley & Sons, 1986

94. Mechanical Designers' Workbook, "Fastening, Joining & Connecting", J. Shigley and C. Mischke, McGraw-Hill 1986

95. Mechanical Designers' Workbook, "Mechanisms", J. Shigley and C. Mischke, McGraw-Hill 1986

96. Mechanical Designers' Workbook, "Power Transmission Elements", J. Shigley and C. Mischke, McGraw-Hill 1986

97. " The Chemical Engineering Guide to Compressors", Richard Greene, McGraw Hill Publications Company, 1984

98. Raymond J. Drago, "Rating the Load Capacity of Involute Splines", Machine Design, February 12, 1976

99. David L. McCarthy, "A Better Way to Rate Gears", Machine Design, March 7, 1996

100. Fittings and Flanges, Hydraulics & Pneumatics Magazine,

101. John H. Bickford, "An Introduction to the Design and Behavior of Bolted Joints", Marcel Dekker, Inc., 1990

102. Dan Seger, Niagara Gear Corporation, "Inside Splines" , Gear Solutions, January 2005

103. Mechanical Designers' Workbook, "Gearing", J. Shigley and C. Mischke, McGraw-Hill 1986

104. Raymond J. Drago, "Fundamentals of Gear Design", Butterworth Publishers, 1988

105. OREDA Offshore Reliability Data, 5th Edition Det Norske Veritas, N-1363 Hovik, Norway 2009 ISBN 978-82-14-04830-8

106. Design and Analysis of Machine Elements, Douglas Wright, Department of Mechanical and Materials Engineering, The University of Western Australia

107. Salzman, R.H. and Reaburn, S. M. 'Probabilistic Modeling for Timing Belt Fatigue Life Predictions Using Accelerated Testing' Int. J. of Materials & Product Technology, 2001

108. Daryl Beatty, Dow Chemical Company, "Oil analysis Boosts Compressor Reliability". *Practicing Oil Analysis Magazine,* November 2004

109. Robert Moffatt, Gast Manufacturing Corp., "Prolonging Compressor Life", Machine Design, May 11, 1978

110. John L. Wright, "Chains for Drives and Conveyors – Lube 'Em to Last", Machinery Lubrication Magazine, March 2002

111. Johnson Electric, Solenoid Technical Data, May 2008

112. NSK Product Guide – Bearings 2008, NSK Americas, Inc.

113. Meine van der Meulen, "On the Use of Smart Sensors, Common Cause Failure and the Need for Diversity", Center for Software Reliability, City University, London, 2008

114. Zhanyang Zhang and Miriam R. Tausner, "Using Markov Process to Model Wireless Sensor Network Life Expectancy with QoS Constraints, Department of Computer Science, College of Staten Island/City University of New York, 2008

115. Partab Jeswani and John Bleda, "A Predictive Process for Spring Failure Rates in Automotive Parts Application", General Motors Corporation, SAE Technical Paper 910356, February 1991

116. Dr. Gerhard G. Antony, "How to Determine the MTBF of Gearboxes", Power Transmission Engineering, April 2008

117. Jerry L. Lyons, Lyons' Valve Designer's Handbook, Van Nostrand Reinhold Company, 1982

118. MIL-STD-1629, "Procedures for Performing a Failure Mode, Effects and Criticality Analysis", August 1998

119. Jack A. Collins, Henry Busby and George Stabb, Mechanical Design of Machine Elements and Machines, the Ohio State University, John Wiley & Sons, 2010

120. Tyler G. Hicks, Handbook of Mechanical Engineering Calculations, McGraw-Hill, 2006

121. Bearing Failure Modes, NAEC-92-153, Naval Air Engineering Center, Lakehurst, NJ 1983

122. "How dirt and Water Slash Bearing Life", Richard C. Beercheck, Machine Design, July 6, 1978

123. Centrifugal Pump & Mechanical Seal Manual, William J. McNally, 2009

124. Parker O-Ring Handbook, 2001 Edition, Catalog ORD 5700/US, Parker Hannifin Corporation

125. "Improving the Reliability of Mechanical Seals", Michael Huebner, Chemical Engineering Progress, November 2005

126. Maintenance and Application Guide for Control Relays and Timers, Electric Power Research Institute, EPRI TR 102067s, December 1993

127. Explaining Motor Failure, Austin Bonnett and Chuck Young EASA, EC&M Magazine, October 1, 2004

128. "Rotary Screw or Reciprocating Air Compressor: Which One is Right?", Bryan Fasano and Randy Davis, Gardner Denver, Plant Engineering, September 1998

129. "Run Times", August 2005, Dale B. Andrews, Lawrence Pumps, Inc.

130. "A Practical Guide to Compressor Technology", Second Edition, Heinz P. Bloch, John Wiley & Sons, 2006

131. “Tidal Current Turbine Reliability: Power Take-off Train Models and Evaluation, C. Iliev and D. Val, Third International Conference on Ocean Energy, October 2010

132. “Temperature Monitoring Is Key to Motor Reliability”, Thomas H. Bishop, Electrical Apparatus Service Association, Maintenance Technology Magazine, July 2004

133. “Performance Prediction of Centrifugal Pumps and Compressors”, 25th Annual Gas Turbine conference and 22nd annual Fluids Engineering Conference Proceedings ASME 1980

134. “ The Chemical Engineering Guide to Compressors”, Richard Greene, McGraw Hill Publications Company, 1984

135. Handbook of Spring Design, Spring Manufacturers Institute, Inc. 2002

INDEX

RELIABILITY PREDICTION PROCEDURES FOR MECHANICAL EQUIPMENT

CARDEROCKDIV, NSWC-11

The index provides a rapid way of locating reliability information for a specific subject within the Handbook. If an entire chapter is devoted to that subject, the Chapter number only is referenced. If the item is part of one or more Chapters, the location(s) within the Chapter are listed. If you can not locate a particular part in the Handbook, please contact the individual identified in the Preface.