CHAPTER 1

INTRODUCTION

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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.

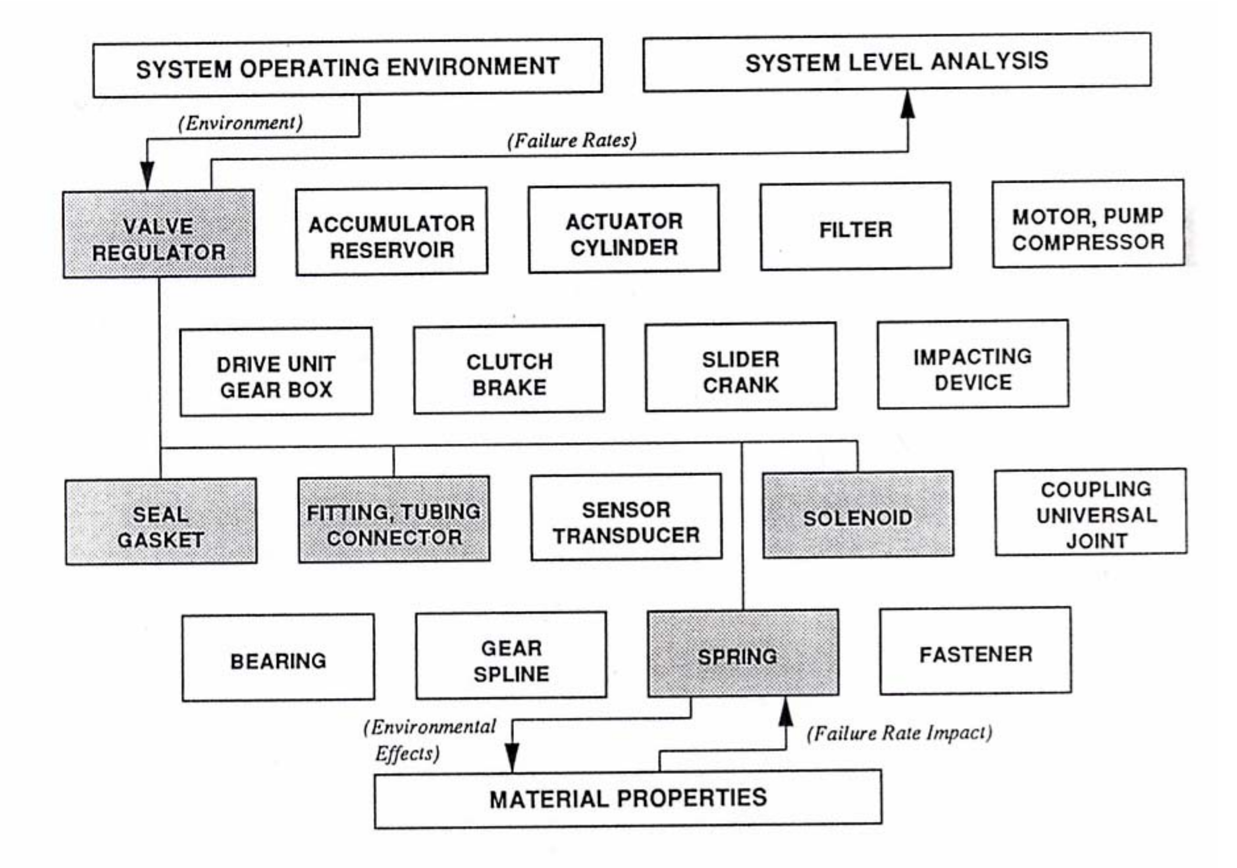

Figure 1.1 Mechanical Components and Parts

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.

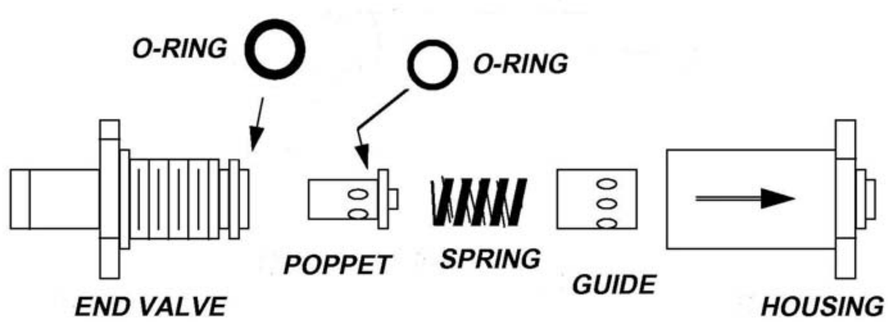

Figure 1.2 Valve Assembly

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, $lb\text{-}min/in^2$

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.

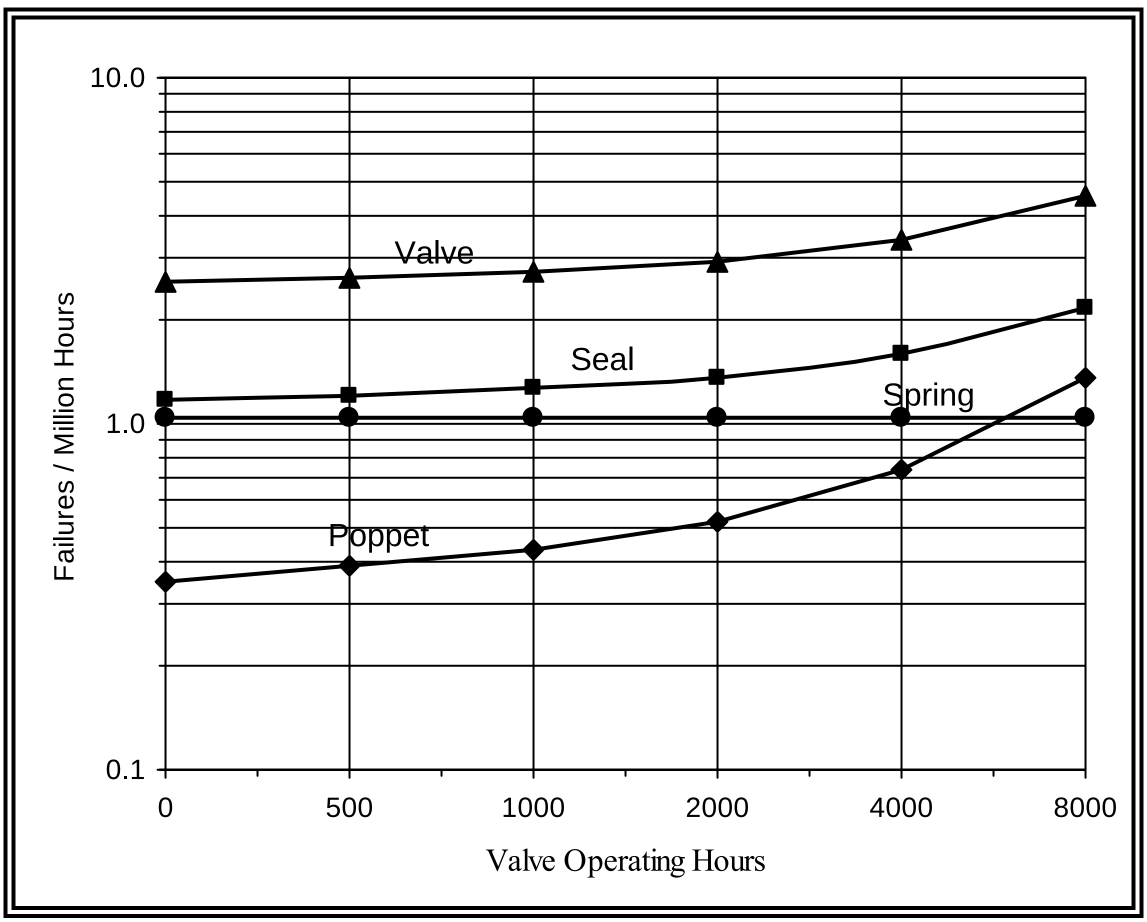

Figure 1.3 Combination of Component Failure Rates

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:

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