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241-SY-101 MIXER PUMP LIFETIME EXPECTANCY

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Abstract: The purpose of WHC-SD-WM-TI-726, Rev 0 241-SY-101 Mixer Pump Lifetime Expectancy is to determine a best estimate of the mean lifetime of the non-repairable (located in the waste) essential features of the hydrogen mitigation mixer pump presently installed in 101SY. The estimated mean lifetime is 9.1 years. This report does not demonstrate operation of the entire pump assembly within the Tank Farm "safety envelope".

It was recognized by the Defense Nuclear Facilities Safety Board (DNFSB) this test pump was not specifically designed for long term service in tank 101SY. In June 95 the DNFSB visited Hanford and ask the question, "how long will this test pump last and how will the essential features fail?" During the 2 day meeting with the DNFSB it was discussed and defined within the meeting just exactly what essential features of the pump must operate. These essential features would allow the pump to operate for the purpose of extending the window for replacement. Operating with only essential features would definitely be outside the operating safety envelope and would require a waiver. There are three essential features:

- The pump itself (i.e. the impeller and motor) must operate
- Nozzles and discharges leg must remain unplugged
- The pump can be re-aimed, new waste targeted, even if manually

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241-SY-101 MIXER PUMP LIFETIME EXPECTANCY

FINAL REPORT

December 8, 1995

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1.0 EXECUTIVE SUMMARY

A test mixer pump was installed in Hanford Waste Storage Tank 241-SY-101 in 1993. The mixer pump acts to mitigate periodic burping of hydrogen gas in the tank, which produce temporary tank concentrations of hydrogen above the lower flammability limit, and provide a potential mechanism (hydrogen burn) for public exposure to radiation. The pump was designed to last 2 to 3 years until planned operations could dilute the waste and transfer it to new tanks in a safe configuration. A decision not to build the new tanks provided impetus to continue running the pump as long as possible and also to estimate the remaining life of the pump.

The objective of this study is to estimate the remaining lifetime of the presently installed test pump. The methodology employed is a combination of fault-tree analysis and calculation of the joint probability function for time of first failure of a system subject to multiple failure modes. A Failure Modes and Effects Analysis (FMEA) was performed on selected system components. Quantification calculations were supported by data from tests specifically designed to obtain needed data, from pump operating experience, and from information supplied by component manufacturers.

Results showed that the best-estimate pump lifetime is approximately 9 years, with seal failures and lubricating oil failures being the dominant failure modes. Of the parameters controlling lifetime calculations, the lubricating oil lifetime and the common-mode parameter β of the seal system (the factor that determines the fraction of redundant failures which fail essentially simultaneously by a common cause) are the ones that most significantly affect the calculated results. A sensitivity study of these two parameters indicated that a worst case situation could produce a lifetime as low as around 6 years and that the most optimistic range of these variables would predict a lifetime of about 12 years.

2.0 INTRODUCTION

2.1 Background

Before 1993, the liquid waste of Hanford Waste Storage Tank 241-SY-101 had created a significant safety concern: periodic burping of hydrogen gas, which was produced by radioactive decay of tank contents. The concentration of hydrogen gas in the space above the liquid following such events frequently exceeded the lower flammability limit of hydrogen creating a potential for hydrogen burn and subsequent environmental release of radioactive contaminants.

A successful mitigation scheme, implemented in July 1993, was the installation of an in-tank test mixer pump which released the hydrogen at a controlled and non-hazardous rate. In order to meet a restrictive schedule for proving a successful mitigation, the design life was expected to be 2 to 3 years to coincide with the introduction of new tanks for diluting the waste. A decision not to build new tanks meant that the current tank storage had to continue and the projected pump lifetime became a major concern.

Tank SY-101's unique operating conditions include a radiation field of 850 R/hr based on in-tank core samples.

The pump has now been operating successfully for more than 2 years and has allowed experimentation to determine that successful mitigation of hydrogen burping can be accomplished with three 25-minute pumping operations per week. This cycle translates to 65 hours of annual operation and a pump duty factor of 7.44×10^{-3}.

The objective of this report is to determine the lifetime of essential components of the present pump, that is, to estimate how many more years of successful operation can be expected. This report documents the phase 3 results of a three-phase program to determine the life of the test mixer pump. The first phase, which has been completed and reported on (Ref.1), focused on the incremental degradation of the test mixer pump as a result of the tank severe environment. The phase 2 objective was to measure pump power cable performance degradation due to radiation and thermal effects via tests of cable samples and thermal aging tests of Buna-N "O" Rings. The results of phases 1 and 2 have been incorporated into a phase 3 model for estimating the expected pump lifetime.

2.2 Scope

The scope of this investigation is to determine a best estimate of the mean lifetime of the essential features of the hydrogen mitigation mixer pump presently installed in Tank SY-101. This report does not demonstrate operation of the entire pump assembly within the Tank Farm "safety envelope".

The SY-101 mixer pump as shown in Figure 1 consists of the following subsystems:

- Motor (includes power cable)
- Pump
- Pump structure (includes pump column, pump inlet and outlet, inlet and outlet nozzles)
- Instrumentation and control systems.

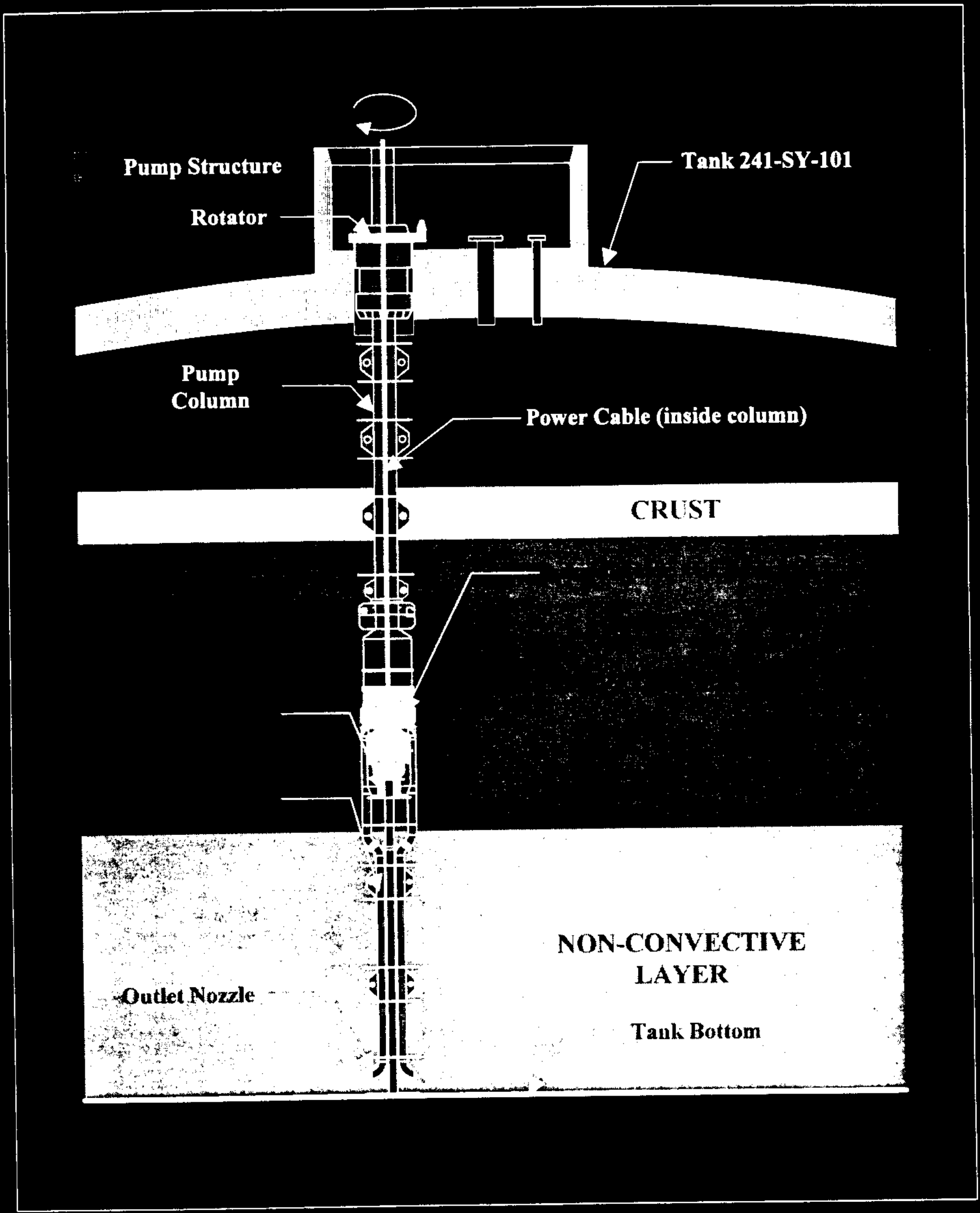

Figure 1. 241-SY-101 Mixer Pump Installation

The motor, pump, part of the pump structure, and some of the instruments are immersed below the waste tank liquid level and cannot be repaired in case of a failure. The control system is located outside the tank and can readily be repaired. The scope of the project is to estimate the life of the mixer pump by estimating the failure rates of the motor, pump and the part of the pump structure located below the tank dome. The determination of instrumentation failure rates is outside the scope of this project because instrument failure only temporarily delays operation of the pump.

The pump and motor subsystems have been broken down further into individual components for the determination of their respective failure rates. Table 1 shows the mixer pump components along with the respective manufacturers.

Table 1 MIXER PUMP COMPONENTS

subsystems	components	manufacturer
Motor	Power Cable Insulation	Bay Associates (BY92-482)
	Motor Winding	Reliance Electric
	Motor Oil	Chevron (GST-ISO 100)
	Double Thrust Bearing	MRC
Pump	Radial Bearing	SKF
	Double Mechanical Seal	John Crane
	Buna-N "O" Ring	Parker (Distributor)
	EP "O" Ring	Parker (Distributor)
	Impeller	Hazelton Pump Inc.
Pump Structure	Pump supporting column, discharge legs	WHC

2.2.1 Essential Features

It was recognized by the Defense Nuclear Facilities Safety Board (DNFSB) that this test pump was not specifically designed for long-term service in Tank SY-101. In June 1995 the DNFSB visited Hanford and asked the question, "how long will this test pump last and how will the essential features fail?" During the meeting with the DNFSB it was discussed and defined just exactly what essential features of the pump must operate. These essential features would allow the pump to operate for the purpose of extending the window for replacement. Operating with only essential features would definitely be outside the operating safety envelope and would require a waiver. There are three essential features:

- The pump itself (i.e., the impeller and motor) must operate.
- Nozzles and discharges leg must remain unplugged.
- The pump can be re-aimed, new waste targeted, even if manually.

Clearing a nozzle and discharge leg has only metal tubing which is inside the tank. The high pressure pump(s) to clear the nozzle/leg is above ground. Therefore, this feature is not the subject of this evaluation. Rotation or re-aiming the pump is done entirely above ground. Therefore, these features are not subject to this lifetime evaluation. Lifetime of the DACS, variable speed drive, and other items outside the tank are beyond the scope of the investigation.

2.2.2 Material Life and Failure Modes & Effects Analysis (FMEA)

The initial examination of pump's essential features longevity was directed at material radiation resistance, and to this end a detailed source term and shielding calculation was done. The investigation then focused on items in the pump itself and how they reacted to the radiation, heat, and chemical environment:

- The 480 volt power cable insulation
- Motor electrical insulation
- Motor oil
- Mechanical seal
- Radial and thrust bearings
- Impeller and pump structure

A FMEA was then performed on these six items, how each might fail, and how its failure would impact other items.

2.2.3 Severe Environment Qualification

A companion document referenced by this Lifetime Expectancy Report is the Severe Environment Qualification report (Ref.1). This report documents actual radiation and thermal qualification testing performed on the power cable and thermal testing on the Buna-N "O" rings in the mechanical seal. Life of other items was taken from manufacturer's or published data.

3.0 METHODOLOGY

3.1 Logic Model for Calculating System Lifetime

System failures stem from failures of the individual components making up the specific systems. Accurate failure rates of systems can be determined either from statistical data on failures of the system as a whole, or by synthesizing from data on failure rate (or probability) of the individual components making up the total system. Frequently, one wishes to predict failure of systems which have not been operational long enough and have not experienced enough failures to produce an adequate failure database. Such is the case with the mixer pump, so synthesis from component data is required to determine its failure rate (i.e., its expected lifetime). Fault-tree methodology is useful for synthesizing system failures from component failures. A brief description of the fault tree methodology follows. Refs. 2 and 3 give more details to the interested reader.

Fault trees provide a graphical bookkeeping means of summarizing, categorizing and portraying the individual contributors to system failure. Most component failure probabilities are combined by "and" gates and "or" gates. The failure probability of a system whose failure components are connected by "or" gates (i.e., in which any one of the component failures fails the system) is the approximate sum of the individual probabilities. The failure probability of a system whose failure components are connected by "and" gates (i.e., in which the failure of all components is required to fail the system) is the exact product of the individual probabilities. A component whose failure cannot, by itself, with no additional failures, cause a system failure is known as a "redundant" component. Adding redundant components to a system is a common means of increasing system reliability.

3.2 Predicting Pump Failure Times

The preceding section addresses only overall failure probability, not failure timing. This study is concerned with estimating the time of failure. The overall prediction of the time of failure of the pump and the uncertainty in the prediction are determined by probabilistically combining the statistical representation of each failure mode. This section provides a minimum background to the reader who maybe unfamiliar with this topic.

3.2.1 Mean Time to Failure

The determination of mean time-to-failure (MTTF, also denoted mean time between failures, MTBF) of systems and components has always been a central issue in reliability analysis. Methods of its prediction are well established and, for relatively simple mechanisms (relays, valves, etc.), the failure behavior is generally well known and the MTTF is well known as a statistical distribution. However, because these behaviors are well known does not mean the time-to-failure of any given component can be accurately predicted, just as the lifetime of any one individual cannot be accurately predicted. Not all components of identical design and manufacturer fail at the same time and this variation is expressed as the time-dependent failure probability, of which Figures 2 and 3 are typical. It is these variations that produce the "distribution functions."

Figure 2 is the so-called Gaussian, or "normal" distribution which typifies the bulk of equipment failures. For complex systems designed for high reliability, the "lognormal" distribution of Figure 3 typically applies. Mathematically these are, respectively,

$$f(x) = \frac{1}{\sigma\sqrt{2\pi}} \exp\frac{-(x-\mu)^2}{2\sigma^2}$$

and

$$f(y) = \frac{1}{y\sigma_x\sqrt{2\pi}} \exp\frac{-(\ln y - \mu_x)^2}{2\sigma_x^2}$$

In these relations, f is the relative frequency (probability of occurrence) of the event (as, for example, a failure), x or y is the independent variable (time in the case of the distribution of failure probabilities), σ is the standard deviation (see definition, below), and μ is the mean of the normal distribution. Each lognormal distribution has a normal distribution associated with it and the subscripts x in the lognormal distribution expression suggest that the subscripted parameter is associated with that normal distribution. These are only two of many distribution functions that can be used to represent system failure distributions, but a great many component and system failures can be accurately characterized by one of these two.

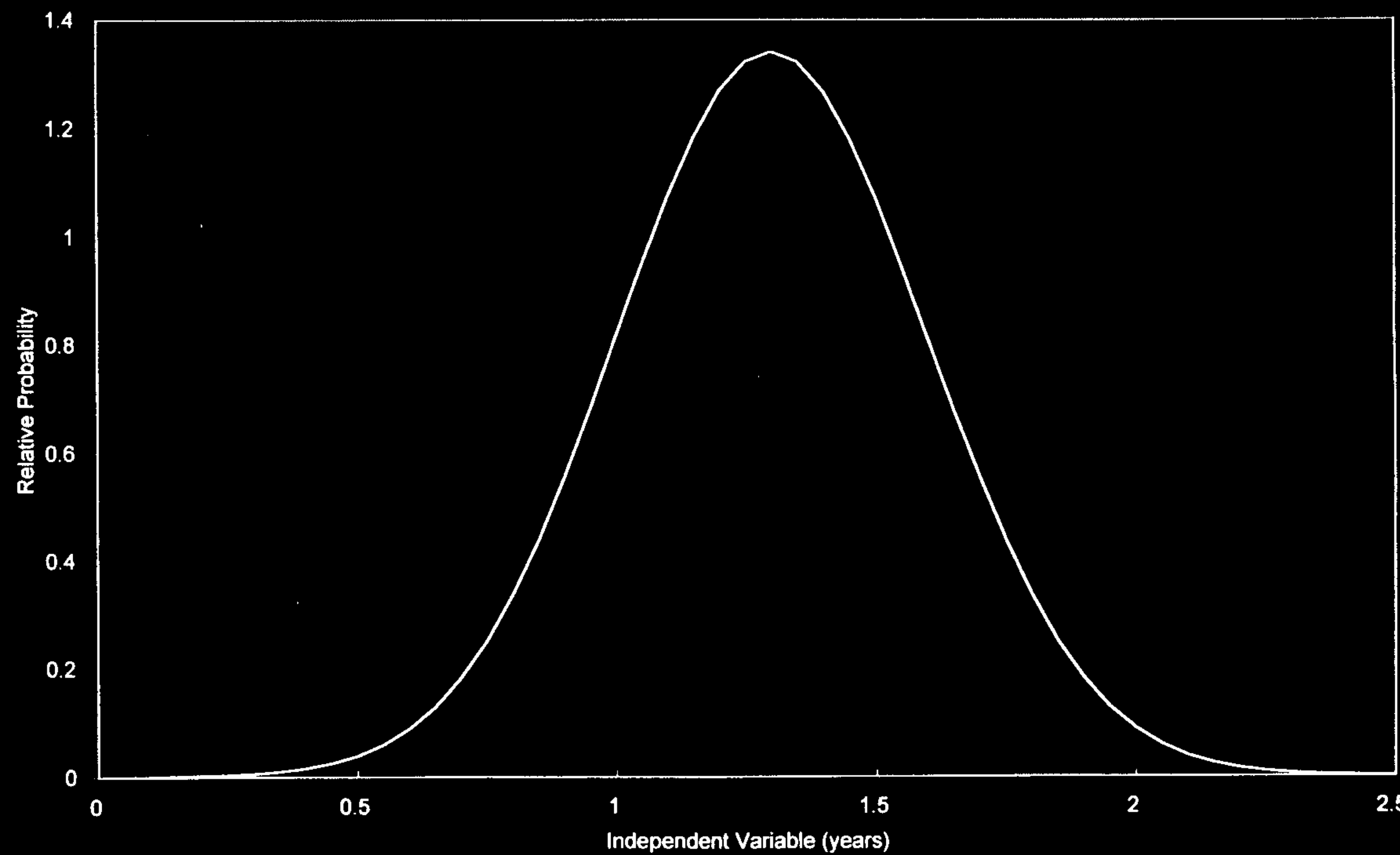

Figure 2. Normal distribution probability function.

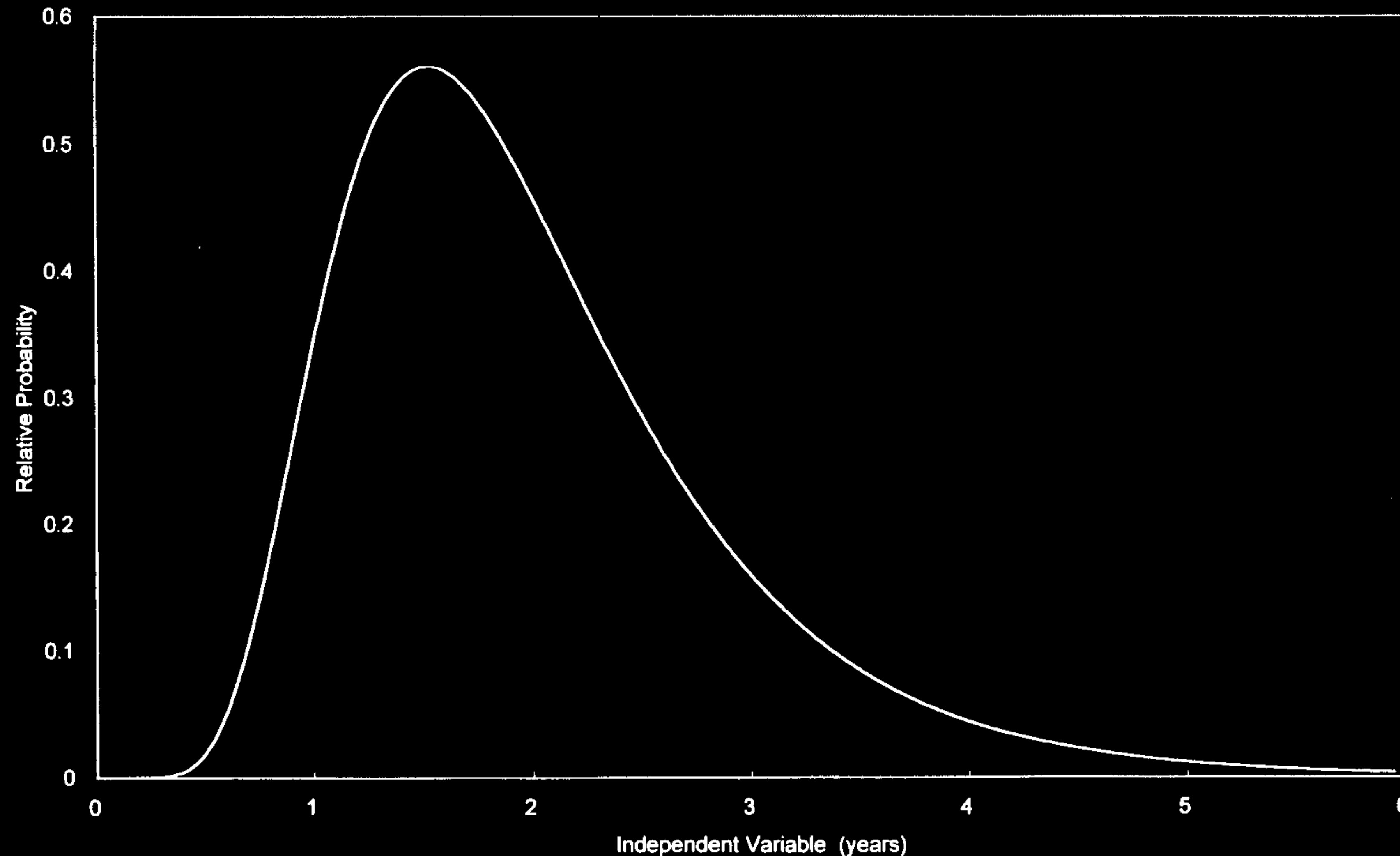

Figure 3. **Lognormal distribution probability function.**

In the absence of knowledge of the true failure distribution of equipment, the choice of one of these two will rarely lead to significant error (Ref. 4) because the failure properties are dominated by the mean and variance of the distribution rather than the distribution type. Tabulated failure data (Refs. 5 & 6) provide estimates of the mean failure rate and a measure of its dispersion (error) for a range of equipment and sub-elements parts. Even through the reported dispersion errors may depend more on the statistical sample size than the variation about the mean, they provide a starting point of estimates of failure rate mean and uncertainty. The Operational Reliability Analysis Program (ORAP) (Ref. 5) database provides the standard deviation as a measure of dispersion and the Savannah River Site (SRS) database (Ref. 6) provides an error factor as a measure of dispersion. ORAP assumes a normal distribution for failure and SRS assumes a lognormal. ORAP serves the gas turbine industry and SRS data relates to the production and processing of special nuclear materials.

3.2.2 Estimating Components Failure Times

In the nomenclature of statistics, all distributions are characterized by a "central tendency" and a measure of how the total distribution varies from this central tendency. The central tendency is that property that tends to concentrate the distribution, usually near the mid-point of the range of the distribution with the range being the total span of the independent variable. Measures of this central tendency are mean, median, and mode, which refer, respectively, to the average value of the independent variable of the distribution; to that value of the independent variable below which 50 percent of the dependent variable values (i.e., probability) lie, so it is often called the 50th-percentile value (see definition below); and, finally, to that value of the dependent variable having the highest value of the distribution function, f. On Figure 2, which represents a symmetric distribution, these three measures coincide.

The usual measurement of how the total distribution varies from the "central value" is by the "variance" or by its square root, the "standard deviation", denoted by σ. Mathematically, the mean and the variance are, respectively:

$$\mu = \int_{-\infty}^{\infty} t f(t) dt$$

$$\sigma^2 = \int_{-\infty}^{\infty} (t-\mu)^2 f(t) dt$$

The mean is the first moment of the distribution about 0 and the variance is the second moment around the mean.

The accuracy of prediction of the mean is specified by "confidence limits", which define the range of the independent variable (time, in this case) associated with a given percent confidence. For example, one might indicate that the failure of a given pump operating under specific conditions occurs between 4 and 5 years with 90 percent confidence. Confidence limits are determined from the "percentiles" of the distribution. The xth percentile is that point on the distribution below which x percent of the probability lies. The 5th and 95th percentiles for various normal distributions are illustrated in Figure 4. The 90 percent confidence band of a distribution is that range between the 5th and 95th

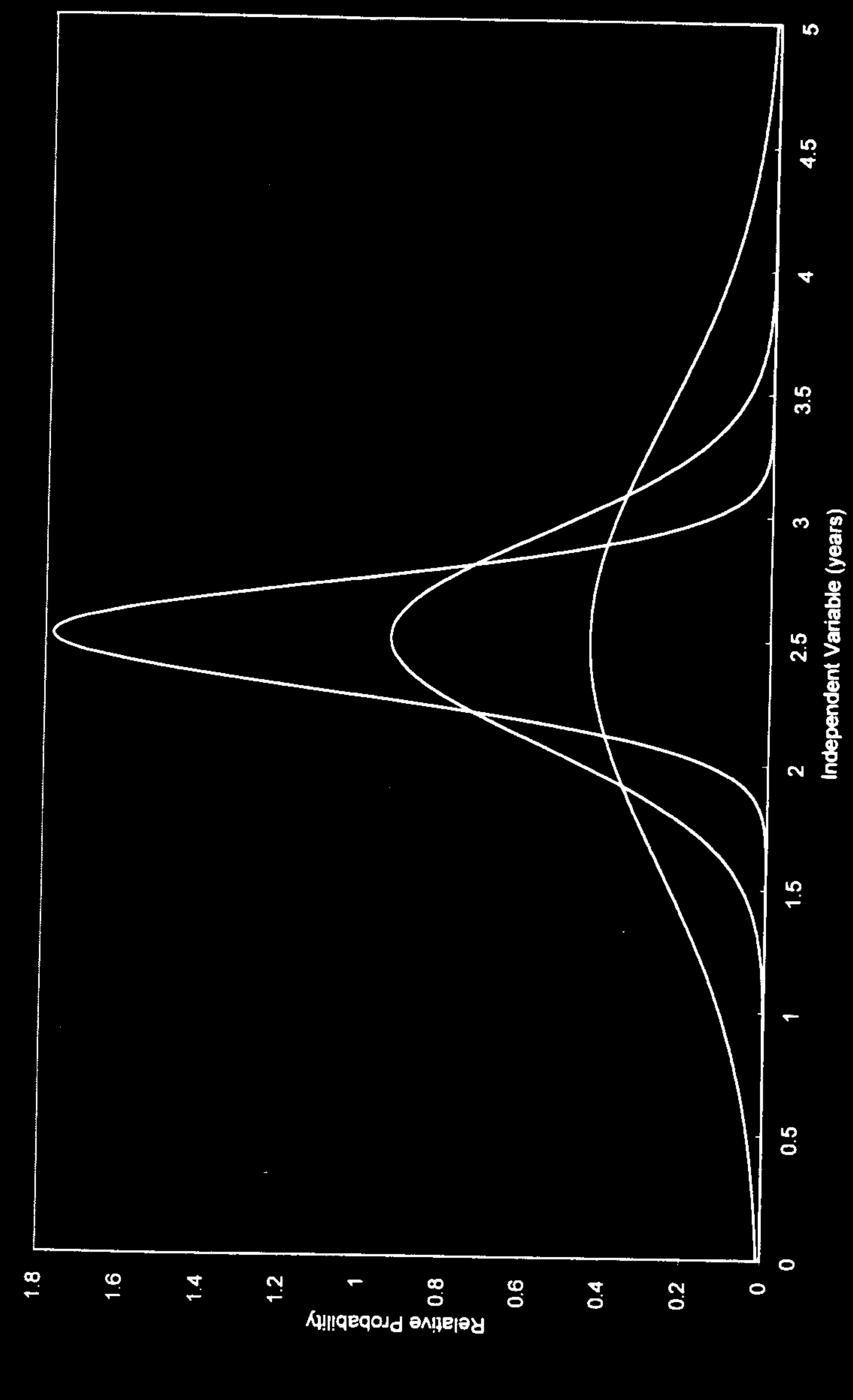

Figure 4. Variance effects in a normal distribution probability function.

percentiles. From Figure 4, one can see that the 90 percent confidence band for a normal distribution with a mean of 2.5 and a variance of 0.8 is six times as great as one having the same mean with a variance of 0.05.

With lognormal distributions, it is common to express the accuracy with the "error factor" which is the ratio of the 95th to the 50th percentile (which is the same as the ratio of the 50th to the 5th, or the square root of the ratio of the 95th to the 5th).

It is important to know the accuracy of prediction in order to respond appropriately to the failure information. It would be fruitless, for example, to attempt preventive maintenance on a system whose failure prediction had an extremely wide error band associated with it.

The reader who wishes to delve deeper into the statistics of failure is directed to Ref. 4.

3.2.3 Estimating System Failure Time and Uncertainties

For prediction of the failure time of the mixer pump and its associated error, several different independent modes of failure of individual components having different means and variances must be combined. One common way of estimating of the mean failure time is simply the reciprocal of the failure frequency determined by the operative fault tree. However, such an approach provides no error estimate, and additionally it underestimates the time to the first failure in a series system (series systems are those connected by an "or" gate in a fault tree.). To overcome this situation, it is necessary to construct the appropriate joint probability distribution function from the following relation; which applies to non-repairable systems that represent the mixer pump maintenance scenario; and which are connected in series:

$$p_c(t) = \sum_i p_i(t) \prod_{j \neq i} \left[1 - P_j(t)\right]$$

where $p_c(t)$, $p_i(t)$, $p_j(t)$ are the t-dependent failure probabilities for the system, the ith and the jth failure modes, respectively; and

$$P_j(t) = \int_{-\infty}^{t} p_j(t)\, dt$$

or the probability that the system will have failed by mode j at time t if the mode j were acting independently, so that $1 - P_j(t)$ is the probability that the system will not have failed at time t by mode j. The summation and the continued product are over all failure modes. This relation illustrates the intuitively obvious fact that the composite failure is dominated by the early failure modes. The factor $1 - P_j(t)$ approaches zero when the mean of the j distribution is significantly less than that of the i distribution

For this study the system distribution is calculated numerically with the QUATTRO PRO[1] spreadsheet program to obtain both the mean and the variance.

3.3 Required Assumptions

Often complete information required for a failure distribution cannot be found and assumptions are required. In order to assure that failure frequencies are not underestimated, these assumptions need to be conservative. The gathered data for the pump provides either a range or a single value. When the range is provided, it is assumed that the lower limit is the 5th percentile and the upper is the 95th percentile which defines the variance (or error factor) for either a normal or lognormal distribution. For sensitivity

[1] Quattro is a trademark of Borland International, Inc.

studies, it is assumed that the variance or the error factor is larger than those of the available data and mean failure times plus the 5th and 95th percentiles are calculated on the basis of the larger error.

For the two redundant seal systems modeled the failure time was estimated as the reciprocal of the failure rate from the fault tree rather than the more exact model provided by the joint probability distribution. With the common-cause effects considered it was judged that this estimate would not lead to serious errors.

It is important to note that for the over all pump failure probability determination the joint distribution function was used. This distribution does not require the constant failure rate assumption.

4.0 COMPONENT DESCRIPTIONS AND FAILURE MODES

The determination of failure rate for the mixer pump has been developed based on failure rates for the individual components shown in Table 1. Failure modes analysis was performed on each component to determine the cause of failure. The cause of failures could be one of, or a combination of, the following failure factors:

- Mechanical
- Electrical
- Chemical
- Thermal
- Nuclear Radiation Effect.

Failure data are determined for each failure factor of each component. In the following sections, each component is described and the respective failure modes and failure rate data are determined. For this study, all the failure data on these specific mechanisms are assumed to be complete failures (i.e., the pumping operation is stopped).

The pump is a centrifugal mixing pump manufactured by Hazelton Pumps Inc. The pump shaft is an integral unit with the electric motor shaft. A double thrust bearing above the motor and a radial bearing between the motor and the pump support the unit. A double mechanical seal protects the motor from waste fluid entrained into the motor. The centrifugal impeller is attached to the shaft with a key-slot and held in place with the threaded end of the shaft and a keyed nut.

The impeller can fail by breaking vanes and by loss of attachment to the drive shaft. The latter mode fails the pumping operation, of course, but for the vane rupture without the pieces binding and preventing pump rotation, pumping is degraded but does not cease.

4.1 Power Cable Insulation

The mechanism to cause mixer pump failure could be the degradation of the insulation material. The insulation material is silicone rubber. The insulation thickness is about 1.9 mm. The physical and thermal properties of this material were synthesized from information provided by manufacturers. Three insulated conductors and one ground conductor are housed in a steel conduit with internal diameter of 6.27 cm (2.469 in.) and external diameter of 7.3 cm (2.875 in.). The gap between the conductors and the conduit is filled with nitrogen gas pressurized to 10 psig.

During the pumping operation, the electrical current of 158 amp passes through three copper conductors. The conductor diameter is 1.22 cm (0.48 in.), with corresponding resistivity of 4 x 10^{-4} ohm/m. This results in heat generation in each conductor of about 9.9 Watt/m.

The pump operates for 25 minutes in each run which translates to a duty factor of 7.44 x 10^{-3}. During pump operation, the copper temperature increases, which in turn increases the insulation temperature. Due to the gap, it is unlikely that steady state condition can be reached in 25 minutes. It can reasonably be assumed that the conduit temperature will not rise appreciably during the 25 minutes and therefore we assume that the heat generated by the electrical current increases only the copper and insulation temperatures.

An estimate for the insulation temperature is heat content calculations for two limiting cases. In the first case, we assume that all the heat generated in the conductor remains in the conductor during the 25 minutes of operation. This sets a maximum temperature of 87.8°C (190°F) for the insulation (assuming 47.2°C ambient temperature). In the second case, we assume that all the heat generated in the conductor will raise the temperature of both the conductor and the insulation, which results in temperature rise of about 26.7°C (48°F). The insulation temperature in this case will be about 73.9°C (165°F). The insulation performance degrades at higher temperatures, and therefore, one of the failure modes of the insulation could be due to thermal effects.

The silicone rubber performance also degrades due to nuclear radiation. Loss of tensile strength (dielectric cracks) is the major cause of failure.

4.2 Motor Winding

4.2.1 Background Information And Assumptions

The lifetime of the winding and motor depend on details of construction, specifications, testing results, modifications, operation, limits to operation, and operating environment. Therefore, it is important to consider these aspects on the temperature of the winding in the evaluation of the pump lifetime estimate.

Construction - The motor, built by Reliance Motor Co., consists of a liquid-cooled partial rotor and stator combination. The motor contains a number of features that effect the temperature of the winding.

Specifications - The motor name plate provides the following information:

> The 150 HP motor is designed for liquid cooling in a 447TY frame. The stator winding is a wire wound, 3 phase, 460 volt, 6 pole, Y connection, with Class H (180^{o} C) insulation. At a frequency of 60 HZ, the full load current is 176 amperes for a speed of 1180 RPM with a service factor of 1.15, and a liquid (oil) gap of 0.04 inches.

Basic test results - Hazelton incorporated this motor into a 10 inch GN type SSB mixer pump supplying the pump, frame, housing and seals. The basic Reliance test motor, designed with the same insulation and thermal characteristics as the pump motor, reached a 40^{o} C rise as the steady state condition for operation on a test floor where external air cooling acted as the heat sink.

In-situ testing - The complete insulation system from the power supply to the motor winding consists of the motor insulation plus power cables between the top of the pump and motor phases. This insulation system was initially tested with a 500 volt Megger that indicated infinity from the phase wires to ground for all phases. This measurement compares with the tests made on the test floor indicating good quality windings at the beginning of the motor's operating life.

Modifications - After the rotor and stator were manufactured by Reliance, the Hazelton Pump, Inc. fit the stator into a submersible pump casing for use in the Hanford waste tank. The casing included a double mechanical seal system with an intermediate barrier oil system between the bearing oil and waste in the tank. Bearing lubrication oil sealed in the motor cavity also serves to enhance cooling of the motor winding. The liquid-cooled rotor uses a smooth rotor end plate to reduce windage, although a 5% adjustment for increase in losses due to oil cooling was noted during the pump test at Hazelton Pump (Ref. 7). This increases the steady state winding temperature. The motor was delivered to WHC, where it was modified for installation at the Hanford Site (Refs. 7, 8).

Operational protection - The mixer pump is controlled with a variable frequency drive (VFD) that includes a preset acceleration speed controller. During startup operations, the pump-motor takes approximately 5 seconds to get to full speed. The control system includes an Eaton dynamic variable torque drive (model #AF520107-0480), rated at 200 HP, 252 amp, and 480 volts. The VFD controller generates square wave pulses. The variable frequency control system also contains DC bus chokes and electronic protective features against: AC line transients, ground faults, input line over loads to electronic devices, motor overload, output contactor shorts, over-current, under and over-voltage, phase loss, and switching interruptions. These features protect the winding from excessive temperatures under faulted conditions.

Operational control - This control system also allows for careful control of the increase in speed which avoids large inrush currents during startup. The motor oil-filled stator-rotor housing is monitored for moisture and temperature. The pump instruments also measure the vibration, pressure at the pump discharge volute, and strain in the pump support column. Also included in the rotation assembly is a position resolver with back up limit switches to control and limit rotation speed, thereby preventing over-rotation and damage to the power and instrumentation cables. These sensors alert operators or the automatic software controls to interrupt power to the motor in cases where the winding is threatened with higher temperatures.

Operating environment - The operators typically run the motor at about 1000 rpm (50 cycles @ 158 amps). Under full speed, the motor ammeter has indicated from 156 to 181 amperes at different times using two different sets of calibrated ammeters. The start up and run data indicate that the oil temperature increases from 47°C to 87°C during the initial 25 minute run time[1]. This is a rather sharp increase in temperature considering the available waste heat sink, which is assumed to be at a stable temperature of 47°C. The pump motor in this application does not reach a steady state condition within the time of 25 minutes. Data show that the oil temperature increases at an average rate of 1.6°C per minute. Since lifetime estimates are sensitive to temperature during operation, operational conditions that limit temperature transients must be considered in the lifetime estimate.

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[1] The in-situ measured data differed from the measured factory test values, which confirms that the operating conditions differ significantly from the initial factory test conditions. This must be taken into account in the lifetime estimate.

4.2.2 Qualitative Factors Affecting Winding Life

Some factors that affect the thermal performance and cause the motor temperature to increase above the rated conditions are:

1. Using a reduced cooling configuration over the test conditions will increase the temperature of the winding. This was not measured during factory tests.

2. Use of variable frequency square wave pulses can increase losses in the winding over that expected for a sine wave. This was not measured during factory tests.

3. High viscosity material in the motor (oil filled winding and bearing chambers) increases the windage factor, thereby increasing current (e. g., 5%), as measured during factory tests.

4. Potential for increasing the current based on pump material loading that drags the pump can increase the current. This was not measured during factory tests, but may increase gradually as the machine ages.

Factors that reduce the temperature are:

1. Oil is cooled by heat transfer to the waste in the tank [temperature at 47° C].

2. The temperature is kept below the 87° C limit by monitoring and control of the motor.

3. Frequency control system for starting and running the motor holds down inrush current.

4.2.3 Estimate of the hot spot winding temperature.

The hot spot temperature (Tl) is defined as the ambient temperature plus normal operating temperature rise plus a standard allowance for the local hot spot. On the basis of test data, the externally measurable full load operating temperature rise under test floor conditions for this type of motor is estimated to be 40°C. This rise is multiplied by 1.5 to account for the additional losses and reduced cooling effects identified above. This adjustment accounts for the differences under items 1 and 2; that is, if the installed configuration could be tested on the test floor the measured rise would be 60°C. A standard 55°C local hot spot correction is conservatively added to the hot spot temperature assessment for estimating the life. The 55°C local hot spot accounts for weakness in the chemical coating on the laminations which can produce a local hot spot within the stator winding that is not easily detectable during testing. The hot spot temperature is therefore equal to:

$$Tl = 47 + 1.5 \times 40 + 55 = 162^{\circ}\ C$$

The initial lifetime estimate is based on an assumption that the motor is operated at this temperature continuously. As can be seen in Section 6.2, actual operation results in a significant increase in the thermal induced failure lifetime to greater than 1000 years.

4.3 Motor Oil

The motor oil cools the motor bearings and in turn is being cooled by the waste flowing around the pump housing during pump operations. The oil is Chevron Turbine Oil GST ISO 100. Since the oil can not be changed during the pump lifetime, Hazleton Pumps, Inc. has recommended that the oil temperature be maintained below 87.8°C (190°F) during pump operation (Ref. 9). During operational measurements, the bulk oil temperature reaches around 87.2°C (189°F). (Ref. 10) after 25 minutes of operation. The oil temperature starts to drop after pump shutdown. It reaches a minimum of 50°C (122°F). The average oil temperature during pump shutdown is about 53.9°C (129°F) if the pump operates three times per week. Therefore, the oil thermal lifetime should be evaluated both at 87.8°C and 53.9°C.

The oil pessure is about 5 psi above the waste pressure (the hydrostatic head of the waste at the seals is about 5 psi above atmospheric pressure). The oil pressure is kept at this level by nitrogen gas pumped into the pump column.

4.4 Double Mechanical Seal

This section describes the basis for estimating the life of the seal system. Review of the design drawings and discussions with representatives of John Crane Company, Hazelton Pump, Inc. and Westinghouse Hanford integration designers provide information for performing the FMEA in Section 5 and describing the system in enough detail to support a detailed estimate of the lifetime.

4.4.1 Background Information

The basic seal system design was developed by John Crane Company. The system consists of an upper and lower seal and a barrier chamber filled with cooling oil. The upper seal acts in conjunction with "O" rings as the barrier between the bearing oil and the barrier oil. Likewise the lower seal acts as a barrier between the process fluid (waste) and the barrier oil. The seal integrity can be managed using either a closed or open system to control the barrier fluid pressure. The specific Hanford design is based on system pressures where the barrier system is maintained at a slightly higher pressure than the process fluid static head and the bearing oil pressures. Other pressure configurations are permitted as long as the pressure difference does not exceed the seal system spring pressure which would lift the rotating seal and permit leakage.

According to the seal manufacturer John Crane, the rotating surfaces made of silicon carbide (SiC) are held in place with springs that result in at least 17 psig on the seal surface. The pressure rated on the John Crane seal drawing is 40 psig. This standard system is provided in parts to the pump manufacturer to install on the pump. The basic design life for a continuously operating seal will meet a minimum 2- year life based on manufacturer data for no leakage - at full pump speed. The pump motor operation of 65 hours per year does not challenge this lifetime measure on either time or velocity.

After the seals were manufactured by John Crane, they were installed by Hazelton Pump Inc. into the submersible pump for use in the Hanford waste tank. The barrier oil system selected for the seals was a closed system. This configuration was tested at Hazelton Pump Inc. and demonstrated appropriate behavior as an entire unit.

the seals were reassembled, by a factory expert and re-tested before installation in the waste tank. The upper seal has a normal pressure difference of 10 psi due to the nitrogen pressure and the lower seal has a normal pressure difference of 5 psi due to the hydrostatic head of the waste. Both are well below the 40 psig spring pressure on the SiC seals.

4.4.2 Qualitative factors affecting seal life

Some factors that reduce seal performance, and cause excessive wear are:

1. Loss of "O" ring leading to by-pass leakage.
2. Increased pressure on the rotating contact surfaces.
3. Large number of starts and stops per year.
4. Operation under conditions differing from the design point.

Factors that increase seal life are:

1. Keeping the seal velocity based on 1000 rpm far below the rated velocity based on 1800 rpm through use of frequency controlled system.
2. Using the slow starting speed provides much lower wear than for full voltage, 60 cycle starts.
3. Providing barrier oil cooling at a constant temperature of 47°C in the tank which cools the seals.
4. Monitoring the temperature to stay below the 87° C limit.
5. Use of SiC.

4.5 Secondary Seals

Three different "O" ring seals have been utilized in the construction of the mixer pump. These seals differ in their materials. They are made of 321 stainless steel, ethylene propylene (EP) and Buna-N. Each of these materials has different thermal characterization, which results in different thermal lifetimes.

The "O" rings temperature is close to tank ambient temperature (47.2°C) when the pump is not operating. During the 25 minutes of pump operation, the "O" rings temperatures will likely increase, but they will not increase beyond the maximum oil temperature which is around 87.8°C. The seal lifetimes are estimated both at 87.8°C and 47.2°C.

4.6 Pump Structure

The pump structure consists of the carbon steel pump column and legs, Alloy 744 (CD_4M CU) cast stainless steel jets and various type of 304L, 316 and 316L components. (Ref. 1) These structures are designed conservatively for any possible mechanical stress. The only failure mechanism considered credible for this component is chemical corrosion. Corrosion is a long-term effect compared to other failure mechanisms but is included in the analysis.

5.0 FAILURE MODES AND EFFECTS ANALYSIS (FMEA)

The purpose of this FMEA analysis is to qualitatively identify failures of the mixing pump system that affect the pump lifetime on an essential component-by-component basis. This examination also identifies how the system is protected from a variety of postulated failure modes which, in effect, helps extend the expected life of the motor. The FMEA process begins by considering the major components of the pump as basic elements, postulating failure modes and then demonstrating features in the design that control or reduce the effects of the postulated failure mode. This process helps identify the interaction between elements and supports the logic for the overall combination of failures in the reliability model that is used to quantitatively predict the overall life of the mixer pump.

The FMEA depicted in Table 2 uses a recommended format found in the IEEE 352 standard. Table 2 summarizes the qualitative assessment of failure modes for each element of the mixer pump.

The initial reliability model conservatively combined the results of each failure mode through an "or" gate, taking no credit for any form of redundancy. In the initial reliability model, loss of the seal system was found to be the dominant contributor to the pump failure. The FMEA was expanded in the area of seals to more precisely examine the failures associated with the seal system and examine the potential for redundant features that protect the system from a single failure. If such redundancies can be identified, "and" gates considering common cause failures can be used to combine the failure modes in the reliability model. The FMEA also permits differentiation between the failures that result in degraded pump operation by defeating redundant elements without loss of the overall function and those failures that leave the pump inoperable.

The more detailed FMEA examination of seals revealed that redundancy is highly dependent upon the selected operating conditions. The current design has a closed system whose volume totals 18 1/2 gallons. The barrier oil chamber contains about 5 1/2 gallons of oil and is essentially full. The barrier pressure chamber is lower than either the waste and bearing oil pressures. This means that only in-leakage to the barrier oil system could occur. If the upper seal fails, bearing oil would drain into the barrier oil, which is driven by the 10 psig pressure differential and could result in loss of lubrication to the thrust bearing. If the lower seal fails, waste driven by a 5 psig differential is assumed to cause failure of the lower seal by abrasion. If such leakage continued unabated, the failures would remove redundancy, since breach of any barrier results in a pump system failure mode. However, continuous leaks from either the bearing or waste environment cannot continue into the barrier chamber, which is a closed system. It is clear that the closed system would soon reach a pressure balance with the leaking seal, permitting only small amounts of waste or bearing oil to mix with the barrier oil. Thus, the pump could continue to operate in this degraded mode, until a second failure involving a leak from the closed piping system occurred. Thus, leakage from the "O" ring seals or the primary seals in either the upper or the lower assembly have a degree of redundancy, if operation can continue with small amounts of waste mixed with the barrier oil. Thus, failure of the upper and lower seals may be represented as a partially redundant system with a rather high common cause failure fraction of 20 to 40%.

If the barrier oil system was operated as an open system with external pressure control to keep the barrier seal pressure higher than the bearing and waste pressures, the reliability model could be considered redundant, but would be subjected to operator errors in maintaining the oil system. This would result in a potentially longer life, but with a greater uncertainty of early failure due to human errors. In the case of the open system, the pump

TABLE 2 MIXER PUMP FMEA

Date:	9/1/95			Page 1 of 6		
Plant:	Hanford Tank Farm			System: Waste tank 241-SY-101		
Reference:	SY-101 Hydrogen mixing pump			Analyst(s): Hannaman & Johnson		
Item	**Identification**	**Description**	**Failure modes**	**Effects**	**Safeguards**	**Actions**
Bay Associates · BY92-482	Cable Insulation	Silicone Rubber	Short or cable to cable faults due to current overload, high voltage, or excessive mechanical forces	High, Reduced or no current to the motor, potential for sparking in the cable housing, and fire mitigated by nitrogen atmosphere- pump failure	Adequate cable size using 125% margin (need some margin on the 176 amp. adjusted for temperature and length of cable).	Megger test the insulation
			Ground through insulation is due to thermal aging, high voltage, high current or mechanical load	High, Reduced or no current to the motor, potential for sparking in housing at the insulation failure to ground points between cable and tank structure - pump failure	Initial ground test voltages according to standard qualification tests	Use frequency control starting system
			Loose contacts or connections	local heating that leads to more rapid insulation aging - pump failure to start	Use of all copper cables	Perform continuity checks
Chevron GST ISO 32	Barrier Oil for Seals	Light oil	Accumulation of moisture and acids	Increase in corrosion on seals - pump degraded then failure	Closed system prevents outside contamination	No actions because of closed system
Chevron GST ISO 100	Oil for bearing lubrication and cooling Motor	Lubricating oil	Formation of oil sludge	Reduction of insulation strength in winding area, and increase wear on bearings, possible reduction in heat transfer from motor - pump degraded	Contamination of oil protected by the double mechanical seal system and barrier oil	Monitor start up current profiles, and temperature conditions
			Decrease in viscosity due to increased temperature	Increase in Bearing Temperature due to lack of lubricating ability - pump degraded	Protective trips keep temperature within operating range less than 190 F	Monitor temperature conditions

TABLE 2 MIXER PUMP FMEA (CON'T)

Date	9/1/95			Page 2 of 6		
Item	**Identification**	**Description**	**Failure modes**	**Effects**	**Safeguards**	**Actions**
Chevron GST ISO 100	Oil for bearing lubrication and cooling Motor	Lubricating oil	Increase in viscosity due to exposure to waste or oxygen	Increase in bearing temperature due to increase in motor windage - pump degradation, then failure	Contamination of oil protected by double mechanical seal, and pressurized nitrogen blanket	Monitor temperature conditions
Reliance Electric Class H	Motor winding stator	Iron laminations, magnet wire, and silicone insulation	Turn-to-turn short	Increase in current for the same load, imbalance in phase current - pump vibration then rapid failure	Class H 180 C insulation, Class H dip and Bake varnish	Monitor current, pump flow & frequency, Variable Frequency Drive (VFD) over-current and phase-loss protection and fuses
			Ground Fault	Current overload in one phase, pump fails to start	Class H 180 C insulation, Class H dip and Bake varnish	Monitor current, protective trips, VFD ground fault protection and fuses
			Loose slot insulation due to aging, dry out, temperature, radiation, chemical attack.	Material in air gap can jam rotor, fail to start	Class H 180 C insulation, Class H dip and Bake varnish, oil bath cooling, nitrogen blanket in housing.	Monitor current speed and discharge pressure, and protective trips
			Break down of chemical insulation coating between stator laminations	Melts iron and causes very hot point for rapid electrical insulation breakdown - pump failure	Grain oriented laminated steel, Class H dip and Bake varnish	Monitor insulation quality - meggers tests, protective trips
Reliance Electric Squirrel Cage	Rotor	Iron laminations, steel shaft, and cast rotor circuit	Open rotor circuit	Lack of torque, heating on end rings and increase in oil temperature - pump degraded operation	Cast end rings and slot material	Monitor start up times and current flow

TABLE 2 MIXER PUMP FMEA (CON'T)

Date	9/1/95			Page 3 of 6		
Item	**Identification**	**Description**	**Failure modes**	**Effects**	**Safeguards**	**Actions**
Reliance Electric Squirrel Cage	Rotor (cont)	Iron laminations, steel shaft, and cast rotor circuit	Break down of chemical insulation coating on rotor laminations	Melts iron and causes hot point, and change in power factor through imbalance in rotor circuit- pump failure	Key to shaft prevents slippage	Monitor vibration, phase current balance, and start up times
			Bent shaft or loose shaft/rotor connection	Excessive vibration - pump degraded operation	Use of shrink fit plus key provides basis for rugged rotor	Monitor vibration
			Axial mis-alignment with stator	Increase in current, increase or decrease on thrust bearing load - pump degraded, loss of expected life	End play allowance allows for slight mis-alignments	Single point bearing lock, design and temperature monitoring
SKF 8320	Double thrust Bearing/ housing	Steel races and ball bearings	Wrong tolerance	Excessive vibration - pump failure	Initial testing	Operational monitoring
			Axial mis-alignment in housing	Excessive Temperature - pump degraded, loss of life due to early wear-out	End play tolerance	Single point bearing lock, design and temperature monitoring
			Loss of lubrication or contamination	Excessive Temperature - pump degraded, loss of life due to early wear-out	Large volume of oil	Limited to a 25 min. operating cycle
			Pitting of balls, races due to inadequate grounding	Excessive Temperature - pump degraded, loss of life due to early wear-out	Insulated frame	Limited to a 25 min. operating cycle

TABLE 2 MIXER PUMP FMEA (CON'T)

Date	9/1/95			Page 4 of 6		
Item	**Identification**	**Description**	**Failure modes**	**Effects**	**Safeguards**	**Actions**
SKF 321 S	Radial Ball Bearing	Steel races and ball bearings	Becomes loose	Vibration then stator rotor interference - pump failure	Protected lubrication environment and maintaining oil below 190 F.	Monitor temperature
			Loss of lubrication or introduce contamination	Excessive Temperature - pump degraded loss of life due to early wear-out	Double seals limit corrosion	No actions because of closed system
			Loss of end play tolerance due to binding or chemical attack	Rapid temp. increase during operation - pump degraded, loss of life	Shaft pinned at one location - double thrust bearing, double mechanical Seal protects lubricating oil, stabilized ph waste environment limits chemical attack on metal parts	Monitor start up current
			Circumferential misalignment	Excessive temperature results in early wear-out - pump failure	Pre-installation testing	Initial operation verifies OK
Double mechanical seal (John Crane CF-SP-98539)	Lower primary rotating seal system	SiC to SiC interface	Leaks at rotating Surface, due to wear at start-up and running	Permits waste to enter barrier oil chamber leading to degraded operation and then pump failure	Slow speed starts and barrier oil cooling limits seal wear, closed barrier system limits amounts of in leakage as pressures balance	No actions because of closed system
			Binds the shaft	Locks rotor during operation - pump failure	Use of SiC seals and barrier oil cooling limits wear	Monitor start up current & RPM
Double mechanical seal (John Crane CF-SP-98539)	Upper primary rotating seal system	SiC to SiC interface	Leaks at rotating Surface, due to wear at start-up and running	Permits motor oil to enter barrier oil chamber which drains lubrication from bearing housing leading to rapid pump failure	Slow speed starts and barrier oil cooling limits seal wear, closed barrier system limits amounts of in leakage as pressures balance	Monitor startup current & Motor RPM
			Binds the shaft	Locks rotor during operation - pump failure	Use of SiC seals and barrier oil cooling limits wear	Monitor start up current & Motor RPM

TABLE 2 MIXER PUMP FMEA (CON'T)

Date	9/1/95			Page 5 of 6		
Item	**Identification**	**Description**	**Failure modes**	**Effects**	**Safeguards**	**Actions**
Seal Barrier Oil Supply	Oil supply system and boundary	Piping, pumps, fittings and seals	Loss of fluid from barrier oil system through "O" rings, pipe fittings, etc., losses of fluids from barrier oil system	Loss of closed system integrity permits waste to mix with barrier oil when pressure is low, primary seal damage follows due to abrasion, then rapid pump failure	The static "O" rings and rotating Seals provide barriers between the barrier oil supply and waste	No actions permitted in the closed system
			Barrier oil pressure too low	Low barrier pressure permits bearing oil leakage in upper seal, and waste leakage in lower seal motor pump damage follows rapidly	Rotating SiC primary seal permits limited operation without sufficient oil pressure. This allows time for external recovery of nitrogen pressure system.	No actions permitted in the closed system
			Barrier oil pressure too high	Oil leakage forced through primary seals, damage to mechanical rotating seals, rapid loss of barrier fluid and rapid pump failure	Closed system prevents any mechanism from increasing pressure above the motor bearing or waste pressures & seal design under pre-wear conditions can withstand 5 to 10 times the normal pressure	No actions permitted in the closed system
"O" ring 245 John Crane	Upper seal Buna - N "O"rings	Boundary between motor oil and barrier oil	Leaks due to dry out & loss of flexibility caused by excessive temperature, oxidation, hydrolysis, radiation, or mechanical forces	Leakage of oil from 10 psig motor chamber to atmospheric barrier oil chamber, pump failure due to failure of thrust bearing	One Buna N "O" ring in the upper seal maintain 10 psig pressure difference between the bearing oil and the barrier oil.	No actions

TABLE 2 MIXER PUMP FMEA (CON'T)

Date	9/1/95			Page 6 of 6		
Item	**Identification**	**Description**	**Failure modes**	**Effects**	**Safeguards**	**Actions**
"O"rings 251 Hazelton & 240 John Crane	Lower seal Buna-N "O"rings	Boundary between waste and barrier oil	Leaks due to dry out & loss of flexibility caused by excessive temperature, oxidation, hydrolysis, radiation, or mechanical forces	Leakage of caustic waste at 5 psig into barrier oil chamber causes abrasive failure of the seals - pump degraded, loss of life due to accelerated wear-out	Retaining rings and the torturous path limit leakage, waste degradation attack on the lower mechanical primary seal may be slow	No actions because of closed system
"O"rings - Hazelton 265 & 262	Housing mount Buna-N /Ethylene Propylene "O" rings	Boundary between Motor oil and waste tank	Leaks due to dry out & loss of flexibility caused by excessive temperature, oxidation, hydrolysis, radiation, or mechanical forces	Oil leakage from the bearing chamber will cause a loss of motor lubrication leading to pump failure	Redundant seals limit the potential for a leak from the motor oil chamber to the waste tank	No actions
Hazelton 7403.3 SS	Impeller	Stainless Steel	Corrosion	Loss of blades, lock rotor prior to start-pump failure	Appropriate clearances in design	Minimize stress using variable frequency startup
			Slips on shaft	Fails to pump	Rotor keyed to shaft	Minimize stress by keeping speed below rated speed (1800 RPM)
			Loss of blade (high frequency cycling)	Damage the pump vanes, loss of head - pump failure	Margin in pump impeller mechanical design	Pump flow monitoring

would continue to operate as long as the barrier oil pressure can be maintained at a level sufficient to maintain controlled oil leakage through the seal into the waste. This degraded operation should keep the pump seals from further degradation, as long as the amount of waste in-leakage is small and causes only a slow degradation of the upper seal. Failure would occur, if the waste in the lower seal caused significant abrasion. If both the upper and lower seals fail, significant oil loss through either the "O" ring seals or the upper primary seal would drain the bearing oil into the waste until bearing lubrication was lost.

Another failure mode of the seals is binding that produces a locked rotor situation. As long as conditions of the pump are monitored (through careful monitoring of the startup current and speed), a more accurate prediction of the probability of failure in the next cycle can be developed. With a small amount of leakage, degraded operation could continue as long as the barrier oil system maintains enough cooling and seal lubrication to prevent severe seal damage. While the primary seal failure mode of binding is rare for the type of seals used, it is a common cause failure, and therefore, in the reliability model must be modeled with an "or" gate. The failure rate associated with this failure mode is much lower than other failure modes like excessive wear as discussed before.

In summary, the effect of most failures is to render the pump inoperable. A few failure modes cause increased degradation of the system, and the pump is protected by a degree of redundancy. Several failure modes can cause some electrical sparking within the pump housing, but none of them can affect the waste tank environment unless an additional structural failure of pump housing is assumed.

6.0 FAILURE DATABASE

The failure database for each pump component has been compiled using experimental evidence, analysis, and information received from component manufacturers. The data base was organized in terms of mechanical, electrical, thermal, chemical and nuclear radiation effects on failure rates. The methods of estimating the failure rate of each component are discussed below with an estimate of the failure rate.

6.1 Power Cable Insulation

The cable insulation performance could degrade due to either nuclear radiation effect or/and thermal effect. The performance degradation due to radiation and thermal effects have been measured as part of the Phase II of this program (Ref.1). The results indicate that the cable can last about 40 years before any sizable thermal performance degradation can occur. The nuclear performance degradation is about 30 years (Ref. 1).

6.2 Motor Winding

The motor winding insulation performance could degrade due combinations of radiation and thermal effects. Mechanical vibration effects on the winding are negligible. The lifetime due to radiation effect has been estimated analytically (Ref. 1) and is about 150 years.

Using the information from Reference 11, the rated life from testing data for a class H winding at 180° C is 60,000 hours with a lower bound of 18,000 hours. This is adjusted to the operating temperature of the windings during the application using Arrhenius model. The equation is derived from the idea that insulation degradation is governed by a chemical reaction rate at a constant temperature. Each class of insulation system has a different reaction rate.

ln(tR/tl) = f/k ((1/TR)-(1/Tl))

where

tR = the rated life at TR
tl = the time at Temperature Tl
f = 1.38 eV from ref. 1 for Class H 180° C winding systems
k = Boltzmann constant = .8617E-4 eV/°K
TR = the reference temperature (180° C)
Tl = the operating hot spot temperature (162° C)

With continuous operation at these conditions for 8760 hours per year, the lower bound of rated life is 9 years and the expected life is 30 years. If the actual operational conditions at the these temperatures are considered (e. g., 65 hours per year), the lower bound lifetime estimate for motor winding thermal effects increases to 1200 years. This estimate does not consider mechanical effects such as vibration, material in the air gap, and chemical corrosion factors in the oil that can lead to winding failure. Key assumptions are that the operation is limited to 25 minutes to ensure that the winding temperature stays within the limits of the Tl estimate. Routine Megger tests of the pump column power cables and motor could be used to establish a historical pattern for predicting the suitability for service and verifying the remaining life estimates.

6.3 Motor Oil

Oil can lose its lubricating properties through an accumulation of moisture and acids, through the formulation of sludge, thermal effects, exposure to radiation, and introduction of suspended materials. The thermal and radiation effects have been evaluated through independent qualification testing to be greater than 30 and 150 years respectively for the conditions of mixer pump operation (Ref. 1). When synergistically considering the operating conditions of the oil, the independent test values were modified to account for failure modes identified in the FMEA and from operating experience. For example, oil within electrical equipment is provided to maintain both lubrication as well as dielectric strength over the entire range of operating temperatures. Such electrical equipment may experience special lubrication oil breakdown at about 15 years, based on observed experience of vertical oil lubricated motors in harsh operating environments at high temperatures. The 15 year lifetime is our base case mean value. The lower 5 % confidence bound is estimated to be 6 years for the base case. Lifetime of the bearing system in this design is reduced if impurities from the insulation material and suspended particles reduce the lubricating properties of the oil. During the initial startup tests outside the waste tank at Hanford the oil was drained and replaced. Under the limited tests no impurities were found. Sampling of oil impurities in this manner does not account for the synergistic effects that are expected over a long time period under the current operating conditions.

Since the behavior of lubricating oil in this application has considerable uncertainty with regard to the synergistic effects on its lifetime, a sensitivity study was carried out to evaluate variations in the oil failure rate on the mixer pump annual failure probability. The base case estimate was increased and decreased by 50% to evaluate the effects of the uncertainty about the synergistic effects on the overall mixer pump lifetime estimates.

6.4 Double Thrust and Radial Bearings

The radiation effect and mechanical factors could degrade the bearing performance. The lifetime due to radiation effect has been estimated analytically (Ref.1) and it is about 60 years for both bearings. The double thrust bearing life information due to mechanical factors (wear, vibration) was received from Hazelton Pump Inc., and it is about 24,700 hours of operation under maximum loading of 8,200 lbs. This translates into 380 years assuming 65 hours of annual operation. This lifetime will be reduced to 137 years due to frequent start-stop cycles of the pump. The radial bearing lifetime is much longer, since it is not subject to axial loading.

6.5 Double Mechanical Seal

6.5.1 Failure Rates of Seal Components

The double mechanical seal is composed of mainly primary seals (SiC) and secondary seals ("O" rings).

The primary seal performance degradation is mainly due to mechanical factors like wear and vibration. The expected lifetime of the seal is 2 to 10 years of actual continuous operation according to the John Crane Company, manufacturer of the double mechanical seal.

There are five "O" rings that have been used in the double mechanical seal. Two of these rings separate the motor oil from the barrier oil and the other three separate the barrier oil from the waste. There are two other "O" rings that have been used in the pump structure to separate the motor oil from the waste.

The following statements summarize results of a series of telephone conversations with John Crane Company (mechanical seal suppliers) and Parker Corp. ("O" ring suppliers) about the lifetime of the Buna-N "O" ring:

a. Neither company is familiar with any thermal degradation performance data at room or higher temperatures.

b. The shelf life is about 6 years. This means that if the rings stay on the shelf for more than 6 years, the manufacturer recommends that the seals be discarded and not be used. They don't know of any scientific basis for this lifetime.

c. They will not accept "O" rings from manufacturers that are older than 2 years. Therefore, the maximum cumulative shelf life could be around 8 years.

d. The shelf life numbers are per military specification.

The lifetime of the "O" ring due to radiation effect has been estimated analytically (Ref.1) and it is about 50 years. The lifetime due to thermal effect is about 10 years based on the temperature aging test recently completed (Ref. 1).

6.5.2 Failure Rate of the Seal System

For the purpose of lifetime estimating, the seal system consists of the following elements: five static and two dynamic "O" ring seals made from Buna-N material, the four SiC mating surfaces in the upper and lower primary seals, the springs holding the primary seals in place, the barrier oil fluid, and the integrity of the closed oil cooling structure. The lifetime estimate must also consider the operational conditions that lead to wear from operating and start-stop cycles.

The system can be split into three specific parts to consider the effects of redundancy and common cause failures on the overall pump failure rate. These parts are: the upper primary seal, the lower primary seal, and the motor housing "O" ring by-pass. A basic sketch of the seal system is shown in Figure 5.

The redundancy model for the seal system is based on both the common cause and redundant features of each element of the seal design. The aim of this model is to better consider the features of the design that act as redundant elements for the purpose of this pump lifetime estimate.

The effect of a common cause failure is to defeat redundancy in otherwise independent systems by a coupling mechanism that propagates between the components. Common cause failures by definition include dependent failures that are coupled by a common mechanism (Ref.12). In the case of the seals, there are a number of redundant elements whose failure under specific conditions would not cause a pump failure, because the redundant element can continue to perform the required function (e.g., small leakage of waste into barrier oil or motor oil into barrier oil). Examples of common cause seal failures that contribute to pump failure are: system shocks such as a high pressure that blows both seals out permitting motor oil to drain into waste, spatial coupling where high vibration of the shaft can cause mechanical failure of both rotating seals, environmental stresses such as high temperature that causes very rapid wear on both the mechanical and Buna-N "O" ring, functional coupling within the waste tank that introduces abrasive material causing rapid wear on both mechanical seals, causal failures where the failure of one element of a redundant system introduces load or stress on the other element(s), which leads to premature cascading failures, or operational actions by the operators such as overpressurizing the nitrogen chamber causing oil to by-pass the mechanical seals.

In failure format, the probability of failure for a simple redundant system with common cause failures can be represented as (Ref. 15)

$$P = 1 - 2e^{-\lambda t} + e^{-(2-\beta)\lambda t}$$

where β is the fraction of common cause failures to all failures, λ is the failure rate and t is time since the operation began. The beta factor model separates the common cause and independent failures so that their treatment is explicitly accounted for in the overall reliability model of the seal system. Generic data have been collected on large and small redundant systems and organized to quantify the contributions to common cause failures (Ref. 12). Analysts must examine the systems for each contribution to the loss of redundancy when assigning the common cause failure fraction. In the case of the seals a high fraction (0.28) was assigned to account for the close spatial and environmental coupling starting with recommended values for pumps, which is a factor of 2 greater than the typical average for all redundant systems of $\beta = 0.1$. This standard common cause failure fraction of 0.1 has been applied to the double "O" ring seals between the bearing oil and the waste (Ref. 12). A representative fault tree diagram for the redundancy model is shown in Figure 6. The value of β will be changed in section 7 to evaluate its sensitivity

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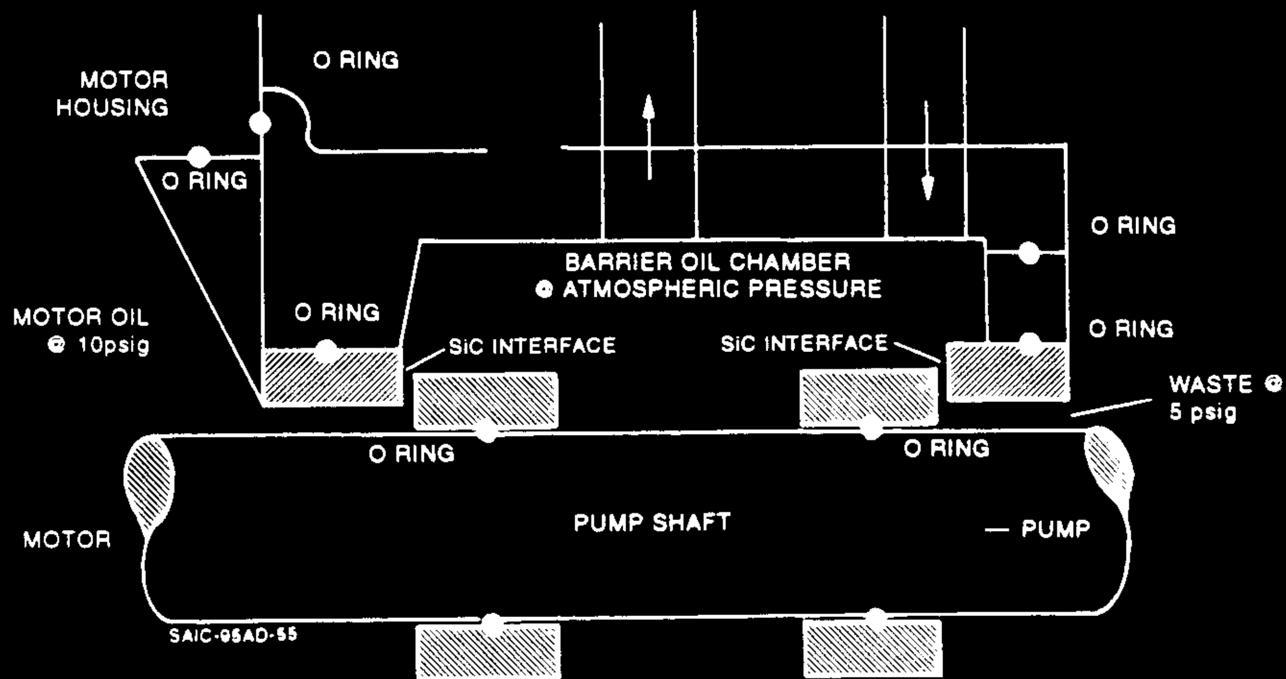

Figure 5. Sketch of the Double Mechanical Seal and "O" Rings.

redundant systems and organized to quantify the contributions to common cause failures (Ref. 12). Analysts must examine the systems for each contribution to the loss of redundancy when assigning the common cause failure fraction. In the case of the seals a high fraction (0.28) was assigned to account for the close spatial and environmental coupling starting with recommended values for pumps, which is a factor of 2 greater than the typical average for all redundant systems of $\beta = 0.1$. This standard common cause failure fraction of 0.1 has been applied to the double "O" ring seals between the bearing oil and the waste (Ref. 12). A representative fault tree diagram for the redundancy model is shown in Figure 6. The value of β will be changed in Section 7 to evaluate its sensitivity effect on the mixer pump lifetime.

There are two primary seals with identical failure rate (λ_p)and seven "O" rings with identical failure rate (λ_o). λ_p consists of two failure components: a run failure rate, λ_r, and a demand failure rate, λ_d. The run failure rate provided by the designer is 2 to 10 years assuming continuous operation, and the demand failure rate can be estimated from generic pump failure data sources. We assume that the seals contribute about 10% to the overall pump failure on demand defined in typical component data bases that defines the pump as the overall system. Our survey indicates the demand failure rate for seal-caused failures is in range of 3×10^{-4} per demand. Thus, the failure rate is approximately equal to:

$$\lambda_p \cong \lambda_d * \# \text{ starts} + \lambda_r * \text{hours/per year of operation}$$

$$\lambda_p = 3\times10^{-4} \text{ per cycle} * 156 \text{ cycles/per year} + 1/2 \text{ year} * 65/8760 \text{ hours operation/per year}$$

$$\lambda_p = .05 \text{ per year}$$

Application of the above equation results in the following lifetime estimates for the total seal system.

$P_{\text{double mechanical system failure}}$:

$$= 1 - 2e^{-[(.32+.23)/2]*1} + e^{-(2-.28)(.32+.23)/2*1}$$

$$= 0.103 \text{ in a year}$$

The values 0.23 and 0.32 come from Figure 6.

The estimated life for each "O" ring in this seal system is about 10 years based on the temperature aging test recently completed.

$$\lambda_o = 1/10 \text{ years} = .1 \text{ per year.}$$

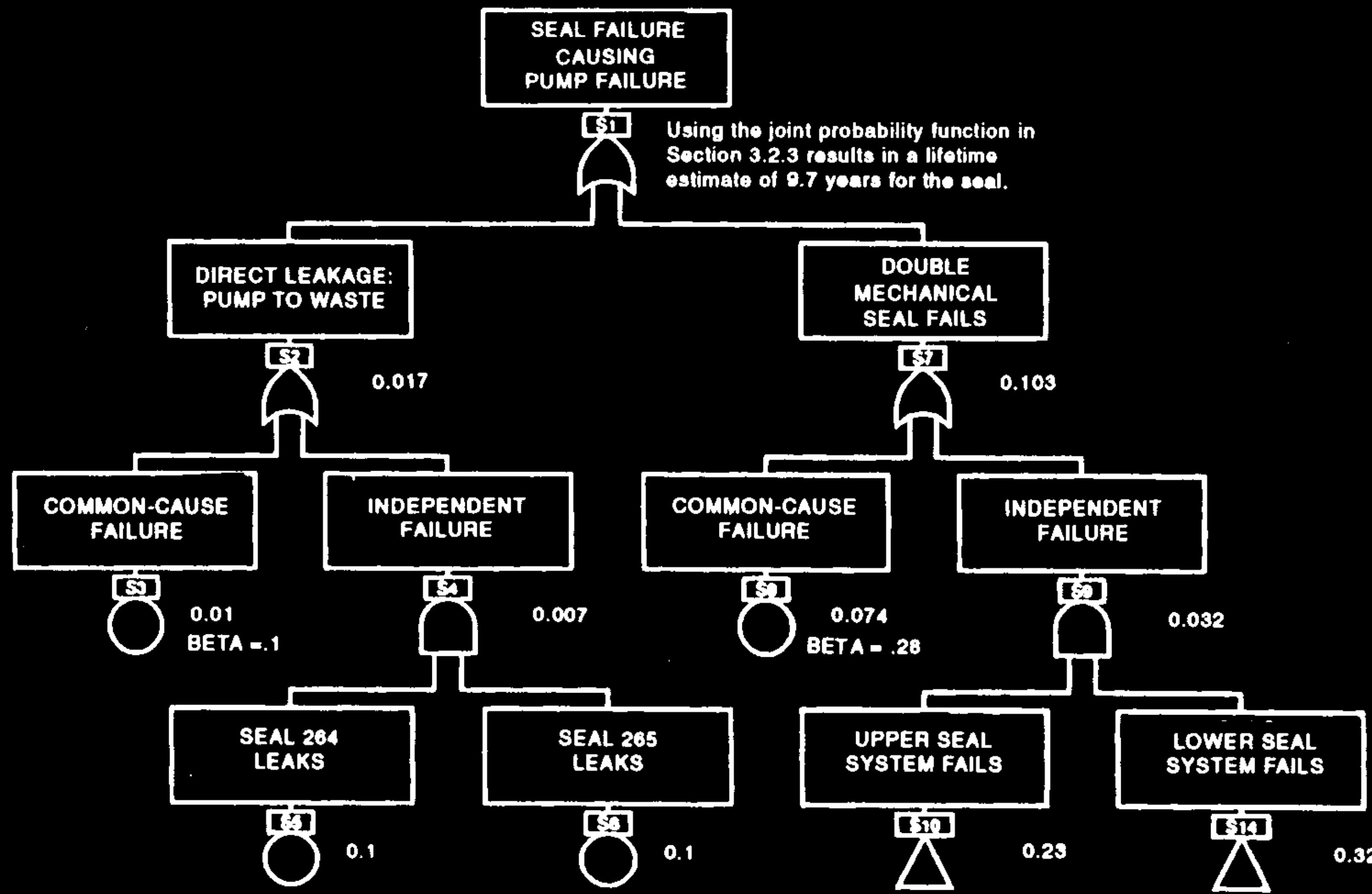

Figure 6. Seal System Fault Tree. (The number on the side of each box represents frequency of failure).

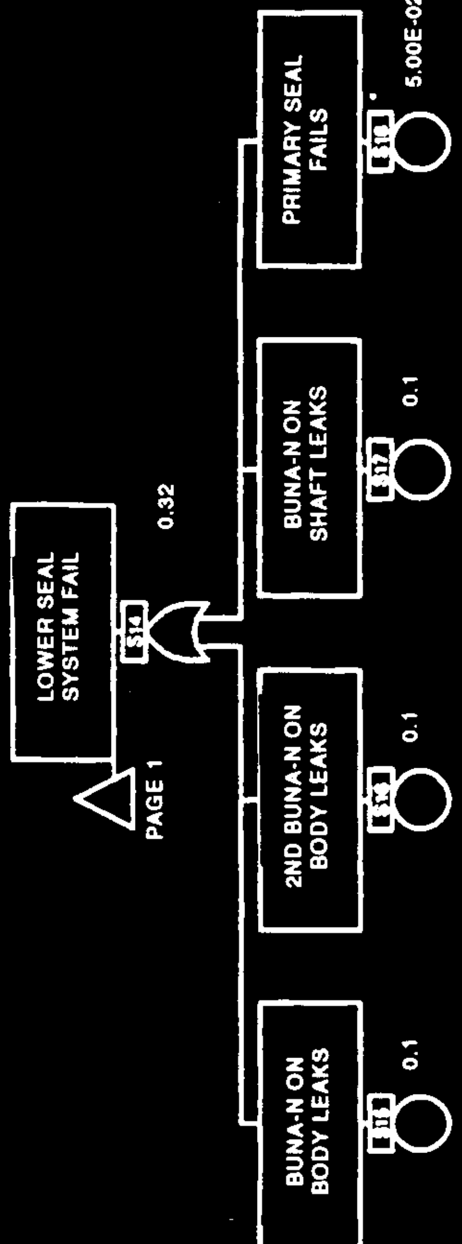

Figure 6. Seal System Fault Tree. (continued)

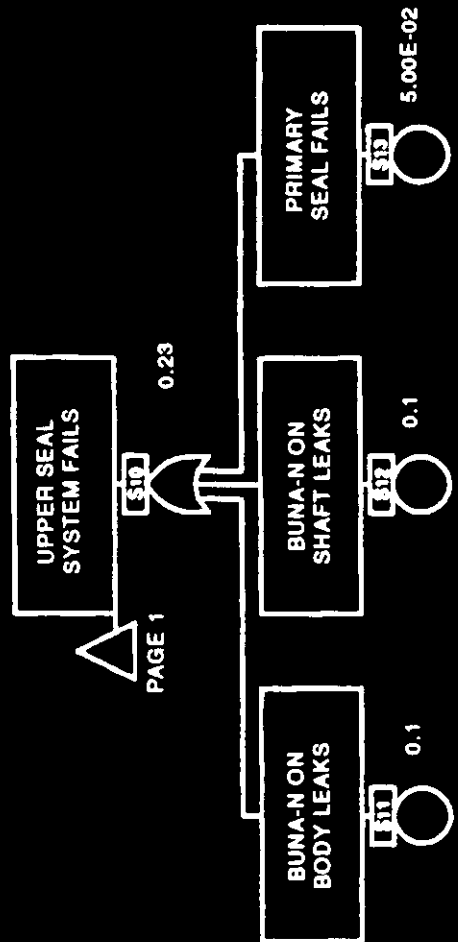

Figure 6. Seal System Fault Tree. (continued)

P_{Oring} by-pass failure:

$$= 1-2e^{-.1} + e^{-(2-.1) \times .1}$$

$$= 0.017 \text{ in a year}$$

The total seal system failure was calculated using the joint probability function as described in Section 3.2.3 which results in an estimated lifetime of 9.7 year. Thus, the estimated life of the seal system is about 9.7 years when the redundancy features qualitatively identified in the FMEA are considered in a reliability model.

The 9.7 years of expected life time for the seal system will be used in Section 7 to estimate the expected lifetime of the test pump.

6.6 Other "O" Rings

Besides the Buna-N "O" rings in the seal system, there are other "O" rings installed in other parts of the pump that are made of stainless steel and ethylene propylene (EP) "O" rings. The S.S. "O" ring can withstand the tank ambient temperature, and 87.8°C (190°F) maximum temperature during pump operation for many years and will not be a weak link in the system.

The EP ring has expected lifetime of 10,000 hours at 132°C (270°F) (Ref.13) which is equivalent to 154 years of actual pump operation.

The seal lifetime during pump shutdown can be estimated assuming the Arrhenius model for thermal aging (Ref.14). Using this model and the knowledge of activation energy for EP, which is about 0.95 eV (Ref.14), the seal lifetime is estimated to be around 1500 years. Therefore it can be concluded that EP seal thermal characteristics will not be a limiting step toward the failure of the pump.

The lifetime of the EP ring due to radiation effect has been estimated analytically (Ref.1) to be negligible.

6.7 Structural Material

The key structural element susceptible to performance degradation is the pump column due to corrosion. (Chemical reaction of the carbon steel with waste). The lifetime has been estimated analytically (Ref. 1) to be about 25 years. The impeller is assumed to be bounded by this estimate.

6.8 Summary Table

The failure rate database has been summarized in Table 3. It is important to note that all the numbers in this table, except the ones annotated by a number 3, are based on a calendar year.

Table 3 DATA FOR MEAN TIME TO FAILURE

Components	Material	Radiation (Yr.)	Thermal (Yr.)	Chemical (Yr.)	Mech. (Yr.)	Elec. (Yr.)
Cable Insulation	Si-Rubber	30	40	N/A(1)	N/A	N/A
Oil Lubricating for motor	GST-ISO-100	150	15(2)	N/A	N/A	N/A
Motor Winding	Class H	150	40	N/A	N/A	N/A
Double Thrust Bearing	S.S	60	N/A	N/A	2.8 (3)	N/A
Radial Bearing	S.S	60	N/A	N/A	» 2.8(3)	N/A
Double Mechanical Seal	SiC	N/A	N/A	N/A	2-10 (3)	N/A
"O" Ring	Buna-N	50	10	N/A	N/A	N/A
"O" Ring	EP	N/A	N/A	N/A	N/A	N/A
Pump Structural	Carbon Steel	N/A	N/A	25	N/A	N/A

1) Not Significant
2) Adjusted for synergistic effects (see section 6.3)
3) Continuous Operation

7.0 SYSTEM FAILURE RESULTS

7.1 Base-Case Results

From Table 3 it is seen that all failure times except two are very long. Those that actually contribute significantly are motor oil at a mean of 15 years and the seal system at a mean of 9.7 years. The seal system failure time was recalculated following accelerated testing of 5 Buna-N seals whose mean adjusted failure rate was slightly in excess of 10 years. A 10 year value was modeled to take into account the synergistic effects.

A statistical evaluation of the Buna-N "O" ring failure times showed a COV of only 0.04 for these measurements. However, this evaluation applies only to the measured lifetimes and does not address how well the true lifetime is modeled by this test. To assure that adequate modeling uncertainty has been included, this COV value was increased by a factor of 5 (to 0.2) to calculate the base case.
The remaining base case numbers include:

1) Oil lifetime at 15 years with a COV of 0.373; this COV was chosen to ensure that the 5th percentile value doesn't exceed 6 years as derived in Section 6.3.

2) Structural lifetime at 25 years with a 0.2 COV; these values are considered very conservative for the structural case.

3) A cable insulation and motor winding lifetime of 30 years each with associated COVs of 0.25; these are very conservative modeling values chosen to ensure the effect is not underestimated.

With these values comprising the base case, a mixer pump composite failure interval was calculated assuming normal distributions for the component failures. Distributions are shown in Figure 7. The COV was then doubled and recalculated. Double COV results are shown in Figure 8. Analogous results were calculated assuming the distributions are lognormal and the associated distributions for the two cases are shown in Figures 9 and 10. Results for all four cases (in terms of means, and various percentiles in the failure time) are displayed in Table 4.

Table 4 CALCULATED PUMP FAILURE TIMES (YRS)

Case	Mean	5th Pct	50th Pct	95th Pct
Normal, Base Case	9.1	5.2	9.2	12.4
Normal, Double COV*	7.7	1.2	7.9	14.2
Lognormal, Base Case	9.1	6.2	9.1	12.9
Lognormal, Double ef*	6.9	2.2	5.8	11

*COV is coefficient of variation for normal distributions (ratio of standard deviation to mean); ef is error factor for lognormal distributions (ratio of 95th to 50th percentile).

If the motor oil could be changed periodically, the mixer pump mean lifetime for the base case would increase to about 9.6 years from 9.1 years.

7.2 Sensitivity

The two parameters felt to have the most significant combination of impact on pump lifetime are the β factor for common mode effects for the seal system and the effective lifetime of the pump motor oil. Therefore, a sensitivity analysis on the expected variation of pump lifetime as a function of variation of these two parameters was performed. The base case for these two parameters was determined to be 15 years for the oil life and 0.28 for β. A 50 percent variation above and below the base case is considered to bracket the range of potential variation for these parameters. Results are shown in Table 5.

Table 5 INFLUENCE OF COMMON-MODE β FACTOR AND OIL LIFETIME ON MIXER PUMP LIFETIME

Oil Life(yrs)	7.5	15	22.5
β Values			
0.14	7.4	11.1	11.9
0.28	7	9.1	9.4
0.42	6.4	7.5	7.7

The β factor is that applied to the composite Buna-N/SiC dynamic seal system which prevents leaking past the pump shaft as described in Section 6.5.2.

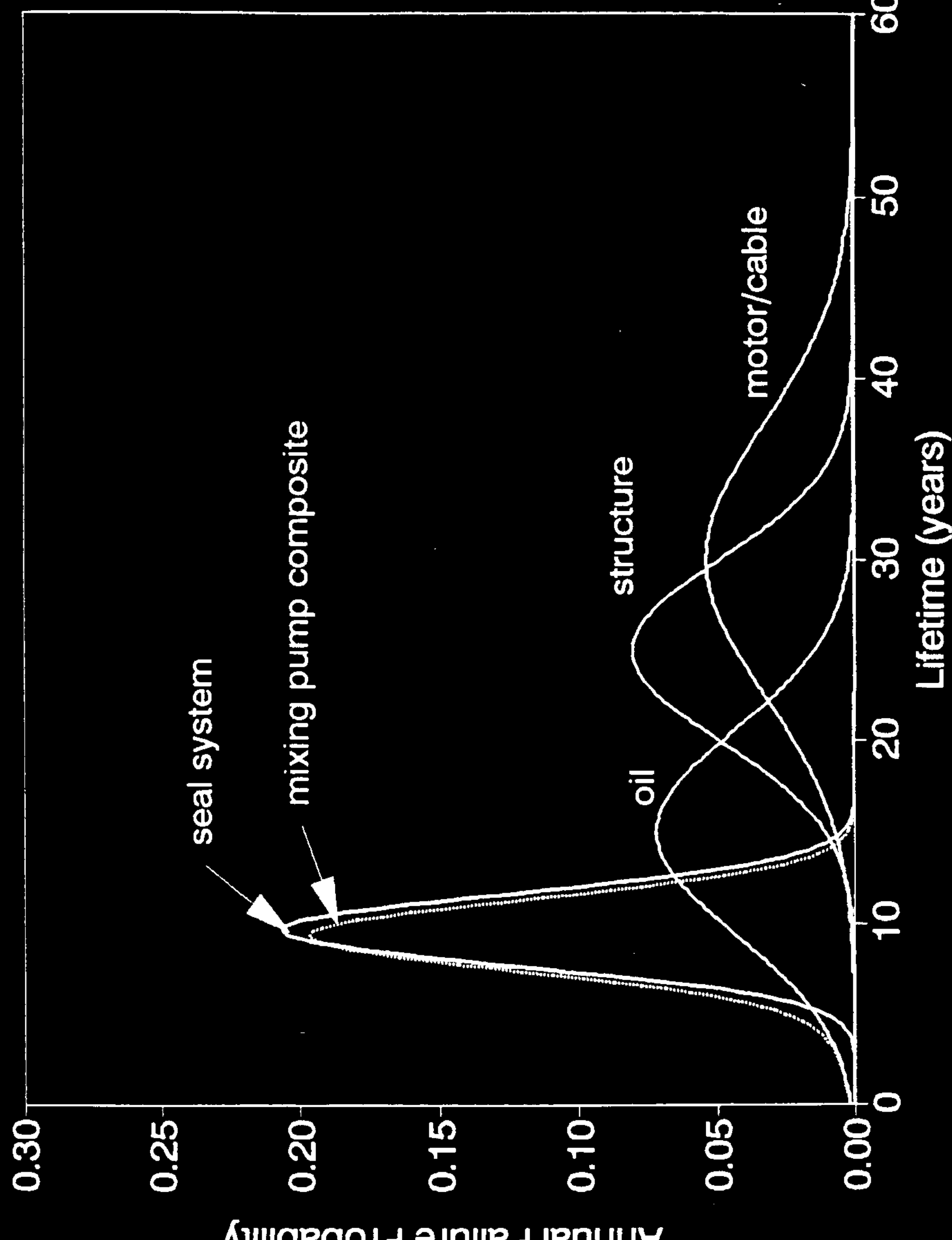

Figure 7. Base Case Normal Failure Distribution.

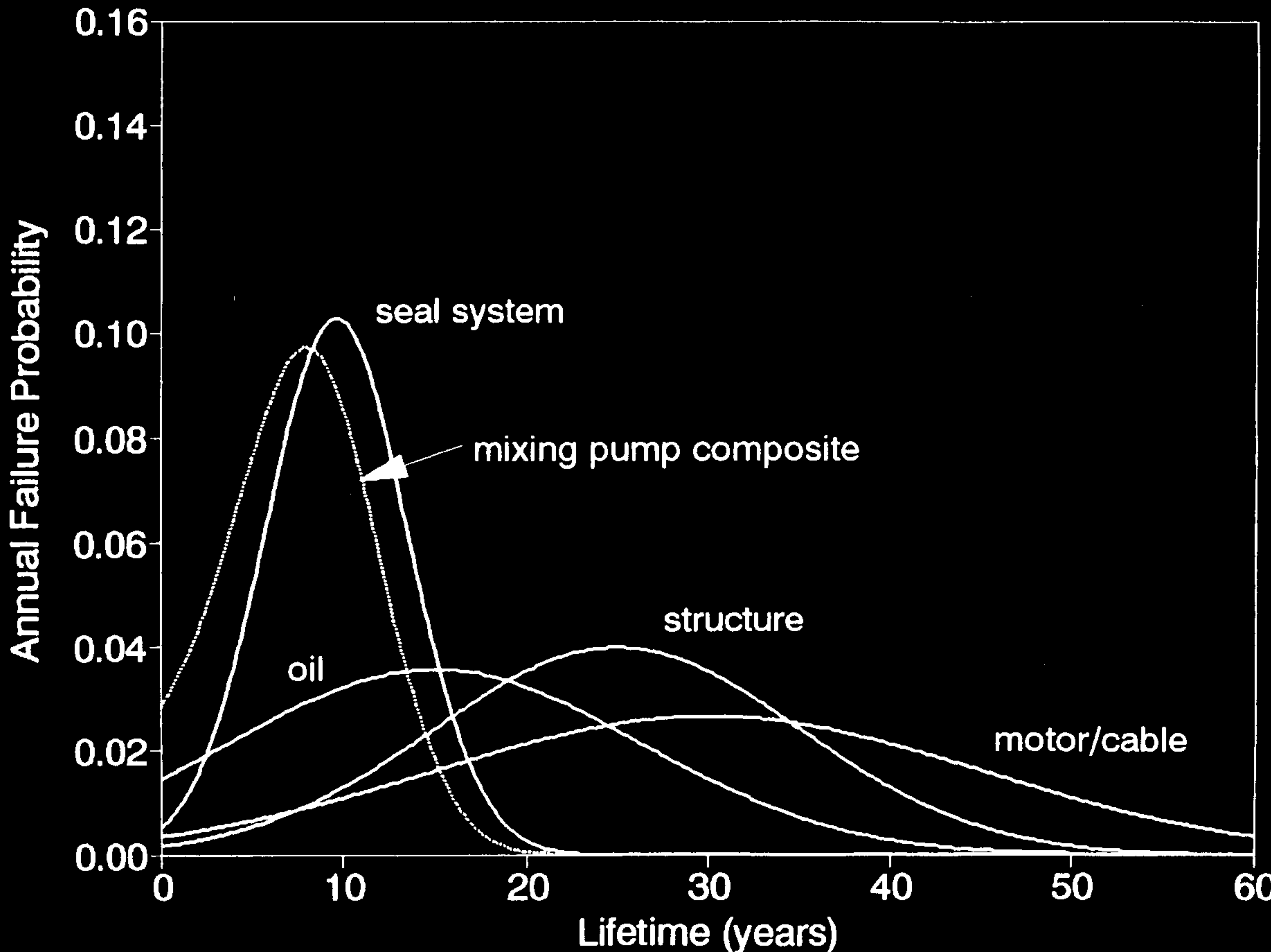

Figure 8. Base Case Normal Failure Distribution, COV Doubled.

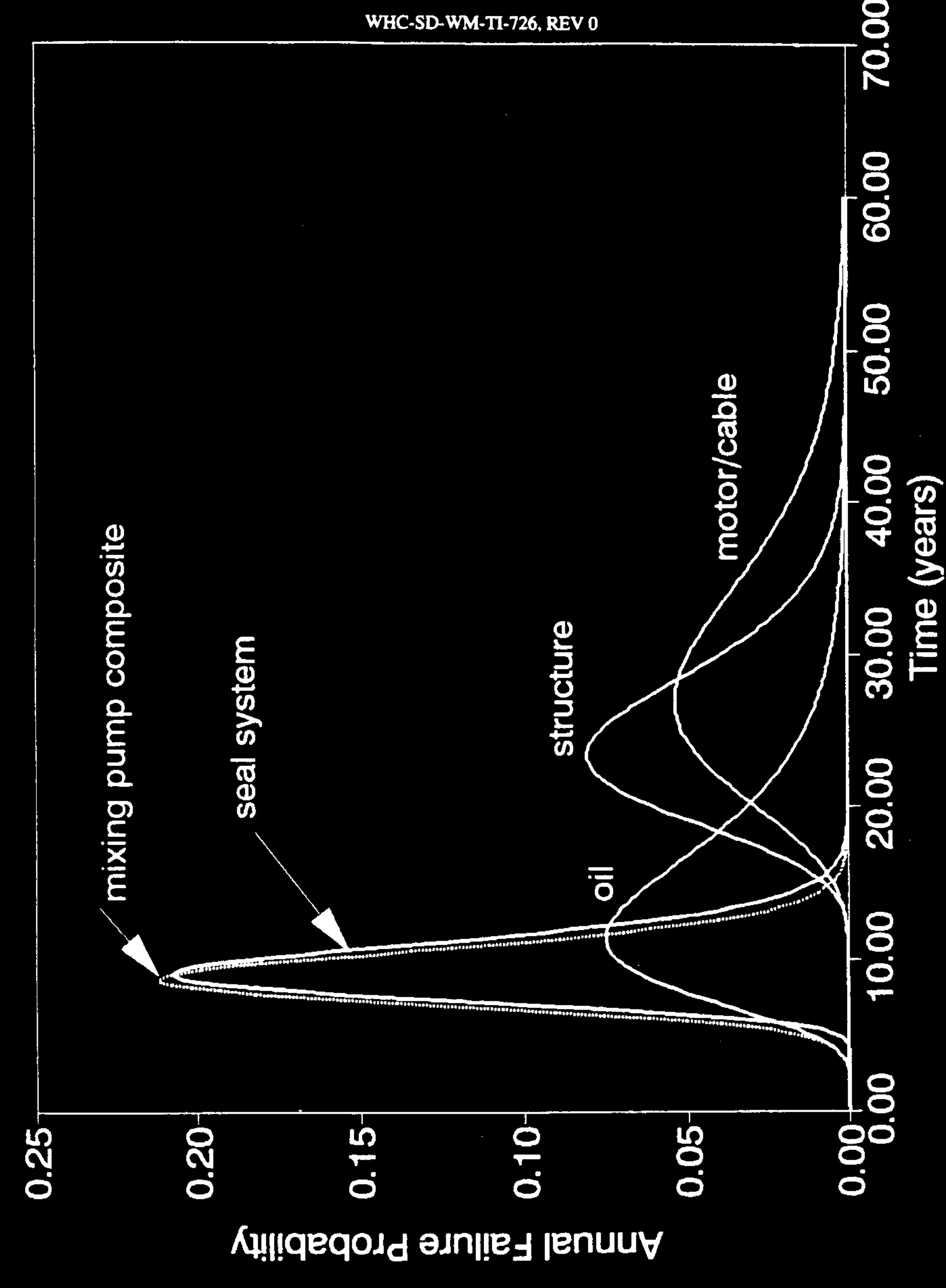

Figure 9. Base Case Lognormal Distribution.

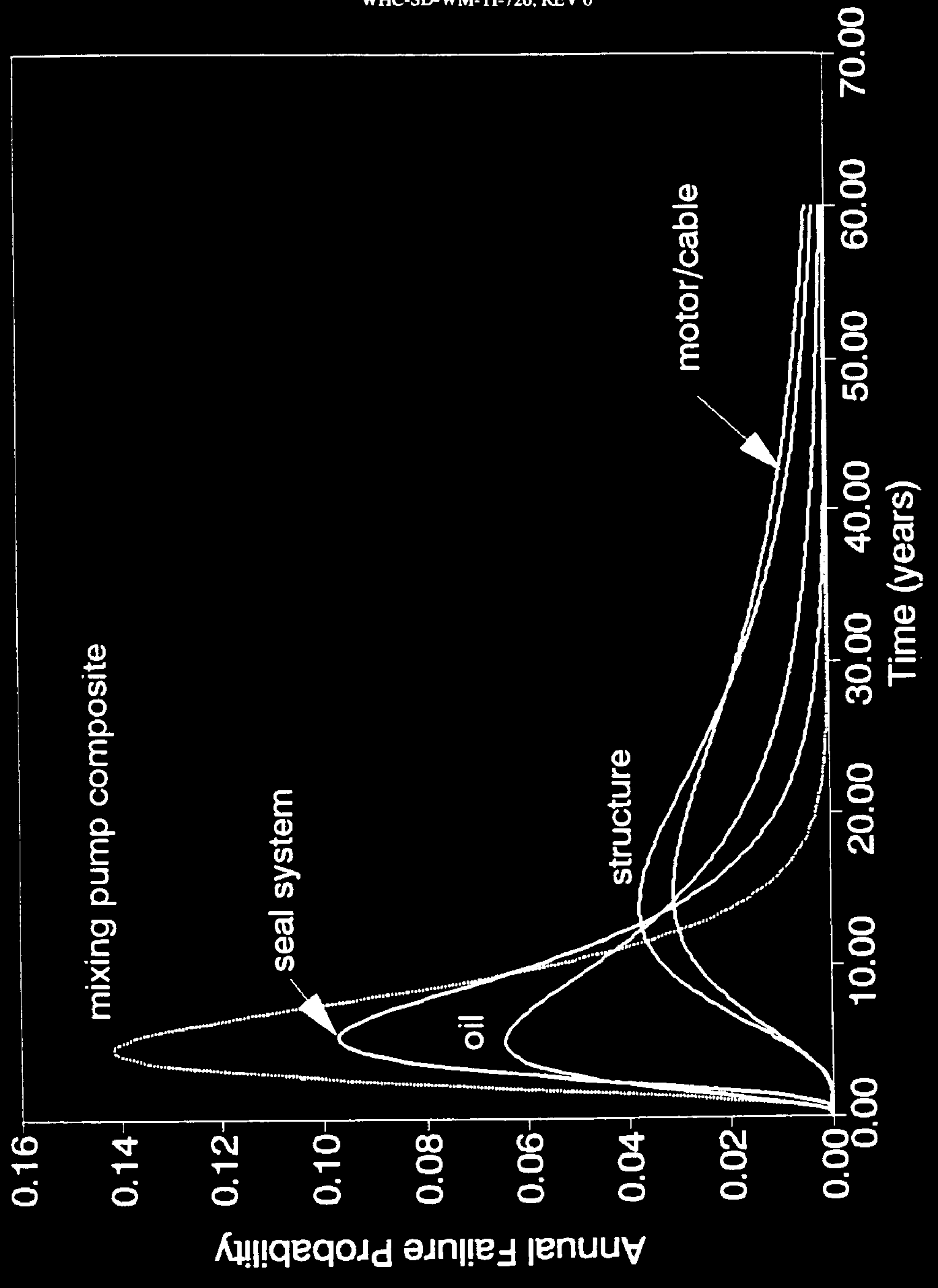

Figure 10. Base Case Lognormal Distribution, Error Factor Doubled.

It is obvious from Table 5 that the worst combination of β and oil life would predict a lower-bound pump lifetime at 6.4 years.

8.0 CONCLUSIONS/RECOMMENDATIONS

Seal and motor oil failures could be the most likely cause of pump failure. Depending on the modeling assumptions, the expected lifetime could range from about 6 to around 12 years.

Because the data and information used to accurately evaluate β are not available, a sensitivity study was carried out. According to the sensitivity study results, changes in β change the mixer pump lifetime estimate from 7.5 to 11.1 years.

Because data on the synergistic effects of oil are not available, a sensitivity study was performed. This shows that the effect of changes in oil lifetime change the results from 7 to 9.5 years.

Seal lifetime and oil lifetime definitely dominate the failure frequency. The lifetimes for the other modes are such that they contribute insignificantly to effective pump lifetime. Both these dominant modes, however, are critically dependent on temperature and the analysis is based on the assumption that heating is limited to that which occurs in 25 minutes of continuous pump operation (i.e., 87.8°C). This prediction, then, is dependent on controlling the operation such that this temperature level is maintained.

It is also recommended that monitoring of the motor condition be continued with megger readings as well as noting maximum temperature rise during the transients. If these begin to drift in the wrong direction, changes in the power controller wave form should be considered to reduce heating and load loss in the motor.

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241-SY-101 Mixer Pump Lifetime Expectancy

Name	MSIN	ATTACH-MENT	
JW Lentsch	S7-15	X	
WW Jenkins	S2-24	X	
JR White	H5-09	X	
LS Krogsrud	R3-08	X	
K Chen	S7-54	X	
CA Groendyke	S7-54	X	
RE Merriman	E6-27	X	
CP Shaw	H5-09	X	
KF Armatis	H5-09	X	
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