Enclosure 2

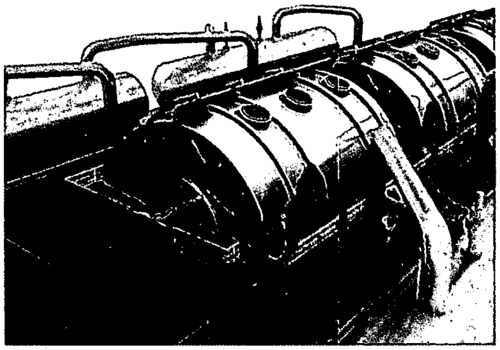
MFN 09-484 Supplement 1

GE-ST "ESBWR Steam Turbine – Low Pressure Rotor Missile Generation Probability Analysis" ST-56834/N-P, Revision 2

Public Version

NON-PROPRIETARY VERSION

Economically Simplified Boiling Water Reactor
Steam Turbine - Low Pressure Rotor
Missile Generation Probability Analysis



ST-56834/N-P, Revision 2
Dated - September 14, 2010

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TABLE OF CONTENTS

1. SUMMARY AND OVERVIEW 6

1.1 RESULTS 6

1.2 METHODOLOGY 7

1.2.1 Probability of Turbine Overspeed 7

1.2.2 Rotor Burst Probability 7

1.2.3 Probability of Casing Penetration 8

1.2.4 Key Parameters of Numerical Evaluation 8

2. TURBINE-GENERATOR DESCRIPTION 9

2.1 GENERAL DESCRIPTION 9

2.1.1 Main Stop and Control Valves 9

2.1.2 Combined Intermediate Stop and Intercept Valves 10

2.1.3 Extraction Non-return Valves 10

2.2 TURBINE OVERSPEED PROTECTION SYSTEM 10

2.3 TURBINE PROTECTION SYSTEM 12

2.4 TESTING 13

3. TURBINE INTEGRITY 15

3.1 MATERIALS SELECTION 15

3.1.1 Fracture Toughness 15

3.1.2 High Temperature Properties 17

3.1.3 Pre-Service Inspection and Testing 17

4. TURBINE GENERATION MISSILE PROBABILITY 18

4.1 FRACTURE MECHANICS 18

4.2 CYCLIC PROPAGATION OF UNDETECTED FORGING DEFECTS 19

4.2.1 Cyclic Crack Growth 19

4.2.2 Undetected Flaw Size 20

4.2.3 Cyclic Profile 20

4.3 STRESS CORROSION CRACKING 21

4.4 DUCTILE ROTOR BURST 23

5. OVERSPEED PROBABILITY 24

5.1 PROBABILITY CALCULATION 24

5.1.1 Steam Valve Arrangement 25

5.1.2 Steam Valve Model 25

5.1.2.1 Valve failure rates 27

5.1.3 Extraction systems 29

5.1.4 Hydraulic Model 29

5.1.5 Emergency Trip System 31

5.1.6 Turbine Generator Control System (MARK™ VIe) 31

6. CASING PENETRATION 34

6.1 COMPONENTS 34

6.2 CASING PENETRATION CALCULATIONS 35

7. OVERALL PROBABILITY DETERMINATION 37

7.1 PROBABILITY OF BRITTLE FRACTURE ($K_I > K_{IC}$) 37

7.2 PROBABILITY OF DUCTILE TENSILE FAILURE 42

7.3 NORMAL OPERATION 42

7.4 ABNORMAL OPERATION 44

8. BORELESS (SOLID) ROTOR RESULTS SUMMARY 48

9. BORED ROTOR RESULTS SUMMARY 50

10. IN-SERVICE INSPECTIONS 52

10.1 IN-SERVICE MAINTENANCE AND INSPECTION OF TURBINE ROTORS52

10.1.1 Rotor Dovetail Inspections 52

10.1.2 In-service Inspection of Turbine Valves 52

REFERENCES 55

1. SUMMARY AND OVERVIEW

The evaluation of turbine missile effects is commonly characterized by the following equation:

$$P_4 = P_1 \times P_2 \times P_3$$

Where P_4 = annual probability of unacceptable damage resulting from a turbine missile

P_1 = annual probability of turbine failure resulting in the ejection of turbine rotor (or internal structure) fragments through the turbine casing.

P_2 = the probability that a turbine missile strikes a critical plant target, given generation.

P_3 = the probability that the critical target is unacceptably damaged, given a missile strike.

The NRC licensing guidelines (Regulatory Guide 1.115 and NUREG-1048) use this formulation to describe hypothetical turbine missiles and specifies that the probability of unacceptable damage from turbine missiles should be less than or equal to 1 in 10 million per year (i.e., P_4 should be $\leq 1\times10^{-7}$ per year per plant). Further definition in the guidelines, due to uncertainties associated with the calculation of P_2 and P_3, state the product of strike and damage probabilities to be 1×10^{-3} per year for a favorable oriented turbine, and 1×10^{-2} per year for an un-favorable oriented turbine. The total turbine missile generation probability (P_1) requirements should be less than 1×10^{-4} per year for a favorable oriented turbine, and 1×10^{-5} per year for an un-favorable oriented turbine.

This report describes the methodology and assumptions used to determine the annual missile generation probability (P_1) for the General Electric ESBWR N3R-6F52 steam turbine. The methodology of this report is consistent with earlier work performed by General Electric (Reference 1), which focused on shrunk-on wheels of the low-pressure turbines as the critical source of turbine missiles. The methodology has been updated to include integral (monoblock) rotors and includes a fracture mechanics burst model that considers the uncertainty in rotor crack behavior, material properties, inspection results, and normal/abnormal control system behavior. This report assesses the missile probability for the low-pressure (LP) turbine only; the high-pressure turbine and generator designs are such that generation of a missile external to the structure is significantly less likely.

1.1 RESULTS

The missile calculations summarized in Sections 8 and 9 show the annual missile probability limit (P_1) of 1×10^{-5} can be met for the entire target 60-year design life of ESBWR, subject to testing, managed inspections and implementation of any specified corrective actions identified as a result of the tests/inspections. Early in life, the missile probability is dominated by overspeed due to a valve failure event. Later in life the limiting missile generating mode becomes Stress Corrosion Cracking (SCC). As described in Section 8 and 9, SCC characteristics (initiation and growth) assumed in this report are based on historical field experience of older LP rotors featuring shrunk-on wheels that exhibited bore keyway SCC. The use of this data is considered to be conservative when applied to the ESBWR LP monoblock rotors as discussed later in this report.

General Electric has a long established practice of recommending periodic in-service inspection of turbine components as part of maintenance programs. The basis of the missile probability analysis assumes that all General Electric maintenance recommendations and operating practices are satisfied and defines that the ESBWR turbine components should be inspected at frequency not to exceed 12 years. The results of the in-service inspections will be used to update unit specific missile probability estimates and to adjust the re-inspection frequency to a shorter interval, if considered necessary based on observed results.

1.2 METHODOLOGY

The methodology of missile generation probability analysis deals with one element of the overall missile issue, which is the probability (P_1) of generating a turbine missile from the LP turbine external to the LP inner casing and LP hood structure. The methodology for determination of missile generation probability contains three major components:

- The probability of the turbine attaining speeds higher than those occurring during normal operation (overspeed),
- The estimation of rotor burst probability as a function of speed, and
- The probability of a rotor fragment penetrating the turbine casing and thus generating an external missile.

1.2.1 Probability of Turbine Overspeed

The probability of a rotor burst and the probability that a fragment will penetrate the turbine casing are both dependent on the speed at which rotor burst is assumed to occur. Under normal operating conditions, the turbine speed is close to the rated speed (1800 rpm). When an abnormal event occurs, such as a full load rejection and failure of elements of the control system, turbine speeds significantly higher than the rated speed may occur. A major component of the analysis is to estimate the probability of attaining various overspeed levels.

1.2.2 Rotor Burst Probability

One rotor failure mode considered is brittle burst, specifically as the result of a crack located in the radial-axial plane growing to a critical size. Brittle burst scenarios addressed are: 1) an undetected internal forging flaw that grows cyclically to critical size, and 2) a time dependent SCC that initiates on the outer body surface and grows to a critical size. A second failure mode due to tensile failure is also included in the methodology. This ductile failure mode contributes to the rotor burst probability particularly during abnormally high overspeed occurrences.

1.2.3 Probability of Casing Penetration

The third major component of the missile probability analysis methodology deals with the probability of a rotor burst fragment penetrating the turbine casing. Calculations employ the energy analysis method already developed by General Electric, described in Reference 1, Section 5.

This method considers the kinetic energy of the assumed fragment at the instant of burst as well as the energy absorbing capability of the stationary components of the low-pressure turbine.

1.2.4 Key Parameters of Numerical Evaluation

Undetected Flaw Size - General Electric nuclear LP rotor forgings are manufactured by the highest quality suppliers and are subjected to both volumetric and finish surface non-destructive inspection using methods and techniques that are well established in the nuclear power industry. Assumptions about possible undetected flaws are consistent with industry practice and the possible undetected flaws are conservatively assumed to be cracks.

Cyclic Crack Growth - Cyclic crack growth behavior was determined from General Electric NiCrMoV specimen tests. For this analysis, a finite number of operational cycles per year is assumed.

SCC Crack Initiation and Growth – Historical experience has shown that SCC can occur in NiCrMoV at locations featuring tensile stress and in contact with wet steam. This missile generation probability analysis considers the possibility that an SCC crack initiates on the outer rotor surface, specifically at bucket attachment dovetail stress concentrations, and grows radially inward to a critical size. Statistical assumptions about both SCC initiation time and SCC growth rate are based on General Electric measurements of stress corrosion cracks found in keyways of shrunk-on wheels (Reference 1, Appendices D and E).

Fracture Toughness –ASTM toughness tests of NiCrMoV material is the basis for the critical stress intensity versus excess temperature correlation used in the analysis. These results are from tests conducted by General Electric and others.

Rotor Operating Conditions - Rotor stresses are derived from finite element analysis. Both mechanical (rotational) and thermal stresses are considered. Temperatures are derived from the overall unit heat balance.

Turbine Control System Reliability - The failure rates of various turbine controls system components are estimated from actual field experience, laboratory studies, and commercial data.

Casing Penetration Behavior - The missile penetration probabilities used in the analysis are based on the published energy method developed by General Electric. This method has been validated by full-scale tests, beginning in 1969 and the method is discussed in greater detail in Reference 1, Section 5.

2. TURBINE-GENERATOR DESCRIPTION

2.1 GENERAL DESCRIPTION

The turbine generator (TG) consists of an 1800-rpm turbine, external Moisture Separator Reheaters, generator, static excitation system, controls, and associated subsystems. The N3R-6F52 turbine for the ESBWR standard plant consists of a double-flow, high-pressure turbine, and three double-flow low-pressure turbines in tandem, each with 52-inch last stage buckets.

The high-pressure turbine has extraction points for reheating steam and high-pressure feedwater heating. Moisture separation and reheating of the high-pressure turbine exhaust steam is performed by external Moisture Separator Reheaters (MSRs). Two MSRs are located on each side of the TG centerline. The steam then passes through the three low-pressure turbines, each with extraction points for the low pressure stages of feedwater heating, and exhausts into the main condensers. In addition to the moisture separators in the external MSRs, the turbine steam path has provisions for removing some additional moisture and routing it to extraction lines.

The generator is a direct driven, three-phase, 60 Hz, 1800 rpm synchronous generator with a water-cooled armature winding and hydrogen-cooled rotor.

The TG uses a General Electric MARK™ VIe[1] Turbine Generator Control System (TGCS), which is a digital monitoring and control system. The TGCS, in coordination with the Steam Bypass and Pressure Control (SB&PC) system, controls the turbine speed, load, and flow for startup and normal operations. The control system operates the turbine stop valves, control valves, and combined intermediate stop and intercept valves. TG supervisory instrumentation is provided for operational analysis and malfunction diagnosis.

TG accessories include the bearing lubrication oil system, turbine hydraulic system, turning gear, hydrogen gas control system, seal oil system, stator cooling water system, exhaust hood spray system, turbine gland seal system, excitation system, and turbine supervisory instrument system.

2.1.1 Main Stop and Control Valves

Four main stop and four control valves admit steam to the high-pressure turbine. The primary function of the main stop valves is to quickly shut off the steam flow to the turbine under trip conditions. The primary function of the control valves is to control steam flow to the turbine in response to the TGCS.

The main stop valves are hydraulically operated in an open-closed mode either by the turbine overspeed protection system in response to a turbine trip signal, or by a test solenoid valve and a fast acting solenoid valve for periodic testing. The disks are unbalanced and cannot open against full differential pressure. A bypass is provided to pressurize the below seat areas of the four valves and supply steam for turbine casing and steam chest warming. Springs in the valves are designed to improve the closing

[1] The MARK turbine controls system is a trademark of General Electric.

time response of the main stop valve under the abnormal conditions. An equalizing header is provided between the stop valves, upstream of the control valves. Each main stop valve is designed to accept a steam strainer to limit foreign material from entering the control valves and turbine.

The control valves are designed to provide steam shutoff adequate for turbine speed control. The valves are of sufficient size, relative to their cracking pressure, to require a partial balancing. Each control valve is hydraulically operated by high-pressure fire-resistant fluid supplied through a servo valve.

2.1.2 Combined Intermediate Stop and Intercept Valves

Hydraulically operated combined intermediate stop and intercept valves are provided in each hot reheat line just upstream of the inlet to each of the LP turbine inlets (six total). The combined intermediate and intercept valves control steam flow to each of the LP turbines in response to the TGCS. Each combined valve includes two independently operated valve discs in series for open-closed operation.

2.1.3 Extraction Non-return Valves

Upon loss of load, the steam contained downstream of the turbine extractions can flow back into the turbine, across the remaining turbine stages, and into the condenser. Associated condensate can flash to steam under this condition and contribute to the backflow of steam or can be entrained with the steam flow and damage the turbines. Non-return valves are employed in selected extraction lines to minimize potential for these conditions to contribute to the turbine overspeed.

2.2 TURBINE OVERSPEED PROTECTION SYSTEM

The normal speed control system comprises the first line of defense against turbine overspeed. This system includes the main control valves, intercept valves, and fast-acting valve-closing functions within the TGCS. The normal speed control unit utilizes three speed signals. Loss of any two of these speed signals initiates a turbine trip via the Emergency Trip System (ETS). Under normal operation, an increase in speed above the speed setpoint initiates the closing of the control and intercept valves in proportion to the speed increase. Rapid turbine acceleration resulting from a sudden loss of load at higher power levels initiates the fast-acting solenoids via the normal speed control system. The fast-acting solenoids rapidly close the main control and intercept valves irrespective of the current turbine speed.

The normal speed control system is designed to limit peak overspeed resulting from a loss of full load, to at least 1% below the overspeed trip setpoint. Typically, this peak speed is in a range of 106-109% of rated speed, and the overspeed trip setpoint is approximately 110% of rated speed. All turbine steam stop, control and intercept valves are fully testable during normal operation. The fast closing feature, provided by action of the fast-acting solenoids, is testable during normal operation.

Normal speed control is supplemented by the power load unbalance function. The power load unbalance function can protect the turbine from an overspeed trip condition in the event of full load rejection. The power load unbalance function looks for an unbalance between mechanical power and electrical load. Under specific load rejection conditions, the power load unbalance will initiate main control valve and intercept valve fast closing functions to prevent rapid acceleration and a subsequent turbine trip.

If the normal speed control and power load unbalance function should fail, the overspeed trip devices close the main stop and control valves, and the combined intercept and intermediate stop valves. This turbine overspeed protection system comprises the second line of defense against turbine overspeed. It is both redundant and diverse.

Redundancy comes from the use of multiple speed probes, multiple controllers, and multiple trip solenoid valves. The turbine hydraulic trip solenoid valve hydraulic circuits are arranged in a dual, "two-out-of-three", de-energize to trip configuration. Any power interruption to either set of the two-out-of-three trip solenoid valves in the Emergency Trip Device (ETD) results in a turbine trip.

Diversity is provided by separate sets of physically isolated primary and emergency overspeed protection controllers. The primary overspeed trip and emergency overspeed trip controllers are independent and diverse by providing unique hardware and logic design and implementation. Power to the trip solenoids is interrupted by either the primary overspeed protection controllers or by the emergency overspeed protection controllers. An overspeed trip results if either set of redundant controllers determines an overspeed condition exists. Power interruption to the turbine control cabinet (which also supplies power to the trip solenoids) results in a "fail-safe" turbine trip. The trip solenoid valve and associated controller are fully testable during normal operation.

For an actual overspeed trip condition, the primary overspeed controllers exchange and vote their individual speed inputs so each controller executes its protective algorithm on the consensus speed value. Each primary overspeed controller de-energizes trip solenoid valves in a two-out-of-three logic arrangement. The two-out-of-three logic precludes a single failure in any of the three controllers from blocking trip initiation.

A different implementation and operation takes place in the three completely separate and individual emergency overspeed trip controllers. Each of the three emergency controllers has a dedicated power supply and operates completely separate from each of the other emergency overspeed trip controllers. The three emergency controllers operate independently from the primary overspeed trip controllers. In the event of an overspeed condition, the emergency controllers individually detect and determine speed, and de-energize trip solenoid valves in a two-out-of-three logic arrangement.

The overspeed protection system is designed to ensure that failure of the normal speed control system does not result in turbine speed exceeding ~120% of rated speed. The components and circuits comprising the turbine overspeed protection system are testable when the turbine is in operation.

The overspeed sensing devices are located in the turbine front bearing standard, and are therefore protected from the effects of missiles or pipe breakage. The hydraulic lines are fail-safe; if one is broken, loss of hydraulic pressure results in a turbine trip. The ETDs are also fail-safe. Each trip solenoid transfers to the trip state on a loss of control power, resulting in a turbine trip. These features provide inherent protection against failure of the overspeed protection system caused by low trajectory missiles or postulated piping failures.

Each turbine extraction line is reviewed for potential energy and contribution to overspeed. The number and type of extraction non-return valves required for each extraction line are specified based on the enthalpy and mass of steam and water in the extraction line and feedwater heater. Higher energy lines are provided with power-assisted open, spring-assisted closed non-return valves, controlled by air relay dump valves, which in turn, are activated by the ETS. The air relay dump valves, actuated on a turbine trip, dump air from the extraction non-return valve actuators to provide rapid closing via actuator spring force. The closing time of the extraction non-return valves is sufficient to minimize extraction steam contribution to the turbine overspeed event.

The following component redundancies are employed to guard against excessive overspeed:

(1) Main stop valves/Control valves,

(2) Intermediate stop valves/Intercept valves,

(3) Normal speed control/Primary overspeed trip/Power Load unbalance/Emergency overspeed trip,

(4) Fast-acting solenoid valves/Emergency trip fluid system (ETS),

(5) Extraction non-return valves (as needed).

The main stop valves and control valves provide full redundancy in that these valves are in series and have independent control signals and operating mechanisms. Closure of all four-stop valves or all four-control valves shuts off all main steam flow to the high-pressure turbine. The intermediate stop and intercept valves are also in series and have independent control signals and operating mechanisms. Closure of either valve or both valves in each of the six sets of intermediate stop and intercept valves effectively shuts off intermediate steam flow to the three low pressure turbines. This arrangement is such that failure of a single valve to close does not result in turbine speed exceeding ~120% of rated speed.

2.3 TURBINE PROTECTION SYSTEM

In addition to the overspeed trip signals discussed, the ETS closes the main stop and control valves and the intermediate stop and intercept valves to shut down the turbine on the following signals.

- Emergency trip in control room,
- Moisture Separator high level,

- High condenser pressure,
- Low lube oil pressure,
- Low pressure turbine exhaust hood high temperature,
- High reactor water level,
- Thrust bearing wear,
- Emergency trip at front standard,
- Loss of stator coolant (if runback fails),
- Low hydraulic fluid pressure,
- Selected generator trips,
- Loss of TGCS electrical power,
- Excessive turbine shaft vibration,
- Loss of two speed signals – either two Normal Speed Control or two Emergency,
- Loss of two or more SB&PC System channels,
- Closure of Main Steam Isolation Valves,
- Differential and/or Rotor Expansion.

When the ETS is activated, it overrides all operating signals and trips (closes) the main stop and control valves, and combined intermediate stop and intercept valves.

2.4 TESTING

The primary and emergency overspeed trip circuits and devices are tested remotely at or above rated speed by means of controls in the Main Control Room (MCR). Operation of the overspeed protection devices under controlled speed conditions is checked at initial turbine startup and after each refueling or major maintenance outage, consistent with General Electric recommendations to perform off-line (actual) overspeed testing every 6 to 24 months. In some cases, operation of the overspeed protection devices can be tested just prior to shutdown. This eliminates the need to test overspeed protection devices during the subsequent startup if no maintenance is performed that affects the overspeed trip circuits and devices.

Main stop, main control, intermediate stop, and intercept valves are exercised at intervals as required by the turbine missile probability analysis (120-days) by closing each valve and observing the remote valve position indicator for fully closed position status. This test also verifies operation of the fast close function of each main stop and main control valve during the last few percent of valve stem travel. Fast closure of the intermediate stop and intercept valves is tested in a similar way if they are required to have a fast close function that is different from the test exercise.

Access to required areas outside of the turbine shielding is provided on the turbine floor under operating conditions.

Provisions are included for testing each of the following devices while the unit is operating:

- Main stop valves and main control valves,
- Low pressure turbine intermediate stop and intercept valves,
- Testable Turbine Extraction non-return valves important to overspeed protection,
- Lubricating oil pumps,
- Hydraulic fluid pumps,
- Emergency Trip Device,
- Power-Load Unbalance circuits,
- Other test/inspections identified by the periodic operational test summary that impact reliability of the overspeed protection system.

3. TURBINE INTEGRITY

3.1 MATERIALS SELECTION

LP turbine rotors forgings are NiCrMoV alloy material in accordance with General Electric specification B50A373B8 or equivalent Specification with more restrictive chemistry requirements (to be issued later)).. The material properties of the rotor forgings are optimized to ensure excellent fracture toughness. Highly refined manufacturing processes are designed to minimize melt related defects and impurities. Undesirable elements, such as sulfur and phosphorus, are controlled to the lowest practical concentrations consistent with good melting practice. Rotors have the lowest Fracture Appearance Transition Temperatures (FATT) and highest Charpy V-notch energies obtainable, on a consistent basis from material at the sizes and strength levels used. The FATT temperature, as obtained from Charpy tests performed in accordance with ASTM A-370, is no higher than -1.1°C (30°F) for a large integral forging. The Charpy V-notch energy at the minimum operating temperature is at least 6.23 kg-m (45 ft-lbf) for a large integral rotor forging.

3.1.1 Fracture Toughness

The fracture toughness of a material is measured by the critical stress intensity factor K_{IC}. The toughness of NiCrMoV material has been found to vary with temperature. At lower temperatures, lower toughness occurs and at high temperatures, an increase in toughness occurs until a maximum is reached. The maximum value is commonly referred to as the upper shelf. The fracture appearance varies from a brittle appearance at low temperature to a ductile appearance at high temperature. FATT is the ductile-to-brittle transition temperature at which the fracture surface appearance is 50% plastically deformed indicating ductile failure and 50% cleaved indicating brittle failure.

Fracture toughness has been correlated to the measured excess temperature (metal temperature minus the FATT). As a result, General Electric does not require fracture toughness testing on production rotors. Deep seated FATT testing, however, will be performed for each production rotor forging during routine material acceptance testing and the fracture toughness at the forging centerline will be derived based on historical correlations, as shown in Figure 3-1. For missile generation probability calculations, a normally distributed FATT featuring a –30°F mean and a 30°F standard deviation is assumed.

The correlation between critical stress intensity K_{IC}, and FATT is shown in Figure 3-2. The fracture toughness is approximated by two independent mathematical representations, one for the low excess temperature region and another representing the upper shelf of fracture toughness in the high excess temperature region. For the low excess temperature region, the best fit through the center of the data is the semi-log expression:

[[

{A}]]

Figure 3-1 Fracture Appearance Transition Temperature (FATT) vs. Radial Distance

[[------------------------------------{A}]] (eq. 3-1)

where:

X =excess temperature

And excess temperature is measured in degrees Fahrenheit

The standard deviation is calculated according to the equation:

[[------------------------------------{A}]] (eq.3-2)

where:

ln= natural logarithm

σ_c = standard deviation

X = excess temperature

For the upper shelf region, a Rolfe –Novak relationship (Reference 2) has been applied to estimate fracture toughness. As shown in Figure 3-2, a log mean value of **[[-------- {A}]]**square root inch and log standard deviation = **[[---- {A}]]** are used.

For missile analysis calculations, the two regions (upper and lower shelf) are treated as statistically independent. The probability of burst is calculated for each region and the two probabilities are combined.

[[

{A}]]

Figure 3-2 NiCrMoV Toughness Curve

3.1.2 High Temperature Properties

The operating temperature range of the low-pressure rotors is below the stress rupture temperature range of the materials used. Therefore, creep-rupture is not considered to be a failure mechanism for these components.

3.1.3 Pre-Service Inspection and Testing

The pre-service inspection procedures and acceptance criteria specific to the rotating parts of the steam turbine are as follows:

- Rotor forgings undergo 100% volumetric (ultrasonic) inspection subject to established inspection methods and acceptance criteria that are equivalent to or more restrictive than those specified for Class 1 components in ASME Code Sections III and V. Subsurface sonic indications are not accepted if found to compromise the integrity of the unit.
- The entire finish machined outer rotor periphery including the bore surface (if present) is subjected to magnetic particle test or liquid penetrant examination. Surface indications are evaluated and removed if found to compromise the integrity of the unit during its service life.
- Each fully bladed turbine rotor assembly is factory spin-tested at 120% of rated speed.

4. TURBINE GENERATION MISSILE PROBABILITY

The determination of the probability of the generation of a missile is based on probability of overspeed (P_A), the probability of rotor burst (P_B), and probability of casing penetration (P_C):

$$P_1 = P_A \times P_B \times P_C \qquad \text{(eq. 4-1)}$$

where:

P_A = Probability of achieving a speed of concern or a particular speed {Brittle Burst could happen at any speed. For cases of rated speed or below $P_A = 1.0$} (Section 5)

P_B = Probability of rotor burst (Section 4)

P_C = Probability of casing penetration (Section 6)

The methodology described in this report is for determination of P_1, the probability of turbine missile generation external to the turbine casing. Earlier work (Reference 1) focused on shrunk-on-wheels as the critical source of missiles. Independent calculations of P_1 for each wheel were combined to obtain an overall value of P_1 for each LP rotor. In this analysis report, the integral ESBWR rotor is also assumed to be made-up of independent disks and an overall rotor missile probability obtained by combining individual disk probabilities. The NRC annual missile probability limit of 1×10^{-5} is assumed to apply to the overall unit.

Rotor burst is assumed to occur when a crack oriented in the radial-axial plane reaches a critical final size. Missiles occur when rotor burst fragments (of sufficient mass and energy) penetrate both the inner casing and outer hood. Three rotor body burst scenarios are considered:

1. Brittle fracture resulting from cyclic fatigue propagation of undetected internal forging defects
2. Brittle fracture resulting from Stress Corrosion Cracking (SCC); specifically external axial entry dovetail surface cracks that initiate and grow radially inward towards the center of the rotor
3. Tensile rupture attributed to gross overspeed

4.1 FRACTURE MECHANICS

From Linear Elastic Fracture Mechanics (LEFM), the crack tip stress intensity factor K_I is defined as follows:

$$K_I = C \times \sigma \times \sqrt{\pi \times a} \qquad \text{(eq. 4-2)}$$

where:

C = crack shape factor

σ = applied tangential stress

a = characteristic crack depth

Brittle Fracture occurs when crack tip stress intensity exceeds the material fracture toughness K_{IC}. Stated mathematically, burst occurs when: $K_I \geq K_{IC}$.

4.2 CYCLIC PROPAGATION OF UNDETECTED FORGING DEFECTS

The analysis focuses on the rotor bore. Bore[2] tangential stress is due to both mechanical (rotation) and thermal loading. Radial thermal gradients encountered during a start up result in tensile thermal stresses having greatest magnitude at the bore surface. The maximum value of mechanical tangential stress (due to rotation) is also found at the bore.

Bored rotor tangential stress, derived using Finite Element Analysis (FEA), is summarized in Table 4-1. Mechanical stresses are evaluated at 1800 rpm (rated speed). Thermal stresses are dependent on the time rate of change in steam conditions leaving the Moisture Separator Reheater (MSR) during a start-up. Values shown represent a worst-case MSR temperature ramp rate developed for the starting & loading of the ESBWR steam turbine that will be used in the development of the controls logic. For probabilistic calculations, values shown in Table 4-1 are assumed to be log normally distributed with a 0.05 log normal standard deviation.

Table 4-1 ESBWR Bore Stress

Stage	1800 RPM Bore Mechanical Stress (ksi)	Maximum Bore Thermal Stress (ksi)
1	[[---	---
2	---	---
3	---	---
4	---	---
5	---	---
6	---	---
7	---	---{A}]]

*Bore stress for a solid (boreless) rotor is 50% of the above values

4.2.1 Cyclic Crack Growth

Cyclic crack growth testing of NiCrMoV by General Electric is represented using the traditional Paris form:

$$da/dN = B \times (\Delta K)^n \qquad \text{(eq. 4-3)}$$

where:

[2] Both bored and solid rotors can be treated with this analysis. Bored rotors incur larger stress and have less uncertainty regarding FATT and crack detection. Solid rotors are discussed in greater detail in Section 9.

ΔK = cyclic range in applied stress intensity

Variability in growth rate is captured by variation in the coefficient B. A log normally distributed coefficient (B) featuring an average value of **[[---- {A}]]**and standard deviation of **[[------ {A}]]** along with an exponent (n) value of **[[------- {A}]]**, best represents the test data.

4.2.2 Undetected Flaw Size

Bore surfaces are subjected to both magnetic particle and ultrasonic inspection. Undetected bore surface crack size is a function of both measurement sensitivity and repeatability. For analysis of the bored rotor, an average undetected bore surface crack is assumed to be semi-circular in shape with an average size "*a*" (i.e., radius) = **[[----------- {A}]]** and crack tip stress intensity shape factor C = 0.73 (Reference 3). For a solid rotor, an average undetected embedded crack is assumed to be semi-circular in shape with an average size "a" = **[[------------{A}]]** and crack intensity shape factor of C = 0.64 (Reference 4). Undetected cracks are assumed to be log normally distributed with a log normal standard deviation of **[[------- {A}]]** for size (*a*). The crack shape factor is also assumed to be log normally distributed with a log normal standard deviation of **[[----- {A}]]**.

4.2.3 Cyclic Profile

Because missile probability is reported on an annual basis, crack growth must be calculated on an annual basis. The annual cyclic loading profile assumed for this analysis is shown in Table 4-2.

During each start/stop cycle, the maximum value of tensile stress (peak value of the cyclic applied stress range) at the bore is assumed equal to the sum of the mechanical and thermal stresses shown in Table 4-1. By comparison, load swings are limited to a 50% drop in power during which speed is unchanged and MSR exit temperature is maintained very close to rated condition. Finite element simulation of the load swing revealed <5ksi change in bore stress. For crack growth calculations, the maximum value of tensile stress (peak value of the cyclic applied stress range) during a load swing is assumed equal to the mechanical stress shown in Table 4-1 plus 5 ksi.

Table 4-2 - ESBWR Cyclic Life Content

Cycle Description	Annual Rate
Starts/Stops	7
Load Swings	285

4.3 STRESS CORROSION CRACKING

SCC requires the combination of a corrosive environment, a susceptible material, and tensile stress. The SCC susceptibility of NiCrMoV shrunk on wheel rotor designs in a wet steam environment is well documented (Reference 1). SCC burst scenarios for the ESBWR rotor consider outer surface tangential stress concentrations as potential SCC crack initiation sites. Burst is assumed to occur when a crack growing radially inward (towards the center of the rotor) reaches a critical size.

As shown in Figure 4-1, ESBWR LP rotors feature both tangential entry dovetails (Stages 1 through 4) and axial entry dovetails (Stages 5 through 7).

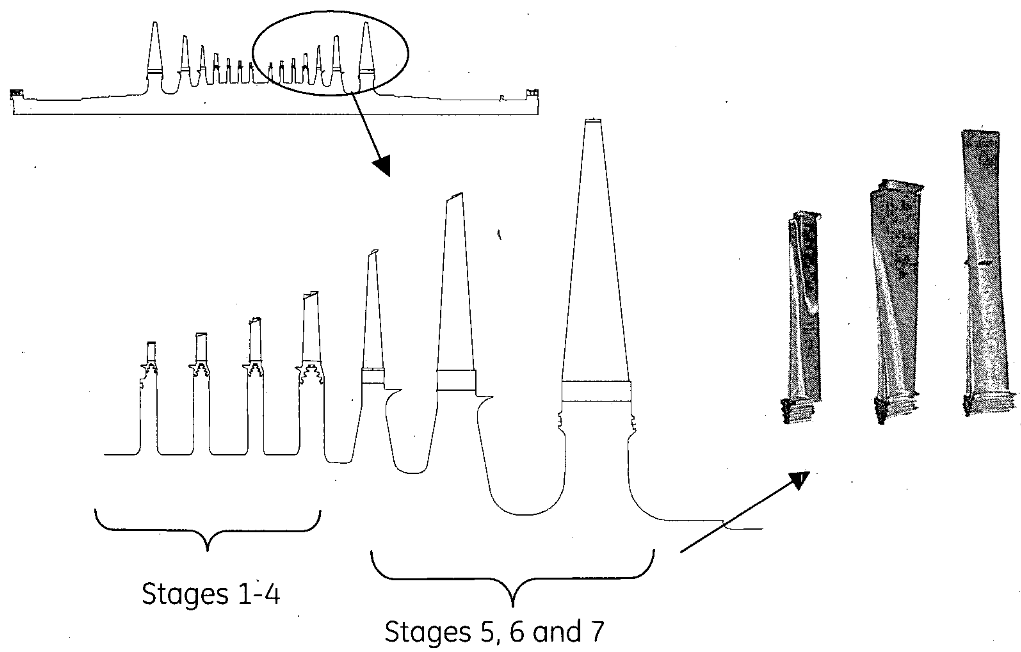

Figure 4-1 LP Rotor Dovetail Configurations

Prior 1960s and 1970s era BWR rotors also featured tangential entry dovetails. SCC cracks found in the wheel dovetail hook fillet radii of prior tangential entry designs have been confined to the radial circumferential and axial circumferential planes. As such, tangential entry dovetail SCC cracks are considered to be a maintenance issue rather than a rotor burst risk.

Cracks forming in the slot bottoms of axial entry dovetails (stages 5, 6, and 7) and oriented in the radial axial plane are considered a rotor burst risk. Shrunk-on-wheel keyway SCC crack statistical behavior including time to initiation and growth rate (summarized below) is assumed. Because ESBWR axial entry dovetail slot bottoms feature dramatically lower tangential stress (vs. shrunk-on-wheel keyways), use of shrunk-on-wheel initiation and growth characteristics is considered conservative.

A summary of the key statistical distributions derived from analysis of cracked nuclear shrunk-on-wheel keyways (Reference 1) and featured in this study include:

Dovetail Crack Initiation Behavior:

$F_I(t) = 1 - e^{-(t/\lambda)^k}$ (Weibull distribution) (eq. 4-4)

where:

$F_I(t)$ = cumulative probability by time t

t = time in years

λ = Weibull scale parameter for initiation defined as:

[[--{A}]]

[[--- {A}]]

exp( ) = base of the natural logarithm raised to power of the value in brackets

T= dovetail temperature during normal operation measured in °F (

Dovetail Crack Growth:

$F_G(G) = 1 - e^{-(G/\lambda_G)^k}$ (Weibull Distribution) (eq. 4-5)

where :

G = growth rate in inches/year

λ_G = Weibull scale parameter defined as:

[[--{A}]]

exp( ) = base of the natural logarithm raised to power of the value in brackets

[[--- {A}]]

A dovetail slot bottom SCC crack is conservatively assumed to extend the full axial length of the dovetail. Stress intensity, or K_I calculations at the dovetail slot bottom assume a crack shape factor, or C, equal to 1.12 (Reference 5). The shape factor is assumed to be log normally distributed with standard deviation of 0.02. Tangential stresses input to the dovetail SCC stress intensity calculation were obtained from finite element analysis. Dovetail tangential stress is assumed to be log normally distributed with a standard deviation of 0.02.

4.4 DUCTILE ROTOR BURST

The probability of ductile burst is determined from the Average Tangential Stress (ATS) of each rotor stage and the Ultimate Tensile Strength (UTS). Failure is assumed to occur when ATS equals or exceeds a fraction of the UTS, as described below.

The ATS values for the ESBWR rotor shown in Table 4-3 are assumed to be log normally distributed with 0.02 log normal standard deviation.

The dependence of UTS on temperature is assumed to be log normally distributed with 0.02 log normal standard deviation and best fit:

$$UTS = UTSRT - (.032 \times (T - 70)) \qquad \text{(eq. 4-6)}$$

where:

$UTSRT$ = room temperature UTS (ksi)

T = metal temperature °F

Tensile failure is assumed to occur when:

$$ATS \geq R \times UTS \qquad \text{(eq. 4-7)}$$

The ratio R is assumed to be log normally distributed with 0.85 mean value and 0.05 log normal standard deviation.

Table 4-3 ESBWR Average Tangential Stress (1800 rpm)		
Stage	Bored Rotor ATS (ksi)	Boreless Rotor ATS (ksi)
1	[[----	----
2	----	----
3	----	----
4	----	----
5	----	----
6	----	----
7	----	---- {A}]]

5. OVERSPEED PROBABILITY

The overspeed probability calculation methodology described in Reference 1 has been used for the current study to compute overspeed probability. The overspeed analysis considers the characteristics of the turbine control system, the steam turbine unit configuration, and test requirements for the steam valves and other overspeed protection devices/systems.

The operating modes of the TG have been separated into two event categories: normal and abnormal. The normal events consist of expected operating conditions at or below rated speed and the actual overspeed trip test. All normal events are assumed to occur on a regular or planned basis. Any event that is a result of a load rejection and failure of the overspeed protection system is classified as an abnormal condition and will be considered within this overspeed analysis. Abnormal events are unplanned and have a small probability of occurrence. Due to the incorporation of fail-safe designs and testing procedures, speed levels greater than the peak overspeed (see Subsection 2.2) can be achieved only through a combination of multiple failure scenarios as well as a load rejection. Overspeed may be divided into two groups,

1.) Controlled overspeed whereby the normal control system, emergency trip or the back-up trip act to close off direct steam paths, and

2.) Runaway overspeed in which a direct steam path from supply to the turbine section exists due to Main Steam Stop and Control Valves and/or Combined Intermediate Stop and Intercept Valve remaining open

The probability of turbine overspeed depends on the probability that the main steam valves and/or intermediate steam valves fail to close when required. There are two primary factors that affect this probability. The first is the basic design of the overspeed protection system, including characteristics of the steam valves and associated electronic and hydraulic systems. The second factor is the knowledge of whether the system's ability to perform has been compromised. The probability calculation assumes that the turbine operator follows General Electric recommendations; especially those concerned with steam valve and trip system test intervals and hydraulic fluid sampling and maintenance requirements. The analysis assumes that appropriate action is taken when the results of these tests so indicate.

5.1 PROBABILITY CALCULATION

The overspeed analysis presented herein for ESBWR considers a steam turbine system design consistent with the Reference 1 typical General Electric 3-hood steam turbine design comprised of an Electro Hydraulic Control System (EHC) with stainless steel trip valves, and titanium cooler. This design is closest to the ESBWR configuration and the failure rates used in the analysis are deemed conservative when compared to the current design upgrades employed on ESBWR (i.e. MARK™ VIe control system, modified trip system, new hydraulic fluid conditioning equipment etc.). The resulting assessment of conservatism will be explained in detail for the valves, hydraulics and controls arrangements discussed in the following sections.

5.1.1 Steam Valve Arrangement

Turbine control systems manufactured by General Electric provide two independent valve groups for defense against overspeed in each admission line to the turbine as shown below.

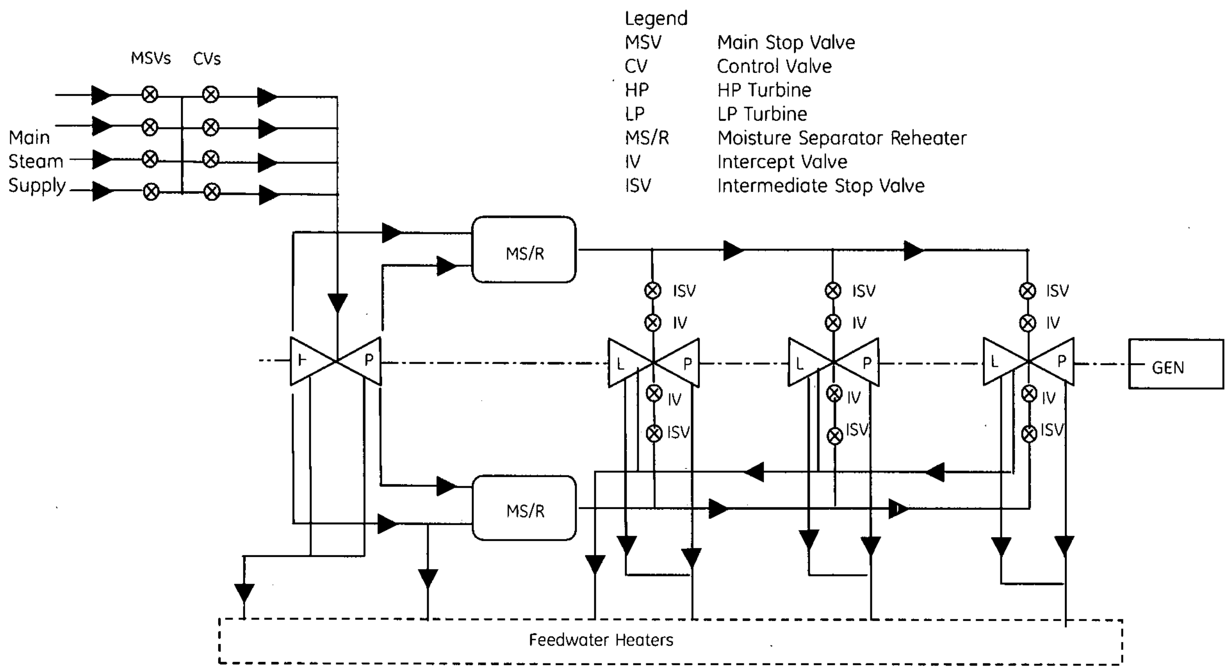

Figure 3 Typical Arrangement

The normal overspeed control system closes the main control valves and intercept valves in proportion to the increase in speed above the overspeed set point, and the ETS closes both valve groups via fast acting solenoids upon a rapid turbine acceleration, irrespective of the current turbine speed. Steam from the nuclear steam supply is admitted through the main stop valves then, enters a manifold, continuing through the main control valves to the high-pressure turbine. The manifold ahead of the control valves permits in-service testing of the stop valves with little effect on load. The control valves can be individually tested in service by reducing load down to approximately 85% rated load.

5.1.2 Steam Valve Model

The steam valve model calculates the probability of the steam valves failing to close when signaled closed. The model considers two steam paths, the main steam path and the reheat steam path. The main steam path represents the steam line from the nuclear steam supply, through the main stop and control valves, to the high-pressure turbine section. The reheat steam path represents the steam line from the MSR, through the combined valves, to the low-pressure turbines. The probability of overspeed and the speed level reached is a function of the combinations of failure modes for stop, control, and combined intermediate stop and intercept valves.

The valves controlling the steam flow can be in one of four possible valve responses: fast closed, closed by servos, tripped closed, or open. The combination of valve responses results in ten events, as outlined below:

Table 5-1 Steam Path Events for Hydraulic Steam Valve Model		
Event Number	**Main Steam Path Valving**	**Reheat Steam Path Valving**
1	Open	Open or Closed by servos
2	Closed by servos	Open
3	Closed by servos	Closed by servos
4	Closed by servos	Tripped or fast closed
5	Tripped	Tripped
6	Tripped	Fast closed
7	Tripped or fast closed	Open
8	Tripped or fast closed	Closed by servos
9	Fast closed	Tripped
10	Fast closed	Fast closed

Probability equations were derived for each event: only the significant events are shown below, the symbol definitions have also been given for clarity:

Table 5-2 Symbol Definition EHC Hydraulic and Steam Valve Models	
Symbol	**Definition**
SV	Probability of stuck stop valve (steam side)
CV	Probability of stuck control valve (steam side)
IV	Probability of stuck intercept valve (steam side)
RV	Probability of stuck intermediate stop valve (steam side)
MV	Probability of stuck mechanical trip valve
EV	Probability of stuck electrical trip valve
FC	Probability of stuck fast acting valve (on control valve)
FI	Probability of stuck fast acting valve (on intercept valve)
SC	Probability of stuck servo valve (on control valve)
SI	Probability of stuck servo valve (on intercept valve)
NC	Total number of control valves
NS	Total number of stop valves
NP	Total number of pairs of combined valves
nc	Number of control valves failed to close
np	Number of pairs of combined valves failed to close
PNI	Probability of no indication of a problem (Hydraulic model only)

Table 5-3 Probability Equations Steam Valve Model	
Event Number	Probability of Overspeed
[[---	-------------------
---	-------------------------
---	-----------------------
---	-----------------------
---	------------------------
---	-----------------------
	----------------------- {A}]]

The steam side model is concerned only with incidents where the steam valves fail to close (e.g., events 1, 5, 6, 7, 9 and 10). The hydraulic and electronic systems are assumed to function properly. The steam valves are assumed to have a constant failure rate and are returned to a "like-new" condition after testing.

Fundamentally, the ESBWR valves are equivalent to those found on the current fleet. The valve bodies and internals are all of the same geometry and design. The only visible difference is the use of direct actuators on the main steam control and the intercept valves. However, these actuators have the same design features and functionality of nuclear actuators of prior designs counterparts (spring fail close, hydraulic piston open, servo valve, solenoid valves, dump valves for fast closure). The valve designs for ESBWR and associated overspeed probability are therefore, considered equivalent to the current steam valve model equations used in Reference 1.

5.1.2.1 Valve failure rates

The steam valve failure rates have been calculated using the same methodology followed in the Reference 1 and later General Electric assessments. Steam valve failure rates are 50% confidence values, based on a chi-square test.

The original failure data used in the Reference 1 report, the failure data collected for the 1993 Valve Test Interval Extension Supplement and failure data collected for a valve test interval assessment performed in 2008 have been combined to determine updated steam valve failure rates. The data included actual steam valve test intervals utilized in the field, valve test profiles and procedures, and valve inspection period and procedures. The steam valve failure rates have been calculated using the same approach followed in the 1984 and 1993 reports and recent assessments completed in 2008 and have been incorporated into the steam valve model. The 2008 valve failure rate reflects [[--- {A}]] of additional operation since the 1993 data and identified that no additional valve failures were experienced.

The resulting updated failure rates for the main stop and control valves and the combined stop and intercept valves for the 1993 and 2008 failure rates is as follows:

- Main Stop Valve

1993 [[--- ----------------------------------{{A}}]]

2008 [[--- ----------------------------------{{A}}]]

Total [[--- -------------------------{{A}}]]

- Main Control valve

1993 [[--- ------------------------------------{{A}}]]

2008 [[--- ------------------------------------ {{A}}]]

Total [[--- --------------------------- {{A}}]]

- Intermediate Stop Valve

1993 [[--- --{{A}}]]

2008 [[--- -- {{A}}]]

Total [[--- --------------------------- {{A}}]]

- Intercept Valve

1993 [[--- -- {{A}}]]

2008 [[--- -- {{A}}]]

Total [[-- --------------------{{A}}]]

The overspeed probability resulting from valve failure rates was assessed for valve test intervals of 90-days and 120-days. General Electric recommendation for valve test intervals for existing nuclear units is currently quarterly (90-days). Additional failure rate data at the greater valve test frequency would need to be collected over several years to improve the confidence in the valve reliability estimates for intervals greater than 90-days. However, approximately the same level of missile probability risk is realized for a valve test frequency of 120-days (with the updated failure rates) versus a 90-day test interval with the older failure rates. The values presented in this report are based on a 120-day valve test interval and are considered acceptable based on the conservatism of the model and the additional valve failure rate data obtained since the 1993 report, while maintaining a similar level of risk as compared with a 90-day valve test interval.

5.1.3 Extraction systems

Extraction systems differ in the amounts of energy available that could contribute to the overspeed of the TG. Depending on the amount of energy in the extraction line, various check valve schemes are employed, varying from no check valve in the line to two power actuated check valves.

Each turbine extraction line is reviewed for potential energy and contribution to overspeed. The number and type of extraction non-return valves required for each extraction line are specified based on the enthalpy and mass of steam and water in the extraction line and feedwater heater. Higher energy lines are provided with power-assisted open, spring-assisted closed non-return valves, controlled by air relay dump valves, which in turn, are activated by the ETS. The closing time of the extraction non-return valves is sufficient to minimize extraction steam contribution to the turbine overspeed event.

The model is concerned with turbine overspeed caused by load loss with valves stuck such that the trip system cannot function. An overspeed event caused by a stuck check valve, allowing steam to feed back from extraction processes is excluded from the probability model. Although the entrained energy associated with a stuck extraction line check valve is not inconsequential; it is not included in this assessment as the entrained energy associated with load loss and stuck MS/CV and CIVs is considered more limiting for overspeed calculations.

5.1.4 Hydraulic Model

The hydraulic model is concerned with hydraulic component failures due to the common failure mode, hydraulic fluid contamination. The steam valves and electronic system are assumed to function properly.

All components of the hydraulic system are assumed to have a constant failure rate after an initial delay time, during which, the probability of failure is zero. Testing of hydraulic components is not assumed to restore the component to "like new" condition, but is assumed to verify that the system is functional with no noticeable degradation. It is assumed that the components are restored to "like new" condition during each major planned outage. The component failure rates used in the hydraulic model are consistent with the values in the Reference 1 report.

Mathematical modeling is required for the hydraulic system to define the probability of occurrence of certain combinations of failures (given that hydraulic fluid contamination is present), to produce an overspeed during a load loss. In addition, the methodology assumes that there may be no indication of such failures (e.g. through testing), or no response by the customer, if there is an indication. When the probability of multiple failure combinations and the probability of no indication are combined the result gives the probability that multiple failure combinations of the hydraulic system can occur at any time (given that hydraulic fluid contamination is present) and that there is no indication of such failures or the indication goes unheeded by the customer. Combining these two probabilities may be accomplished by multiplication since the probability of no indication is a conditional probability with respect to the multiple failure combinations.

Table 5-4 Probability Equations EHC Hydraulic Fluid Model	
Event Number	Probability of Overspeed .
[[---	-------------------------------------
---	-------------------------------------
---	-------------------------------------
---	-------------------------------------
---	-------------------------------------
---	-------------------------------------
---	-------------------------------------
---	-------------------------------------
---	-------------------------------------
---	-- {A}]]

As indicated above, the resultant probability is a conditional probability in that it assumes that hydraulic fluid contamination is present. The contamination is assumed to originate from two sources.

- hydraulic pump failure and corresponding filter failure causes silt to enter hydraulic valves, or
- hydraulic fluid cooler failure allows water into the system which can eventually lead to rust in the hydraulic valves.

It is important to note that the rust is assumed to develop at the valves and the filter does not affect the results of the hydraulic cooler model.

An overspeed cannot occur in the hydraulic models without a load loss. In order to have an overspeed without a load loss a failure has to occur which drives the steam valves open. No such hydraulic failures are postulated in the models. In summary, for an overspeed to occur in the hydraulic model, there has to be: 1.) a pump failure and a filter failure, and a combination of hydraulic valve failures, and no indication of a failure or an indication with no customer response, and a load loss or; 2.) a cooler failure, and a combination of hydraulic valve failures, and no indication of a failure or an indication with no customer response, and a load loss.

The ESBWR hydraulic system will include titanium hydraulic fluid coolers. The effect of titanium hydraulic fluid coolers is significant in the modeling. This cooler configuration mitigates rusting issues far better than older cooler designs, therefore having a lower failure rate.

ESBWR will also incorporate new hydraulic fluid conditioning equipment that replaces the old "Selexsorb and Fuller's Earth" conditioning media with an ion exchanger. Oxidation and acid formation combine to breakdown the fluid and causes sticking and varnishing of spool valves in the system. Ion exchanger media better controls acid production in the hydraulic fluid and does not release metal soaps into the fluid (like the old conditioning media), which lead to increased air retention and oxidation in the fluid. A dry air blanket system has also been implemented, blowing over the top of the fluid in the reservoir to remove water from the system, as water directly impacts acid production in the hydraulic fluid. This equipment has been validated in the field and will mitigate fluid contamination, resulting in the assessment that the current hydraulic system model is considered to be equivalent or conservative.

5.1.5 Emergency Trip System

Although sharing the same basic hydraulic model and common failure modes, the previous nuclear steam turbine designs differ in the Emergency Trip System layout. Extensive analyses have been conducted for various nuclear turbine controls retrofit projects, quantifying the replacement of the existing control system, EHC and mechanical trip systems, with the MARK™ VIe and duplex TMR Emergency Trip Devices (ETDs). The previous EHC overspeed protection system consisted of two hydraulic trip valves; the mechanical trip valve (MTV) and the electrical trip valve (ETV) which have since been upgraded to a duplex TMR electronic trip device block.

If the normal speed control and power load unbalance function should fail on an ESBWR unit, the emergency trip system closes the main and intermediate stop valves. This turbine overspeed protection system comprises the second line of defense against turbine overspeed. The turbine hydraulic trip solenoid valve hydraulic circuits are arranged in a dual, "two-out-of-three," de-energize to trip configuration. Any power interruption to either set of the two-out-of-three trip solenoid valves in the ETD results in a turbine trip. The ETD is also fail-safe. Each trip solenoid transfers to the trip state on a loss of control power, resulting in a turbine trip.

The on-line test of the ETDs provides the capability of individually de-energizing each of the redundant ETDs and verifying its correct operation without tripping the unit. Only one ETD can be tested at a time, the controls logic allows the remaining ETDs to trip the turbine if necessary.

5.1.6 Turbine Generator Control System (MARK™ VIe)

The ESBWR MARK™ VIe overspeed analysis is leveraged from the MARK™ VIe Reliability, Availability & Maintainability (RAM) analyses completed in support of control system upgrades to existing nuclear fleet units. These prior analyses include major systems, subsystems, and components of the control system including the turbine controller, various Input/Output (I/O) devices (i.e. LVDTs, servo valves, pressure transducers, speed pickups, and proximity probes), and a modified front standard trip system. The modified front standard trip system replaces the previous mechanical overspeed trip function with a simple, diverse electronic/hydraulic trip system utilizing independent speed probes and an electric trip device block.

The multiple overspeed protection methods and resulting redundancies incorporated into the ESBWR control system are described in Subsection 2.2. Several of the major ESBWR controls improvements are highlighted below:

- Primary speed pickups are each connected to their own Primary Turbine Protection I/O boards.
- Emergency speed pickups are fanned to redundant sets of Emergency Protection I/O boards (Emergency Protection Terminal Board, SPRO & Turbine Emergency Trip Terminal Board).
- Each of the TMR ETD trip system legs are connected to separate Primary Trip Terminal Boards & Turbine Emergency Trip Terminal Boards
- Dual Power Load Unbalance boards, serving dual Relay Output Terminal Board I/O relay outputs for fast-acting solenoid valves (FASV's, with crossover pressure transmitters fanned to multiple Analog Input Terminal Boards.
- AC source selector, preserves power source to FASV's in event loss of one AC input.

The model considers two failure modes; 1) the first is the failure of the control system driving the control valves open during either start-up or during overspeed testing, and 2) the failure of the overspeed trip systems and the failure of the control system driving the control valves open just prior to a loss of load event. Only the latter condition results in an overspeed event greater than 120% speed.

The MARK™ VIe TGCS, ETD, and all associated I/O, excluding valves were evaluated for this same failure mode. Special consideration was given to common mode failures since this was determined to be the hidden driving cause of system failure. Valves were not included since they are considered separately in the overspeed model.

Since common mode failures were considered the critical driver to system failure, special attention was given to determine their probability. Completion of a Failure Modes and Effects Analysis (FMEA) identified that the types of common mode failures did not have a predictable nature and would require a unique computation method. IEC-61508 part 6 has an internationally accepted methodology for computing common mode failures as a function of the predicted failure rates of each of the components. This method was chosen because of the global acceptance of the process and the quantitative nature of the results.

The failure rates used in the probability model are based on several sources. The MARK™ VIe TGCS electronics models are based on Bell Communications Research (Bellcore) parts count predictions. The Bellcore predictions were developed assuming a 35°C ambient temperature and a 50% applied stress level. The ETD system, I/O transducers, and sensors were modeled utilizing existing GE Turbine fleet data. All failure rates were considered constant over time, which is represented by an exponential failure distribution.

The probability models include redundancy at the system level and, in a few cases, at the circuit level. The model considers the three simultaneous conditions necessary to occur to result in an overspeed event:

1) Loss of Load,
2) Failure of the Primary and Emergency Overspeed Trip system, and
3) Failure of the Control system, which results in driving the control valves open.

For this analysis, it was assumed that a load loss occurs once per year. The conditions that were critical in modeling the ETD trip system included in the model are:

- One of two parallel ETD sub-systems required to protect/trip the turbine,
- ETDs in each subsystem are arranged in a two-out-of-three configuration and remain energized/valve closed during normal operation,
- Weekly online functional testing of ETDs is performed,
- Functional testing is conducted on a two-year period during refueling outages, For ETD system components that are not tested via the weekly ETD test,

The primary and emergency overspeed protection system is dominated by undetectable common mode failures. An assessment of the system in accordance with IEC61508-6 results in an assumed β-Factor of 1% for the controls and 2% for the I/O and ETD. This means that 1% (or 2%) of the sum of component failure rates for all parts in the protection system represent the common mode failure probability. Of this percentage, 1% is not expected to be detectable. The Diagnostic coverage is assumed to be 99% for the logic and speed sensors and 60% for ETDs.

The TGCS failure probability resulting in driving a control valve open is also dominated by undetectable common mode failures. An assessment of the system in accordance with IEC61508-1 results in an assumed β-Factor of 1% for the controls and 2% for the I/O and ETD. Here again 1% (or 2%) of all component failure rates in the redundant control system represent the probability for the system failure, driving the valve open. Of this percentage, 1% is not expected to be detectable. The Diagnostic Coverage is assumed to be 99% for the logic and 60% for ETDs.

Ultimately, the results of the analyses for the MARK™ VIe TGCS and associated trip system upgrade yield a lower overspeed probability when compared to the existing fleet control and mechanical trip systems. However, for conservatism, the probability of an overspeed event due to ESBWR MARK™ VIe control system retains the probability identified in Reference (1).

6. CASING PENETRATION

The probability of casing penetration values used in the overall missile probability analysis are based on the missile energy analysis previously developed by General Electric and verified by full scale casing penetration tests sponsored by the Electric Power Research Institute (Reference 1). The casing escape probability is a function of burst speed and includes uncertainty of both fragment behavior and stationary structure energy absorption capability.

6.1 COMPONENTS

A schematic section of the turbine components included in the casing penetration model is shown in Figure 6-1. Individual wheel burst fragments are assumed to impact only those stationary components that lie directly in the path of the exiting fragment. The missile is assumed to be a 120-degree fragment of the wheel. During collision, kinetic energy is dissipated during fracture of the diaphragm and penetration of the inner casing and outer exhaust hood.

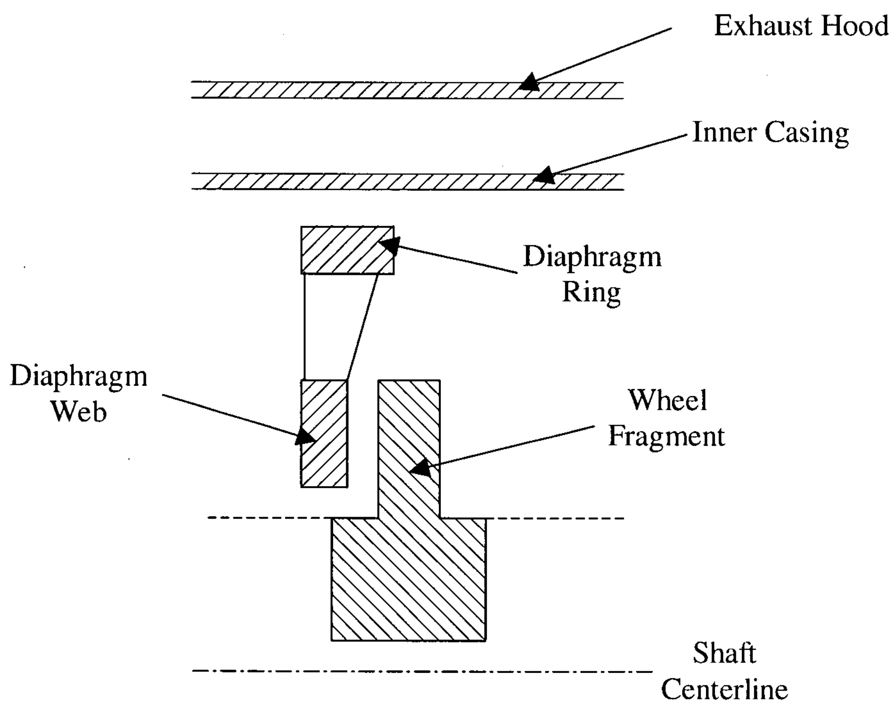

Figure 6-1 Components Included in Penetration Stage Model

6.2 CASING PENETRATION CALCULATIONS

The initial translational and angular velocity of the wheel fragment is determined based on conservation of momentum. The linear velocity of the fragment equals the radius of the fragment multiplied by the wheel angular velocity (prior to burst). The angular velocity of the fragment is assumed equal to the angular velocity of the wheel prior to burst. Translational kinetic energy equals one half the product of mass times translational velocity squared. Likewise, rotational kinetic energy is assumed equal to one half the product of polar moment of inertia and angular velocity squared:

$$KE_{tr} = \frac{1}{2} \times M_f \times V_f^{\,2} \qquad \text{(eq. 6-1)}$$

$$KE_{ro} = \frac{1}{2} \times J_f \times \omega_f^{\,2} \qquad \text{(eq. 6-2)}$$

As summarized in Reference 1, diaphragm web and ring fragments are assumed to be created and accelerated as a result of collision. The retained wheel fragment energy is:

$$KE_{ff} = KE_{fi} \times \left[\frac{\dfrac{2M_f}{M_w} - 1}{\dfrac{2M_f}{M_w} + 1} \right]^2 \qquad \text{(eq. 6-3)}$$

where:

KE_{ff} = energy of wheel fragment after collision

KE_{fi} = energy of wheel fragment before collision

M_f = mass of wheel fragment

M_w = mass of diaphragm fragments that are directly in the path of the wheel fragments

The energy lost by the wheel fragment during penetration of the inner casing wrapper and exhaust hood is calculated from the empirical relation developed by Moore (Reference 6) commonly referred to as the "Stanford Formula". The Stanford formula applies to missiles having a right circular solid shape that impact a flat plate with the axis of the cylinder normal to the plate. The energy loss during penetration (ft-lbs) is a function of both the plate ultimate tensile strength (UTS) and thickness (T) as follows:

$$KE_{loss} = UTS \times DIA \times T_c \times (.344T_c + .008DIA) \qquad \text{(eq. 6-4)}$$

where:

UTS = casing material ultimate tensile strength (psi)

DIA = equivalent circular diameter of fragment (in)

T_c= casing thickness (in)

KE_{loss} =Energy Lost by Fragment (ft-lbs)

Based on the preceding discussion and further consideration of fragment size, orientation during penetration, impact fragment direction, energy at fracture and energy absorption, a statistical spread in escaping or external energy is derived. The probability of casing penetration is taken to be the probability that the external energy is greater than zero.

7. OVERALL PROBABILITY DETERMINATION

7.1 PROBABILITY OF BRITTLE FRACTURE ($K_I \geq K_{IC}$)

The probability that the rotor will burst by time t_2 is the probability that K_I is greater than or equal to K_{IC} by time t_2. For each turbine stage, multiplying the following three conditional probabilities:

- Crack depth existing
- Excess temperature occurring
- $K_I \geq K_{IC}$ given the crack depth and excess temperature

and integrating over the whole range of possible rotor material excess temperatures gives the probability that the given crack depth will cause a burst by time t_2. A further integration over all possible crack sizes will give the probability of a rotor burst by time t_2 for any depth of crack. Time is a factor since the crack model assumes that the crack size increases with time. Stated mathematically:

$$P_b(t_2|\sigma,T) = \int_0^{\infty}\int_{-\infty}^{\infty}\int_{-\infty}^{\infty} I(LogK_1 \geq LogK_{1c}) f_e(y) f_f(X) f_a(a)\,dy\,dX\,da \qquad \text{(eq. 7-1)}$$

where:

$P_b(t_2|\sigma,T)=$ cumulative burst probability at time $=t_2$,given stress (σ) and temperature (T)

$I(LogK_1 \geq LogK_{1c})=$ 1 if y, X, and a are such that Log $K_1 \geq$ Log K_{1c}.
=0 otherwise

$f_e(y)=$ the probability density of the error function

$f_f(X)$ = the probability density of excess temperature

$f_a(a)$ = the probability density of crack depth

$\sigma, T =$ Operational stress at the desired speed and temperature for the stage in question. (Each turbine stage has a unique combination of stress and temperature.)

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7.2 PROBABILITY OF DUCTILE TENSILE FAILURE

The probability of wheel burst in the ductile mode is determined from the Average Tangential Stress (ATS) and the Ultimate Tensile Strength (UTS). Failure occurs when the stress equals a set ratio of strength as described in subsection 4.4. As noted in subsection 4.4, UTS, ATS, and the ratio R are all assumed to be log normally distributed.

The ultimate tensile strength of the wheel material varies with the temperature of the material and the stress in the wheel varies with the speed of rotation. Therefore, the probability of wheel burst in the ductile mode is conditional on the temperature of the wheel and on the speed of rotation.

7.3 NORMAL OPERATION

The probability of a rotor burst is a function of time, start/stop cycles, temperature, and speed. Time and cycle count determine the overall crack size distribution. Larger cracks lead to higher probability of brittle failure. Temperature is important in the determination of the rotor material toughness. Lower temperature will reduce the toughness and increase the probability of brittle burst. Speed is important in determining the stress intensity factor since a higher speed results in larger wheel stresses and larger stress intensity factors and therefore leads to a higher burst probability. Larger stresses also increase the ductile failure probability. Thus, the parameters of time, temperature, and speed must be properly accounted for in the overall probability determination.

During operation, low-pressure rotor disks are subjected to various speeds and temperatures. Each start/stop operating cycle shown in the Table 4-3 is assumed to consist of the elements shown in Table 7-1. The occurrence of these conditions is considered normal and is assigned a probability of one.

Table 7-1 Conditions in a Normal Operating Cycle		
Description	**Speed/Rated Speed**	**Min. Temp (°F)**
Shutdown	0	50
Turning Gear	0.001	50
Start (Unsync'd)	1	75
Loading (Including Thermal Stress)	1	75
Part Load	1	120
Full Load	1	225
Unloading	1	120
O/S Trip Test	1.11	120
Coastdown	1	120

Cumulative burst probability (equation 7-1) is then calculated for each element of the start/stop cycle shown in Table 7-1 and the maximum value determined as a function of time for each stage. Figure 7-1 illustrates the maximum predicted cumulative burst probability for a typical single LP stage derived from equation 7-1.

This cumulative burst probability (example Figure 7-1) defines a one-time burst probability given no prior stressing of the material. The normal annual burst probability is the annual failure rate given that no failure has occurred previously. This value is determined from the instantaneous slope of the cumulative burst probability curve (example Figure 7-1) at the time of interest for the stage in question.

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Figure 7-1 Maximimum Cumulative Burst Probability of Typical Stage (Normal Operation)

Stated mathematically, annual normal burst probability is derived from the cumulative burst probability as follows:

$$P_{BN}(t) = \frac{d}{dt}\left[\frac{P_{BMAX}(t) - P_{BMAX}(0)}{1 - P_{BMAX}(0)}\right] \quad \text{(eq.7-17)}$$

where:

$P_{BN}(t) =$ annual normal burst probability at time = t

t = time of interest, in years

PBMAX(t) = maximum individual value of rotor burst cumulative probability at time = *t* (for the 9 normal "events")

PBMAX(0) = highest value of initial rotor burst cumulative probability at time t = 0 (for the 9 normal "events")

During numerical evaluation, the annual normal cumulative burst probability is calculated separately for the dovetail, body, and tensile mechanisms and then combined for each stage and time of interest.

7.4 ABNORMAL OPERATION

An abnormal event is defined as an occurrence of a control system failure and a full load rejection by the turbine generator that causes operation above the normal operating speed. Each abnormal event has a maximum speed and an annual probability of occurrence as shown in Table 7-2. The values shown in Table 7-2 reflect four month (120-days) valve testing frequency.

Table 7-2 Typical Data for Abnormal Events

Event Number[3]	Speed/Rated Speed	Abnormal Event Annual Probability[4]
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[3] Reference Table 1.5-1

[4] Values are based on the results of the abnormal probability model discussed in Section 1.5.

When the abnormal event occurs, speed increases from normal running speed and continues until it reaches the maximum speed for the particular event. Burst will not occur until the cumulative burst probability exceeds the level attained during normal operation. The probability that the stage will burst at this temperature, given the occurrence of the abnormal event, is simply the difference between the conditional burst probability at the maximum speed and that at the worst cumulative burst probability encountered during normal operation. A wheel stage will only burst due to an abnormal event if it has not burst during prior normal operation (conditional probability). Therefore, this probability difference must be divided by the probability that no burst has occurred during normal operation:

$$P_{BA}(t) = \sum_i P_{Ai} \times \left[\frac{P_{BAi}(t|\sigma_i, T) - P_{BMAX}(t)}{1 - P_{BMAX}(t)} \right] \quad \text{(eq.7-18)}$$

where:

$P_{BA}(t) =$ annual abnormal burst probability at time =*t*

$P_{Ai} =$ annual probability of occurrence of abnormal event *i*. (ref. Table 7-2)

$P_{BAi}(t|\sigma_i, T) =$ annual cumulative burst probability at time = *t*, given that the abnormal event (*i*) occurs

$P_{BMAX}(t) =$ cumulative burst probability for the worst normal operating condition

σ_i = stress corresponding to the maximum speed of the event (reference Table 4-1 and noting that mechanical stresses increase with the square of speed)

T = minimum temperature during normal operation

During numerical evaluation, the annual abnormal cumulative burst probability. $P_{BAi}(t|\sigma_i, T)$, is calculated separately for the dovetail, body and tensile mechanisms and then combined prior to the event summation indicated by equation 7-18. In addition, each stage is evaluated separately.

The probability of burst during abnormal operation is conditional on the stage not bursting during normal operation by the time in question. When the normal and abnormal individual burst probabilitites are combined, the probability of burst during abnormal operation must be multiplied by the probability of no stage burst during normal operation. The total single stage annual burst probability is then determined as follows:

$$P_B(t) = P_{BN}(t) + [1 - P_{BN}(t)] \times P_{BA}(t) \qquad \text{(eq. 7-19)}$$

where:

$P_B(t) =$ annual stage burst probability at time = t

Figure 7-2 illustrates the total annual burst probability predicted for the same stage shown in Figure 7-1 using the above procedures and the abnormal event probabilities shown in Table 7-2.

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Figure 7-2 Annual Burst Proability for Typical Stage (Normal and Abnormal Operation)

Missile probability calculation requires the addition of casing escape probability to Equations 7-17 and 7-18 as follows:

$$P_{MN}(t) = P_{EPB\max} \times P_{BN(t)} \qquad \text{(eq. 7-20)}$$

where:

$P_{MN}(t) =$ annual stage missile probability at time = t associated with normal operation

$P_{EPB\max} =$ stage casing escape probability given that burst has occurred during the worst normal operating condition

$$P_{MA}(t) = \sum_{i} P_{Ei} \times P_{Ai} \times \left[\frac{P_{BAi}(t|\sigma_i, T) - P_{BMAX}(t)}{1 - P_{BMAX}(t)} \right] \qquad \text{(eq. 7-21)}$$

where :

P_{Ei} = conditional casing escape probability given that a burst has occurred for the stage in question during abnormal event i (reference Table 7-2)

$P_{MA}(t)$ = annual stage missile probability at time $= t$ associated with the abnormal event i

Similar to Equation 7-19, the probability of generating a turbine missile external to the casing during abnormal operation is conditional on the stage not bursting during normal operation by the time in question. When the normal and abnormal missile probabilities are combined, the probability for an external missile during abnormal operation must again be multiplied by the probability of no stage burst during normal operation:

$$P_M(t) = P_{MN}(t) + [1 - P_{BN}(t)] \times P_{MA}(t) = P_1 \qquad \text{(eq. 7-22)}$$

where:

$P_M(t)$ = annual stage missile probability at time $= t$

As noted, the rotor is assumed to be made up of a series of independent wheel stages and individual stage burst probabilities are combined to obtain the probability that at least one wheel stage on the rotor will burst. Missile probabilities are similarly combined.

8. BORELESS (SOLID) ROTOR RESULTS SUMMARY

The calculated annual missile probability for an individual ESBWR turbine containing three statistically independent solid LP rotors (42 stages in total) and valve testing in 120-day (4 month) intervals is shown in Figure 8-1. The annual missile probability (P_1) at the 12-year inspection frequency is 1.2×10^{-7} and remains less than 1×10^{-5} for greater than 50 years of turbine operation, with the assumption that no corrective actions are taken to address any observed crack indications in the dovetail slot bottom.

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Figure 8-1 Unit Featuring Solid Rotors: Annual Missile Probability

The missile probability assessment described in Sections 6 and 7 applies to both a solid (boreless) and bored rotor. The statistical distribution of the crack size is dependent on the bore configuration. For the solid rotor, the rotor body undetected flaw is assumed to be a half penny shaped embedded crack that is oriented in the radial axial plane. By comparison, the bored rotor undetected flaw is assumed to be a half penny shaped surface crack. A smaller undetected crack can be claimed because a bored rotor is inspected from both the outer periphery and the inner periphery (e.g., bore surface). In contrast, the solid rotor is only inspected from the outer periphery. These concepts are summarized in subsection 4.2.2. Section 9 provides a comparison of results between a bored and solid rotor. The bored rotor stresses shown in Tables 4-1 and 4-3 are reflected in Figure 8-1 as is the undetected flaw assumptions

The flat region of Figure 8-1 (up to **[[--- {A}]]** years) is dominated by valve failure (event #1 in Table 7-2). The probability shown in this region is associated with gross tensile failure during the subsequent overspeeding of the turbine. The curve is flat in this region because 1) rotor tensile strength is time invariant and 2) the probability of the valve failure is dependent only on the in-service valve-testing interval.

The gradual increase in missile probability beginning after **[[--- {A}]]** years reflects the increased probability with time of a dovetail SCC crack reaching critical size. The conservative SCC crack growth model described in subsection 4.3 results in a gradual increase in predicted crack depth with time. The gradual increase in missile probability is a reflection of the expected increase in crack size with time.

The SCC crack initiation and growth distributions shown in Equations 4-4 and 4-5 and used in the calculations summarized in Figure 8-1 are based on historical field experience with shrunk-on wheel bore keyway cracks. The use of this data is considered to be conservative when applied to the ESBWR LP monoblock rotors. The justification for this assessment is based on: 1.) The ESBWR rotors feature greatly reduced concentrated tensile stress magnitudes in critical locations (vs. earlier shrunk-on wheel keyways), and 2.) critical locations will be shot-peened and thus feature a beneficial layer of compressive residual stress at or below the outer surface (SCC characteristics utilized in this report are from non shot-peened shrunk-on wheels).

These conservative assumptions of SCC initiation and growth result in a mean predicted SCC crack length of 3 inches in the 5th stage dovetail slot bottom after 50 years of accumulated turbine operation (the worst case location). The calculated 3 inch mean length is within the crack detection capability for the planned testing described in Section 10. At that time, the missile probability analysis will be updated to reflect both the actual measured crack size (if a crack does exist) and the expected uncertainty in the crack measurement. The inspection results will allow a strategy to be developed for the inspection frequency to allow continued operation of the unit while ensuring that the NRC limits of missile probability are not exceeded.

9. BORED ROTOR RESULTS SUMMARY

Similar to the results presented in Section 8, the calculated annual missile probability for an individual ESBWR turbine containing three statistically independent bored LP rotors (42 stages in total) and valve testing in 120-day (4 month) intervals is shown in Figure 9-1 The annual missile probability (P_1) at the 12 year inspection frequency is 9.3×10^{-7} and remains less than 1×10^{-5} for greater than 50 years of accumulated turbine operation, with the assumption that no corrective actions are taken to address any observed crack indications in the dovetail slot bottom.

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Figure 9-1 Unit Featuring Bored Rotors: Annual Missile Probability

The bored rotor stresses shown in Tables 4-1and 4-3 are reflected in Figure 9-1 as is the undetected flaw assumptions summarized in subsection 4.2.2.

The flat region of the curve (up to **[[--- {A}]]** years) is dominated by valve failure (event #1 in Table 7-2). The probability in this region is associated with gross tensile failure during the subsequent overspeeding of the turbine. The curve is flat in this region because 1) rotor tensile strength is time invariant and 2) the probability of the valve failure is dependent only on the in-service valve-testing interval.

The gradual increase in missile probability beginning after **[[--- {A}]]** years reflects the increased probability with time of a dovetail SCC crack reaching critical size. The conservative SCC crack growth model described in section 4.3 results in a gradual increase in predicted crack depth with time. The gradual increase in missile probability is a reflection of the expected increase in crack size with time. Again it should be noted that the SCC crack initiation and growth distributions shown in Equations 4-4 and 4-5 and used in the calculations summarized in Figure 8-1 are based on historical field experience with shrunk-on wheel bore keyway cracks. The use of this data is considered to be conservative when applied to the ESBWR LP monoblock rotors. The justification for this assessment is based on: 1.) The ESBWR rotors feature greatly reduced concentrated tensile stress magnitudes in critical locations (vs. earlier shrunk-on wheel keyways), and 2.) critical locations will be shot-peened and thus feature a beneficial layer of compressive residual stress at or below the outer surface (SCC characteristics utilized in this report are from non shot-peened shrunk-on wheels).

These conservative assumptions of SCC initiation and growth result in a mean predicted SCC crack length of 3 inches in the 5th stage dovetail slot bottom after 50 years of accumulated turbine operation (the worst case location). The calculated 3-inch mean length is within the crack detection capability for the planned testing described in Section 10. At that time, the missile probability analysis will be updated to reflect both the actual measured crack size (if a crack does exist) and the expected uncertainty in the crack measurement. The inspection results will allow a strategy to be developed for the inspection frequency to allow continued operation of the unit while ensuring that the NRC limits of missile probability are not exceeded.

Figure 9-2 provides a comparison of the missile probability between a bored and solid rotor. In addition to the differences in assumed undetected crack size described in Section 8, the solid rotor features a slightly lower average tangential stress as summarized in Table 4-3.

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Figure 9-2 Bored vs. Boreless Rotors - Unit Annual Missile Probability

The bored and boreless designs feature negligible difference in dovetail stress and temperature (any given stage). Thus, the probability of a burst due to dovetail SCC is the same for either design. The bored rotor's slightly greater overall missile probability in the upswing region is due to the secondary contribution of the gross tensile failure, which has greater probability in the bored rotor.

10. IN-SERVICE INSPECTIONS

10.1 IN-SERVICE MAINTENANCE AND INSPECTION OF TURBINE ROTORS

The in-service maintenance and inspection program for the turbine assembly and accessories includes the complete inspection of all normally inaccessible parts such as couplings, coupling bolts, turbine blades and turbine rotors. Inspections typically coincide with refueling outages and may be sequenced by section (i.e., LPA, LPB, LPC) provided that individual sections are inspected at least once within the recommended 12-year period. Inspection consists of visual, surface, and volumetric examinations. Engineering disposition of any anomalies shall include consideration and recalculation (if necessary) of missile probability. As mentioned in Section 1, refinement of the ESBWR specific SCC model is dependent on in-service inspection measurements.

10.1.1 Rotor Dovetail Inspections

Compared to older vintage designs, ESBWR rotor dovetails feature significantly lower tensile stress magnitude. General Electric believes the combination of lower stress and shot peening will result in significantly longer initiation time for SCC. General Electric, however, recommends periodic dovetail inspection at a frequency not to exceed 12 years of accumulated operation. The examination of wheel dovetails can be performed with buckets assembled to the rotor.

Surface inspection of tangential entry dovetails (Stages 1 thru 5), is not possible because of the externally mounted bucket. For these stages, General Electric will continue to offer utilities a patented phased array inspection service that has been extensively calibrated and refined in General Electric's fossil and nuclear fleet.

Unlike the tangential entry dovetails, surface inspection of the axial entry wheel dovetails (via magnetic particle) on the wheel faces is possible with assembled buckets. For the axial entry dovetails, General Electric recommends surface inspection of the dovetail faces (both upstream and downstream) as well as ultrasonic examination of the dovetail slot bottoms. The latter will include both pulse/echo and pitch/catch type inspection with transducers and sensors located on both upstream and downstream wheel faces. Indications and/or reflectors found during any of these tests (both tangential entry and axial entry stages) will require engineering disposition and may result in a recommendation by General Electric to remove buckets and perform more extensive surface testing.

The inspection results will be used to update the SCC initiation and growth rates used in the calculation as required. Re-inspection intervals (after the initial 12 year inspection) are condition based and dependent on the previous inspection results.

10.1.2 In-service Inspection of Turbine Valves

All main stop valves, control valves, extraction non-return valves important to overspeed protection (although not specifically required to support the assumptions of the missile probability analysis), intermediate stop, and intercept valves are tested under load. Test controls installed in the Main Control Room (MCR) permit full stroking of the stop valves, control valves, and intermediate stop and intercept valves. Valve position indication is provided in the MCR. Some load reduction may be necessary before testing main stop and control valves, intermediate stop and intercept valves.

Main stop, main control, intermediate stop, and intercept valves are exercised at intervals as required by the turbine missile probability analysis (120-days) by closing each valve and observing the remote valve position indicator for fully closed position status. This test also verifies operation of the fast closure function of each main stop and main control valve during the last few percent of valve stem travel. Fast closure of the intermediate stop and intercept valves is tested in a similar way if they are required to have a fast close function that is different from the test exercise.

A tightness test of the main stop and main control valves may be performed as required. A tightness test is normally performed by checking the coast down characteristics of the turbine from no load with each set of four main stop and main control valves closed alternately. As alternative methods, warm up steam may be used as an indicator or the turbine speed may be monitored when on the turning gear while opening each set of four main stop and main control valves alternately.

All main stop valves, main control valves, and intermediate stop and intercept valves are disassembled and visually inspected once during a three refueling shutdown cycle such that all of the valve are inspected at least once within a six-year period. Currently, General Electric recommends valve inspections/maintenance be performed every three to five years. The inspections are conducted for:

- Wear of linkages and valve stem packing;
- Erosion of valve seats and stems;
- Deposits on stems and other valve parts, which could interfere with valve operation; and
- Distortions, misalignment or cracks.

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REFERENCES

1) General Electric Co. Large Steam Turbine-Generator Department, "Probability of Missile Generation in General Electric Nuclear Turbines", January 1984 [NRC approval NUREG-1048, Supplement 6 dated July 1986, Appendix U]
2) Rolfe, S.T. and Novak, S.R. "Slow Bend KIC Testing of Medium Strength High Toughness Steels", American Society for Testing and Materials, STP 463, 1970 p.124
3) JC Newman and IS Raju, "Stress Intensity Factor Equations for Cracks in Three Dimensional Finite Bodies Subject to Tension and Bending Loads", Langley Research Center, Hampton, Virginia 1984
4) RC Shah and AS Kobayshi, "Stress Intensity Factor for an Elliptical Crack Approaching the Surface of a Plate in Bending", Stress Analysis and Growth of Cracks, Proceedings of the 1971 National Symposium on Fracture Mechanics, Part I, ASTM STP 513, American Society of Testing and Materials, 1972, pp 3-21.
5) Hiroshi Tada, "The Stress Analysis of Cracks Handbook", Del Research Corporation, Hellertown, Pa 1973
6) Moore, C.V., "The Design of Barricades for Hazardous Pressure Systems" Nuclear Engineering and Design, Vol. 5, No. 1, 1967

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Enclosure 3

MFN 09-484 Supplement 1

Affidavit

GE-Energy

AFFIDAVIT

I, Damodar Padhi, state as follows:

(1) I am the General Manager-Steam Turbine Engineering, have been delegated the function of reviewing the information described in paragraph (2) which is sought to be withheld, and have been authorized to apply for its withholding.

(2) The information sought to be withheld is the "ESBWR Steam Turbine – Low Pressure Rotor Missile Generation Probability Analysis" ST-56834/P, Revision 2 dated September 14, 2010. The GE proprietary information in contained within the report is delineated by a **[[**text of proprietary information $^{\{A\}}$**]]**. Figures and large equation objects are identified with bold red double square brackets before and after the object. In each case, the superscript notation $^{\{A\}}$ refers to Paragraph (3) of this affidavit, which provides the basis for the proprietary determination. A non-proprietary version of this report has been provided titled, GE "ESBWR Steam Turbine – Low Pressure Rotor Missile Generation Probability Analysis" ST-56834/N-P, Revision 2 - Public Version, dated September 14, 2010.

(3) In making this application (via GEH submittal letter) for withholding of proprietary information of which GE is the owner, GE relies upon the exemption from disclosure set forth in the Freedom of Information Act ("FOIA"), 5 USC Sec. 552(b)(4), and the Trade Secrets Act, 18 USC Sec. 1905, and NRC regulations 10 CFR 9.17(a)(4), and 2.390(a)(4) for "trade secrets" (Exemption 4). The material for which exemption from disclosure is here sought also qualify under the narrower definition of "trade secret," within the meanings assigned to those terms for purposes of FOIA Exemption 4 in, respectively, <u>Critical Mass Energy Project v. Nuclear Regulatory Commission,</u> 975F2d871 (DC Cir. 1992), and <u>Public Citizen Health Research Group v. FDA</u>, 704F2d1280 (DC Cir. 1983).

(4) Some examples of categories of information that fit into the definition of proprietary information are:

a. Information that discloses a process, method, or apparatus, including supporting data and analyses, where prevention of its use by GE competitors without license from GE constitutes a competitive economic advantage over other companies;
b. Information which, if used by a competitor, would reduce his expenditure of resources or improve his competitive position in the design, manufacture, shipment, installation, assurance of quality, or licensing of a similar product;
c. Information which reveals aspects of past, present, or future GE customer-funded development plans and programs, resulting in potential products to GE;
d. Information that discloses patentable subject matter for which it may be desirable to obtain patent protection.

The information sought to be withheld is considered to be proprietary for the reasons set forth in paragraphs (4)a, and (4)b, above.

(5) To address 10 CFR 2.390(b)(4), the information sought to be withheld is being submitted to NRC in confidence. The information is of a sort customarily held in confidence by GE, and is in fact so held. The information sought to be withheld has, to the best of my knowledge and belief, consistently been held in confidence by GE, no public disclosure has been made, and it is not available in public sources. All disclosures to third parties including any required transmittals to NRC, have been made, or must be made, pursuant to regulatory provisions or proprietary agreements, which provide for maintenance of the information in confidence. Its initial designation as proprietary information, and the subsequent steps taken to prevent its unauthorized disclosure, are as set forth in paragraphs (6) and (7) following.

(6) Initial approval of proprietary treatment of a document is made by the Chief Engineer – Steam Turbines, the person most likely to be acquainted with the value and sensitivity of the information in relation to industry knowledge, or subject to the terms under which it was licensed to GE. Access to such documents within GE is limited on a "need to know" basis.

(7) The procedure for approval of external release of such a document typically requires review by the staff manager, project manager, principal scientist or other equivalent authority, by the manager of the cognizant marketing function (or his delegate), and by the Legal Operation, for technical content, competitive effect, and determination of the accuracy of the proprietary designation. Disclosures outside GEH or GE-Steam Turbine are limited to regulatory bodies, customers, and potential customers, and their agents, suppliers, and licensees, and others with a legitimate need for the information, and then only in accordance with appropriate regulatory provisions or non-disclosure agreements.

(8) The information identified in paragraph (2), above, is classified as proprietary because it identifies detailed GE ESBWR steam turbine design information. GE utilized prior design information and experience from its Turbine-Generator fleet with significant resource allocation in developing the system over several years at a substantial cost.

The development of the evaluation process along with the interpretation and application of the analytical results is derived from the extensive experience database that constitutes a major GE asset.

(9) Public disclosure of the information sought to be withheld is likely to cause substantial harm to GE's competitive position and foreclose or reduce the availability of profit-making opportunities. The information is part of GE comprehensive BWR Turbine-Generator safety and technology base, and its commercial value extends beyond the original development cost. The value of the technology base goes beyond the extensive physical database and analytical methodology and includes development of the expertise to determine and apply the appropriate evaluation process. In addition, the technology base includes the value derived from providing analyses done with NRC-approved methods.

The research, development, engineering, analytical and NRC review costs comprise a substantial investment of time and money by GE.

The precise value of the expertise to devise an evaluation process and apply the correct analytical methodology is difficult to quantify, but it is clearly substantial.

GE's competitive advantage will be lost if its competitors are able to use the results of the GE experience to normalize or verify their own process or if they are able to claim an equivalent understanding by demonstrating that they can arrive at the same or similar conclusions.

The value of this information to GE would be lost if the information were disclosed to the public. Making such information available to competitors without their having been required to undertake a similar expenditure of resources would unfairly provide competitors with a windfall, and deprive GE of the opportunity to exercise its competitive advantage to seek an adequate return on its large investment in developing these very valuable analytical tools.

I declare under penalty of perjury that the foregoing affidavit and the matters stated therein are true and correct to the best of my knowledge, information, and belief.

Executed on this 17 day of September 2010.

Damodar Padhi
General Mgr, Steam Turbine Engineering
GE-Energy Engineering