Dry Cask Storage; Safely Moving Spent Nuclear Fuel - 11169

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ABSTRACT

Few special projects at nuclear power plants receive more regulatory oversight than the long term storage of spent nuclear fuel. Working with the regulator and many US plants, Konecranes Nuclear Equipment and Services (P&H Cranes) has been safely moving spent nuclear fuel in casks from the plant's fuel pools to storage pads at numerous nuclear power generating facilities. This process includes the use of new and modified Single Failure Proof Cranes (SFP crane), special rigging equipment and sophisticated Cask Transporters to safely transport the cask to its interim resting place. This paper will discuss recent safety improvements in the equipment used to move the highly radioactive material from the spent fuel pool to the long term storage pad including the industry standards that guide the design, manufacturing and testing of the Overhead (SFP) Crane and the Vertical Cask Transporter (VCT).

OVERHEAD CRANE

The overhead crane in either the Spent Fuel Building of a Pressurized Water Reactor (PWR) or Reactor Building of a Boiling Water Reactor (BWR) is one of the first tools used in getting the fuel to the pad. The crane is responsible for handling the cask and accurately positioning it underwater next to the spent fuel racks where it is filled with spent fuel. Once the cask is filled, the crane performs a number of steps to assemble the cask, and then lifts and transports the cask to a location where it can leave the containment building. During these moves of cask components and the assembled cask, numerous heavy loads are transported over and/or near the spent fuel racks and therefore the process falls under the requirements of NUREG 0612, Control of Heavy Loads (Reference 1) and other regulations.

NUREG 0612 provided a path forward for users to define the requirements of the overhead crane, specifically the Single Failure Proof Criteria, but implemented standards that are approximately 30 years old. NUREG 0612 provided one path for the user to make the crane a Single Failure Proof device per the requirements of NUREG 0554 (Reference 2). NUREG 0554 provided the user direction, but was very qualitative in nature, leaving design areas open to interpretation. Figure 1 provides a flow path on how the standards governing this issue evolved over the years, and shows a major flaw since the primary document was NUREG 0554. Two major steps which occurred to help eliminate this ambiguity included the issuance of ASME NOG–1–2004 (Reference 3) and the adoption of NOG–1–2004 by the Nuclear Regulatory Commission (Reference 4). For plants with existing cranes, this provided a quantifiable and detailed direction for the implementation of Single Failure Proof Criteria plus a path to use their original design basis to meet the regulator's requirements. The most common solution is to replace the trolley with a NOG–1–2004 compliant assembly, and then qualify the bridge to NUREG 0554. The plant can then bundle the entire crane into a NUREG–0554 compliant assembly to meet the original design basis of their plant and thus only requires a 10CFR50.59 Changes, Tests and Experiments process to enable them to proceed with Spent Fuel Cask handling.

For new cranes or new facilities, the use of NOG–1–2004 provided them a comprehensive methodology to obtain approval for the new facility or application. The entire crane could be built per NOG–1, and then the plant could issue a 50.59 since NOG–1–2004 is an acceptable alternative to NUREG 0554 per the directives in Reference 4.

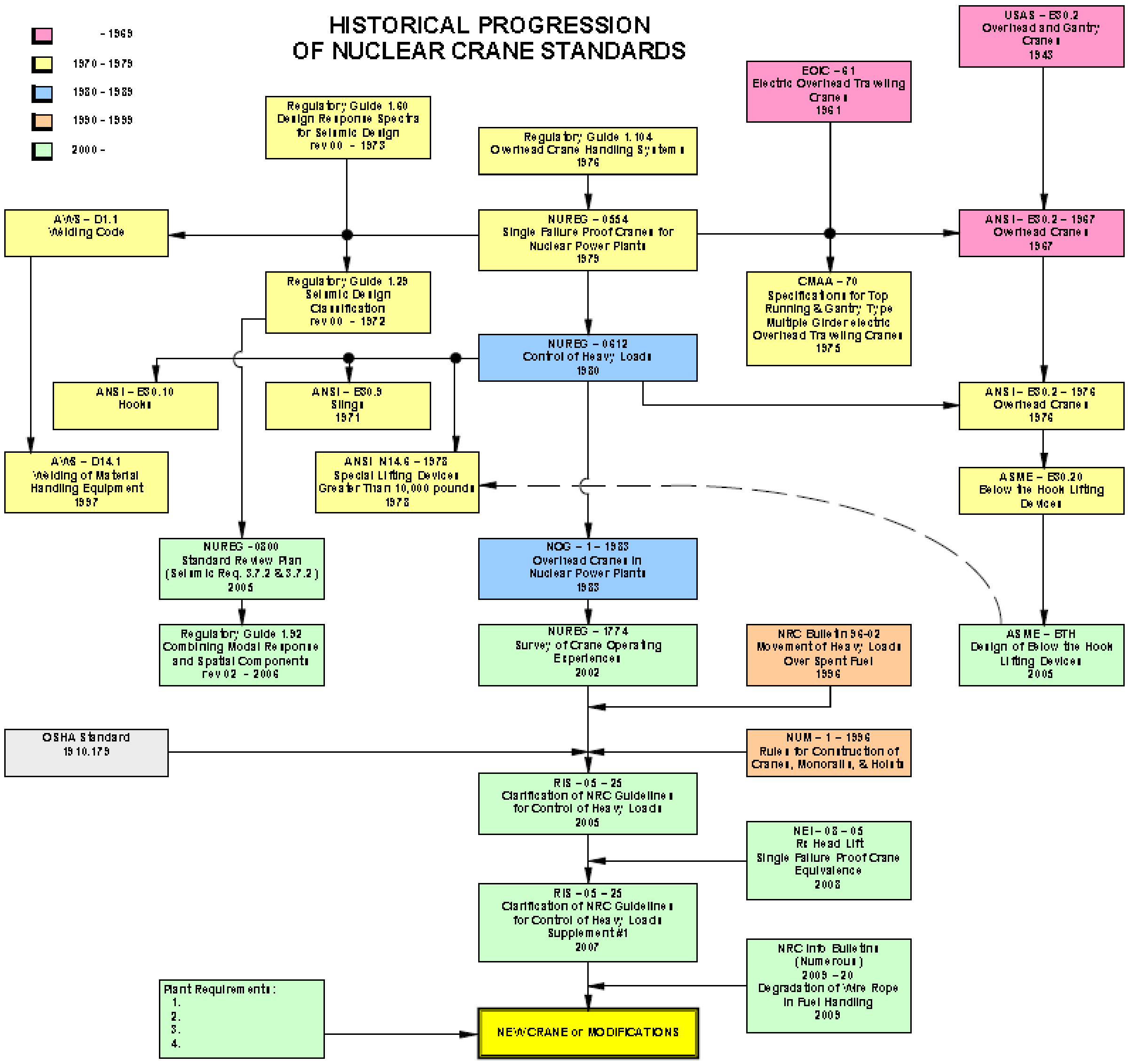

Figure 1 – Historical Progression of US Nuclear Crane Standards

A Single Failure Proof crane is "*designed such that any credible failure of a single component will not result in the loss of capability to stop and hold the critical load within facility acceptable excursion limits.* (NOG–1, Reference 3)" To meet this rigorous requirement, numerous unique safety features are implemented to satisfy the criteria, but the overriding design philosophy is defense–in–depth. The entire electrical and mechanical approach is to ensure there is always a minimum of one back up to every safety system or mechanical device that holds the load. This includes wire ropes, gearboxes, brakes, limit switches and load weighing devices, to name a few.

One example of the defense–in–depth methodology is the hoist assembly which is configured in a "closed kinematic loop" as shown in ASME NOG–1–2004 and the French EDF standards (Figure 2.) It uses dual drive components so that if one item in the drive train fails, the "other" side of the drive train can handle the load. Each side of the drive train is fully designed to handle the load in as much the calculations are performed assuming only one gear box, shaft, brake, etc. are available during worst case scenarios. As an example, if the shaft breaks at the input of the gearbox on the right side of the trolley, the other gearbox will receive the torque (drive input) from the motor and can raise or lower the load. As another example, if the drum shaft breaks on one side, the other gearbox and brake can stop hoist rotation, hold the load indefinitely and safely lower the load when ready.

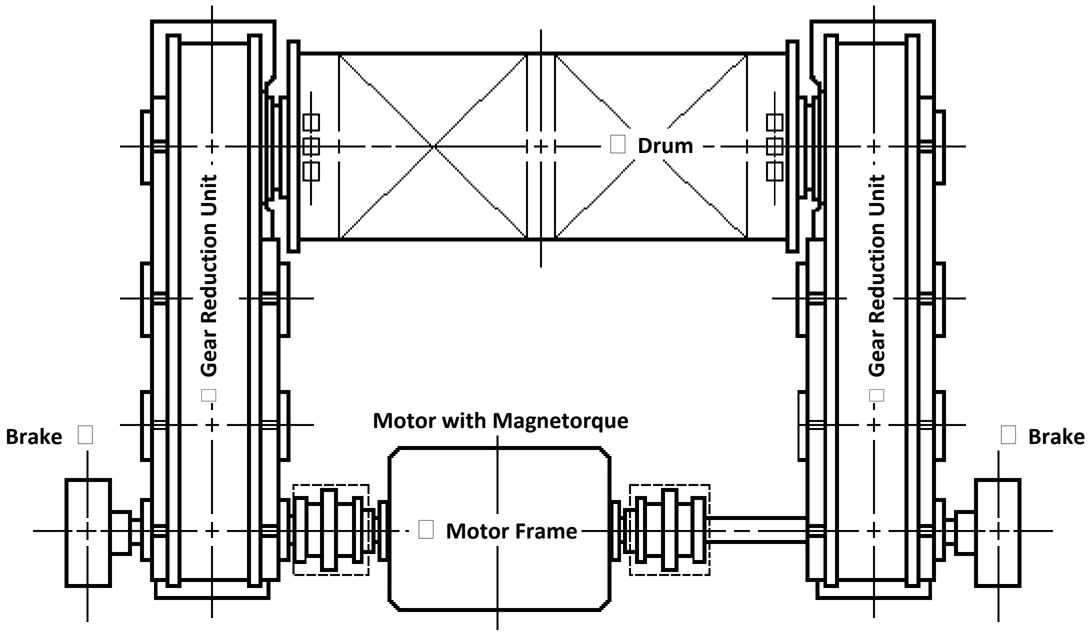

Figure 2: Closed Kinematic Loop (NOG–1, Figure 5416.1–2)

Another example of defense–in–depth is the rope configuration. NOG–1–2004 has very specific design criteria including that each rope is required to have a maximum load that is <10% of the breaking strength of the actual rope and additional quantifiable maximum allowable loading after a rope failure. In one reeving scenario, four ropes are used to lift the load. The ropes are specially engineered to maximize the breaking strength while minimizing the amount of drum length required for the application. The reeving is designed as an 8 part system with four individual ropes (Figure 3). In this reeving pattern, one end of the rope is dead–ended on the drum, while the other end is dead–ended at the Upper Block. This conservative safety factor (safety margin) is based on evaluations by the nuclear industry, Navy Crane Center, and other crane groups showing the most common reason for loss of load at nuclear power plants was rope failure.

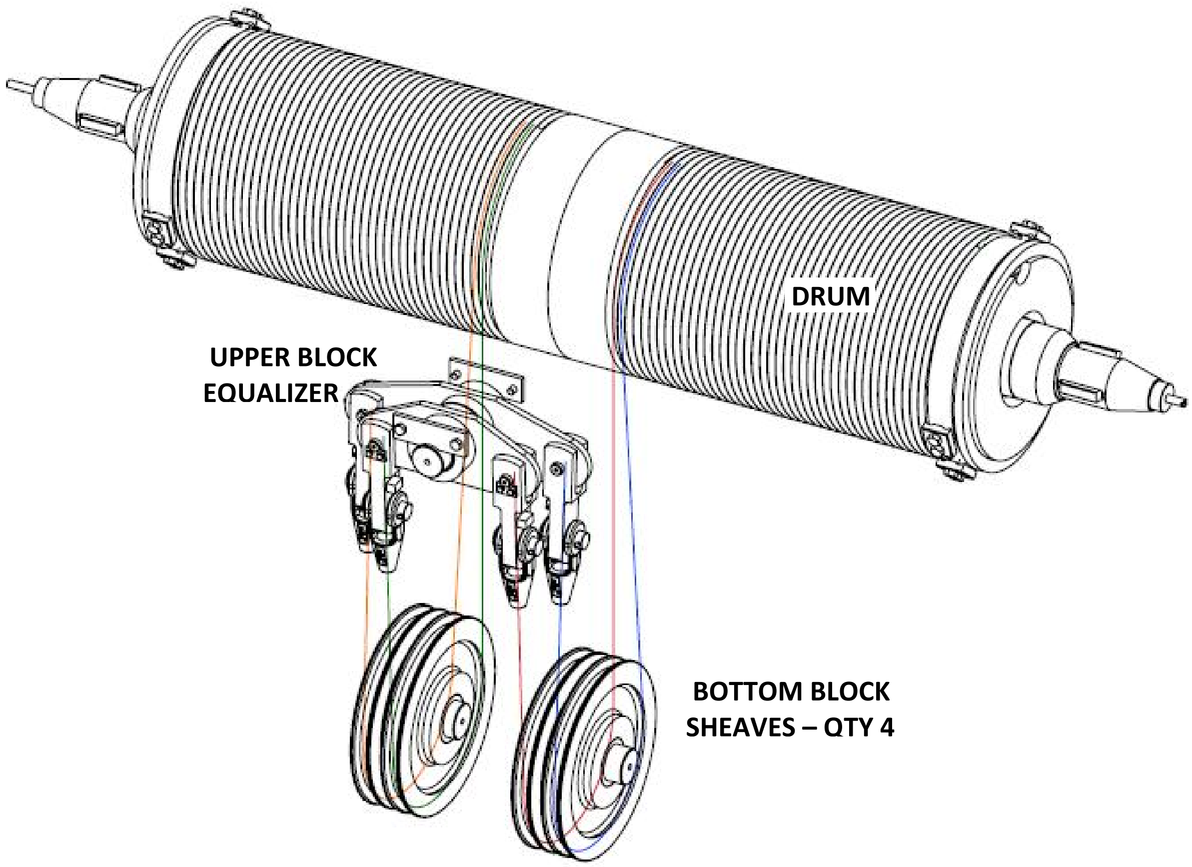

Figure 3 - Reeving Diagram

The Upper Block assembly (Figure 4) also has very specific design criteria since it is the epicenter of the mechanical safety system to prevent failure as a result of "Two Blocking" or an Overload (crane hook is snagged on something or the operator tries to pick up an object that is over the capacity of the crane.) Two Blocking is when the bottom block is run up past the upper limits and contacts the trolley frame, instantly causing the ropes to go into tension and reach ultimate failure. The first part of the Upper Block is the equalizer, which provides necessary play in the system to account for unbalanced ropes in normal operation. Each rope has an open spelter socket attached to one end of the rope, and is in turn pinned to the equalizer using a special load weighing pin (electronic load weighing) that individually monitors the actual load on each rope. The equalizer is then mounted to the torque arm along with two hydraulic cylinders which provide the energy absorption mechanism for a rope over–load condition. The hydraulic system is a passive system, so even during the loss of power the mechanism is ready to protect the load.

When an overload occurs, it pulls the torque arm toward the overload and simultaneously compresses the hydraulic cylinders. The action of the cylinders retracting gives the electrical control system time to react to the emergency situation and shut down the hoisting system. Even in this control circuit, there are many systems in parallel performing the same function to shut down the hoisting motion (defense–in–depth.)

Figure 4 – Energy Absorbing Block for Reeving System

- Load Cell – The electronic load cells are in series with each individual rope and sense the tension. At a preset limit, the load cell removes power from the motor drives which engages the shoe brakes (trips the safety circuit.)
- Torque Arm position switches – The torque arm (Figure 4) lowers as the cylinders compress. This action trips mechanical limit switches and actuates the safety circuit.
- Hydraulic Pressure switch in cylinders – As the cylinders compress, the fluid flows out of the cylinder via a relief valve so pressure is maintained for controlled movement. A pressure switch is directly connected to the cylinder to detect the high pressure and trips the safety circuit.
- Limit switches in cylinders – Connected to the rod of the cylinders are limit switches that detect abnormal stroking of the cylinder which subsequently trips the safety circuit.
- Drive mis–match parameters – The hoist electronic drives (such as a Variable Frequency Drive or DC drive) have features to fault out the drive if there is an inconsistency between the hoist motor encoder and the position of the operator input.
- Drive over–current – The electronic drives of the hoist will detect an over current condition and trip the safety circuit when the motor is trying to work against an overload or a two blocking condition.

In addition to tripping the electrical safety circuit which removes power from the drives and sets the brakes, another safety feature that can be implemented includes an Eddy Current Brake or Magnatorque. The basic function of a Magnetorque load brake is to provide a variable retarding torque for control purposes and is often incorporated with an alternator for emergency backup lowering. The Magnetorque load brake consists of a stationary field assembly (stator) and a simple, low–carbon steel rotor. The rotor is keyed to a shaft that is mounted between two anti–friction bearings. A fan, bolted to the rotor, draws air through the unit to remove heat from the rotor and field assembly. The rotor design allows for thermal expansion without outward expansion. This permits a small air gap to be maintained between the rotor and the stator, which is essential for fast response to control current changes.

The Magnetorque Load brake works as follows; braking torque is produced by electromagnetic fields. The electromagnetic fields are created by a D.C. control current flowing through the field coil. The coil is encased in a toroidal pole assembly, which is arranged to form a large number of alternate north and south magnetic poles. The large number of magnetic poles produces a strong magnetic field from a relatively low level current. As the rotor revolves through the electromagnetic field, eddy currents are induced in the rotor. The magnetic fields produced by these eddy currents oppose the magnetic fields induced in the field coil. Thus, a resistance to rotation is created. The strength of the electromagnetic fields, and, therefore, the amount of resistance to rotor rotation, is dependent upon the level of the D.C. current applied to the field coil. Therefore, the amount of retarding torque can be varied simply by varying the level of control current.

A permanent–magnet alternator is installed on many Magnatorques for emergency lowering. Its function is to provide excitation current in the event of simultaneous power loss and motor brake failure. This rectified D.C. excitation current will enable the Magnetorque brake to retard the lowering speed of a load that was off the floor at the time of a power loss. Normally the holding brakes will hold the load during an emergency and the Magnatorque will be in stand–by. When the operator is ready to lower the load during a loss–of–power scenario, he will slowly release the holding brakes (one–at–a–time) and allow the Magnatorque to safely lower the load at a controlled speed. This is much safer than the old method of "feathering" the brakes, which was prone to causing overheating and resulting in a run–away load.

Single failure proof criterion also dictates the crane shall hold the load before, during and after a seismic event, often called an Operational Basis Earthquake (OBE) and a Safe Shutdown Earthquake (SSE). Since the crane's bridge is a thin plate structure, a 3D shell model is generated of the bridge from the mid surfaces of the plates which produces an accurate and efficient finite element model to analyze. Depending on the complexity of the trolley, either a shell or a solid model of the trolley is created and connect to the bridge girders by using constraint equations adhering to the boundary condition requirements in ASME NOG–1–2004. The weight of the miscellaneous items such as motors, electrical enclosures, couplings, wheels etc are also loaded in the crane model as mass elements in order to provide the exact total weight of the structure.

Figure 5 shows a 125 ton fuel handling crane built for a Pressured Water Reactor in which the trolley is made of solid elements and the bridge of shell elements.

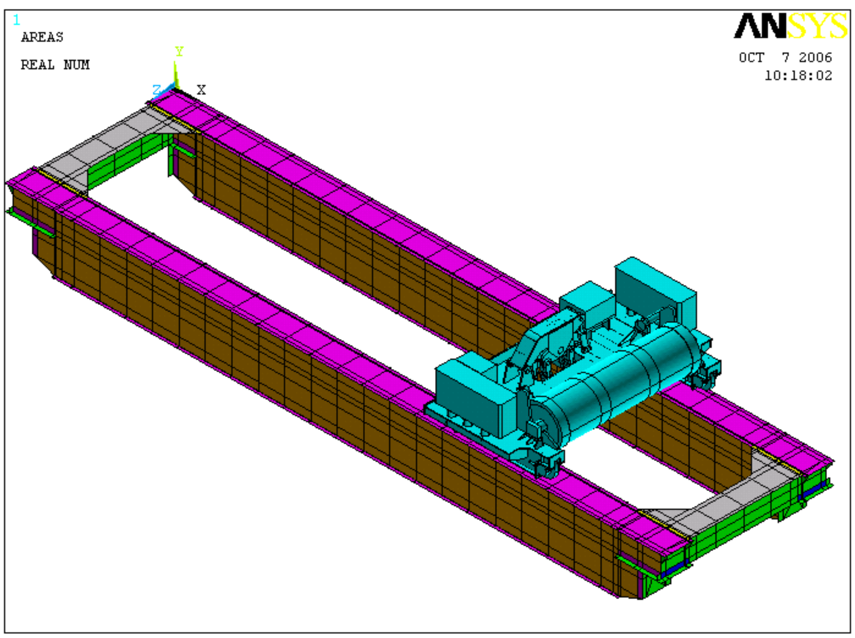

Figure 5: 125 ton Fuel Handling Crane Model

A standard methodology uses response spectrum method for seismic analysis and calculates the crane's response for three different trolley positions with and without having maximum capacity load on the hook. Prior to the spectrum analysis, a Block Lanczos Method modal analysis is performed up to 33 Hz frequency which provides mass participation ratio around 90% for most overhead cranes. Modal analysis is a technique to calculate the crane's natural frequencies, mode shapes and mode participation ratios. To compensate for the amount of the mass that is not participated in the modal solution, an additional analysis is performed to incorporate this mass. Missing mass calculations are performed separately for all three earthquake directions to determine the additional load per U.S. Nuclear Regulatory Guide1.92 (Reference 5). The missing mass method constitutes the total effect of all the system mass that is not included in the modes with frequencies above 33 Hertz. The system response to the missing mass is calculated by performing a static analysis for an applied load that equals the missing mass multiplied by the highest gravitational acceleration value (ZPA) above 33 Hertz in the provided spectrums and then the additional loads are added individually to the previous loading that came from the initial spectrum analysis. In parallel with this effort, it would be expected to examine all the cranes mode shapes and determine the dominant crane frequencies with respect to the mode coefficients to calculate the equivalent g loads for all directions. This can then be used to calculate the effect of the seismic event on the mechanical components.

When the modal solution is complete, a spectrum analysis is performed which computes the response of the structure to a given spectrum at each natural frequency. Grouping or Square–Root–Sum–of–the–Squares (SRSS) method is commonly used for mode combination depending on the existence of closely spaced modes. Two consecutive modes are defined as closely spaced if their natural frequencies differ from each other by 10% or less of the lower frequency. If closely spaced modes are present, the standard is to use grouping method; if closely spaced modes are not present, the standard is to use SRSS method for mode combination per ASME NOG–1–2004 requirement. Once the responses of the crane for each direction are determined, the total seismic response is calculated by the SRSS method and added to the absolute value of the static results. Note that sometimes different methods are used to combine loads based on the original design basis of the plant and their commitments to the regulator.

NOG–1–2004 sets high safety margins for design of various structural and mechanical components of the crane in the normal (operational) and seismic (extreme environmental) condition. A safety factor of 2 is used between the normal stresses and yield strength considering an additional 15% impact factor for normal operating conditions for structural components plus additional margins for degradation or single failure (critical) components. Crane designs are developed to meet the NOG–1–2004 requirements for seismic conditions which ensure that the combined stresses during the seismic event stay below 90 % of the yield strength of the material.

Seismic application methodologies are well defined in NOG–1–2004 which provides an excellent guidance for performing high tech dynamic applications for seismic evaluations. As the frequency behavior of the structure is essential for the seismic response, criteria for evaluation of the crane decoupled from the building are evaluated. Evaluation of different trolley and hoist configurations are also defined for analysis which covers all possible earthquake scenarios and generates high reliability for seismic applications.

Dynamic behavior of the building and crane are dependent on each other when the mass and stiffness properties meet NOG–1–2004 criteria. In Figure 6A, a coupled model of a crane and fuel handling building is analyzed to ensure the safety of the crane in extreme environmental conditions. Linear spectrum analysis is carried out for a number of bridge positions on the building runway along with different trolley and hook positions to thoroughly examine crane safety. A non–linear time history analysis is also carried out to evaluate the slack rope condition to understand the magnitude of the impact loads on the bridge girders.

Secondary restraints are advocated by NOG–1–2004 to take the horizontal and vertical loads resulting from extreme environmental conditions to ensure that the crane stays on the runway girder. This results in high load transfer from the crane to the runway girder. Figure 6B shows a detailed model of runway girders used in different plants to perform a local stress check resulting from crane wheel loads during a seismic event.

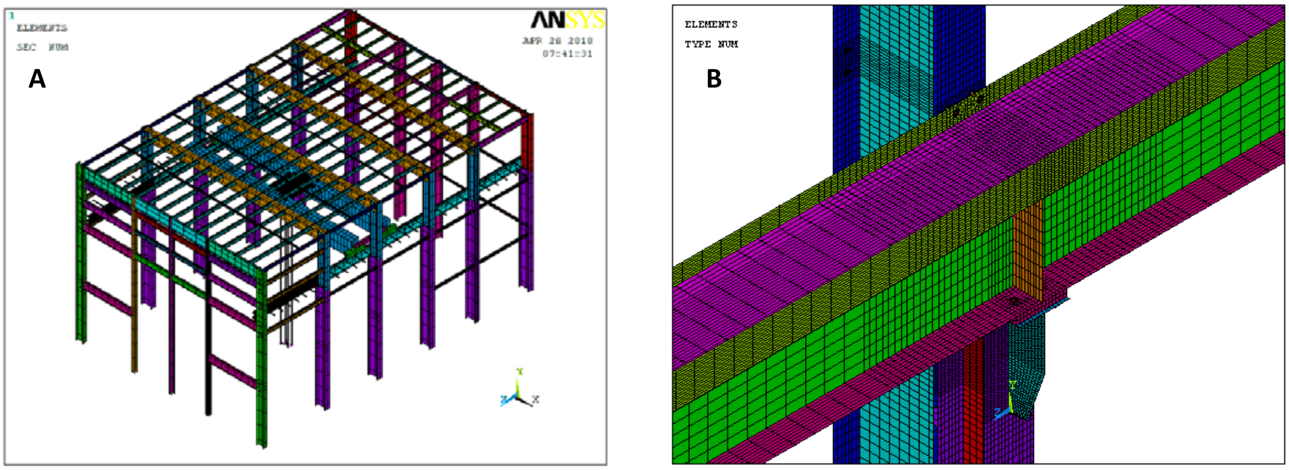

Figure 6 – Bridge runway analysis

The seismic analysis provides a picture of the stresses and the amount of movement of the crane (uplift or lateral movement) during the seismic event. To maintain the crane on the girder and runway rails, both the bridge and trolley may be fitted with Seismic Restraints. The restraints are mounted to ensure they do not interfere with the normal operation of the crane, but will restrict the movement of the crane during a seismic or abnormal event. Preferably, they are fitted to interact with the structure supporting the rail giving approximately 1” of space to allow the crane to operate normally, which results in approximately 1/2” spacing between when the flange interfaces with the rail and before the restraints engage in the lateral plane of motion. Per NUREG–0554, “*If a seismic event comparable to a safe shutdown earthquake (SSE) occurs, the bridge should remain on the runway with brakes applied, and the trolley should remain on the crane girders with brakes applied.*” NOG–1–2004, ¶4153.6, Boundary Conditions at Trolley and Runway Rails mimics the same words with “*The crane bridge (including gantry legs, if applicable) and trolley shall be provided with devices so that they remain on their respective runways during and after a seismic event.*” The crane does not need to operate after a SSE, but does need to remain in place and maintain control of the load.

During a seismic event, the trolley and bridge may move laterally and the flanges of the wheels will engage their respective rail head before the seismic restraints are contacted. The wheel is a heat treated forged steel component and will withstand a significant impact load to help maintain the crane on the rail. The forged rail is secured per the requirements of CMAA–70 (Reference 6) with closely spaced clips to help maintain the rail’s integrity to side loading during normal and abnormal events. However, the standard methodology of defense–in–depth as prescribed by both NUREG 0554 and NOG–1–2004 is applied by utilizing back–ups to ensure the bridge and trolley remain on their respective runways during and after a seismic event. If the wheel flange, rail or rail clips yield, the seismic restraints will interface with an entirely different structural component to maintain the crane on the runway.

If the seismic restraints contact their respective runway structure, the crane will still be on the rails since there is only approximately 1/2” of margin between the flange contacting the rail and the restraints contacting the structure (runway for bridge, girder top plate for trolley.) The crane will still be resting on the rail since the wheel flange width is approximately 1/2”. Additionally, the end truck can sit on the rail since the end truck steel structure is designed to prevent the crane from dropping more than 1” due to a wheel shaft break. The restraints will maintain the crane’s position within a small envelope and hence on top of the rail, even if the rail is deformed.

Studying the behavior of yielding of wheel flange or rail is a highly non–linear dynamic problem along with the probabilistic nature of variables involved in the system including machinery tolerance in the shafts, gears, motors, bearings, and wheel sliding. Therefore the methodology used by many users and authorized by NOG–1–2004 is to allow a linear analysis, where “*The restraint devices shall be considered to be in contact with the resisting structure in establishing boundary conditions used in the analysis for the crane*.” (NOG–1–2004 ¶4153.6). This is a conservative approach taken to address a probabilistic non–linear dynamic problem in a simple way using the limiting value of loads and gaps present in the system with a minimal number of components taking the entire load.

The use of the seismic restraints for the crane provides a redundant and conservative methodology for ensuring the crane meets the requirements of NUREG 0554 by maintaining the crane components on the runway. With a defense–in–depth strategy, the crane seismic restraints provide a simple approach that does not necessitate future maintenance or adjustments to ensure the plant maintains control of a load during an abnormal event.

All the analysis, design and special features are vitally important to the safe handling of spent nuclear fuel, but do very little good if the materials are incorrect or not in compliance with what are specified by the engineers. NUREG 0554 requires a “quality assurance program to be established to the extent necessary to verify the design, fabrication, installation, testing, and operation of crane handling systems for safe handling of critical loads.” The NUREG continues on instructing the crane manufacturer to have a quality assurance program consistent with the pertinent provisions of Regulatory Guide 1.28, "Quality Assurance Program Requirements (Design and Construction)" to ensure the site assembly, installation, and testing of the crane meets the design basis and standards. NOG–1, Section 7000 provides further detailed instructions on what should be tested, inspected and the documentation that is expected.

VERTICAL CASK TRANSPORTER

The principles of defense–in–depth for the safety systems discussed for the SFP Overhead Crane are mirrored in the next piece of equipment handling the spent fuel cask. However, unlike the overhead crane, the VCT has relatively few industry standards to guide its design, manufacturing and testing. The most frequently specified reference is ANSI N14.6, "Special Lifting Devices for Shipping Containers Weighing 10,000 pounds (4500 kg) or More" (Reference 7), which was published in 1993, and has not been revised since. In combination with this standard, the family of ASME B30 standards is frequently used to provide further guidance; specifically ASME BTH (Reference 8) to supply additional analytical methodology that N14.6 is either missing or whose ambiguity requires clarification.

ANSI N14.6 subscribes to the Single Failure Proof methodology and therefore the VCT has been evaluated to ensure its critical load is safe during all postulated failures of a component and other environmental conditions such as seismic events, tornados, hurricanes or mis–operations. One of the most important components critical to the mission of the VCT is the cross beam assembly, (horizontal load beams that connect the cask to the VCT). This welded structure is designed to N14.6 including material testing, non–destructive testing of all welds, and extensive load testing. The beam is equipped with hydraulic sliders that move horizontally to allow for a variety of different diameter casks to be handled (Figure 7.) The cross beam is vertically raised and lowered by a pair of welded structural towers also designed to ANSI N14.6.

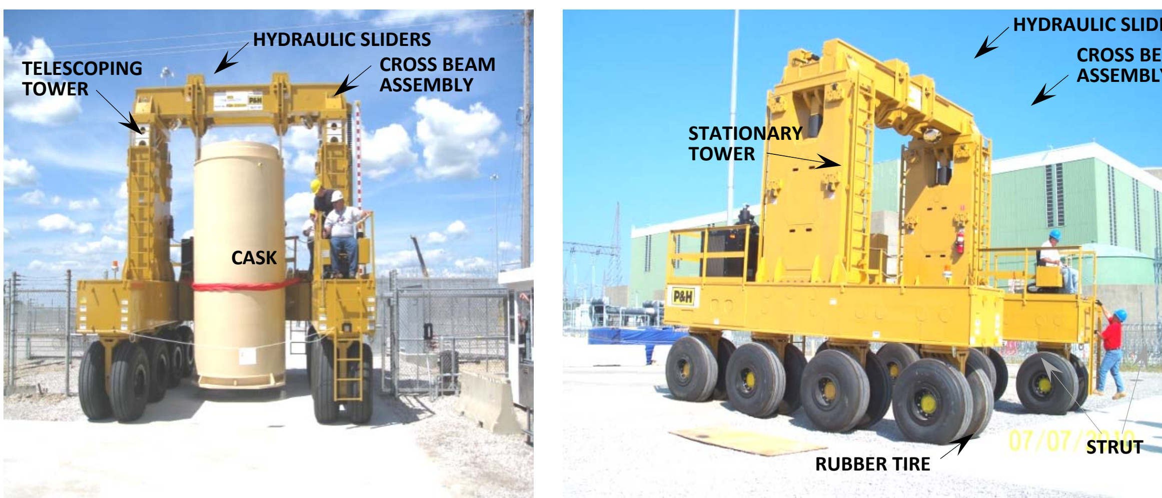

Figure 7 – Rubber Wheeled Vertical Cask Transporter (VCT)

Inside each tower are dual hydraulic cylinders and a safety catcher. The system is specifically designed to safely hold the load in position upon loss of electrical power or hydraulic flow using simple counterbalance valves and other commercial hydraulic components. On loss of pressure, the commercial safety catcher will hold the load. The VCT employs two safety catchers (Figure 8, one in each tower) for redundant load drop protection. The safety catcher rod (1) is surrounded by the housing (2) in which several wedged clamping jaws (3), each with one slide lining (4) and one brake lining (5), are assembled. When pressure (p) is applied to the plungers (8), the clamping jaws are held in a raised position so that the rod can move freely. The springs (6) are compressed in this position. The safety catcher becomes effective as soon as pressure is released from the plungers. The action of the springs causes the clamping jaws to clamp the rod firmly, thus securing the load. The clamping force, however, is not built up until the rod has been moved by the load. Due to the self–intensifying static friction at the rod, the clamping jaws are drawn into the clamping position at their stops (7) after having moved the distance "e" (approximately 1/2".)

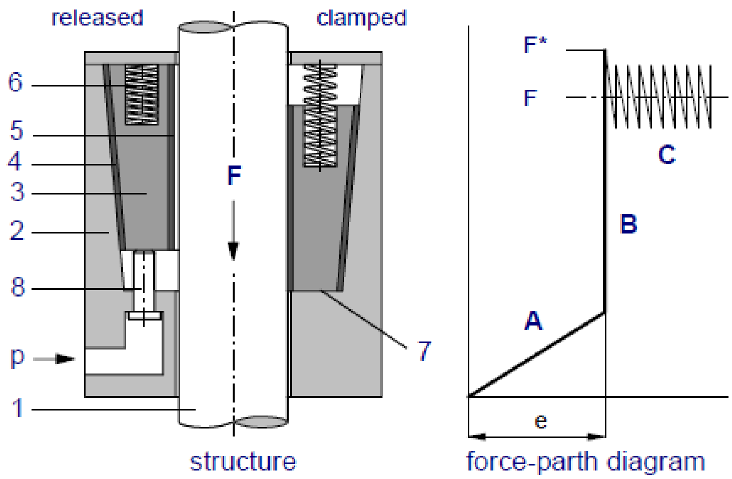

Figure 8 – Safety Catchers

Clamping is released by an upward movement of the rod through path "e", by applying pressure corresponding to the load being lifted. Thus the release operation is only possible if the pressure system is intact. Excess pressure (e.g. for breaking loose) is normally not required.

The lift towers are designed and built as described to provide sufficient lifting capability and redundant drop protection. The most likely failure mechanism is a loss of hydraulic fluid pressure either because of a break in the system or failure of the pumping system. In either case each tower has independent drop protection and would lock in position because of the loss of hydraulic pressure. In addition, each side is equipped with a manual lowering hydraulic circuit so the operator can lower the cask (towers) safely to the ground in case of loss of power.

The new style of VCT uses rubber tires versus the old track–style system which are frequently used on bull dozers. The rubber tire system provides numerous advantages over its predecessor including significantly improved maneuverability, less ground force being applied to the roadway, and no damage to concrete or asphalt haul paths. The total load (cask weight plus dead weight of VCT) is distributed through 4 sets of dual tires on each side of the VCT (total of 16 tires.) The tires are foam filled to prevent flat–spotting, loss of pressure and blowouts. The foam filling gives the tire additional stability for lower internal pressures and better ground pressures preventing travel surface damage. The completely foam filled tire eliminates the concerns for blow outs (loss of air pressure) which are related to cord/belting damage. In the event there were to be damage to the rubber, the tire would fail in a safe manner (fail–safe) without any affect to the load.

The VCT is used in a ground application where the load capacity is the driving factor. The tires are commercial aircraft tires used on very large planes. The tires are conservatively used in this application since the VCT does not apply the same dynamic conditions that a tire on a large aircraft would see during take-off and especially landing. In addition, the VCT is designed to travel at 0.4 MPH and thus the tires are not subjected to high rotational speeds and the corresponding centrifugal forces seen on airplanes.

The tires also play an important part in the seismic and structural analysis of the VCT (Figure 9.) The unique combination of rubber tires (modulus of elasticity) and low center of gravity make the VCT a very stable platform. Even at the highest cask lift heights and largest grade slope (5% slope with maximum load), the tip over analysis coupled with the seismic (structural) analysis has shown that the VCT remains stable even during the accidents analyzed (i.e., tornado missiles, flood water velocity, and seismic activity).

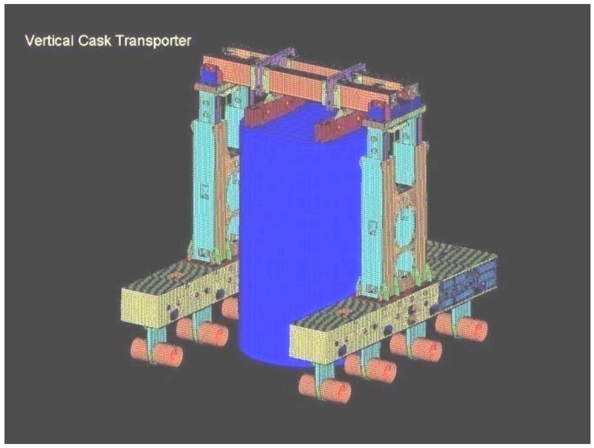

Figure 9 – Seismic Analysis of VCT

The VCT design basis requires a 1.1 factor for stability during a worst case scenario. The seismic analysis of the VCT is performed with the cask at the highest vertical height and frequencies are determined for the device. The finite element solution showed the dominant frequency of the FE model where X direction is 1.437 Hz, and Z direction is 1.438 Hz; in other words, a very low frequency and thus stable piece of equipment. When the forces with respect to the center of gravity are evaluated, the calculated stability factor is approximately 2.35; significantly better than the required 1.1 factor.

Additional safety features on the VCT include:

- Low Fire Load – Quintolubricant 888 is used as the hydraulic fluid, which has a flash point of 275°C (527°F). Coupled with the limit of < 189 liters (50 gallons) of diesel oil, flame resistant wiring and hosing, there is insufficient burnable material to cause any damage to the spent fuel.
- Operator controls have fail safe (dead man's) switches on the controls and on the seat, so if the operator stands up or lets go of the controls, the VCT shuts down.
- A secondary emergency shut off switch on a tether allows a back–up person to shut down the VCT if there are problems.
- Testing has shown the VCT retains its functionality if one of the 8 struts becomes disabled or if up to two tires become disabled.

Safely moving spent fuel at nuclear power plants is the top priority for the plant, personnel, regulator, and the general public. The two major pieces of equipment that handle the cask full of spent fuel have been carefully evaluated with regard to design, manufacture, testing and operation to <u>attempt to anticipate</u> all possible casualties and potential problems. Since it is impossible to foresee the future and anticipate all possible scenarios, the equipment has been designed based on a "defense–in–depth" with multiple backup safety features couple with significantly enhanced design safety factors (margins). Following these industry guidelines, spent nuclear fuel has been transported with these and similar devices event–free for a number of years. Simultaneously there are multiple efforts in progress to continue to enhance the safety of the equipment and improve the governing standards.

REFERENCES

1. NUREG 0612, “Control of Heavy Loads at Nuclear Power Plants”, published by Nuclear Regulatory Commission, July 1980.
2. NUREG 0554, “Single–Failure–Proof Cranes for Nuclear Power Plants”, published by Nuclear Regulatory Commission, May 1979.
3. ASME NOG–1–2004, “Rules for Construction of Overhead and Gantry Cranes (Top Running Bridge, Multiple Girder,” published by The American Society of Mechanical Engineers, Issued May 16, 2005.
4. NRC Regulatory Issue Summary 2005–25, Supplement 1, Clarification of NRC Guidelines for Control of Heavy Loads, published by Nuclear Regulatory Commission, May 29, 2007.
5. Nuclear Regulatory Guide 1.92, Combining Modal Responses and Spatial Components in Seismic Response Analysis, published by the Nuclear Regulatory Commission, July 2006, Rev. 2
6. CMAA–70–2004, Specifications for Top Running Bridge and Gantry Type Multiple Girder Electric Overhead Traveling Cranes, Crane Manufacturers Association of America, Revised 2004.
7. ANSI N14.6–1993, Special Lifting Devices for Shipping Containers Weighing 10,000 pounds (4500 kg) or More, published by the American National Standards Institute, Inc., June 28, 1993.
8. ASME BTH–1–2008, Design of Below–the–Hook Lifting Devices,” published by The American Society of Mechanical Engineers, Issued January 9, 2009.