SEISMIC ANALYSIS ON MODIFIED SERVICE WATER PUMPS

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ABSTRACT

At one nuclear power plant, the original essential service water pumps were replaced by another set of pumps with higher capacity. The latter were transferred from another "idle" nuclear power plant. Due to some important differences between the civil construction and because of seismic spectra being different as well, the pumps required modification and, subsequently, requalification.

The structural and seismic qualification was based on a three-dimensional finite element model. Simulations showed that the replacement pumps would suffer from some structural resonances. Several pump components and the interconnections between them were fortified so as to move the potentially hazardous natural frequencies.

After having applied the proposed adaptations, the calculated dynamic behavior was checked by means of an experimental modal analysis and by field performance tests.

Finally, the replacement pumps including all structural modifications were implemented onsite during the refueling of the plant.

CAPACITY INCREASE OF POWER PLANT

At the nuclear power plant in Cofrentes, Spain, there is an ongoing effort for increasing efficiency. This power plant was originally designed in the 70s and was operated from 1984 on, for a rated output power of 974 MW. An improvement program aims to increase the electrical output to 1080 MW. Additionally, the program continuously evaluates and improves the components of the plant. It also aims to comply with the evolving requirements imposed by the exploitation permit.

Among many others, some important modifications were:

- Installation of a continuous vibration monitoring system on the principal rotating equipment.
- Replacement of the low-pressure rotors of the turbines. Also the auxiliary feedwater turbines were replaced.
- Replacements/modifications of many pumps.

Instead of replacing the existing equipment by completely new machines, it was often more economical to use as much as possible the existing equipment of the nuclear power plant in Valdecaballeros, Spain. The construction of the latter plant started in 1978, but was never finished because of a moratorium on nuclear power.

The plant in Cofrentes remained on the grid while the replacement equipment was requalified and, eventually, adapted. Replacement typically occured during the fuel stop of the plant.

ESSENTIAL SERVICE WATER PUMPS

In March 2002, the installed essential service water pumps, with a capacity of 9000 gpm (980 rpm, head of 120 ft), had to be replaced by the ones from Valdecaballeros, rated for 9426 gpm (980 rpm, head of 160 ft). The pumps in both power plants were designed and classified according to *ASME Boiler & Pressure Vessel Code* (1995), Section III, Class 3, Seismic Category 1.

Due to some differences between the civil construction of the Cofrentes plant and the Valdecaballeros plant, it turned out to be necessary to apply some major modifications to these replacement pumps (Figure 1):

- Installation of a lighter electrical motor
- Redesign of baseplate
- Replacement of the existing spool piece (defined as the part that links the discharge head of the pump with the motor stool) by a shorter one
- High specified nozzle loads for the Valdecaballeros plant replaced considerably lower nozzle loads at existing pumps in the Cofrentes plant.
- Further on, in Valdecaballeros, the existing pumps were installed with three seismic restraints. As these are difficult to access (they need to be removed if the pump must be pulled), it would be an improvement not to have to install these supports in Cofrentes.
- Additionally, seismic ground response spectra in Cofrentes are different from those in Valdecaballeros.

Taking into account all modifications, it was impossible to consider the replacement pumps as duplicates and it was necessary to submit them to a new seismic analysis. Additionally, the structural integrity of the pump structure under various static and additional dynamic load conditions was reassessed.

THEORETICAL ASSESSMENT OF DYNAMIC AND STATIC BEHAVIOR

Concept

The assessment of the integrity of the pump structure was based on a mixture of well established and widely accepted standards and codes, and a modern calculation method.

The dynamic behavior was evaluated according to *ASME Boiler & Pressure Vessel Code* (1995), Section III, Division 1, Appendix N (seismic analysis) and Appendix I (fatigue evaluation). In a first step, however, the natural frequencies and the mode shapes were calculated so as to depict and avoid vibration problems during normal operation of the pump.

The calculations were performed by a parametrical three-dimensional finite element model (Figure 2); the lumped mass-beam model in the existing engineering documents of the original pumps was not adapted.

The implementation of such a model might appear time-consuming, but it offers some important advantages:

- The prediction of the effect of small individual components (for instance: bolts, fortification ribs, even flanges) and of geometrically complicated components (for instance: the part of the discharge column with the 90 degree elbow) is more accurate.
- The sensitivity of the structure with respect to relatively small adaptations can be studied without expensive testing/prototyping/reworking. The parametrical aspect still allows application of "quick" changes to the model.

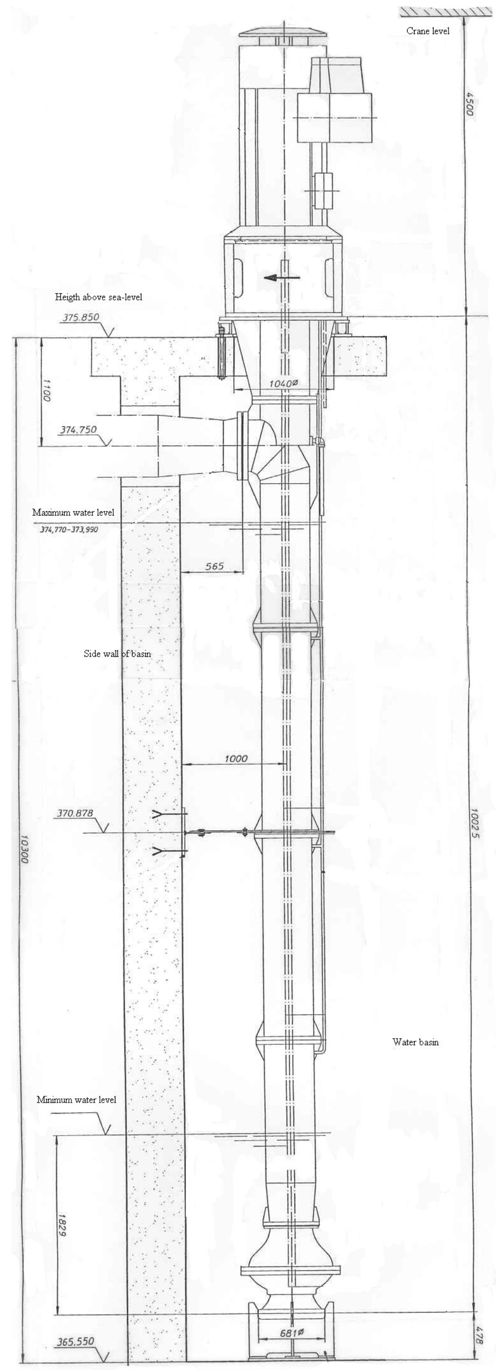

Figure 1. Layout of Modified Essential Service Water Pump (Final Design).

- The same model can be used for the evaluation of the static behavior of the pump. Local stress raisers can be identified, on condition that these are modeled. The model then could replace the pressure boundary calculations according to the formulas of *ASME Boiler & Pressure Vessel Code* (1995), Section III, Division 1, Subsection ND-3400.
- Calculated stresses are not averaged values anymore as they account for details. Consequently, allowable stresses can be set higher.

In a last step, the stress levels resulting from static and dynamic calculations were combined into one set of stress intensity values. These were evaluated according to *ASME Boiler & Pressure Vessel Code* (1995), Section VIII, Division 2, Appendix 4.

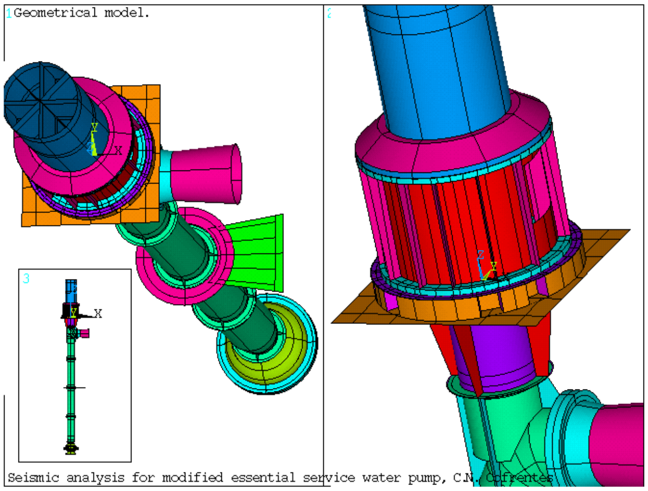

Figure 2. Geometry of Total Pump Structure. (Model consists of, from bottom to top: suction bowl, series of three columns, discharge head (90 degrees), spool piece, fortified baseplate, fortified motor stool, and motor. Inclusive: intermediate and fixed seismic support, convergent-divergent pipe between discharge nozzle of the pump and concrete wall of the basin.)

Despite being more sophisticated than a lumped mass-beam model, this three-dimensional finite element model remains an approximation of reality, which is subjected to a set of limitations:

- Most important is the assumption of linear behavior. This makes calculations relatively straightforward, at the cost of reduced accuracy:
  - Bolts are not modeled into detail. The effect of pretensioning on stress levels in bolts and flanges is not directly accounted for; neither are the friction effects between flanges. The contact areas between flanges then still must be estimated. For most calculations it was assumed that the flanges are tied together at the cross-sections of the bolts only.
  - Eventual small gaps between pump structure and seismic restraints are not modeled. Instead, depending upon the type of calculation, the restraint is either attached to the pump structure or not. Possible impacts of the structure into the seismic restraint cannot be calculated entirely correctly.
  - As this is a vertical pump, the fluid bearings are lightly loaded. Such load condition will give rise to very high orbits of the rotor and might violate some assumptions behind the calculation of bearing coefficients.
  - All materials behave elastically.
- The effect of the water in the discharge column was accounted for; however, the effect of the water in the basin was not implemented.
- The model is a combination of solid elements (some flanges), shell elements (piping, remaining flanges), and beam elements (rotor), and there is also a compromise between computer resources and accuracy.

Estimate of Natural Frequencies

A small model of the electrical motor was generated first. Reed frequencies and mode shapes were calculated and tuned to the results produced by the manufacturer of the motor.

Natural frequencies and mode shapes were then estimated for a "first proposal design" of the pump structure, without any seismic restraint along its discharge column.

The analysis revealed that there would be two sets of natural frequencies in the neighborhood of 1× rpm. Figures 3 and 4 show that one set is closely related with the flexibility of baseplate and motor stool with the heavy motor on top of it; the other set is directly related with the flexibility of the discharge column. Even though the pump could withstand a safe shutdown earthquake (SSE) seismic event, its reliability could be low due to 1× rpm vibrations.

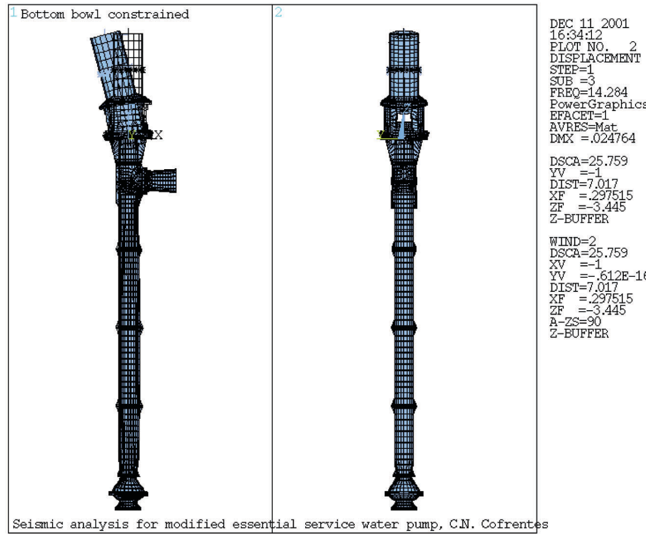

Figure 3. Initial Design of Modified Essential Service Water Pumps (A). Two Important Mode Shapes in Vicinity of 1× RPM (16.33 Hz).

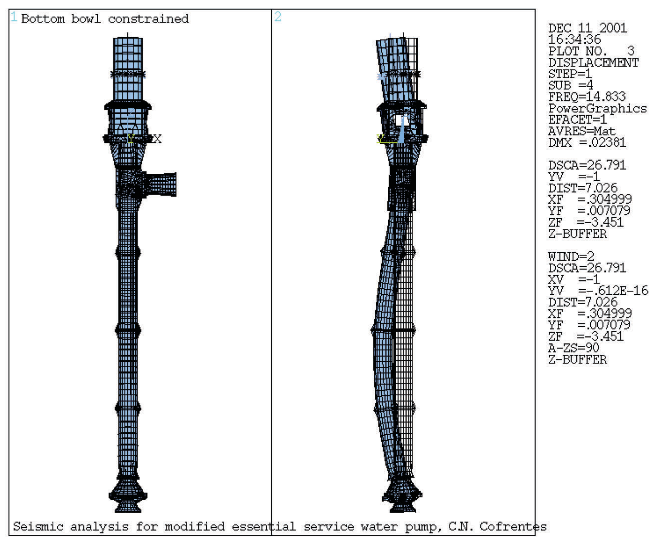

Figure 4. Initial Design of Modified Essential Service Water Pumps (B). Two Important Mode Shapes in Vicinity of 1× RPM (16.33 Hz).

Several simple modifications that would make the pump structure stiffer were then implemented in the finite element model (Table 1, Figure 5). It was possible to adapt the "first proposal design" to such an extent that both sets of natural frequencies in the vicinity of 1× rpm migrate out of the frequency zone 10 Hz to 20 Hz. Therefore:

- The baseplate construction must be fortified. There was some competition between a box-type baseplate construction (Figure 6) and a solid baseplate configuration. The first option was chosen because of shorter lead-time and ease of manufacturing.
- Many radial ribs were added to the motor stool; plate thicknesses of the stool were doubled (refer to Figure 2). The number of bolts between motor stool and baseplate/spool piece was doubled as well.
- A fixed intermediate seismic support, which is rigidly connected to the wall of the basin, was applied in the middle of the discharge column (refer to Figure 1).

Table 1. Effects of Stiffening Pump Structure. Rigid Plane Determined by Centerline of Rotor and Direction Perpendicular to Service Access Holes in Motor Stool. Flexible Plane Perpendicular to Rigid Plane.

	1st **Proposal**, suction bowl not constrained.	1st **Proposal**, suction bowl rigidly constrained	Extra intermediate constraint, fixed in middle of column. Bowl free.	Box-type base. Original motor stool + spool piece. Intermediate fixed constraint. Free suction bowl.	Box-type base & motor stool. No extra bolts in foundation. 16 bolts stool-spool piece. Intermediate constraint. Free bowl.	Box-type base & motor stool. No extra bolts in foundation. 24 bolts motor stool-spool piece. Fixed intermediate seismic support. Free bowl. **Final solution.**	Box-type base & motor stool. 4 extra bolts in foundation. 16 Bolts motor stool-spool piece. Intermediate seismic support. Free suction bowl.	Solid base & fortified motor stool, 16 bolts stool – spool piece. Intermediate support. Free bowl.	Solid base & fortified motor stool, 32 bolts stool-spool piece. Intermediate support. Free bowl.
Pendulum, flexible plane	2.3	-	5.47	5.47	5.47	5.40	5.47	5.46	5.46
Pendulum, rigid plane	2.6	-	5.59	5.59	5.59	5.60	5.59	5.58	5.58
Motor (+ pipe)	13.9	13.4	13.9	16.9	20.90	21.54	21.67	22.11	23.63
Motor	14.7	14.3	14.3	17.6	21.35	21.85	21.96	22.45	24.00
1st bending pipe	15.8	14.8	-	-	-	-	-	-	-
1st bending pipe	17.2	16.0	-	-	-	-	-	-	-

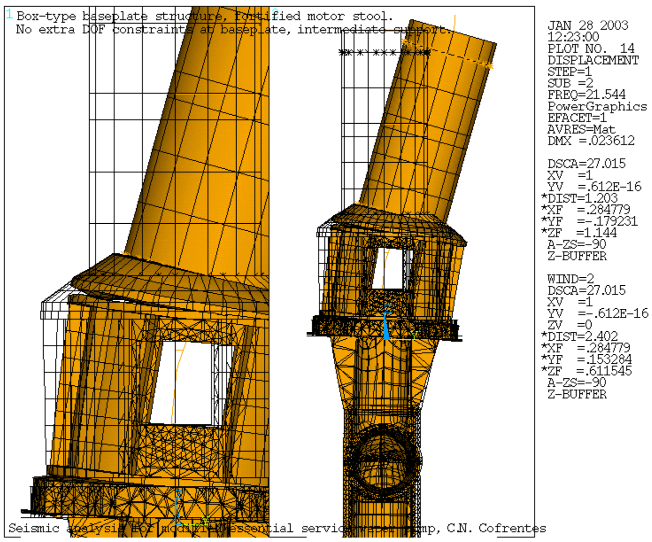

Figure 5. Bending of Motor and Fortified Motor Stool on Fortified (Box-Type) Baseplate. (Detailed view allows observation of how important part of flexibility is generated at flanges and at bolts.)

- No modifications were applied to the structure of the bare pump, so as to avoid additional welding and associated qualifications. Even installation of the intermediate seismic support did not require any welding, as it was already implemented at the pump in the original power plant.

For the most rigid pump configuration, first sets of natural frequencies now occur at 5.5 Hz (pendulum mode of suction bowl) and 23 Hz (bending of motor and motor stool. The frequency shifted more than 8 Hz with respect to the original design, for a total mass of approximately 24,643 lbm while in operation). The final design, however, also accounted for some economical considerations and was slightly more flexible: frequencies were 5.6 Hz and 21.5 Hz.

The dependency of the calculated natural frequencies with respect to foundation stiffness was assessed by means of a sensitivity study (Table 2). Values were extracted from Boswell and Sperry (1981). It is demonstrated that the natural frequencies remain sufficiently far away from 1× rpm, even if there were some flexibility of the concrete foundation.

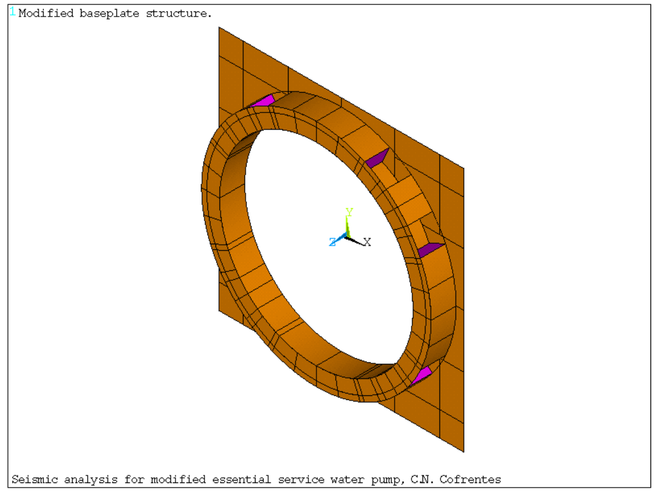

Figure 6. Detail of Fortified Baseplate Structure. Box-Type Structure. (Foundation bolts are accommodated by recesses in outer annular ring. Radial fortification ribs maintain box structure.)

Table 2. Sensitivity of Natural Frequencies of Second Most Rigid Pump Structure to Foundation Stiffness.

	INFINITELY STIFF	$K_{lateral}$ = 1.00e8 lbf/in $K_{torsional}$ = 2.241 e10 lbf.in/rad	$K_{lateral}$ = 0.500e8 lbf/in $K_{torsional}$ = 1.205 e10 lbf.in/rad
Bending of pipe, lower half, flexible plane	5.47	5.47	5.47
Bending of pipe, lower half, rigid plane.	5.59	5.59	5.59
Bending of motor on motor stool + baseplate Flexible plane.	22.11	21.03	20.21
Bending of motor on motor stool + baseplate. Rigid plane	22.45	21.37	20.52

Seismic Behavior

The seismic behavior of the pump was estimated based on:

- The calculated modal parameters of the final model, and
- The specified triaxial ground response spectra (Figure 7) at the connections between the pump and the external world.

All properties—velocities, accelerations and stress intensity levels—were calculated according to the square root of the sum of the squares (SRSS) method.

If the suction bowl were completely free, its displacements would become as high as 0.488 inch, mainly in the radial direction (Figure 8). Relative displacements between impeller and casing then would amount to 49 mils. In this case, the impeller would touch the pump casing, both during an SSE seismic event and at a 50 percent SSE event. Furthermore, important shear effects would occur at some bolts, particularly at those of the intermediate (fixed) seismic support.

A flow strainer was then added as a second (though not fixed to the pump) seismic restraint; the radial clearance between strainer and bowl was limited to 39 mils. Stresses caused by seismic events were then calculated for two cases (this is, in fact, a consequence of the linear approach of modeling):

- As if the suction bowl were fixed to the flow strainer
- As if the suction bowl were free from the flow strainer. This situation would occur during small seismic shocks, and during the first phase of a more serious seismic event.

As the amplitudes of the excitation spectra are low and as no natural frequencies occur in the range 10 to 20 Hz, maximum stress levels are small:

- Typically less than 1015 psi in the main parts of the structure

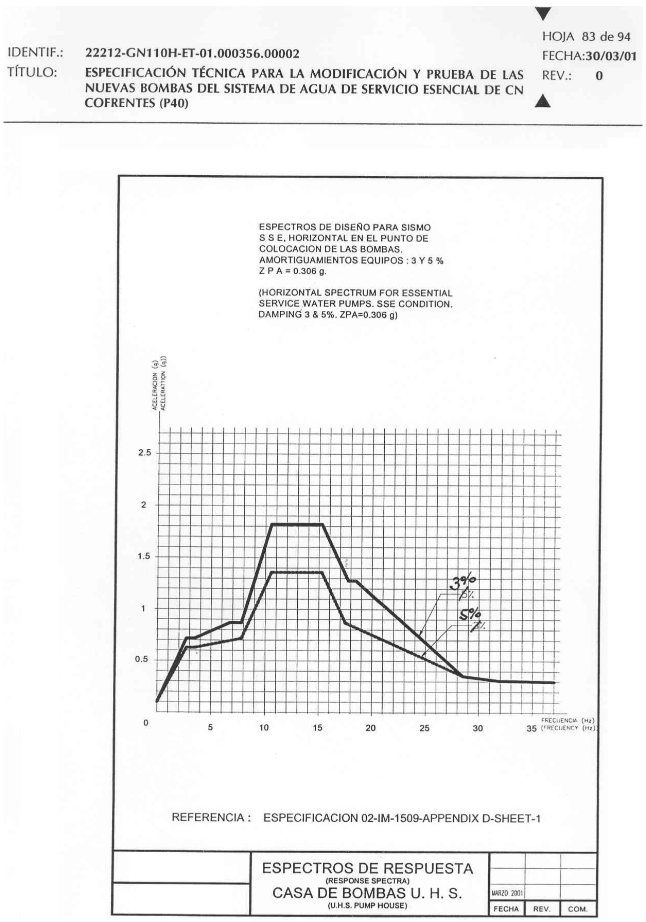

Figure 7. Seismic Ground Spectrum in Horizontal Direction.

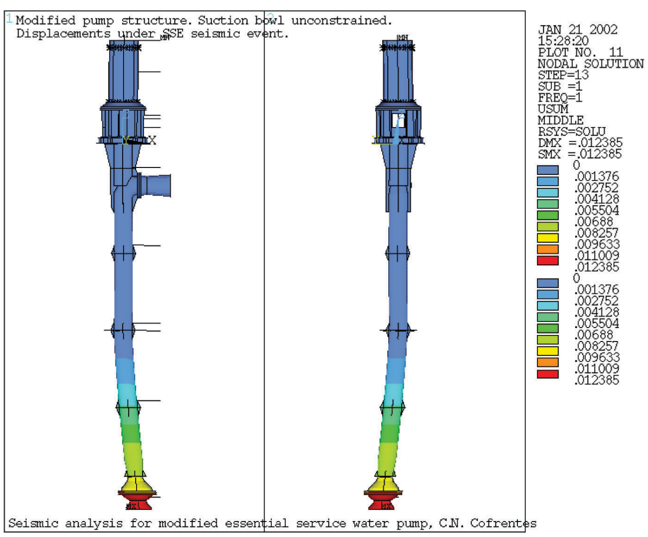

Figure 8. Displacements of Pump Structure under SSE Seismic Event, If Suction Bowl Were Completely Free.

- Maximally 9427 psi in the surroundings of the bolts that join baseplate and foundation

Static Calculations

The static calculations considered the combined effects of:

- Weight
- Motor torque—pump torque
- Drag force caused by wind
- Hydraulic thrust
- Internal pressures

and, most important, two sets of high nozzle loads. Values are listed in Table 3 and are much higher than for the pumps to be replaced. Forces and moments must be applied in two separate load cases and must not be combined (customer specification). Forces or moments are mainly caused by SSE movements of the discharge piping, which is clamped into the concrete wall of the basin (refer to Figure 1) at only 23.62 inches distance from the nozzle of the pump.

Table 3. Loads at Discharge Nozzle of Essential Service Water Pumps.

	Nozzle loads (20") under faulted plant conditions, replacement pump.	Nozzle loads (20") under faulted plant conditions, replacement pump.	Nozzle loads (20") under faulted plant conditions, pump to be replaced.	Nozzle loads (16") per API 610, 8th Edition.
Fx	11605 lbs	--	6500 lbs	1900 lbs
Fy	11605 lbs	--	1000 lbs	2300 lbs
Fz	11605 lbs	--	8500 lbs	1500 lbs
Mx	--	56180 lbs.ft	208 lbs.ft	5400 lbs.ft
My	--	56180 lbs.ft	83 lbs.ft	2700 lbs.ft
Mz	--	56180 lbs.ft	42 lbs.ft	4000 lbs.ft

Calculations with discharge piping detached from the pump nozzle and all static loads applied revealed that:

- Stress levels in the pump structure were maximal (typically 22 ksi) in some local areas of the discharge head and at the extreme edges of the fortification ribs of the spool piece.
- Important shear forces would occur at some bolts—particularly at those connecting the pump with the outer environment (foundation, bolts at discharge nozzle, connections between pump and baseplate structure). Depending on the pretension levels of the bolts, there is a risk of the complete pump structure moving relative to the basin (though stress levels in the bolts remain acceptable). The concrete wall of the basin is heavily loaded by the intermediate seismic constraint.
- Relative displacements between both ends of the discharge piping between the pump nozzle and the concrete wall of the basin would amount to 130 mils.
- There are important deformations of the discharge head/spool piece; the suction bowl touches the flow strainer.

After discussion with the customer, it was agreed that the discharge piping remained clamped within the concrete wall of the basin and that the same set of nozzle loads was maintained. Stresses (typically 6 ksi), forces, and displacements drop significantly as the concrete wall now absorbs most of the nozzle loads.

This simulation also demonstrates once more that such a vertical pump (and more generally any pump) should not be used as an anchor that solves all deformations of the piping systems.

EXPERIMENTAL VALIDATION OF CALCULATIONS

Concept of Experimental Validation

Though the control organism accepted the theoretical calculations proving the structural integrity of the pump, the operator of the power plant requested an additional experimental validation.

The ideal way for verifying the theoretical calculations would be to install the pump structure onsite and to artificially simulate both seismic conditions as high nozzle (plus other) loads while the pump is in operation (more specifically: when it is filled with water).

Vibrations and strains could then be measured all along the pump. This, however, was not practically feasible in the short time available:

- Simulation of seismic events would require developing a case-specific test rig, including a set of large exciters and substructure allowing the attachment of exciters to the baseplate of the pump and to the ground. Additional excitation could be required at the intersection between the discharge piping of the pump and the concrete wall of the basin, at the intermediate seismic support, and eventually at the flow strainer at the suction bowl.
- Such a test rig would likely interfere with the pump structure and introduce additional dynamics. The theoretical model of the pump must be extended so as to generate results that are comparable to the measurements.
- The excitation forces should be applied in such a way that they are representative of the seismic ground spectra in all directions (correct amplitude versus frequency), and that they fulfill the implicit assumptions behind the application of the SRSS method. This would imply a test program on its own.
- Finally, there were important practical constraints:
    - Part of the pump is submerged in water, imposing extra requirements of the exciters and of the instrumentation.
    - As the pump would be in operation, excitation levels should be sufficiently high so as to obtain correct signal-to-noise ratios.
    - Poor accessibility of the water basin
    - Civil works were going on at the basin, and the plant was being refueled.

Therefore, the physical simulation of a seismic event was replaced by a much simpler experimental modal analysis. The modal parameters (natural frequencies, mode shapes, and participation factors) determine completely the dynamic behavior of any linear(ized) structure, including its response to seismic events. Results of the experimental modal analysis are fully compatible with the theoretical analysis, as the calculated response of the pump structure to a seismic event is based on the calculated modal parameters.

Since the dynamic behavior and thus the modal parameters are determined by stiffness, damping, and inertia effects, a correct estimate of the dynamics of the pump structure implicitly involves a correct assessment of its static behavior as well.

The complete pump structure was assembled and tested at the works of the pump contractor (Figure 9). The pump structure under test was similar, though not identical to the pump onsite (the baseplate was not attached, there was no water inside or outside the pump, and there were different connections with the environment).

Figure 9. Test Pump Subjected to Experimental Modal Analysis.

During the test, the pump was laying on two rubber tires. The tires were supposed to act as soft springs that:

- Isolate the test structure from the vibrations in the shop.
- Disturb as little as possible the dynamic behavior of the pump, particularly in the horizontal direction. For this purpose, the tires were also located close to the (calculated before the test on the unconstrained pump) nodal points of the two most important mode pairs.

Although differences are important, the test pump is representative of the pump onsite, and the experimental results can be extrapolated toward the situation at the power plant.

Results of Experimental Modal Analysis

A matrix with frequency response functions (FRF) was assembled for three directions in 66 locations along the pump (Figure 9 and Figure 10).

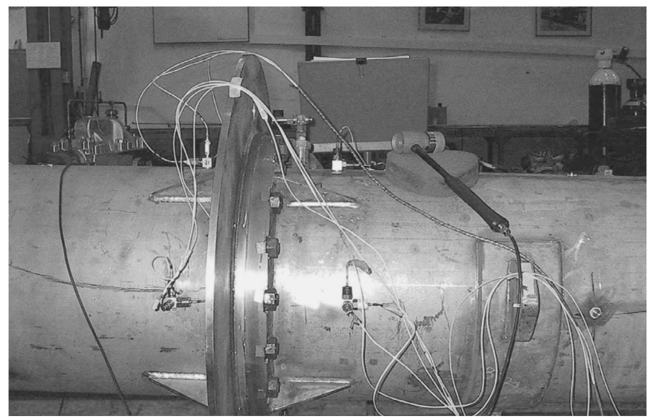

Figure 10. Experimental Modal Analysis. Measurement of FRF. Accelerometers in Three Directions and Calibrated Hammer, at Intermediate Seismic Support.

Figure 11 shows amplitude and phase of the function, in which all measured FRF are summed. Analysis of this summed block function already indicates the existence of many natural frequencies within a limited frequency range and thus a flexible structure. Closely spaced frequencies typically correspond with mode shapes in orthogonal planes.

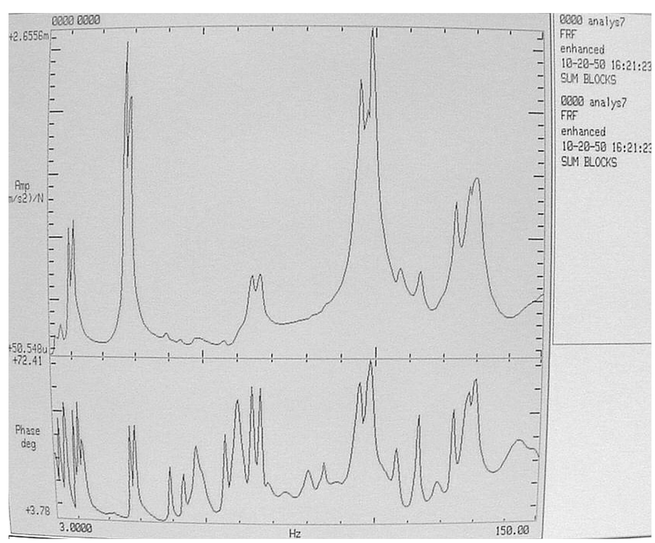

Figure 11. Summed Block Function (Both Amplitude as Phase) of All Measured FRF.

The matrix with the measured FRF was then decomposed into the modal parameters. A wire frame model of the pump allows visualizing of the identified mode shapes. Figures 12 and 13 show first bending and second bending of the pump structure (each mode belonging to one pair).

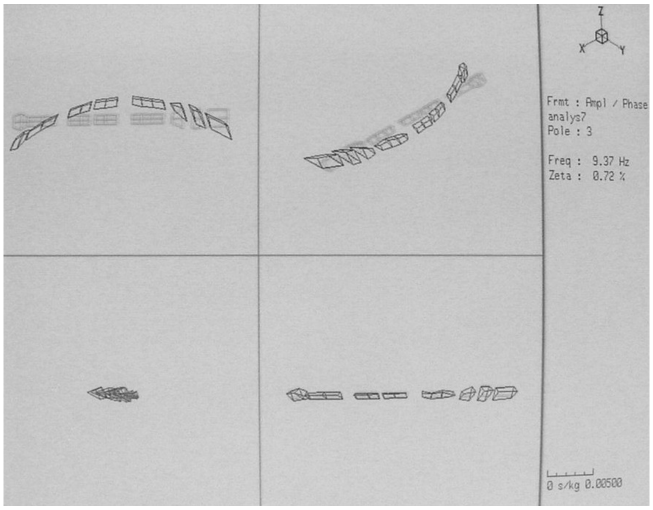

Figure 12. Experimental Modal Analysis. First Bending in Horizontal Plane.

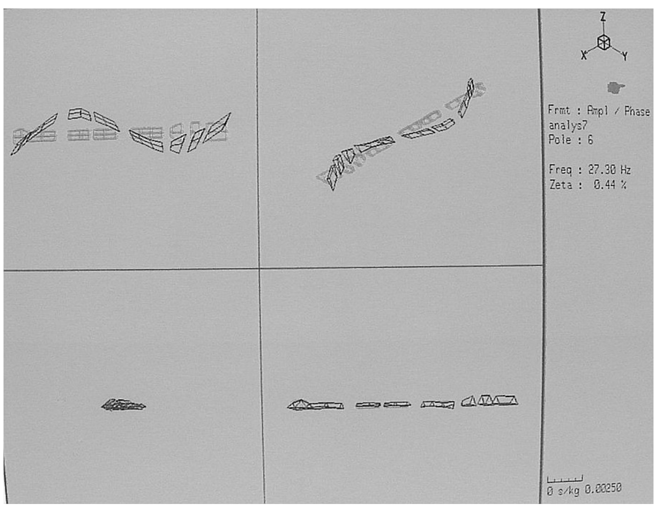

Figure 13. Experimental Modal Analysis. Second Bending in Horizontal Plane.

The natural frequencies of 4.5 Hz (axial direction) and 6.5 Hz (pendulum mode in the horizontal plane) correspond to rigid body modes and show that the rubber tires do have some effect on the dynamics of the test object.

Comparison with Calculated Results

Experimental results were then compared with calculated natural frequencies and mode shapes. In order to account for the differences between the test pump and the pump onsite, the theoretical model of the pump structure at the power plant was modified accordingly.

Since the experiments revealed that the rubber tires do influence the dynamic behavior, it was necessary to:

• Add some springs to the mathematical model and assign stiffness values so as to match measured and predicted dynamic behavior at low frequencies. Stiffness coefficients are low: 120 lbf/in (motor stool) and 86 lbf/in (suction bowl) in the vertical direction, 51 lbf/in and 34 lbf/in in the horizontal direction.

• Compare the theoretical and experimental model not in the frequency envelope of the seismic spectrum, but up to approximately 100 Hz. As the effect of the weak springs decreases with increasing frequencies, it then becomes much easier to distinguish whether deviations between both models are caused by the test rig itself, or by the theoretical calculation.

Table 4 and Figures 12, 13, 14, 15, 16, and 17 demonstrate that there is good correspondence between theory and experiment, both for values of natural frequencies as mode shapes and over a wide frequency range. This match could only be achieved by assuming that a larger part of the surfaces of the flanges was tied together, instead of being tied at the bolts only. In the latter case, there was an underestimate of 10 to 15 percent of theory with respect to experiment.

Table 4. Comparison Between Experimentally and Theoretically Identified Modal Parameters.

Natural frequency, calculated	Natural frequency, measured	Description of mode shape
4.88	4.66	Rigid body motion in axial direction.
5.90	--	Slight bending of motor on motor stool.
6.95	6.32	Rigid rocking in horizontal plane.
9.70	9.37	1st Bending of pump structure, horizontal plane.
12.03	10.82	1st Bending of pump structure, vertical plane.
13.00	12.18	Bending of pump structure. Pump oscillates in rubber tires.
25.98	27.30	2nd Bending of pump structure, horizontal plane.
27.67	28.78	2nd Bending of pump structure, vertical plane.
42.06	39.24	Torsion, expansion of discharge pipe.
59.95	64.19	3rd Bending of pump structure, horizontal plane.
61.46	66.73	3rd Bending of pump structure, vertical plane.
62.57	--	Bending of flange of spool piece, resulting in compression in axial direction.
79.25	80.88	Bending of flange of spool piece, resulting in compression in axial direction.

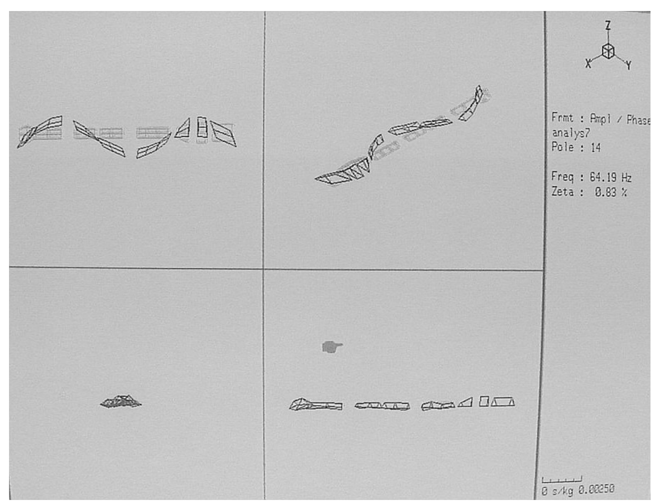

Figure 14. Experimental Modal Analysis. Third Bending in Horizontal Plane.

SHOP PERFORMANCE TESTS AND FIELD PERFORMANCE TESTS

As the replacement pumps had never been in operation before, their hydraulic performance was checked in the shop. Vibration levels were recorded and were very low. After shipment and installation

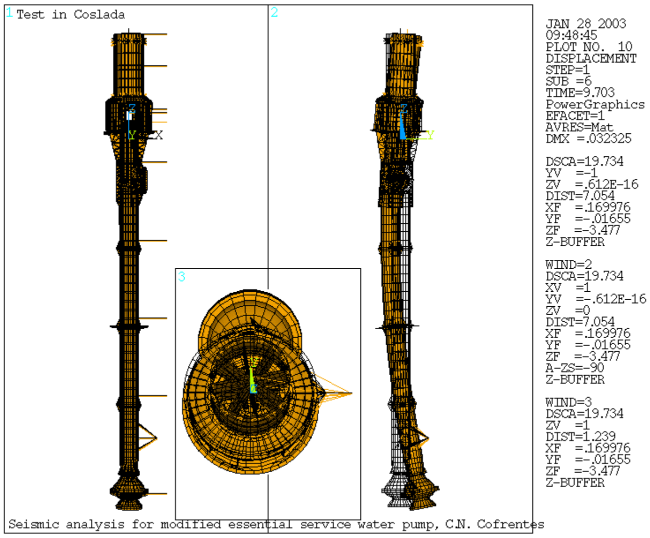

Figure 15. Theoretical Modal Analysis. First Bending in Horizontal Plane.

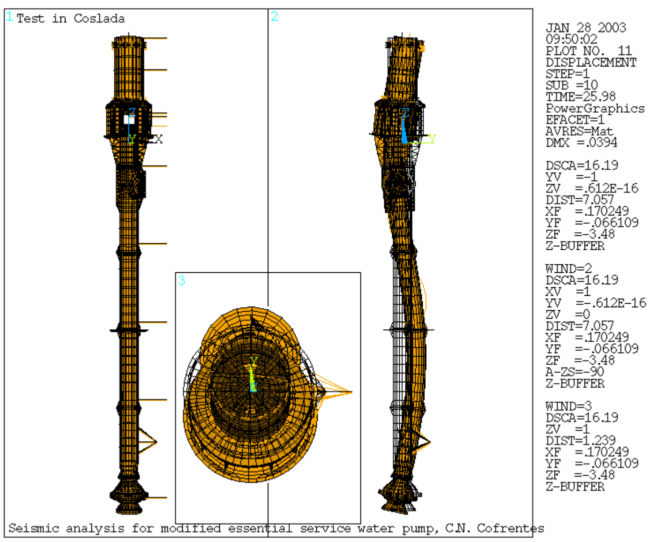

Figure 16. Theoretical Modal Analysis. Second Bending in Horizontal Plane.

onsite (six 24 hour days for each pump), the vibration levels were tested again. All values were satisfactory; overall vibration levels on the lower motor bearing were typically 24 mils/s and 45 mils/s in both radial directions and 13 mils/s in the axial direction.

CONCLUSIONS

The paper demonstrates, for one nuclear power plant, the successful replacement of the existing essential service water pumps by higher capacity pumps from another "idle" nuclear power plant, instead of purchasing completely new units.

This replacement was technically feasible though not evident. Different boundary conditions required purchase of a new motor and modification and requalification of the structure of the pump.

The integrity of the pump structure was evaluated within the framework offered by *ASME Boiler & Pressure Vessel Code* (1995) and IEEE 344 (1987).

A linear three-dimensional finite element parametrical model of the complete machine theoretically calculated its behavior under dynamic and static load conditions. Such an approach might appear time consuming, but it offers important advantages over the lumped mass-beam model:

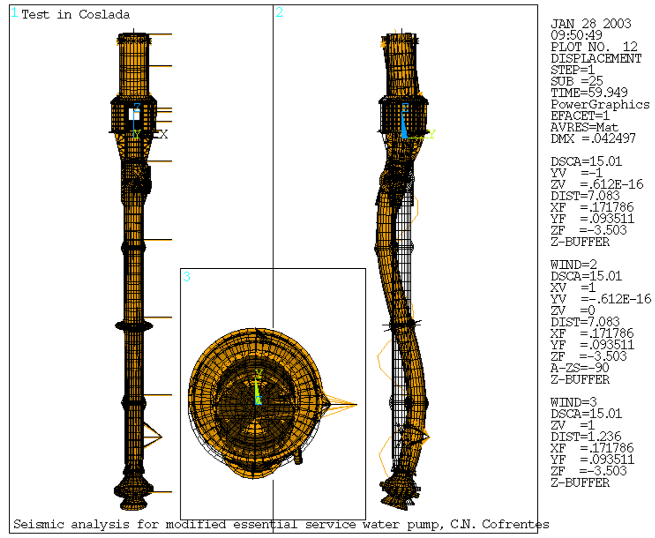

Figure 17. Theoretical Modal Analysis. Third Bending in Horizontal Plane.

- Better approximation of reality, specifically for small items or geometrically complicated components.
- The sensitivity of the structure with respect to relatively small adaptations can be studied by virtual prototyping.
- One single model can be used to estimate seismic/dynamic behavior, to evaluate static behavior and to qualify components for pressure boundary conditions.

The parametrical aspect still allowed application of "quick" changes to the model.

It is demonstrated that the dynamic evaluation of the pump should not only encompass its behavior under seismic events, but an analysis of its natural frequencies as well. The calculations suggested that the initially proposed design could suffer from resonances. The problem was solved by making the pump structure considerably stiffer and by fixing one additional seismic support at the discharge column. Subsequent dynamic and static analysis of relative displacements and stress levels (particularly at the bolts) showed that it was necessary to use the flow strainer as a second seismic restraint. Once calculated results were acceptable, the proposed modifications were physically applied.

In a next step, theoretical estimates of natural frequencies and of mode shapes were verified by an experimental modal analysis. This experiment allows validation of both the seismic and the static calculations. Care must be taken to design test rig and measurements so that these are comparable with the theoretical model. Eventually, the theoretical model must be reviewed in a physically acceptable way, so as to generate results that coincide with the experiment. Correct matching between theory and experiment then provides a solid basis for adaptations and maximizes the confidence level for correct operation in the field.

The combination of —still relatively simple—theoretical calculations and a well-chosen experimental verification, which was continuously backed and supported by an "open mind" cooperation of the user, engineering, and machine contractors, allowed taking the replacement pump out of the idle power plant; to implement some important adaptations; to let it successfully pass all qualifications and performance tests at the pump manufacturers shop without retesting; to pull the old pump and reinstall; and to test the adapted pump onsite within the refueling schedule of the nuclear power plant, without any problem.

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