# Design of Seismic Test Rig for Control Rod Drive Mechanism of Jordan Research & Training Reactor

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## 1. Introduction

Jordan Research & Training Reactor(JRTR) is under development by Korea Atomic Energy Research Institute(KAERI) and its general arrangement is shown in Fig. 1. The reactor assembly is submerged in a reactor pool filled with water and its reactivity is controlled by locations of four control absorber rods(CARs) inside the reactor assembly. Each CAR is driven by a stepping motor installed at the top of the reactor pool and they are connected to each other by a tie rod and an electromagnet. The CARs scram the reactor by de-energizing the electromagnet in the event of a safe shutdown earthquake(SSE). Therefore, the safety function of the control rod drive mechanism(CRDM) which consists of a drive assembly, tie rod and CARs is to drop the CAR into the core within an appropriate time in case of the SSE.

As well known, the operability for complex equipment such as the CRDM during an earthquake is very hard to be demonstrated by analysis and should be verified through tests [1]. However, the CRDM is too long to be simulated and excited on a single test rig as shown in Fig. 1, and hence, the seismic test rig for the CRDM is divided into two parts. One of them simulates the reactor assembly and the guide tube of the CAR, and the other one does the pool wall where the drive assembly is installed. In this paper, design of the latter test rig and how the test is performed are presented.

## 2. Initial design of seismic test rig and its modal analysis

The drive assembly which comprises the stepping motor and the electromagnet is depicted in Fig. 2. It is supported by the upper and lower bracket installed on the top of the reactor pool. The seismic test rig should simulate this environment.

### *2.1 Initial Design of Seismic Test Rig & its Experimental Modal Analysis*

The initial design of seismic test rig is shown in Fig. 3. The drive assembly is installed inside the rig and it is connected to a dummy weight which reflects the dead weight of all the moving parts attached to the drive assembly as well as their maximum dynamic load due to the vertical excitation. The inside of the test rig is filled with water.

The reactor pool top where the upper and lower brackets are installed can be assumed to be rigid enough with a natural frequency well above the cutoff frequency(=33Hz) of zero peak acceleration(ZPA) for the floor response spectrum(FRS). Therefore, the seismic test rig should not have a resonance below the cutoff frequency, which means that the upper and lower brackets should have in-phase motions.

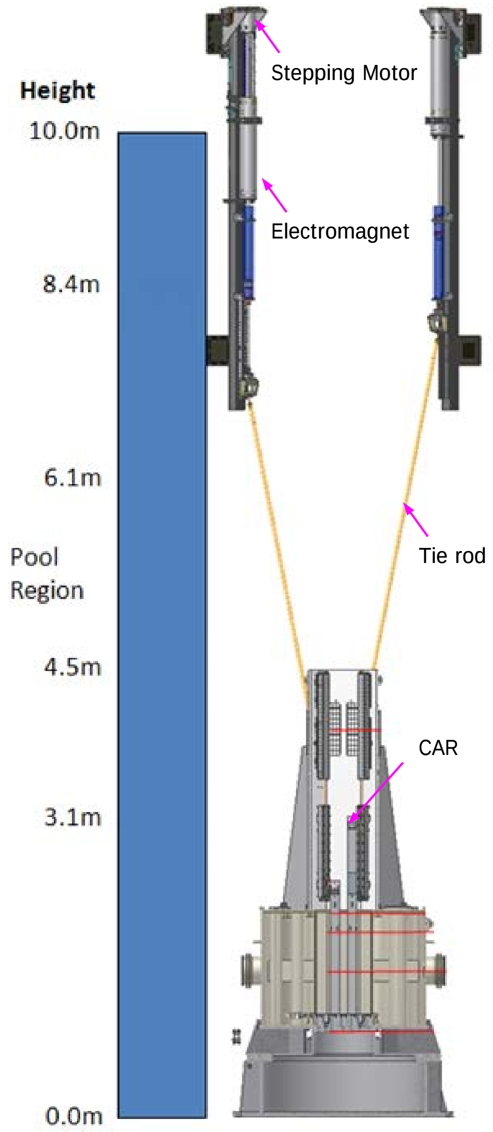



Fig. 1. General arrangement of Jordan Research & Training Reactor and its Control Rod Drive Mechanism

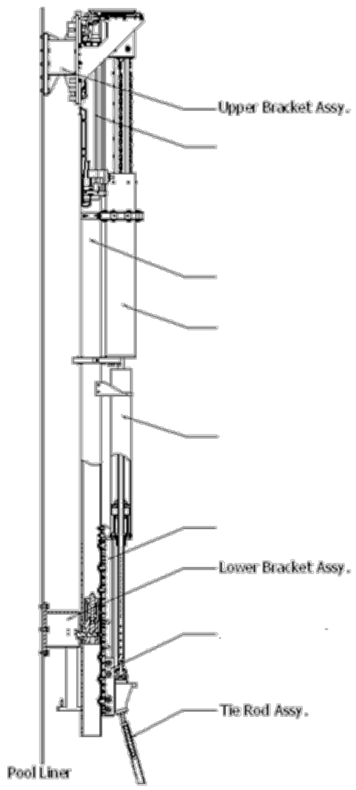



Fig. 2. Drive assembly of control rod drive mechanism

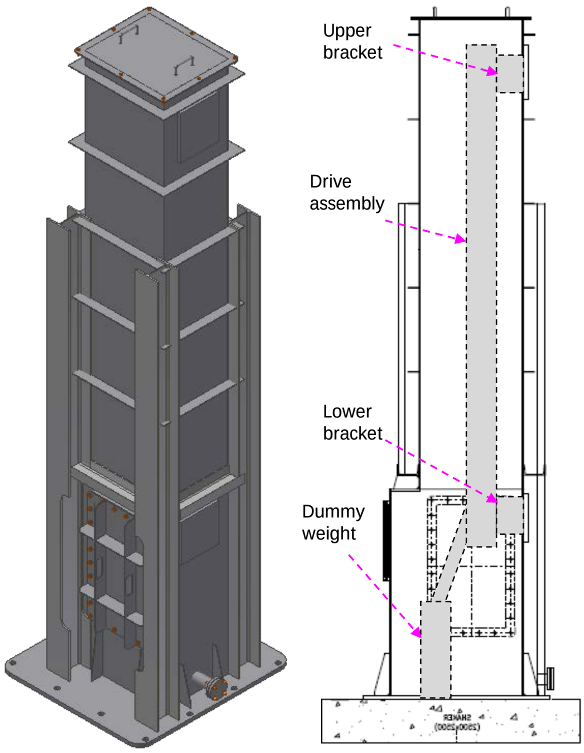
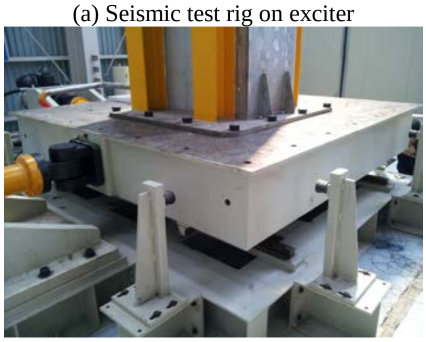



Fig. 3. Initial design of seismic test rig

The photo of the seismic test rig and its exciter which can move in two horizontal directions are shown in Fig. 4. An experimental modal analysis for the test rig was performed without the drive assembly and water inside it. Its first natural frequency was found at 16.3Hz which is much lower than the cutoff frequency(33Hz).

(a) Seismic test rig on exciter



(b) Exciter

Fig. 4. Photo of seismic test rig and exciter

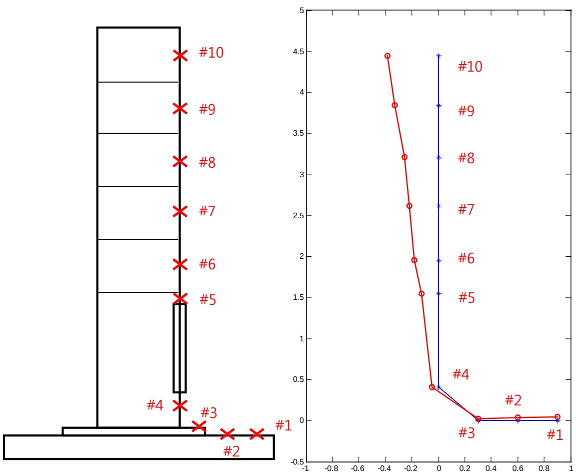
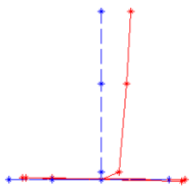
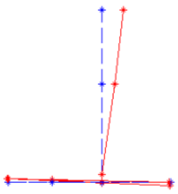


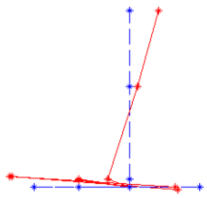

Fig. 5. Mode shape of the seismic test rig at the first natural frequency, 16.3Hz



(a) ODS at 10.0Hz



(b) ODS at 16.3Hz



(c) ODS at 25.0Hz

Fig. 6. Operational deflection shape of the seismic test rig and excitation table

The mode shape of the seismic test rig is shown in Fig. 5, and its operational deflection shape(ODS) is also shown in Fig. 6. At the first natural frequency, the test rig and excitation table rotate without translational movement. They move without rotation below the natural frequency, which means that the upper and lower brackets of the drive assembly have in-phase motions. However, the upper and lower brackets have out-of-phase motions over the natural frequency as shown in Fig. 6(c).

In order to find out how to modify the design of the test rig, a finite element model for the seismic test rig and the excitation table is essential. However, the stiffness of the sliding bearings underneath the excitation table is unknown, which is a very important boundary condition for the excitation table. Therefore, a modal analysis was performed only for the excitation table without the seismic test rig.

The first natural frequency was found at 63Hz through an experimental modal analysis with the mode shape shown in Fig. 7. In the figure, the upper rectangular represents the shaker table and two rods attached to it describe the beams connected to actuators. Furthermore, two lines below the table mean rails underneath the table and other two lines express rails on the ground. As shown in the mode shape, displacements at the end of the upper rails are much larger than those of the lower rails. This means that the stiffness of the sliding bearings in Fig. 8 installed between upper and lower rails is relatively softer than the others.

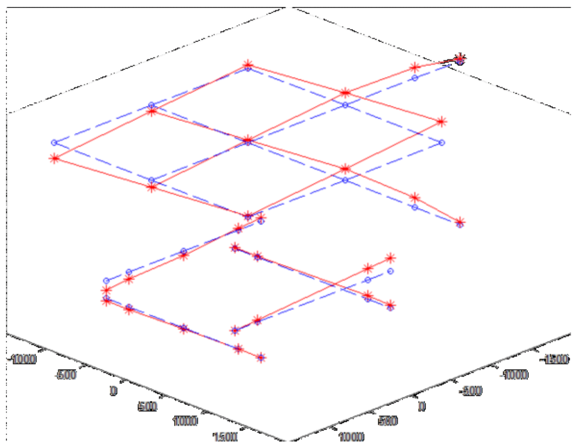


Fig. 7. Mode shape of the excitation table at first natural frequency, 63Hz

Fig. 8. Sliding bearing of excitation table

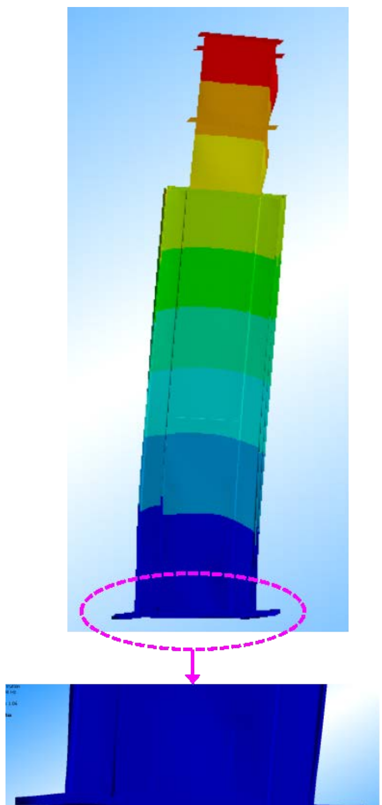


Fig. 9. Mode shape of FE model of test rig at 25Hz

### *2.2 FE Modal Analysis*

As mentioned before, an FE model for the test rig and table is vital for design modification. The model will be established based on the experimental results.

At first, the test rig without the excitation table was examined by FE modal analysis. The test rig has the first natural frequency at 25Hz and its corresponding mode shape is shown in Fig. 9. Even though the test rig without the excitation table was examined, the natural frequency is lower than the criteria, 33Hz. The mode shape shows that the base plate is too thin to provide enough base stiffness. Therefore, the design of the test rig itself should be modified somehow.

An FE model for the excitation table was also obtained based on the test result mentioned in the previous section. The stiffness of the sliding bearings was tuned to $1.6 \times 10^8$ N/m in order to match the first natural frequency of the FE model to the experimental result as shown in Fig. 10.

As mentioned before, the first natural frequency of the test rig itself is lower than the cutoff frequency. Therefore, additional stiffeners were installed to the rig and the modified test rig was assembled into the excitation table. The first natural frequency obtained by FE modal analysis is 18.5Hz as shown in Fig. 11. It was concluded from the result that it was impossible to get the first natural frequency higher than 33Hz without a remarkable change of its configuration.

## 3. Design Modification

One of the major reasons for the low natural frequency is that the center of gravity is very high, and hence, a good way to increase the natural frequency is to lower the height of the center of gravity. When the center of gravity is located at the top of the table level as shown in Fig. 12, a rough modal analysis shows that the first natural frequency is 27.2Hz which is still lower than the cutoff frequency. As mentioned before, another major reason of the low natural frequency is stiffness of the sliding bearing, thus, the bearings were replaced with the stiffer ones. An experimental modal analysis for the excitation table with the stiffer bearings showed the first natural frequency at 83.4Hz which is much higher than before, and the stiffness for the bearing was tuned to $8.26 \times 10^8$ N/m to match the experimental result. The excitation table with the point mass on it and the stiffer sliding bearings shows the first natural frequency at 36.1Hz.

The design of the test rig and the excitation table was modified as shown in Fig. 13 in order to lower its center of gravity. In practice, the ground should be excavated to lower the test rig or the rails and actuators should be relocated on higher level. The modified model showed its first natural frequency at 34.6Hz which is higher than the cutoff frequency, and its mode shape is depicted in Fig. 14.

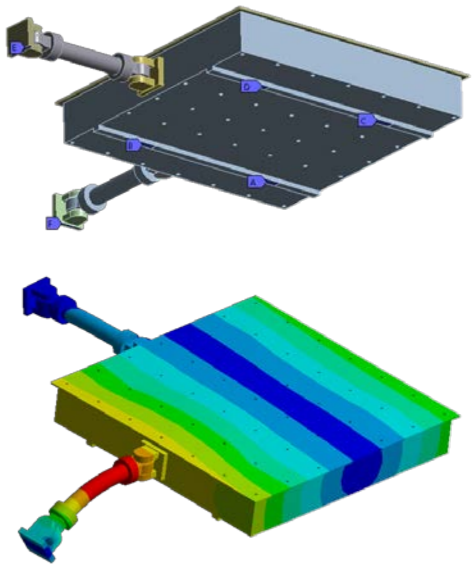


Fig. 10. FE model for the excitation table and its mode shape corresponding to the first natural frequency, 63Hz

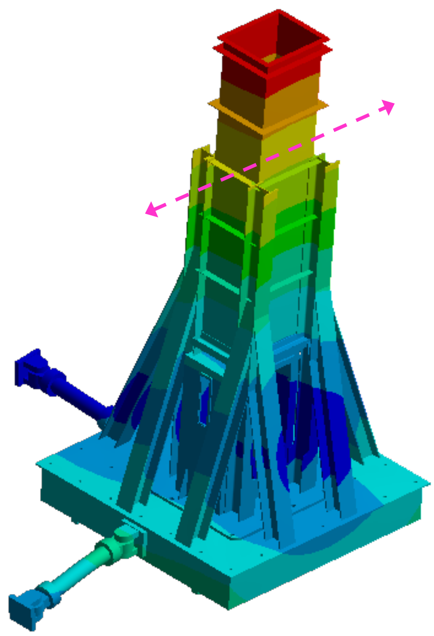


Fig. 11. Mode shape of test rig and excitation table at 18.5Hz

## 3. Conclusions

Initial design of the seismic test rig and excitation table had its first natural frequency at 16.3Hz and could not represent the environment where the CRDM was installed. Therefore, experimental modal analyses were performed and an FE model for the test rig and table was obtained and tuned based on the experimental results. Using the FE model, the design of the test rig and table was modified in order to have higher natural frequency than the cutoff frequency. The goal was achieved by changing its center of gravity and the stiffness of its sliding bearings.

## REFERENCES

[1] ASME QME-1-2002, Qualification of Active Mechanical Equipment Used in Nuclear Power Plants

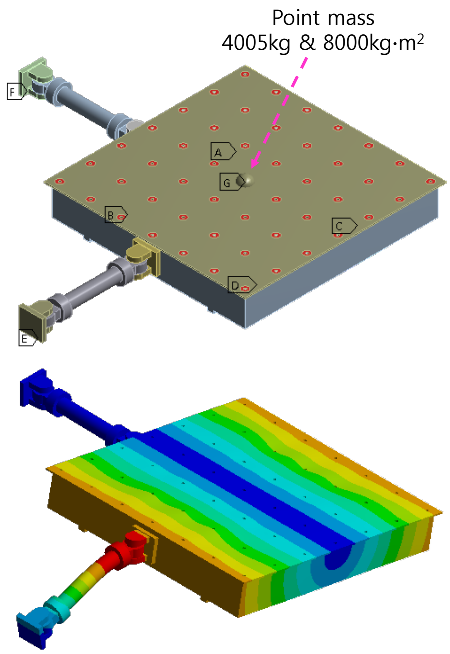



Fig. 12. Excitation table with a point mass on it and its mode shape at 27.2Hz

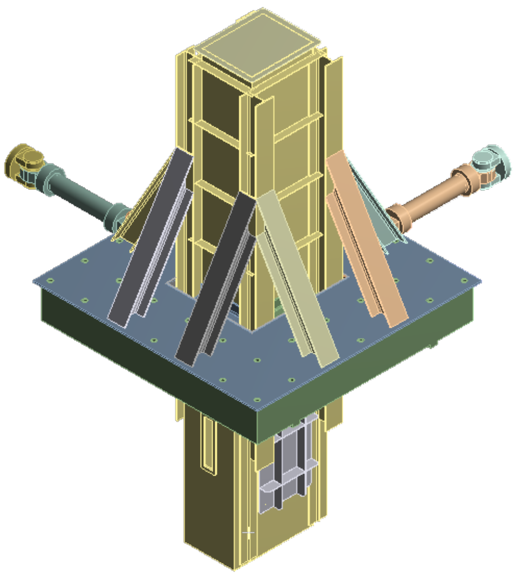


Fig. 13. Modified test rig and excitation table

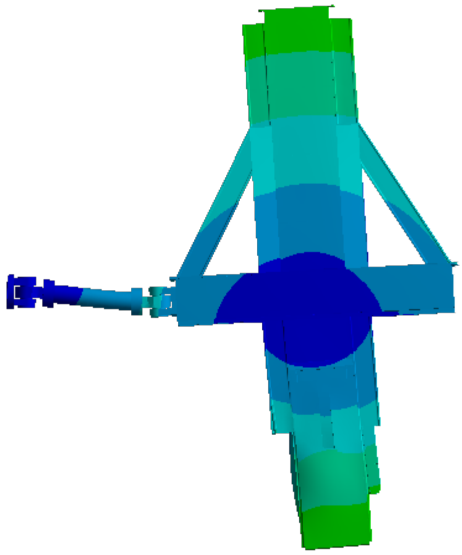


Fig. 14. Mode shape of the modified test rig and excitation table at the first natural frequency