한국표준형 원자력발전소 제어봉집합체 보호구조물의 모우드 특성

Modal Characteristics of Control Element Assembly Shroud for Korean Standard Nuclear Power Plant
(I : Pre-Test Analysis)

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요 약

원자로 내부구조물의 설계시 필요한 동적응답해석을 위하여 각 구조물의 정확한 진동특성을 파악할 필요가 있다. 한국 표준형 원자력발전소를 위하여 설계된 제어봉집합체 보호구조물은 기존의 설계로 부터 많은 설계변경이 있었고, 또 이 구조물은 튜우브와 얇은 판이 사각격자 형태로 이루어져 있고 연결봉에 의해 고정되는 등 매우 복잡한 형태로 구성되어 있어서 해석과 시험에 의한 진동측정 프로그램을 수행할 필요성이 대두되었다. 따라서 본 논문에서는 진동측정 프로그램의 첫 단계로서 범용구조해석코드인 ANSYS 를 이용하여 시험전 해석을 수행하였다. 또 자유도의 수와 얇은 판에 있는 구멍 및 연결봉의 pre-load가 구조물의 자유진동수에 미치는 영향을 검토하였다. 이로부터 결정된 유한요소모델에 대하여 모우드해석을 수행하여 구조물의 고유진동수와 모우드형상을 구하였고, 조화운동해석(Harmonic Analysis)을 행하여 주요모우드에 대한 응답을 측정함으로써 추후에 수행될 진동측정 시험조건 즉 응답측정부위, 측정위치의 수, 측정진동수의 범위 및 가진력의 크기 등을 결정하였다.

ABSTRACT

The design of reactor internals requires the accurate vibration characteristics of each component for subsequent dynamic structural response analysis. For Korean standard nuclear power plant some modifications on the Control Element Assembly shroud from the reference design have been made. Since the shroud is complex in geometry having an array of vertical round tubes and webs in a square grid pattern, and being tied down by preloaded tie rods into position, it is planned to perform a vibration measurement program consisting of both experimental and analytical modal studies upon that component.

To determine the proper test conditions, the pre-test analysis has been performed using the general purpose structural analysis program ANSYS. Also the effects of the number of master degrees of fre-

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이 논문에 대한 토론을 1992년 12월 31일까지 본학회에 보내주시면 1993년 6월호에 그 결과를 게재하겠습니다.

edom, holes in the web and tie-rod preload on the natural frequencies are examined prior to the pre-test analysis. After decision of appropriate finite element model, frequency analysis and harmonic analysis are performed and ideas for the test conditions such as the number of measurement points, their locations, measurement frequency range and the excitation force level are determined.

1. INTRODUCTION

Reactor internals are the assembly of important structures locating inside the reactor pressure vessel and supporting the reactor core from the various external loads, and they should be designed per ASME Boiler and Pressure Vessel Code Section Ⅲ subsection NG.A Control Element Assembly(CEA) shroud is one of the reactor internals structure that locates in the upper plenum of reactor vessel, and provides the passage of CEA insertion and withdrawal, and protects the CEA from the flow-induced loads. The CEA shroud consists of an array of vertical round tubes which are arranged in a square grid pattern. The tubes are joined by welding vertical plates called webs between adjacent tubes. The CEA shroud is mounted on twelve pads on the UGS base plate and is held in position by twelve tie rods which are threaded into the UGS base plate at their lower end. At their upper end, the pre-tensioned tie rods are held by nuts which bear on twelve plugs in the top of twelve of the CEA shroud tubes.

The reference design of CEA shroud has a very big force level considered from the pipe break excitation in the preliminary design stage and therefore there was a significant modification from the reference design(Ref.1) for Korean standard nuclear power plant(Ref.2).

Since the CEA shroud is complex in geometry as mentioned earlier and we need to have correct vibration characteristics to be used in the subsequent dynamic structural response analyses under the seismic and pipe break excitation, it is planned to perform a vibration measurement program on the CEA shroud. To perform the vibration measurements of the CEA shroud, the vibration characteristics of the CEA shroud are determined analytically. The vibration measurement program consists of three phases ; pre-test analysis, vibration testing and post-test analysis. In the pre-test analysis phase the natural frequencies and mode shapes are determined. Also, the responses for the sinusoidal force applied at significant modal frequencies are calculated. This is done to guide the development of the test procedure in such areas as points for load application, force levels required to excite the structure, and locations for response measurement(Fig.1).

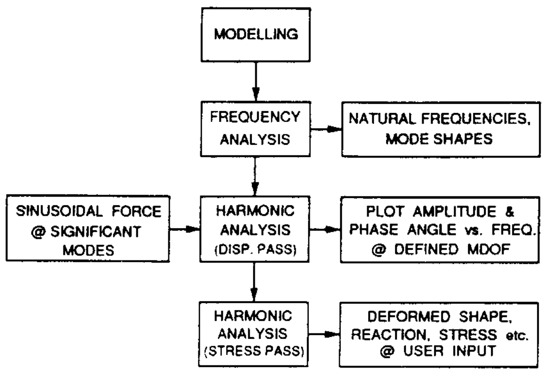

Fig.1 Procedure for Pre-Test Analysis

2. MODEL DEVELOPMENT

A 3-dimensional finite element model of the CEA shroud is developed for use with the ANSYS code(Ref.3). For webs and tubes, 4 node quadrilateral shell element(type STIF 63) is used. For tie rod which is a hollow pipe providing preload on the CEA shroud, elastic straight pipe element(STIF 16) is used. And snubber block that is relatively heavy element is modelled with generalized mass element(STIF 21).

To model the ring behavior for comparison to test, plate elements are used to explicitly represent the web and flanges of the channels. A single element is used through the web depth. The nodes lying on the ring web is made to lie exactly on the circle of mean web radius. The ring mesh for test simulation requires that the outer tubes which are attached to the stiffener rings be modeled with octagonal cross sections for compatibility with the ring detail. The interior tubes are modeled similarly to enable direct comparison of modal characteristics between analysis and test. Using octagonal tube modeling requires additional levels of nodes to maintain the proper tube and web element aspect ratios. Actual tube thicknesses is used.

The tie rod base nodes are fixed in all degrees of freedom to represent the preloaded clamped test conditions. Also the symmetric boundary conditions for the half symmetric model are imposed using the "SYMBC" command of the ANSYS code.

All material properties used in these analyses are taken from Ref.4 and the web containing holes uses the equivalent properties of Young's modulus and Poisson's ratio as calculated using Ref.5. The Young's modulus for the tie rods was adjusted to reflect the stiffening effect of the preload.

The entire model contains 4306 plate elements, 66 pipe elements, 3 mass elements, and 3570 nodes. The model is built in 11 layers of elements connecting 12 levels of nodes. Nodes and elements numbering follows the sequential pattern of the bottom level. The finite element model of the CEA shroud is shown in Fig.2.

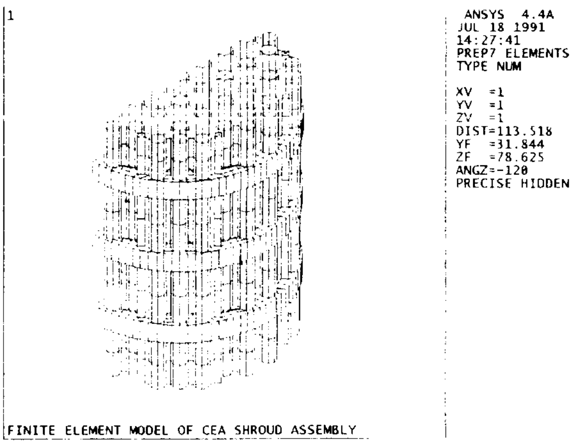

Fig.2 Finite Element Model of CEA Shroud

3. ANALYSIS

Using the finite element model mentioned above, modal analyses including some parametric study for the best analysis setup and harmonic analyses were performed.

3.1 FREQUENCY ANALYSIS

The reduced Householder procedure was used in the ANSYS code for eigenvalue extraction, which needed to define the master degrees of freedom(MDOF). Therefore as the first step, three frequency analysis runs were carried out with the following different MDOF models to determine the proper number of MDOF on this problem :

Model #1 : Number of MDOF=1644
Model #2 : Number of MDOF=1020
Model #3 : Number of MDOF=916

All analysis conditions except the number of MDOF were the same for all three runs, and the first 4 modes and their running time were examined as results.

Secondly, the effect of flow holes in the webs on the natural frequencies were examined with the following model :

Model #2 : Original properties for web material.
Model #4 : Equivalent properties for web material.

Hole effects were incorporated into the model by the use of equivalent material properties, i.e., Young's modulus and Poisson's ratio suggested in Ref.5.

From these preliminary runs it was determined that the following should be used to simulate the test conditions:

- Number of MDOF=1020
- Web material properties=Equivalent properties

Also to reflect the stiffening effect of the tie rod preload, the frequency range is determined theoretically for the fixed-fixed and fixed-pinned beam cases as upper and lower bounds using the typical beam theory as follows.

The tie rod can be approximated as a fixed-fixed beam of length 147 inches or a fixed-pinned beam of length 139 inches. The first mode natural frequency of single-span beam is from Ref.6.

$$f_i=\frac{\lambda_i^2}{2\pi L^2}\sqrt{\frac{EIg}{w}}$$

FIXED-FIXED BEAM

$\lambda_1=4.73$
L=147 inches
$E=28.3\times10^6$ psi
$I=6.28$ in^4
$g=386$ in/sec^2
$w=\rho A=(.289)(\pi/4)(4^2-3.364^2)=1.063$ 1b/in
$\therefore f_1=41.9$ Hz(Actual frequency will be lower due to upper section being more flexible than tube)

FIXED-PINNED BEAM

$\lambda_1=3.927$
L=139 inches
$E=28.3\times10^6$ psi
$I=6.28$ in^4
$g=386$ in/sec^2
$w=\rho A=(.289)(\pi/4)(4^2-3.364^2)=1.063$ 1b/in
$\therefore f_1=32.3$ Hz(Actual frequency will be higher since upper end is not exactly pinned)

A tensile load which is applied to a beam increases the natural frequencies of the beam. The effect of axial load on the natural frequencies can be found by considering the natural frequency parameter to be a function of a non-dimensional load parameter λ_i and the beam boundary conditions;

$$\lambda_i=\lambda_i(\frac{PL^2}{EI}\text{ , boundary conditions}),$$

where P is the axial force. An approximate formula for the effect of axial load on the natural frequency f_i is from Ref.6.

$$\frac{|f_i|_{P\neq0}}{|f_i|_{P=0}}=\sqrt{1+\frac{P}{|P_b|}\frac{\lambda_1^2}{\lambda_i^2}},$$

where λ_1 and λ_i: non-dimensional frequency parameter in the absence of axial load,
P_b : axial load required to buckle the beam,
P : preload(=25000 1bs).

FIXED-FIXED BEAM

From Table 8-4 of Ref.6

$$P_b=\frac{4\pi^2EI}{L^2}=323000\text{ 1bs}$$

$$\frac{f_1}{f}=1.038$$

FIXED-PINNED BEAM

From Table 8-4 of Ref.6

$$P_b=\frac{2.05\pi^2EI}{L^2}=185000\text{ 1bs}$$

$$\frac{f_1}{f}=1.065$$

This results in a range of frequencies of (1.038)(41.9)=43.5 Hz to (1.065)(32.3)=34.4 Hz. To account for the pre-load of tie rod, the average 1.0515(of 1.038 and 1.065) is used for adjusting Young's modulus as follows:

$$\frac{E'}{E}=1.0515^2=1.1025$$

$$E'=1.1025\times 28.3\times 10^6=31.2\times 10^6 \text{ psi}$$

In the finite element model, this preload effect was considered by adjusting Young's modulus of the tie rod to get the bounded natural frequencies as calculated above.

For the final setup, two additional models were made:

Model #5: Detailed Tie Rod model,
Defined additional Tie Rod MDOF on tie rod,

Model #6: Detailed Tie Rod model,
No additional Tie Rod MDOF on tie rod.

Model #6 was made to determine the effect of tie rod modes on the assembly modes.

3.2 HARMONIC ANALYSIS

Since the modal test to be performed may use the excitation techniques of electromagnetic shaker or impulse hammer, this harmonic analysis was performed to determine the required force (excitation) level in the test. An arbitrary sinusoidal force (50 1bs) was applied for the frequency of 0 to 100 Hz at the snubber location and the responses such as displacements and phase angles were determined. Since the half--symmetry model is considered here, the total force applied to the full model will be 100 1bs. The responses were calculated for the two different damping values of 0.1% and 1%, which can give lower and upper bound values.

4. RESULTS AND DISCUSSION

4.1 FREQUENCY

Frequency analyses were performed for the Models #1 through 3 to verify that the natural frequencies of the model are not sensitive to the selection of the MDOF. The modes with a significant participation factor($\geq$0.1) are summarized in Table 1. The result for Model #2 shows that 1020 MDOF can be used to adequately describe the dynamic response of the assembly considering the sensitivity of the frequency values and the computer CPU time. This number of MDOF is used as a base in the ensuing harmonic analysis.

The hole effect of web was determined from the results of Models #2 and 4. The first 4 modes with a significant participation factor are summarized in Table 1. Even though there is little difference in the first mode, higher modes show a significant difference. It is concluded that the hole effect of web and tube should be considered to predict dynamic characteristics accurately, especially for the higher modes.

Two separate runs(Model #5 and Model # 6) were made to see the effect of tie rod modes

Table 1 Summary of Frequencies for Preliminary Runs

MODEL NO.	1		2		3		4	
	FREQ.	P.F.*	FREQ.	P.F.	FREQ.	P.F.	FREQ.	P.F.
MODE 1	29.97	1.0000	30.04	1.0000	30.04	1.0000	29.60	1.0000
MODE 2	108.53	.1838	110.20	.1850	110.45	.1922	107.64	.2072
MODE 3	103.18	.1209	104.71	.1199	105.02	.1154	92.48	.1014
MODE 4	92.76	.1072	94.42	.1000	95.06	.0888	85.87	.0831
Number of MDOF	1644		1020		916		1020	
WEB Young's modulus (psi) Poisson's ratio	2.52×10^7 .30		2.52×10^7 .30		2.52×10^7 .30		2.1168×10^7 .28	
CPU TIME (SEC)**	30113		12663		11044		–	

* Particifation Factor

** on APOLLO DN10000 Workstation

on the assembly mode-with and without MDOF on the tie rod. There is only a slight difference between the two runs. This little difference results from the increase of the number of MDOF rather than the inclusion of tie rod modes. It is generally accepted that a lower frequency value is obtained for a larger number of MDOF. Therefore, it is concluded that the inclusion of tie rod mode does not make any significant difference to the assembly mode.

The mode shapes for Model #5 are shown in Figures 3 through 5. Fig.3 corresponds to the first assembly model or (cos θ) mode which is a typical rocking model. Figures 4 and 5 show the second(cos 2θ) mode and the third(cos 3θ) mode of the assembly, respectively. The first beam modes of the tie rods are in a range of frequencies 36.0 to 37.7 Hz, which is simulated well for the theoretical results of 34.4 to 43.5 Hz. The result files from the frequency analysis of Model #5 were used for harmonic analyses. The result files from the frequency analysis of Model #5 were used for harmonic analyses. The frequencies are summarized in Table 2.

Table 2 Summary of Frequencies of CEA Shroud (Model #5 & #6)

Model #5		Model #6		REMARKS
MODE No.	FREQ.(HZ)	MODE No.	FREQ.(HZ)	
1	31.75	1	32.36	Assembly 1st mode
2	32.04	2	32.76	Assembly 2nd mode
3~14	35.97~37.33	–	–	Tie Rod 1st mode
15	45.57	3	44.96	Assembly 3rd mode
16	51.96	4	51.83	Assembly 4th mode
17	68.92	5	68.84	Assembly 5th mode
18	69.69	6	69.48	Assembly 6th mode
19	75.90	7	75.85	Assembly 7th mode
20	79.11	8	78.62	Assembly 8th mode

From the previous results, it is concluded that the model with 1020 MDOF, equivalent web plate properties, and modified tie rod properties are adequate for the analysis which determines all the necessary information for the ensuing test setup. The frequencies given in Table 2 told the test could be performed in the frequency range of 0~70 Hz to see up to the first 5 modes. The number of measurement points and locations can be decided from the mode shapes shown in Figures 3 through 5. About 10 points along the circumference will be sufficient to get the first 5 modes.

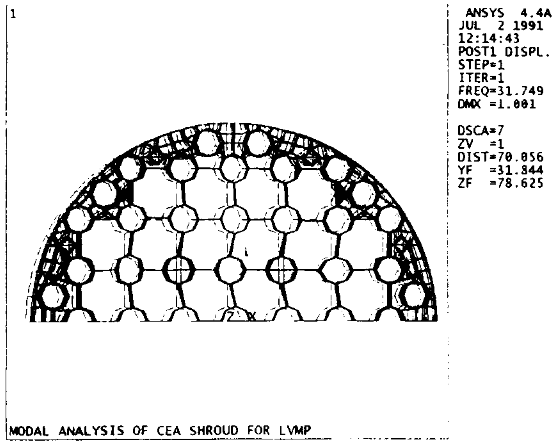

Fig.3 Mode Shape of Assembly 1st Mode

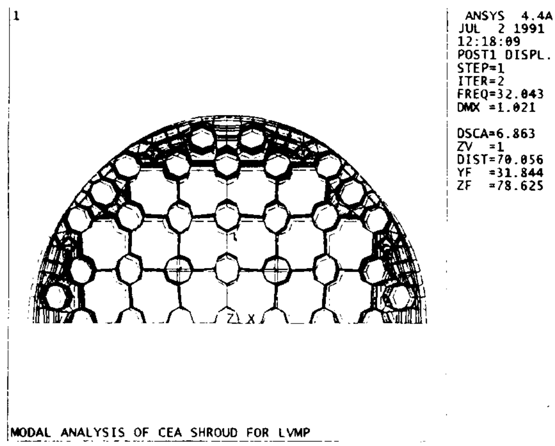

Fig.4 Mode Shape of Assembly 2nd Mode

4.2 DETERMINATION OF FORCE LEVEL

The response motions for the sinusoidal force applied are compared to the corresponding mode shape which it is intended to excite. The motions showed a good agreement with those of the mode shapes. The displacements at the

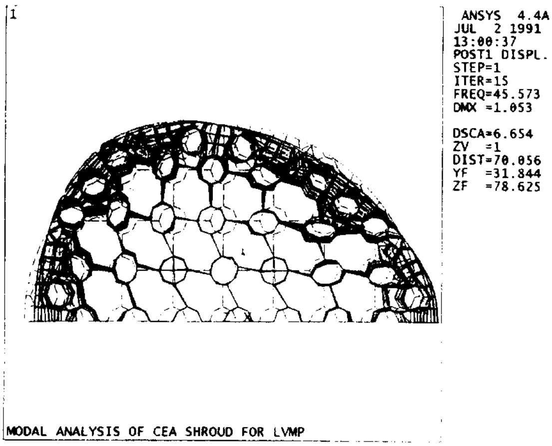

Fig.5 Mode Shape of Assembly 3rd Mode

snubber and other locations are summarized in Table 3, which will be used to determine the locations of response measurements. The maximum stress intensities for the first five modes at a phase angle of 90° are also included in Table 4. All harmonic results are based on an applied load of 100 lbs to the full model. Stress results are used for assurance that no fatigue damage will be done during the test.

Table 4 Maximum Stress Intensities from Harmonic Analyses

FREQUENCY (HZ)	PLATE LOCATION	DAMPING	
		0.1%	1%
31.75	TOP	13108	1570
	MID	3124	446
	BOT	11884	1350
32.04	TOP	9392	1512
	MID	2567	416
	BOT	7902	1239
45.57	TOP	8193	827
	MID	2987	303
	BOT	7413	749
51.95	TOP	1594	161
	MID	403	41
	BOT	1497	151
68.91	TOP	1462	197
	MID	549	162
	BOT	1443	198

Table 3 Displacements of Nodal Points for Snubber and Other Locations from Harmonic Analyses

FREQUENCY	31.749 Hz		32.043 Hz		45.573 Hz		51.958 Hz		68.916 Hz	
DAMPING	0.1%	1%	0.1%	1%	0.1%	1%	0.1%	1%	0.1%	1%
6101 UX	.23993E-1	.18037E-2	.35740E-1	.25628E-2	.18455E-1	.18460E-2	.29220E-2	.32836E-3	.21546E-2	.77276E-3
6149 UX	.24872E-1	.48240E-2	.36260E-1	.49786E-2	.18456E-1	.18551E-2	.29215E-2	.32620E-3	.21662E-2	.79701E-3
6125 UX	.21545E-1	.21547E-2	.23103E-2	.15748E-2	.76796E-2	.76823E-3	.62964E-3	.71427E-4	.17265E-4	.17203E-4
6125 UY	.30307E-2	.26807E-2	.35873E-1	.35903E-2	.12158E-3	.12148E-3	.92288E-4	.92211E-4	.16754E-2	.75685E-3
3391 UX	.17614E-1	.11944E-2	.19180E-1	.13025E-2	.26507E-2	.26895E-3	.31803E-2	.31955E-3	.18297E-2	.14766E-3
3391 UY	.43475E-2	.10788E-2	.18377E-1	.16325E-2	.87223E-2	.87739E-3	.64775E-2	.65383E-3	.24672E-2	.48781E-3
3397 UX	.11319E-1	.11326E-2	.11947E-2	.81478E-3	.27361E-2	.27057E-3	.33425E-2	.33370E-3	.20002E-4	.20046E-4
3397 UY	.21402E-2	.14597E-2	.19440E-1	.19458E-2	.53022E-4	.52983E-4	.32576E-4	.32565E-4	.64338E-2	.49465E-3
3403 UX	.18057E-1	.30148E-2	.19562E-1	.29830E-2	.26510E-2	.27378E-3	.31810E-2	.32425E-3	.18319E-2	.16264E-3
3403 UY	.47933E-2	.17549E-2	.18476E-1	.20657E-2	.87221E-2	.87923E-3	.64778E-2	.65706E-3	.24399E-2	.40699E-3
3458 UX	.25738E-1	.20322E-2	.11915E-1	.13107E-2	.13137E-1	.13089E-2	.60480E-2	.60368E-3	.54001E-2	.49498E-3
3458 UY	.30927E-2	.19026E-2	.26985E-1	.26310E-2	.62087E-2	.62588E-3	.15129E-2	.15420E-3	.93533E-3	.25482E-3
3464 UX	.26014E-1	.32835E-2	.12466E-1	.28725E-2	.13137E-1	.13090E-2	.60474E-2	.59837E-3	.53979E-2	.49084E-3
3464 UY	.34147E-2	.21499E-2	.27015E-1	.27741E-2	.62083E-2	.62757E-3	.15164E-2	.17868E-3	.93753E-3	.25788E-3
3520 UX	.21716E-1	.21711E-2	.23650E-2	.16114E-2	.14710E-1	.14717E-2	.18956E-2	.21018E-3	.32317E-4	.32228E-4
3520 UY	.44696E-2	.30483E-2	.40800E-1	.40833E-2	.13950E-3	.13937E-3	.10706E-3	.10696E-3	.21725E-2	.96808E-3
MAXIMUMS	.26014E-1	.48240E-2	.36260E-1	.49786E-2	.18456E-1	.18551E-2	.64778E-2	.65383E-3	.64338E-2	.96808E-3

(UNIT=INCHES)

Table 5 Force Level Required for Test

COMPONENT	DAMPING (%)	FREQUENCY (HZ) 31.75	32.04	45.57	51.95	68.91
MAX DISP.(inch)	0.1	.26014E−1	.36260E−1	.18456E−1	.64778E−2	.64338E−2
	1	.48240E−2	.49786E−2	.18551E−2	.65383E−3	.96808E−3
ACCEL.(in/sec²)	0.1	1035.2	1469.8	1513.3	690.4	1206.3
	1	192.0	201.8	152.1	69.7	181.5
g's for 100 1bs	0.1	2.6791	3.8038	3.9163	1.7867	3.1220
	1	.4968	.5223	.3936	.1803	.4698
$F_{required}$ for 0.1g's	0.1	3.73	2.63	2.55	5.60	3.20
	1	20.13	19.15	25.41	55.46	21.29

The force level required to excite the structure is calculated. The displacement for a sinusoidal force is $x=A \sin \omega t$. Differentiating gives velocity and acceleration;

$$\dot{x}=A\ \omega \cos \omega t$$

$$\ddot{x}=-A\ \omega^2 \sin \omega t=-\omega^2 x$$

For the case of f=31.749 Hz, ζ=0.1% and F=100 1bs

x=0.026014 inch

$$|\ddot{x}|=\frac{(2\times\pi\times 31.749)^2\times 0.26014}{386.4}=2.6791 \text{ g's}$$

If the accelerometer used for the test has a sensitivity of 0.1 g's, the input force required is

$$F_{required}=\frac{(2\times 50)\times 0.1}{2.6791}=3.73 \text{ 1bs}$$

For other frequency and damping values, the same procedure is applied and the required force in the test is summarized as in Table 5. As shown in Table 5, the force level required is between 2.55 and 55.46 pounds for the accelerometer with a sensitivity of 0.1 g's. The actual force will be determined based on the structural damping value and the number of frequencies to be estimated in the test.

5. CONCLUSIONS

For accurate response collection from the dynamic structural analyses, the finite element model of CEA shroud for Korean standard nuclear power plant was developed. The modal characteristics of the CEA shroud such as natural frequencies and mode shapes were determined analytically. The effects of master degrees of freedom, holes in the webs, and tie-rod preload on the modal characteristics were investigated. Also, the responses for the sinusoidal force applied at significant modal frequencies were calculated. This effort guided the development of the test procedure in such areas as points for load application, force levels required to excite the structure, and locations for response measurement for the ensuing test program.

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