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Screening Method for Flow-induced Vibration of Piping Systems for APR1400 Comprehensive Vibration Assessment Program

APR1400 종합진동평가를 위한 배관시스템의 유동유발진동 간이평가

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Key Words : APR1400(신형경수로1400), CVAP(종합진동평가), Flow-induced Vibration(유동유발진동), Vibration Analysis(진동해석), Main Steam(주증기), Feedwater(주급수) Screening Method(간이해석)

ABSTRACT

The revised U.S. Nuclear Regulatory Commission(NRC), Regulatory Guide(RG) 1.20, rev.3 requires the evaluation of the potential adverse effects from pressure fluctuations and vibrations on piping and components for the reactor coolant, steam, feedwater, and condensate systems. Detailed vibration analyses for the systems attached to the steam generator are very difficult, because these piping systems are very complicated. This paper suggests a screening method for the flow-induced vibration of acoustic resonances and pump-induced vibration of the piping systems attached to the steam generator in order to conduct the APR1400 comprehensive vibration assessment program. This paper seeks to address the areas such as potential vibration sources, and methods to prevent the occurrence of acoustic resonances and pump-induced vibration of piping systems attached to the steam generator, for conducting the APR1400 comprehensive vibration assessment program. The screening method in this paper will be used to estimate the flow-induced vibration of the piping systems attached to the steam generator for the APR1400.

요 약

U.S. Nuclear Regulatory Commission(NRC), Regulatory Guide(RG) 1.20(Rev.3, 2007년)은 원자로 냉각재계통, 주증기, 주급수 및 복수시스템의 주요 배관 및 기기에 대하여 압력변동 및 진동에 의한 잠재적 유해효과에 대한 평가를 요구한다. 그러나 증기발생기와 연결된 주증기, 주급수 및 복수시스템의 주요 배관 전체에 대하여 상세 해석하는 것은 매우 복잡하여 한계가 있다. 이 논문은 APR1400 원전의 종합진동평가(comprehensive vibration assessment program, CVAP)를 수행하기 위하여 증기발생기에 연결된 2차측 주요 배관의 음향공진과 펌프유발진동을 위한 간이평가 방법에 관한 것이다. 이 논문에서는 이러한 배관시스템의 잠재적 진동 원인이 무엇인지, 음향

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공진과 펌프유발진동의 가능성을 예방하기 위한 간이평가 방법은 무엇인지를 고찰하고자 한다. 이 논문은 APR1400 원전 증기발생기와 연결된 주증기 및 주급수 배관의 유동유발진동 간이평가를 위해 사용될 것으로 판단된다.

1. Introduction

The vibration and stress analysis program(1~4) of U.S. Nuclear Regulatory Commission Regulatory Guide(NRC RG) 1.20(5) includes the assessment of steam generator internals in pressurized water reactors that may possibly be adversely affected by flow-induced vibrations. Thus, the piping systems attached to the steam generator, namely the main steam(MS), feedwater(FW), and condensate(CD) systems, are involved since acoustic waves can potentially propagate to the steam generator in these systems. Flow-excited acoustic resonances within the valves, stand pipes, or branch lines can play a significant role in producing mid to high frequency pressure fluctuations and vibrations. These can damage main steam line(MSL) valves and possibly other steam system components also.

The purpose of this paper is to describe the vibration analysis methods pertaining to acoustic resonance and pump-induced vibrations of the piping systems attached to the steam generator, to conduct the APR1400 comprehensive vibration assessment program(CVAP). This paper seeks to address the following questions: What are the potential vibration sources? What are the methods for preventing the occurrence of acoustic resonance and pump-induced vibrations? How to apply the vibration analysis methods on the MS, FW, and CD systems attached to the steam generator for the APR1400 CVAP? The vibration analysis method in this paper will be used to estimate the flow-induced vibrations of the piping systems attached to the steam generator for the APR1400.

2. Expected Flow-excited Pipe Vibration

For water moving in an FW system, it is expected that the resultant piping vibrations from the acoustic sources, such as the flow control valve and pump-induced excitation to be of relatively low frequency and associated with the beam modes of vibration. Low frequency vibrations less than 300 Hz(6) are characteristic of the overall movement of the piping in the lateral beam modes, and may involve the resonance of one or more piping spans.

For the high speed flow in the MSL, high frequency vibrations are likely. High frequency vibrations may involve piping shell wall resonances, or may involve individual components such as vents, drains, instrumentation lines, and valves.

3. Vibration Analysis Method

The screening methodology in the design of piping and valve components can be implemented to prevent the occurrence of acoustic resonance(7). The approach for screening requires an evaluation of the critical points in the system that exhibit the potential for an acoustic wave to be generated. For branch pipings with the potential for acoustic resonance, design measures for avoidance of the acoustic resonance should be taken.

3.1 Flow-excited Acoustic Resonance

The focus of the screening technique is on resonance caused by flow-induced oscillations that may develop in the valve cavities or flow past a standoff pipe(side branch). Shear wave resonance of the cavities occurs when the standing acoustic

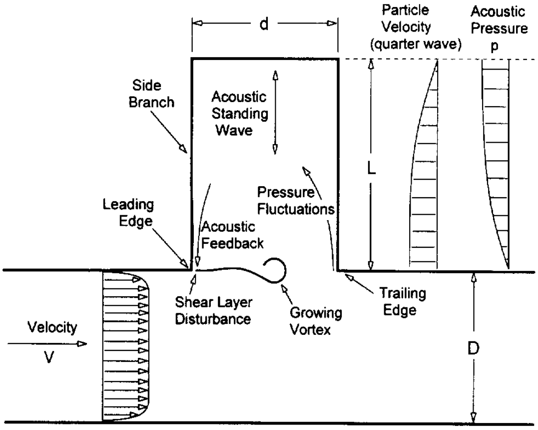

Fig. 1 Flow past a closed side branch

wave in the cavities gets coupled with the vortices shedding off from the leading edge of the cavity's opening, as depicted in Fig. 1. Vortex generation on its own is not detrimental, until amplified by acoustic resonance.

When the vortex shedding frequency becomes close or equal to the acoustic frequency of the cavity, increased pressure fluctuations occur. The resulting resonance pressure oscillation can travel through the affected piping system with little attenuation. The oscillating pressure amplitudes inside the cavity can be much greater than the main piping dynamic pressure.

Experience gained from the safety relief valves (SRVs) in boiled water reactor(BWR) power plants indicates that vibrations and noise occur during the normal plant operation when the SRV is not blowing down, and the pulsation and vibrations exist at a dominant frequency or tone.

The flow-induced oscillations for the onset of resonance are characterized by a non-dimensional frequency, namely the Strouhal number. The following equation defines the Strouhal number:

$$f_s = SV/d \qquad (1)$$

where f_s = vortex shedding frequency(Hz), S = Strouhal number, V = free stream velocity in the pipe, and d = inside diameter at the side branch entrance.

The unsteady flows into and out of the cavity produce a compression of the vertical column, and excite the acoustic depth modes with a frequency, which is expressed as follows:

$$f_a = NC/(4L) \qquad (2)$$

where f_a = acoustic resonance frequency, N = mode number (e.g., 1, 3, 5,$\cdots$), C = acoustic velocity in the flow stream, and L = branch stub length (or standpipe height for the safety relief valve).

When the vortex shedding excitation frequency (Strouhal instability) and the first modal frequency of the cavity coincide (N = 1), an acoustic resonance condition is created, which appears as a loud noise. For the resonant situation ($f_s = f_a$), Eqs. (1) and (2) are used to determine the Strouhal number:

$$S = Cd/(4LV) \qquad (3)$$

These pulsations can be further amplified if the standpipe acoustic mode is coupled with an acoustic mode of the main piping, with velocity anti-node near the side branch entrance. In that case, the acoustic impedances will match, and the side branch resonance will get coupled with the main pipe resonance. Conversely, if the side branch is located near a main piping velocity node(where the pressure is maximum), an impedance mismatch occurs, and the stub standing wave will attenuate.

Flow instability frequencies will be compared to those of acoustic modes in the piping systems and any connected valves.

Baldwin and Simmons(7) determined that the Strouhal number, for which no coupling should occur, is approximately 0.60. Ziada and Shine(8) asserted that the experimental investigations indicated that no coupling should occur for a Strouhal number of 0.62 and above. The apparent discrepancy in the two studies is due to the type and number of side branches investigated.

The evaluations in Baldwin and Simmons[(7)] focused on a single side branch on a pipe, while Ziada and Shine[(8)] extended the scope to cover the interactions between a small set of piping components(single, tandem and coaxial arrangements of closed side-branches). Furthermore, Ziada and Shine[(8)] examined the effect of a 90°-elbow in the pipe, upstream of the side branch. The results of studies by Ziada and Shine[(8)] demonstrate that the value of the Strouhal number at the initiation of resonance varies over a wide range($0.35 < S < 0.62$) depending on the ratio d/D, where D is the main pipe diameter.

Hence, a conservative value of $S_{crit} = 0.63$ is used to mark the critical flow velocity that leads to resonance.

The maximum velocity of the installation should be less than the critical velocity to avoid the occurrence of resonance($V < V_{crit}$ for a safe design) or, for a given velocity in the system, the length of the standpipe should be shorter than the critical length($L<L_{crit}$).

The Mach number M is defined as the ratio of flow speed to the speed of sound, V/C. Thus, substituting M in Eq. (3) with the critical value of the Strouhal number yields the following equation:

$$\left(\frac{d}{L}\right)crit = 4SM \tag{4}$$

and the critical ratio d/L when S=0.63 is given by

$$\left(\frac{d}{L}\right)crit = 2.52M \tag{5}$$

Hence, by design, d/L must be greater than $2.52M$ to avoid resonance for the all the operating conditions. The minimum allowable diameter-to-length ratio of the cavity is governed by the Mach number of the flow past the opening of the side branch.

If the pipe branch sizes and system velocities are already established, then it is to be checked whether the length(L) of a given size standpipe (d) will be acceptable(to avoid resonance) in a given flow stream; the following screening criterion will be used.

$$L < \frac{d}{2.52M} \tag{6}$$

If the criterion in Eq. (6) is not satisfied, design modifications should be adopted to either enlarge the branch diameter at the junction with the main pipe, or shorten the length of the closed standpipe.

Another feature of the branch connection design that can alter the ratio d/L favorably is the use of rounded or chamfered branch entry regions. When the corners of the branch mouth are rounded with a radius of curvature 'r', the critical flow velocity changes in proportion to the blend radius + standpipe inside diameter. The modified diameter ($d + r$) is used as the equivalent diameter, and is substituted in the formula for the acoustic resonance frequency in Eq. (1):

$$f = SV/(d+r) \tag{7}$$

$$L < \frac{(d+r)}{2.52M} \tag{8}$$

The evaluation of the standoff pipes in the piping system and the cavities for safety and relief valves, that have a potential to generate pressure pulses under the range of possible plant operating power levels and flow rates, can be performed as part of the design process to ensure that acoustic resonance does not occur.

3.2 Control Valve and Operator Resonance

Flow control valves(e.g., regulating valves, throttle valves, etc.) generate strong flow vortices that can create pressure fluctuations that can cause pulsations in the downstream piping, and possibly result in pipe fatigue failure. In addition, the vibrating pipe surfaces generate airborne noise that may exceed the noise levels mandated to provide a safe working environment in the overall plant.

The highest frequency at which the energy content of the pressure pulsations flow past the constrictive opening of a control valve can be correlated with the non-dimensional parameter, the Strouhal number, and is called the peak response frequency.

$$S = f_S \frac{D_{jet}}{V_{vc}} \tag{9}$$

where D_{jet} = effective diameter of the restricted flow path past the throttling element(i.e., sometimes called the "plug" in the control valve, and often of proprietary design), V_{vc} = velocity of the fluid through the vena contracta of the restriction.

The Strouhal number is approximately 0.2 over a wide range of pipe Reynolds numbers, based on empirical data(9). This approach is used for water and other nearly incompressible fluids.

The piping can be designed so that the peak response frequency of the valve cannot induce mechanical resonance in the pipe segment, or it can be corrected by adding a pipe support, if necessary.

The high frequency pulsations from the valve flow will travel transversely and interact with the pipe wall. The pipe walls will respond to the excitations by localized flexing in radial shell modes. The magnitude of the response depends on the pipe diameter and wall thickness. If the circumferential radial natural frequencies correspond to the frequencies generated by the valve, stress waves will be generated that travel along the circumference of the pipe.

The stress waves generate undesirable piping deflections that are usually experienced first as high levels of airborne noise. The vibration level of the piping in these modes has been correlated to the sound pressure level(SPL) measurements made one inch from the outer surface of the pipe(10).

Only a small portion of the energy of the jet at the vena contracta of the control valve is available to induce acoustic resonances in the downstream piping(11). However, with certain combinations of acoustic velocities and pipe lengths, the reflected wave from the end of the pipe run can add constructively to the pulsations leaving the valve. This is an acoustic standing wave resonance. The resonant acoustic frequency of a pipe of length L closed at its end is as follows:

$$f_a = \frac{(2n-1)}{4} \frac{c}{L} \tag{10}$$

where c = the acoustic velocity in the pipe, n = 1, 2, 3,⋯ thus the odd harmonics of a quarter wavelength are in resonance(12).

The acoustic resonance would occur if the pipe length matched an integral number of half-wave lengths associated with the excitation frequency.

The length of the piping is also to be checked to ensure that there will not be acoustic resonance at the excitation frequency.

The acoustic velocity used in the calculations above for the resonant acoustic frequency of the piping should consider the distensibility of the pipe wall. This additional flexibility lowers the effective speed of sound in the piping. For infinitely rigid pipe walls, the speed of sound of the fluid is as given below:

$$c = \sqrt{\frac{k}{\rho}} \tag{11}$$

where k = bulk modulus of the fluid, and ρ = fluid density.

Accounting for the finite stiffness of the walls requires modification of the above equation as given below(10,13):

$$c = \sqrt{\frac{k'}{\rho}} \tag{12}$$

where k' = the effective bulk modulus given by the following equation:

$$\frac{1}{k'} = \frac{1}{k} + \frac{d}{tE} \tag{13}$$

where d = the mean diameter of the pipe, t = the pipe thickness, and E = Young's modulus of the pipe.

3.3 Pump-induced Vibration

Pump-induced pressure pulsations are potential sources of steady-state vibrations in the piping. The pulsations originate at the pump, and travel through the entire discharge piping. The effects of the pressure pulsations can be more severe when they coincide with the natural frequency of the piping system downstream of the pumps. Eliminating the pulsations may involve modifying the pump or changing the piping acoustical frequency. For example, the acoustic properties of the piping can be changed through the addition of a pulsation damper and suction stabilizer.

When the pressure pulsations travel through the piping at any instant of time, the pressure on one elbow may not be equal to the pressure on the other elbow of the piping leg, resulting in an unbalanced force in the pipe leg. Moreover, several support failures and vibration failures of the attached instrumentation and other small-branch pipings, as well as excessive vibrations in the suction piping, have been experienced.

The pump-induced pressure pulsations occur at distinct frequencies, which are multiples of the pump speed or multiples of the number of pump plungers, blades, volutes, or diffuser vanes. The potential pulsation frequencies are defined by the following equation:

$$f = \frac{n \times X}{60} \text{ or } \frac{n \times X \times Y}{60} \tag{14}$$

where f = frequency of pressure pulsation(Hz), n = 1, 2, 3,$\cdots$, X = pump rotating speed(rpm), Y = number of pump plungers, blades, volutes, or diffuser vanes(dependent on the pump type).

The appropriate actions required in the design of the piping system to mitigate pump-induced acoustic resonance are, to have a thorough design review of the pump characteristics, and to ensure that the natural frequency of the piping system downstream of the pumps is tuned away from the pulsation frequencies generated by the pump.

4. Conclusion

The analysis method for the flow-induced vibration, presented in this paper, is very useful to confirm the occurrence of potential adverse flow effects of MS, FW, and CD systems for the APR1400 CVAP. The excessive vibrations due to acoustic resonance and pump-induced vibrations as a result of flow in the attached piping systems can be eliminated by ensuring that the piping systems are screened, and such adverse conditions are averted in the design phase itself. These vibration analysis methods will be applied to the CVAP for the APR1400 nuclear power plants.

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