An Evaluation of Fluid-elastic Instability for a FHX Serpentine Tube in PGSFR

Sung-Kyun Kim [a*], Gyeong-Hoi Koo [a]
[a]*Korea Atomic Energy Research Institute, Dukjin-Dong, Yuseong-Gu, Daejeon, South Korea*
[*]*Corresponding author: sungkyun@kaeri.re.kr*

1. Introduction

Even after the reactor shutdown, decay heat is continuously generated from the reactor core due to radioactive decay of fission products. Therefore, proper means for residual heat removal should be provided to avoid a fuel cladding failure and consequential radioactive fission product release caused by undesirable temperature rise during a system transient. In PGSFR (Prototype-Gen IV Sodium-cooled Fast Reactor), there are two independent RHRSs (Residual Heat Removal Systems); ADHRS (Active Decay Heat Removal System) and PDHRS (Passive Decay Heat Removal System), to remove total decay heat from the reactor coolant system without exceeding temperature limits of the reactor core, structures, and relevant system components.

The ADHRS is a safety-grade active system, which is comprised of two independent loops with a single DHX (Decay Heat eXchanger) immersed in hot pool region and a single FHX (Forced-draft sodium-to-air Heat eXchanger) located in upper region of the reactor building. FHX is a shell-and-tube type counter-current flow heat exchanger with serpentine finned-tube arrangement. Liquid sodium flows inside the heat transfer tubes and atmospheric air flows over the finned tubes. The configuration and overall shape of the unit are shown in Fig. 1.

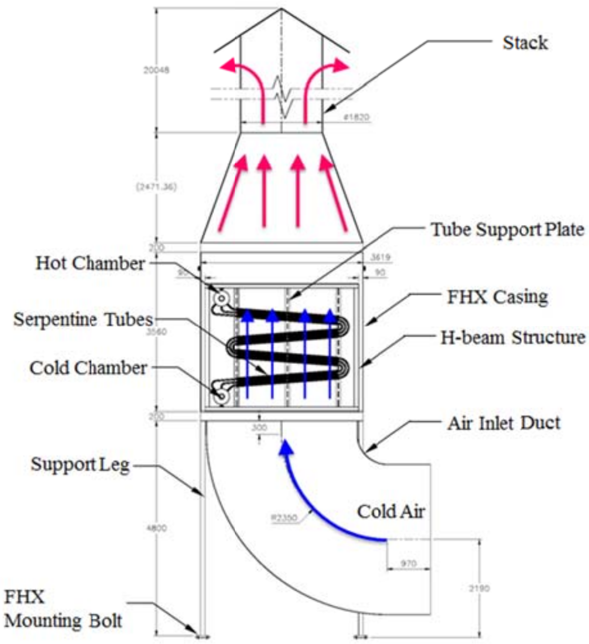


Fig. 1. Configuration and overall shape of FHX

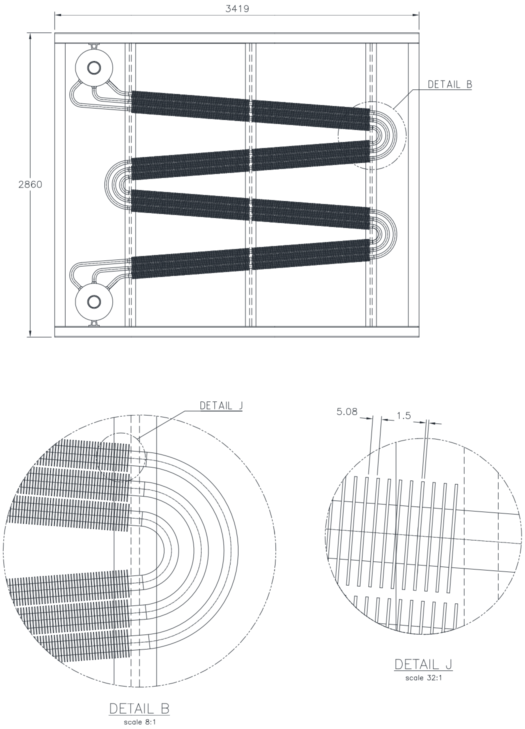


Fig. 2. Serpentine tubes of FHX

As shown in Fig. 2, three TSPs (tube support plates) are installed in the tube bundle for the lateral and vertical supports of tubes because the forced air cross-flow entering into tube bundle would induce their vibrations. On the other hand, the TSP installed in the middle of tube bundle would bring about a disturbance of air flow and cause to generate a turbulent flow in the tube bundle so that it leads to weaken the heat exchanging performance of FHX. Therefore, it is necessary to figure out the characteristics of fluid-elastic instability for the current TSP positions.

In this paper, the fluid–elastic instability for a serpentine tube of FHX in terms of the current design was evaluated by following the ASME code.

2. Methods and Results

2.1 ASME Code to evaluate instability

The ASME BPV Sec.III Appendix [1] reveals that the fluid-elastic instability can be evaluated to compare the critical velocity of a serpentine tube with the inflowing air velocity across tubes. According to the ASME rule, the more the critical velocity is close to the inflowing air velocity, the more increased the instability is [1].

2.2 Method of critical velocity

The critical velocity of a serpentine tube can be obtained by following Eq. (1) which is constituted by a natural frequency, effective mass, damping ratio, and fluid density functions.

$$V_c/f_nD=C[m_t(2\pi\varepsilon_n)/\rho D^2]^a \qquad (1)$$

where, V_c is critical velocity (m/s); f_n is natural frequency of the immersed tube (Hz); D is tube outer diameter (m); m_t is total mass per unit length of tube (kg/m); ε_n is damping ratio; ρ is fluid density (kg/m3); C is mean value (instability coefficient); a is constant value determined with respect to the tube array geometry. a=0.5 has been recommended [1].

2.2.1 Calculation of natural frequency

The natural frequency of a serpentine tube is calculated by using the ANSYS software. For the modeling simplification for the complex geometry, following equivalent density is used as an input data.

m_F (fin mass) = 8.91 kg, m_t (tube mass) = 2.95 kg
$\rho_{eqv}=(m_F + m_t)/ V$=30,750 kg/m^3

Fig. 3 shows the natural frequencies and mode shapes for without and with middle support cases. The results reveal that the fundamental natural frequencies of tube for without and with support cases are 8.5 Hz and 33.7 Hz respectively.

2.2.2 Calculation of total mass

The total mass for critical velocity is calculated as follows.

$m_t = m_A + m_C + m_S$ =5.8 kg/m

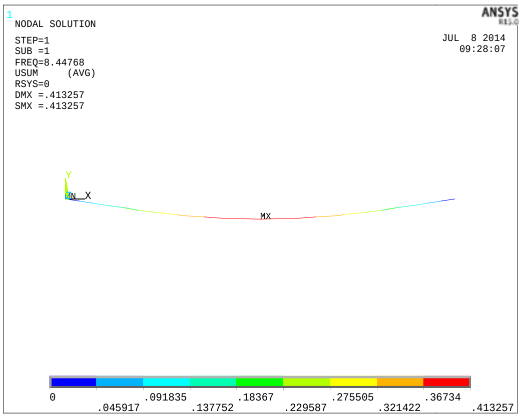
where , m_A (added mass)= ignored
m_C (sodium mass) =0.64 kg/m (ρ_s =864.6 @ 370 ^{o}C[2])
m_S (structure mass)=5.16 kg/m (ρ_{eqv} =30,750 kg/m^3)

2.2.3 Calculation of air density

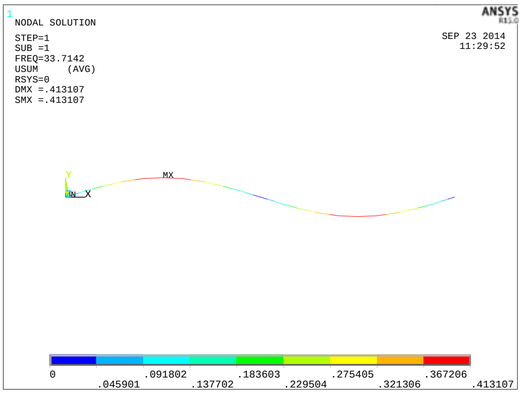
In Eq.(1) the air density flowing across tubes should be known. According to the reference [3], the air density at air temperature 40 ^{o}C and atmospheric pressure is 1.1266 kg/m^3.

2.2.4 Critical damping ratio

The critical damping value in Eq. (1) is determined based on the tube installation. Since serpentine tubes are only constrained in the lateral and vertical directions and are free in the axial direction, the tube installation is considered as high than low in Fig. 4. The critical damping values in the high condition are 0.005 and 0.03, and among both values 0.005 is determined for the conservative evaluation.



(a) without middle support



(b) with middle support

Fig. 3. Natural frequency of serpentine tube

TABLE N-1311-2
GUIDELINES FOR DAMPING OF FLOW-INDUCED VIBRATION

Description of Tube Installation	Fluid Surrounding Tube	Critical Damping Ratio, ζ Low [Note (1)]	Typical Design Value	High [Note (2)]
Thermowells and single span tubes supported by welded or rolled in ends	Liquid and gas	0.0005	0.002	0.005
Multispan heat exchanger tubes supported by passing through oversized holes in plates	Low density gas	0.008	0.017	0.03 [Note (3)]
	Water and other liquids	0.01	0.02	0.03 [Note (3)]

GENERAL NOTE: This table applies to metallic tubes with 0.5 in. (13 mm) to 2.0 in. (50 mm) outside diameter. For tubes passing through oversized holes in support plates, this table applies to typical diametrical clearance between tube outside diameter and tube support inside diameter of 0.010 in. (0.2 mm) to 0.030 in. (0.8 mm).

NOTES:
(1) Low value: For midspan rms vibration amplitude less than 1% of tube diameter and smaller than the diametrical clearance between the tube and the support plate.
(2) High value: For midspan rms amplitudes comparable to or larger than the diametrical clearance between the tube and the support plate. Tube wear can result.
(3) Critical damping ratios 0.03 < ζ < 0.05 can be used if justified by applicable experimental data.

Fig. 4. Guidelines for damping of flow-induced vibration

2.2.5 C (mean value)

The constant C value is determined by tube array pattern as shown in Fig. 5. Tubes of FHX are arranged with a square so that C value is 3.4.

	Triangle	Rotated Triangle	Rotated Square	Square	All
C_{mean}	4.5	4.0	5.8	3.4	4.0

Fig. 5. Mean value of tube array

2.3 Calculation of critical velocity

On the basis of previous obtained parameters the critical velocity of a serpentine tube are as follows.

$$V_c = f_n C[m_t(2\pi\varepsilon_n)/\rho]^a = 11.68 \text{ m/s} \quad (2)$$

For the evaluation of the fluid-elastic instability, air velocity of entering FHX should be known. Since air flow rate of FHX is 4.93 kg/s and the cross section of FHX inlet duct is 6.226 m^2, the air velocity is

$$V_{air} = Q_m/A = Q_v/A\rho = 0.705 \text{ m/s} \quad (3)$$

Comparing Eq.(2) and Eq.(3), it is known that the critical velocity is far from the inflowing air velocity. It is concluded that the fluid-elastic instability is not happen in the current design. This means that the middle TSP is redundant.

3. Conclusions

In this paper, the fluid-elastic instability of a serpentine tube in FHX was evaluated. The analysis results reveal that the possibility of occurring the fluid-elastic instability is very low and the middle TSP seems to be unnecessary.

REFERENCES

[1] ASME Boiler and Pressure Vessel Code, Section Ⅲ Appendices, ASME, pp. 289~302, 2013
[2] RCC-MRx Code 2012 edition
[3] Heat Transfer Engineering Handbook, 4th edition, 1986, JSME