Acoustic and Vibration Stability Analysis of Furnace System in Supercritical Boiler

Hyuk-Min Kwon[1]; Chi-Hoon Cho[2]; Heui-Won Kim[3]

[1,2,3] Advanced Technology Institute, Hyundai Heavy Industries, Co.,

Ulsan, Republic of Korea

ABSTRACT

The furnace of supercritical boiler is tall and flexible because it consists of thousands of tubes connected with thin plates each other. So the furnace has additional outfitting structure to resist internal combustion turbulent flow, external wind load and earthquake forces.

It is difficult to evaluate the dynamic characteristics of the furnace in the design stage due to structure complexity. The evaluation procedure for dynamic characteristics of the furnace was proposed in this paper based on CFD, acoustic resonance calculation and transient forced vibration analysis considering the temperature change of furnace inside. The transient CFD analysis of the furnace combustion has been carried out to obtain the fluctuating pressure which was good agreement with acoustic mode of the furnace. The transient forced vibration analysis has been performed considering combustion fluctuation pressure, which was evaluated with allowable vibration limit. The proposed evaluation procedure has been applied to HHI's supercritical power plant project.

Keywords: Supercritical boiler, Furnace, CFD, Acoustic resonance, Forced vibration analysis

1. INTRODUCTION

Supercritical boilers are constructed of various types of steel panels made of tubes that have pressurized water flowing through them. The combustion fluctuation pressure made inside boiler furnace could be critical factors which generate acoustic and vibration resonance of boiler.

This study includes the results of CFD (Computational Fluid Dynamics) calculation for combustion flow and vibration analysis for the boiler furnace structure of several steam power plants. The purpose of analysis is to investigate the pressure status in the furnace and vibration characteristics of the boiler furnace structure in operation without occurrence of resonances by means of predicting the fluctuating pressure in the combustion turbulent flow and the vibration level of the structure. The quantitative prediction results are performed by three kinds of the analysis methods; CFD with combustion models, acoustic FEM (Finite Element Method) and structure FEM.

2. OBJECTIVES AND PROCEDURE

The objective of this study is to evaluate the structural stability by vibration analysis for furnace structure based on excitation pressure of combustion turbulent flow and consideration of acoustic resonance modes.

The analysis consists of two parts; Acoustic analysis with CFD and vibration analysis. The flow chart of the procedure is shown in Figure 1.

[1] kwonhyukmin@hhi.co.kr
[2] ccho21@hhi.co.kr
[3] khw@hhi.co.kr

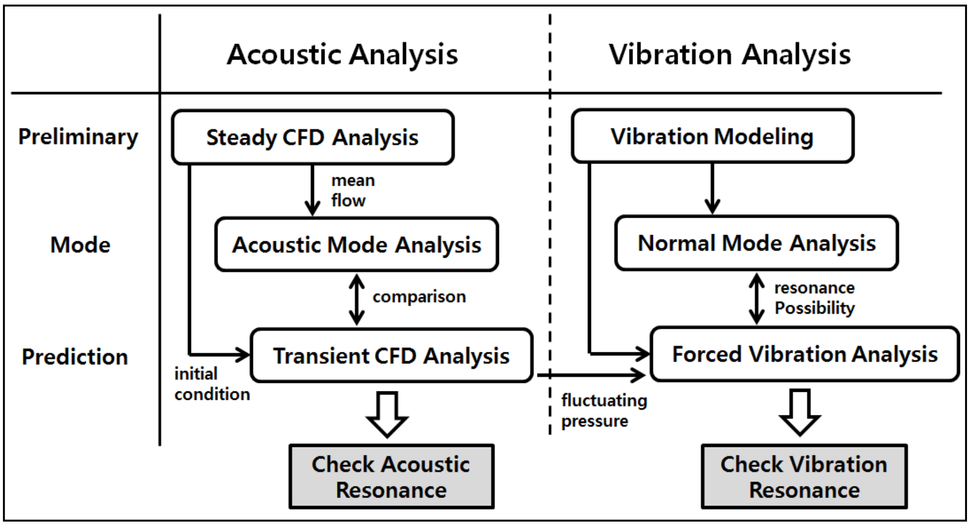


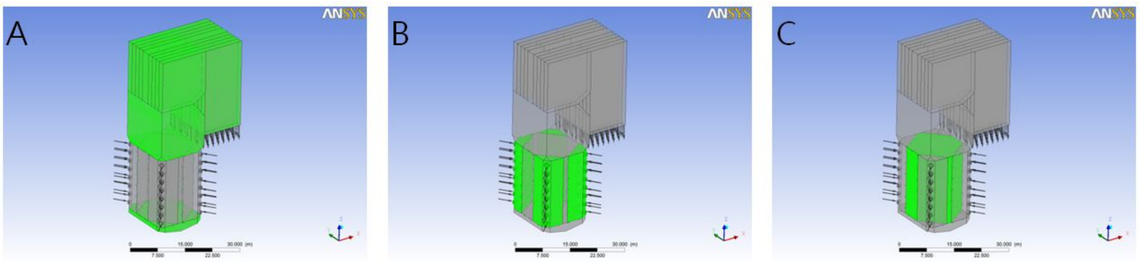
Figure 1 – Flow Chart of Analysis Procedure

3. ACOUSTIC ANALYSIS WITH CFD

3.1 Steady CFD

Steady CFD analysis is performed to obtain the mean flow data required for the acoustic analysis and the initial condition of the transient analysis[1]. The heat exchanger tubes (superheater, reheater and economizer) are simply modeled by thin plates. The CFD domain was separated into three mesh blocks as shown in Figure 2 to obtain a fine mesh effectively.

Combustion zone denoted by C in the figure has the finest mesh and total number of mesh element is about 1.9 million. The mesh was generated by CFD preprocessor, ANSYS Gambit and fluid is modeled as mixture of ideal gases. Moreover, the analysis is validated by comparing the calculated FEGT (Flue Exit Gas Temperature) to the design performance value. Gas properties, temperature and flow velocity of mean flow will be input for acoustic mode analysis.



(a) Non-Combustion Zone (b) Nozzle Injection Zone (c) Combustion Zone

Figure 2 – Mesh blocks for CFD calculation

3.2 Acoustic Mode Analysis

Acoustic mode analysis is to identify the acoustic mode of the fluctuating pressure in the furnace. The input data for the solver is gas properties, temperature and flow velocity which were the results of the steady CFD calculation. They are used to calculate the local propagation speed of acoustic wave. The interpolation between CFD and acoustic grids is performed before solving process. After setting the medium data, acoustic mode calculation was carried out using Acoustic FEM module in Virtual Lab Acoustics program.

The furnace has a shape of bended tube whose fundamental acoustic mode is characterized by its overall length. The fundamental mode frequency is found at 2.4 Hz. The second acoustic mode is determined by the height of furnace and its mode frequency is 6.7 Hz.

3.3 Transient CFD Analysis

Transient CFD analysis was conducted to obtain the fluctuating pressure in the furnace by using the same CFD model and solver, ANSYS CFX. The time step of Δt=0.005 sec and time duration of 5.0 sec is adapted for the transient simulation. The pressure measurement positions are selected as shown in Figure 3. They are grouped by their vertical position; HEIGHT1, HEIGHT4 and HEIGHT5 correspond to bottom, nose and ceil of the furnace, respectively.

The fluctuating pressure was calculated by subtraction of its temporal mean pressure from absolute one. The frequency spectrum of the pressure signal was calculated by Fourier transformation after multiplying of Hanning window to original time signal. From the calculation results as shown Figure 4, two peaks at 2.4 Hz and 7.0 Hz were found in most of the frequency spectrum. The frequencies are in good agreement with 1st and 2nd acoustic modal frequencies, 2.4 Hz and 6.7 Hz. This confirms that the transient CFD calculation analyzed the acoustic pressure wave phenomenon correctly. The maximum value of fluctuating pressure was occurred at HEIGHT 1 which is the center of the firing region, and its amplitude is 400 Pa which is just 0.4 percent of absolute pressure, 100.6kPa, in the furnace as shown in Figure 4. The fluctuating pressure data keeps the oscillation amplitude stable and does not show any trend of amplification with respect to time. Also, this analysis provides the pressure acting on furnace walls to the forced vibration analysis.

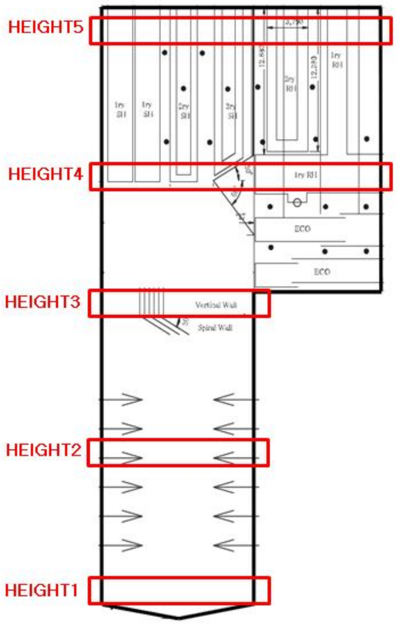


Figure 3 – Pressure Measurement Position

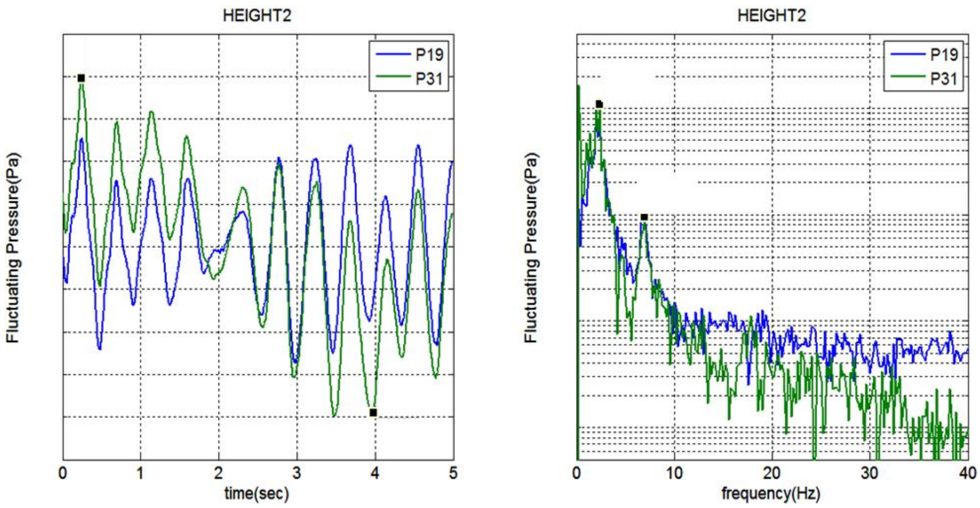


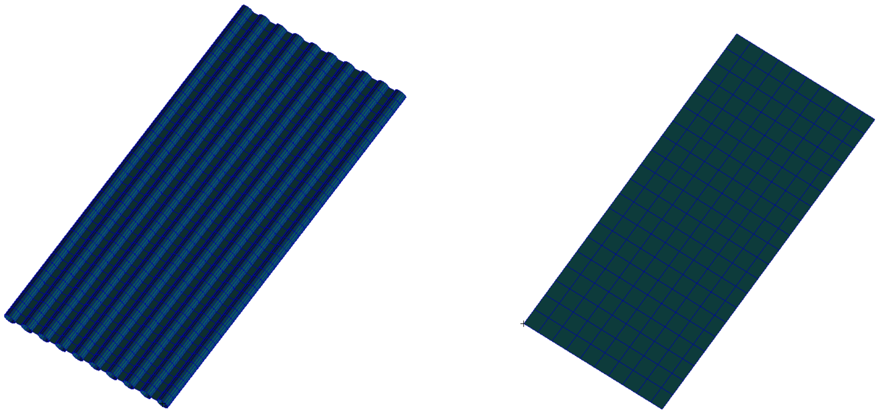
Figure 4 – Pressure Measurement Position

4. VIBRATION ANALYSIS

4.1 Vibration Modeling

Vibration modeling is to construct pressure parts of boiler as FEM modeling which contains the boundary condition. The furnace consists of thousands of wall tube plate as shown in Figure 4(a) that is very difficult to construct analytical model considering its geometry due to computing efficiency[2]. Therefore, the analytical model of the furnace structure was constructed as simplified plate for the vibration analysis as shown in Figure 4(b). Vibration characteristics of simplified plate were confirmed through a series of model tuning process adjusting orthotropic material properties such as elastic and shear modulus.

The vibration characteristics such as mode shapes and their natural frequencies of simplified plate were calculated and adjusted with element modeling and the results were summarized in Table 1.



(a) (b)

Figure 4 – Element model and sample plate

Table 1 – Property adjusting process

	E1=19.5*E5 (N/mm^2)	E2=5900 (N/mm^2)	G = 3.3*E4 (N/mm^2)		
Order	1st	2nd	3th	4th	5th
Modeling (Hz)	25,5	37.5	64.2	70.5	81.2
Plate (Hz)	25.1	38.3	64.4	69.6	80.6
Difference (Hz)	0.4	0.8	0.2	0.9	0.6

Boundary conditions such as Hanger rod, Buckstay and Boiler stopper are modeled considering its counts and stiffness. Especially, buckstay is considered as primary part to stabilize local vibration, which supports the boiler wall tubes against the furnace gas pressure that can be either positive or negative with respect to the local atmospheric pressure. The buckstay is modeled as shown in figure 5. Equivalent buckstay stiffness which is used in bush (1-D) element that has the stiffness in longitudinal and vertical directions is deducted through displacement of modeling when unit force is applied on the buckstay. Also, analytical boiler model with boundary conditions is shown in Figure 6.

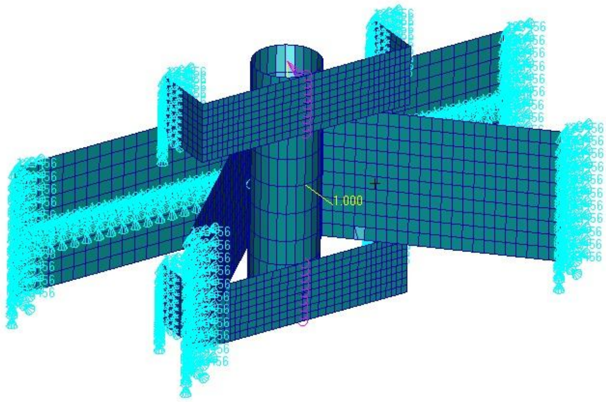


Figure 5 – Buckstay Modeling

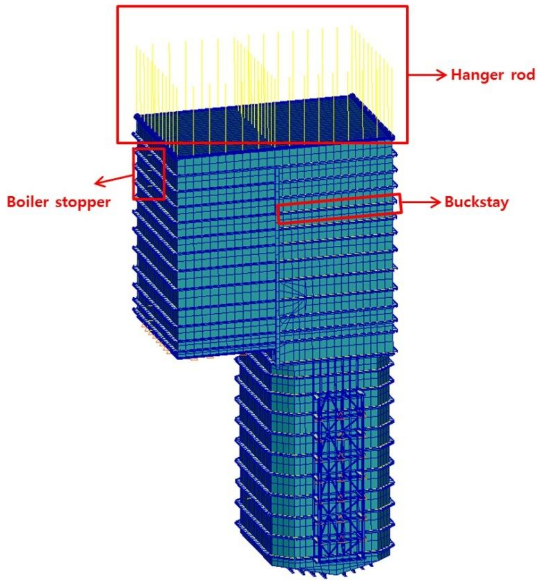


Figure 6 – Vibration Modeling of Boiler

4.2 Normal Mode Analysis

The natural frequencies and their corresponding mode shapes of analytical model were calculated using the general purpose structure analysis program MSC/NASTRAN. The input file for the MSC/NASTRAN was generated from the model data file of the preprocessor, MSC/PATRAN. Frequency range for the eigenvalue solution was set to 0.1 ~ 5.0 Hz to exclude rigid body modes and consider excitation pressure characteristics.

The FRF (Frequency Response Function) is a transfer function, expressed in the frequency-domain and is a mathematical representation of the relationship between the input and the output of a system[3]. It expressed the structural response to an applied force as a function of frequency, which could be information as the corresponding natural frequency. Therefore FRF is an efficient way of figuring out whether resonance occurs or not, through referring input force frequency.

The frequency response function model in linear system is illustrated in Figure 7 and it can be represented by the following equations (1).

$$H(\omega) = \frac{X(\omega)}{F(\omega)} \tag{1}$$

$H(\omega)$: Transfer function
$F(\omega)$: Input force as a function of the angular frequency ω
$X(\omega)$: Displacement response function

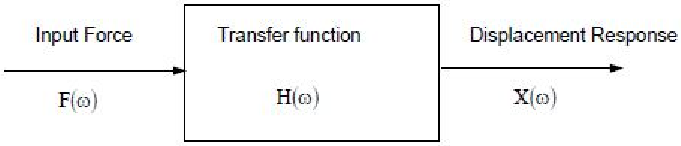


Figure 7 – FRF in linear system

Several vibration modes were identified in resonance region (2Hz~5Hz) according to CFD calculation results and acoustic mode analysis. The natural frequency of representative location that is shown in Figure 8 is different due to the wall tube arrangement and its direction. Therefore, vibration levels at representative location should be confirmed through forced vibration analysis, considering excitation forces based on CFD calculation.

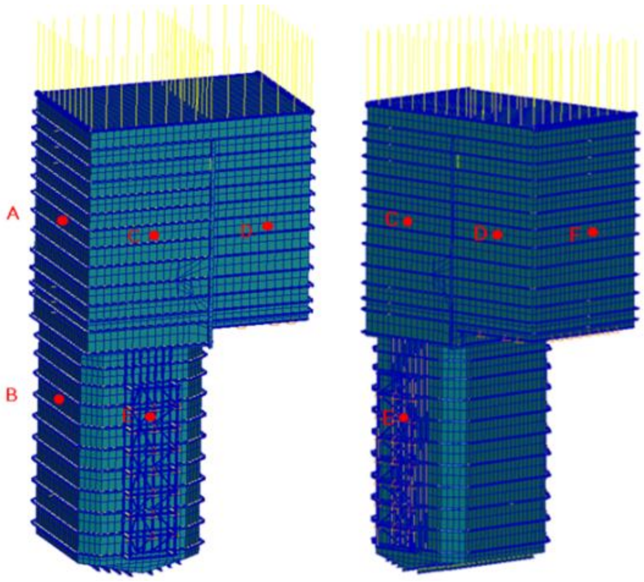


Figure 8 – Representative location for vibration confirmation

4.3 Forced Vibration Analysis

Forced Vibration Analysis using the direct transient method by MSC/NASTRAN was performed considering the pressure fluctuation from CFD calculation.

Transient response analysis is the most general method for computing forced dynamic response. The purpose of a transient response analysis is to compute the behavior of a structure subjected to time-varying excitation. The excitation is explicitly defined in the time domain. All of the applied forces are known at each instant in time. In direct transient response, structural response is computed by solving a set of coupled equations using direct numerical integration as below equation (2).

$$[M]\{\ddot{u}(t)\} + [C]\{\dot{u}(t)\} + [K]\{u(t)\} = \{P(t)\}$$
$$\{\dot{u}_n\} = 1/(2\Delta t)\{\dot{u}_{n+1} - u_{n-1}\}$$
$$\{\ddot{u}_n\} = \frac{1}{\Delta t^2\{u_{n+1} - 2u_n + u_{n-1}\}} \quad (2)$$

Beginning with the dynamic equation of motion in matrix form based on the analysis results, the dominant vibration peak is found at 2.4 Hz which is component of excitation force from CFD calculation. The direct transient analysis was carried out considering fluctuating pressure by MSC.NASTRAN and FFT (Fast Fourier Transform) is also calculated to verify frequency component of vibration responses.

The maximum vibration level of furnace is 6.1 mm/s at F location, secondary pass rear wall, and its frequency is 2.4 Hz from combustion excitation frequency. Vibration levels of other positions are below 4.5mm/s which are considered as low value in plant structures. Therefore it is expected that the furnace structure has no vibration problem in normal operating condition

5. CONCLUSION

The characteristics of fluctuating pressure in the furnace were investigated through CFD calculation. The fluctuation pressure in the furnace has tonal frequencies of 2.4 Hz and 7.0 Hz which are corresponding to 1^{st} and 2^{nd} acoustic modes. The maximum amplitude of the fluctuating pressure is 400 Pa and the amplitude level keeps constant and does not increase with respect to time. This confirms indirectly that there is no high noise level during operation.

The vibration analysis for the furnace structure was performed using FE model and fluctuating pressure. From the normal mode analysis results, the possibility of resonance was not expected considering fluctuating pressure. The overall vibration levels of the boiler are expected to be below 7 mm/sec in average value. From the vibration analysis results using CFD calculation and acoustic study, the furnace structure of super critical boiler is expected to have no vibration problem because there is no thermo-acoustic resonance and no the structure resonance.

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