INTRODUCTION OF JAXA TOOL FOR RANDOM VIBRATIONS PREDICTION AND ITS RECENT UPGRADING

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OVERVIEW

JAXA tool, JANET, available through WEB browser, was developed to predict the acoustic induced random vibration level of equipments mounted on satellite panels a few years ago.
JANET, was a simplified design tool to support system design in the early phase to define the equipment random vibration environment, based on the SEA (Statistical Energy Analysis) concept and equipment dynamic approximation. Prediction error of narrow frequency band as well as spatial distribution were statistically obtained by thousands of acceleration data from test results of historical satellites, and defined conservative envelopes.

Various prediction methods have been developed in JANET: (i) Lewis method, (ii) Point mass impedance method, (iii) Empirical formulation method. In the recent upgrading, a new method combining the asymptotic apparent mass of specific equipment with Lewis method is proposed. This method takes the elastic behavior of satellite equipment instead of rigid mass into consideration. The acoustic excitation experiments for nine real satellites (404 equipment total) are conducted to compare the exiting methods with the new method in statistical sense. The result from the comparison shows that the new method provides most accurate prediction in important frequency range.

1. INTRODUCTION

Payload and equipment mounted on honeycomb panels of artificial satellite are exposed to intense random vibro-acoustic environment during launch. In process of payload design, especially in concept design phase and preliminary design phase, as shown in FIG 1, random vibro-acoustic environment for components and sub-components are defined based upon analytical prediction or experimental database empirical approach. This defined specification describes how payload components vibrate in the acoustic excitation along with certain margin regarding uncertainties. Since payload response under acoustic testing environment is of high frequency in random, empirical approach may not give a satisfactory definition for the nonsimilarity structures. The structural model(SM) is used to revise the initial random vibration level of components from the acoustical testing. The re-definition of random vibration environment at interfaces may impact the design circle, schedule and cost of spacecraft. The web-based JANET tool offers the mechanical design engineer to predict random vibration level of components induced by acoustic environment with a satisfactory accuracy in the preliminary design phase in order to reduce the impact of interface random vibration environment revision after SM(Structural Model) acoustic testing. The vibro-acoustic random vibration analysis tool are composed of three functions: analysis function, in which, three prediction methods (JAXA data based empirical method, NASA Lewis method[1], and Improved Impedance method, which was upgraded in March, 2007), are available with data-based margin; the comparison of PSD results between analysis and test in 1/3 Octave band and narrow band; update the current analysis parameter's data base with the acoustic testing result.

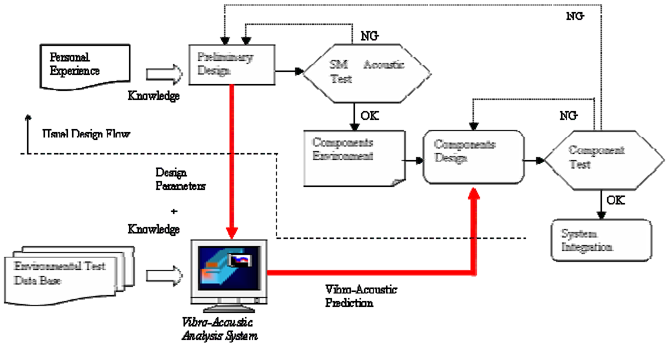

FIG 1. Diagram of random vibration design and verification of spacecraft

2. OUTLINE OF JANET SYSTEM

The JANET system is consists of three major subsystems as described in FIG 2. The first one is the analysis core subsystem, includes three analysis approaches to support the different design purposed according to how detailed parameters the designer has in hand. The second one is the data base which includes the test result of component mounted on a honeycomb panel, the footprint area and mass of the component, material property of honeycomb panel, etc. The third one is the general data base subsystem which can download the ETS (Environmental & Structural Test Data management System), which is the general structural environment testing data base of all JAXA project spacecraft from 1990's.
The web-based JANET system supplies the qualified user to access the system from external JAXA internet by web browser to analysis the vibro-acoustic random response of defined acoustic sound pressure level, search and download the analysis result and compare the analysis result to the test for that the acoustic test has been completed. The limitation of access is defined by the security and agreement of projects. Data-base updating is permitted to the administrator only.
After the login menu, the system main menu supplies the user with data management; go to the next menu for analysis, operation help in friendly GUI.
The analysis menu offers the user with 3 different analyses

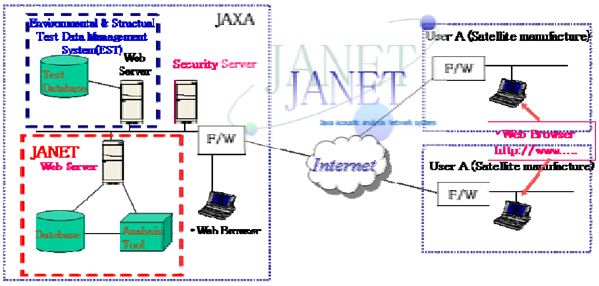

FIG 2. Configuration of JANET systems

approach choices, the methodology of details will be described afterwards. Each analysis approach supplies you to the maximum 5 parametric studies as shown in FIG 3. The parametric studies are the parameters of honeycomb panel (skin parameters, core parameters, component parameters, etc) to find a 'better' design. The acoustic environment load (SPL: Sound Pressure Level) can be defined and default values (AT,QT) for the major launch vehicle in the world are supported in the system. The analysis parameters and results can be stored in CSV format. The analysis result supplies the user with normal (average) and statistical margin (P95/50, P99/90) in 1/3Octave band and narrow band PSD. The applicable low frequency limitation is shown in the graphic expression. Example of graphic expression of three methods are shown in FIG 4. The comparison of analysis and test results and interface specifications of preliminary and critical design are shown in the same graphic as the user chosen.

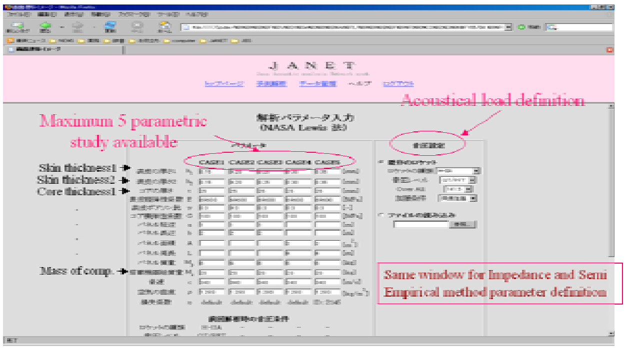

FIG 3. Parameter definition window for parametric study (same window for all analysis approaches)

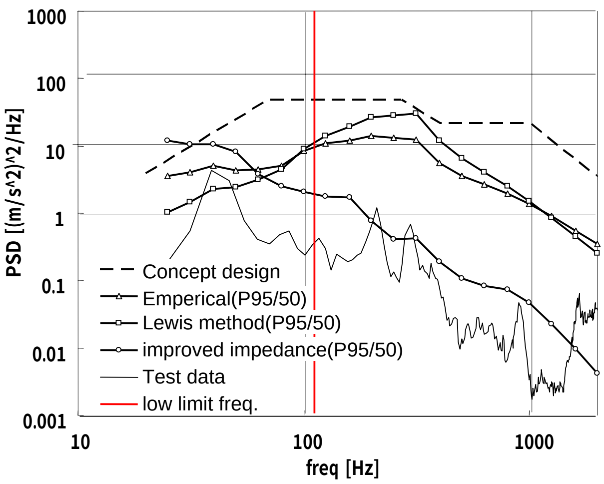

FIG 4. Example of graphic expression of three methods in JANET

3. INTRODUCTION OF ANALYSIS AND UPGRADE METHODS IN JANET

There are three methods available that enables the designer to predict a random vibro-acoustic response of a honeycomb panel and equipment mounted on the panel shown in FIG 5 with reasonable margin in the early stage of design. Three methods are based on the simplified statistical energy approach (SEA) in the assumption that the vibro-acoustic load of panel is dominated by the acoustics rather than the structure. Therefore, the coupling between structural subsystems is neglected in SEA model, the two subsystem of acoustic and panel with instrument components are modeled in SEA. Damping loss factor of honeycomb panel is obtained by 11 honeycombs with different parameters of skin and core thickness, materials, etc. and summarized in the internal data-base in JANET.

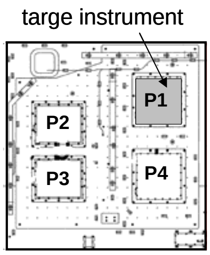

FIG 5. Example of satellite equipment panel

3.1. EMPIRICAL METHOD

This method is induced based on two subsystems of SEA model, to simplify the necessary parameters used for prediction. The coefficient of empirical method, called JANET coefficient is obtained from the acoustical random vibration test data of several satellites. The time-space average acceleration of honeycomb panel with components mounted is calculated from Equation(1).

$$<a^2> = J\_coef \frac{c^2 <p^2>}{\eta \cdot f}\sqrt{\frac{1-\nu^2}{Em^3t^2h}} \qquad (1)$$

where, J_coef is JANET coefficient obtained from several satellites test data, shown in FIG 5 which is a function of frequency, ν is Poisson ration, t is core thickness, E is Young's module of skin, m is the area mass of component panel, f is frequency, h is the average thickness of honeycomb skin, $\eta = 1/f^{0.7}$ is the damping loss factor of panel, which was measured from tens of honeycomb panels.

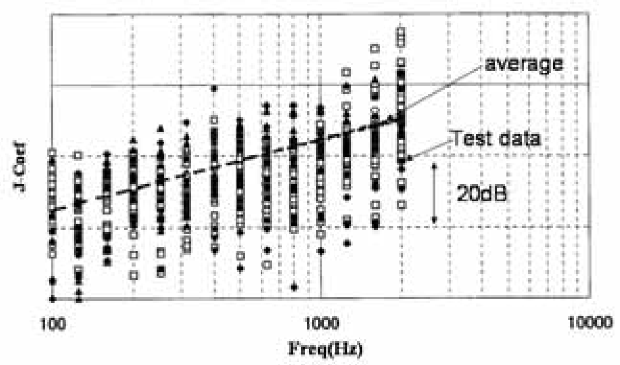

FIG 6. JANET coefficient vs. testing data

3.2. NASA Lewis method

This method assumes that the instruments and harness, pipes etc. mounted on a panel shown in FIG 5 do not change the modal density, critical frequency, damping loss factor of a panel before mounted and may be equivalent to a uniform panel[1]. According to the assumption, the mounted equipment is uniformly melted into the panel and equivalent structural parameters of a uniformed panel are calculated. The time-space average acceleration of the panel may be calculated in Equation(2), when a reverberant sound pressure is loaded.

$$\langle a^2 \rangle = \frac{2\pi^2 c n_2 \langle p^2 \rangle}{\rho_0 (M + M_c)} \times \frac{1}{1 + \eta (M + M_c)\,\omega / 2\rho_0 c S\,\sigma_{rad}} \tag{2}$$

where, n2 is modal density of panel, M, Mc are the masses of panel and component, ρ_0 is air density, σ_{rad} is the radiation coefficient, <a2> and <p2> are the acceleration and acoustic time-space average, ω is frequency, c is speed of sound.

3.3. Improved impedance method (upgraded)

Impedance method assumes that the instrument mounted on panel may be treated by a dumped impedance, which is the apparent mass(force/acceleration) at the mounted interface and the coupled to panel can be calculated by their apparent mass of each. This is based on the principle that when an item of equipment is attached to the panel, the acceleration response at the mounting points of the equipment with and without the equipment present is related by the impedance ratio of panel and equipment. The time-space average acceleration of the panel can be calculated in Equation(3), when a reverberant sound pressure is loaded.

$$\langle a^2 \rangle_L = \left| \frac{1}{1 + Z_m / Z_p} \right|^2 \langle a^2 \rangle \tag{3}$$

where, Z_m and Z_p are the impedance of instrument and panel at the mounted interface individullly, $\langle a^2 \rangle_L$ and $\langle a^2 \rangle$ are the time-space average acceleration of loaded with and without instrument.

The impedance of panel is approximated by $Z_p = 2M\omega n_2 / \pi$ for infinite panel, where M and n_2 are the mass and mode density of infinite panel. The impedance of instrument is approximated by $Z_m = M_c$ for a rigid mass. Therefore, the vibration response of instruments under the sound pressure excitation may be calculated by the NASA Lewis method for those except the target instrument P1 shown in FIG 5 may be applied by Equation(2), after then the target instrument may be calculated from the Equation(3) to obtain formula with Equation(4).

$$\langle a^2 \rangle_L = \frac{2\pi^2 c n_2 \langle p^2 \rangle}{\rho_0 M_p} \times \frac{1}{1 + \eta_2 M_p\,\omega / 2\rho_0 c S\,\sigma_{rad}} \times \frac{1}{1 + \left(\frac{M_c}{M_p} \frac{\pi \omega n_2}{2} \right)^2} \tag{4}$$

Where, M_p is the mass of panel, M_c.

However, the analysis result on the assumption that the instrument was a rigid mass may under-estimate the vibration response due to the over-estimation of apparent mass in high frequency shown in FIG 6. The improved impedance method which treats the instrument impedance as a frequency dependant value obtained from Monte Carlo simulation (500 samples) by 50 degrees of freedom model shown in FIG 8. The impedance of the model defined by force to acceleration(F/A), was calculated and smoothed in 1/1 Octave band frequency corresponding to damping ratio 1%, 5%, 10% and is shown in FIG 9, in which horizontal axis is the frequency normalized to the first mode frequency, vertical axis is the impedance normalized in rigid mass.

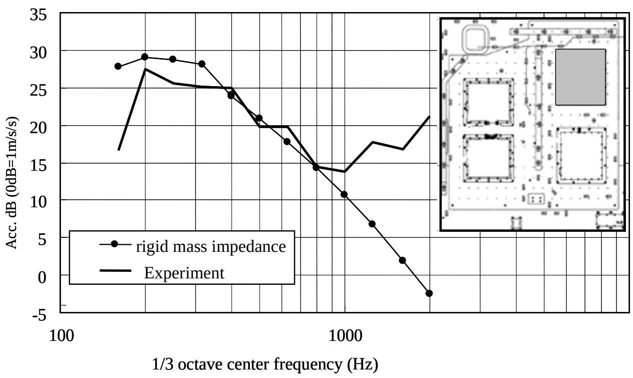

FIG 7. Example of impedance method

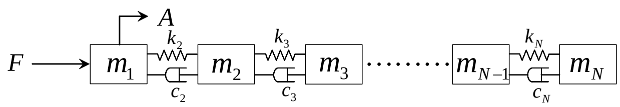

FIG 8. 50 degrees of freedom dedicated to Monte Carlo simulation

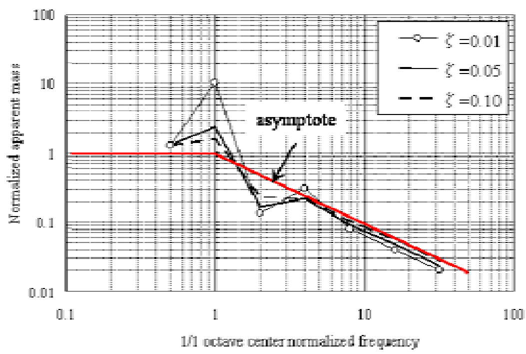

FIG 9. Frequency averaged normalized impedance

The asymptote impedance of instrument can be summarized by the approximation in Equation(5).

$$\overline{M}_c = \begin{cases} M_c, & \omega < \omega_0 \\ M_c(\omega_0/\omega), & \omega \geq \omega_0 \end{cases} \quad (5)$$

Where, M_c is the rigid mass, ω_0 is the first mode frequency in radian.

The improved impedance method uses frequency dependant apparent mass instead of rigid mass and leads the prediction formulation to Equation(6).

$$\langle a^2 \rangle_L = \frac{2\pi^2 c_0 n_2 \langle p^2 \rangle}{\rho_0 (M_{tot} - M_c)} \times \frac{1}{1 + \eta_2 (M_{tot} - M_c)\,\omega / 2\rho_0 c_0 S\, \sigma_{rad}} \times g(\omega) \quad (6)$$

$$g(\omega) = \begin{cases} \dfrac{1}{1 + \left(\dfrac{M_c}{M_p} \dfrac{\pi \omega n_2}{2} \right)^2} & (\omega < \omega_0) \\ \dfrac{1}{1 + \left(\dfrac{\pi \omega_0 n_2 M_c}{2 M_p} \right)^2} & (\omega \geq \omega_0) \end{cases}$$

Where, M_{tot} is the total mass of panel and all instruments, harness on the panel, M_c is the target instrument mass to be predicted.

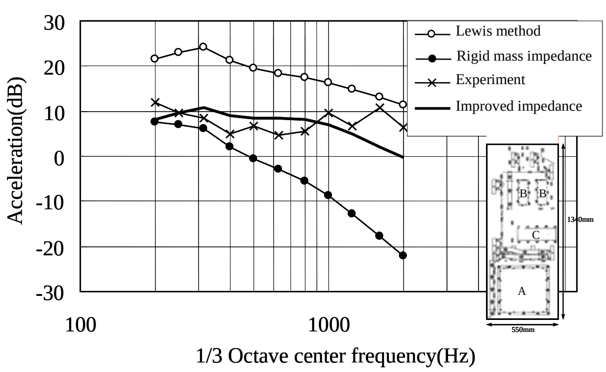

FIG 10. Comparison of methods stated in the paper

An example of comparing results of methods stated in this paper is shown in FIG 10. The improved impedance method which assigns the first mode frequency to 100Hz in the case gives a good agreement to the experiment result than the rigid mass impedance, especially in high frequency range. The prediction accuracy of each method is summarized statistically from nine application satellites including communication, observation and experimental satellite, whose item is listed in Table 1. The statistical time-space average error in logarithm expression is plotted in FIG 11.

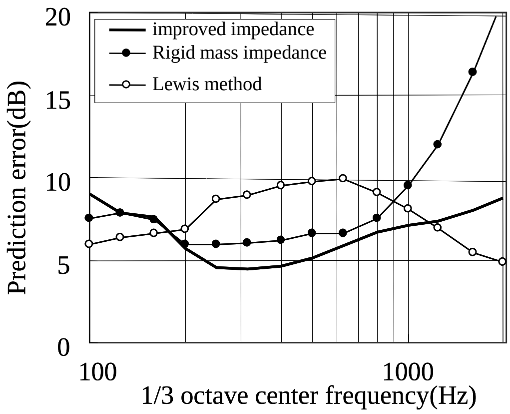

FIG 11. Comparison of prediction errors

It is clarified that the error of increases dramatically in high frequency due to overestimation of the impedance at high frequency, Lewis method offers good estimation between 5-10dB over all frequency, however it is poor at 200-800Hz than rigid mass impedance, improved impedance method gives the best estimation especially in the frequency range 200-1000Hz, in which most instrument's critical frequency exists. The first mode frequency is assigned to 100Hz for improved impedance method for all instruments used in this paper, the estimation result approaching to that of rigid mass impedance. The improved impedance method estimation is a few worse than Lewis method in high frequency above 1000Hz.

4. MARGIN OF ANALYSIS RESULT

The discussion in the previous section is based on time-space average which is the space average value in octave frequency band. However, limit value for each individual instrument in narrow frequency band, which is the interface specification is usually used in space engineering. The definition of interface specification in narrow band PSD(power spectrum density) includes space limit margin and narrow frequency limit margin. These limit margins would be defined by the external data file or internal statistical results from data base. The internal statistical margin (P90/50, P99/90) in JANET is calculated from 90 panels and more than 300 samples from nine satellites of JAXA projects. The statistical result is obtained by the logarithm Gaussian distribution of acceleration in space and in frequency band. The upper-tolerance limit in logarithmic expression (dB) is defined as the value which may not be exceeded at a portion of β percentage of all population, with a confidence coefficient γ, and is given by,

$$P\beta / \gamma = \overline{\mu} + k_{n,\beta,\gamma} \hat{\sigma} \quad (7)$$

Where, n is the number of samples. $\overline{\mu}$ is the logarithm sample average in logarithm(dB), $\hat{\sigma}$ is the logarithm sample standard deviation, $k_{n,\beta,\gamma}$ is the normal tolerance factor given from a cummulative distribution function.

The internal statistical upper-tolerance limits for both 1/3 octave and narrow frequency bands PSD are obtained from 90 panels and more than 300 samples from nine

satellites of JAXA projects shown in TAB 1. The ensemble normalized variance of PSD is obtained and P95/50 PSD margin P95/50 is 3.6 dB in JANET internal definition.
The prediction example for a instrument C with 17kg on a panel with total instruments weight 55kg is shown in FIG 12 by three methods in JANET including space and narrow frequency band margin (P95/50).

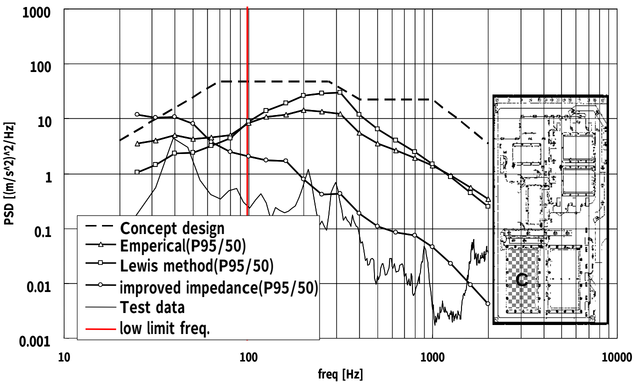

FIG 12. Prediction example by three methods in JANET including space and narrow frequency band margin (P95/50).

5. SUMMARY REMARKS

The web-based JANET tool, developing in March 2005 and upgraded in 2006 with improved impedance method, offers the mechanical design engineer to define the random vibration level of components induced by acoustic environment with a satisfactory accuracy in the preliminary design phase, in order to reduce the impact of interface random vibration environment revision and avoid the risk and cost of satellite design. The vibro-acoustic random vibration analysis tool are composed of three functions depend on the different purpose are available with data-based margin; the comparison of PSD results between analysis and test in 1/3 Octave band and narrow band; update the current analysis parameter's data base with the acoustic testing result. Several projects utilized JANET in the early design.

REFERENCE

[1] Mark E. McNelis, A modified VAPEPS Method for Predicting Vibroacoustic Response of Unreinforced Mass Loaded Honeycomb Panels, NASA-TM-101467 (1989)
[2] S. Ando, Q. Shi, K. Nagahama, M. Saitoh, H. Saegusa, the Prediction of Random Acoustic Vibration of Equipment Mounted on Honeycomb Panel, 5th ESA Aerospace Environmental Testing Symposium, Belgium (June, 2005)

TAB 1. The properties of satellites dedicated to the analysis

Property	Sat. A	Sat. B	Sat. C	Sat. D	Sat. E	Sat. F	Sat. G	Sat. H	Sat. I
Satellite type*	CS	OS	OS	CS	ES	CS	CS	OS	OS
Number of equipment panels	25	12	15	3	5	15	9	7	3
Number of equipment	81	50	31	21	30	88	52	37	14
* CS: communication satellite, OS: observation satellite, ES: experimental satellite									