Applications of Transient Acoustic Simulation in LS-DYNA

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Abstract: The engineering rattle and squeak noises have obvious transient noise characteristics. Using a transient acoustic simulation method to analyze such noise problems is more suitable for studying the mechanism and sound radiation process. In common CAE software, vibration calculation and acoustic calculation are often separated. The vibration results of the finite element model need to be transformed and then mapped to the boundary element for further acoustic calculation. LS-DYNA has integrated the vibration calculation and acoustic calculation in a single executable file, which simplifies the establishment of the model and avoids the possible errors in the mapping and time-frequency transformation process of result data files. In this paper, LS-DYNA is used to solve some transient acoustic problems, starts from the simulation of cantilever beam and conical spring rebound noise, and further to the rattle noise simulation of the car sensor components in the seatbelt retractor under given random vibration excitation, the feasibility and superiority of LS-DYNA in the applications of transient noise simulation is verified.

Keywords: transient noise; rattle noise; seatbelt retractor

摘要： 工程中遇到的碰撞、摩擦噪音，具有明显的瞬态噪音特性。使用瞬态声学仿真方法对此类问题进行分析，能够更准确的模拟噪声问题产生的机理和声辐射过程。在一般的仿真软件中，其振动计算和声学计算往往相互独立，为了求解瞬态噪声，需要将有限元模型的振动结果时频转换后再映射到边界元声学网格上再进行计算。LS-DYNA 将振动计算与声学计算集成在一起，即方便了模型的建立，也避免了在对结果文件进行映射和时频转换过程中可能产生的错误。本文使用 LS-DYNA 软件，从简单的悬臂梁、圆锥弹簧回弹声仿真出发，并进一步对安全带的卷收器车敏组件在给定随机振动激励下的碰撞噪声进行了仿真，验证了 LS-DYNA 在对瞬态噪声问题进行仿真的可行性和优越性。

关键词：瞬态噪声；碰撞噪声；安全带卷收器

0. Introduction

The words squeak and rattle are generally used for describing short duration transient noise in automobile engineering. Squeak is generally caused by stick-slip motion between parts in frictional contact, while rattle refers to the impact between parts. The dynamic of squeak and rattle involve complicated stick-slip, a highly nonlinear phenomenon, or a periodical nonlinear impact. A nonlinear CAE model with complicated contact elements to fully simulate the dynamic contact behavior in needed to solve these problems [1].

LS-DYNA is originally designed for transient dynamic analysis of highly nonlinear problems. LS-DYNA has advantages on its explicit solver and contact problem calculations. Furthermore, LS-DYNA "combines the multi-physics capabilities into one parallel scalable code for solving highly nonlinear transient problems to enable the solution of coupled multi-physics, multi-scale, and multi-stage problems". Most of the functions in LS-DYNA are integrated into one code, instead of being divided into multiple modules. The "one model, one code" concept simplifies the establishment of the model and avoids the possible errors in the mapping and time-frequency transformation process of transforming time domain vibration data files in to frequency domain acoustic boundary conditions.

In this paper, a typical transient acoustic simulation process in LS-DYNA is described. After that, three examples are illustrated. The feasibility and superiority of using LS-DYNA for transient noise simulation is discussed.

1. Transient Acoustic Simulation

1.1 Background theory

Ignoring sound-structure coupling effect, a transient acoustic simulation usually consists of two sequent process: a highly non-linear transient dynamic process in time domain and a linear sound propagation process in frequency domain. The dynamic problem is expressed by partial differential equations (PDEs) of the structure. For computer calculation, the PDEs have to be transformed into their matrix form as:

$$[\mathbf{M}]\{\ddot{\boldsymbol{x}}\} + [\mathbf{C}]\{\dot{\boldsymbol{x}}\} + [\mathbf{K}]\{\boldsymbol{x}\} = \{\boldsymbol{f}\} \tag{1}$$

where $[\mathbf{M}]$, $[\mathbf{C}]$ and $[\mathbf{K}]$ are mass, damping and stiffness matrix of the structure, $\{\boldsymbol{x}\}$ is the displacement vector and $\{\boldsymbol{f}\}$ is the loading vector.

For squeak and rattle problems, the non-linearity of the model basically belongs to the boundary nonlinearity category, which due to the non-linearity of the boundary conditions in contact problems. Eq. (1) will change to:

$$[\mathbf{M}(\boldsymbol{x})]\{\ddot{\boldsymbol{x}}\} + [\mathbf{C}(\boldsymbol{x})]\{\dot{\boldsymbol{x}}\} + [\mathbf{K}(\boldsymbol{x})]\{\boldsymbol{x}\} = \{\boldsymbol{f}\} \tag{2}$$

Explicit analysis aims to solve for acceleration. Once the accelerations are calculated at the *n*-th step, the velocity at $n + 1/2$ step and displacement at $n + 1$ step are calculated accordingly.

The wave equation describing sound propagation in three dimensions is as following:

$$\nabla^2 p - \frac{1}{c^2}\frac{\partial^2 p}{\partial t^2} = 0 \tag{3}$$

where ∇ is the Laplace operator, p is the acoustic pressure, and c is the speed of sound. Eq. (3) can be transformed in to frequency domain as:

$$\nabla^2 p + k^2 p = 0 \tag{4}$$

where $k = \omega/c$ is the wave number, $\omega = 2\pi f$ is the circular frequency.

Based on Green's theorem, the Eq. (4) can be transformed into an integral equation. Then the pressure at any point in the fluid medium can be expressed as an integral of surface pressure and surface velocity of a vibrating structure by following equation:

$$\boldsymbol{P}_Q(\omega) = -\int_s \left(i\rho\omega\boldsymbol{v}_n(\omega)\boldsymbol{G} + \boldsymbol{p}(\omega)\frac{\partial\boldsymbol{G}}{\partial n}\right) d\boldsymbol{S} \tag{5}$$

where $\boldsymbol{P}_Q(\omega)$ is the sound pressure at field point Q, $\boldsymbol{S}$ is the structure surface, $\boldsymbol{v}_n(\omega)$ is the normal vibration velocity on structure surface, $\boldsymbol{p}(\omega)$ is the surface pressure. $\boldsymbol{G}$ is the Green's function [2].

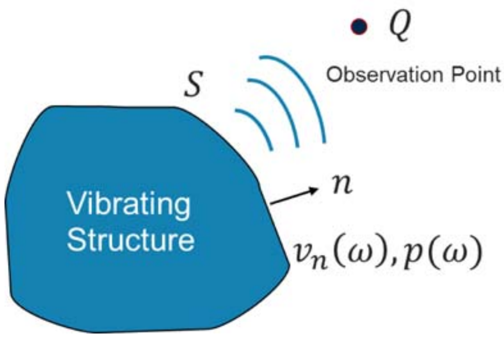

Figure 1. Vibrating structure

1.2 Calculation process

As shown in Fig. 2, for a typical transient acoustic calculation, a normal transient dynamic analysis is firstly done by an explicit solver, the time domain structure velocity or acceleration results are obtained. Then an FFT algorithm is used to transform the time domain velocity or acceleration data into frequency domain. The transformed data can be directly used or mapped to boundary element model and then use the boundary element method for acoustic calculation.

The acoustic calculation is done by using *FREQUECNY_DOMAIN_ACOUSTIC_BEM card. In this keyword, firstly, the boundary element model is defined, including the vibrating structure surface, acoustic medium density, sound speed and the observation point (field point) location. The boundary element method provided by LS-DYNA is done in frequency domain, so the calculation frequency range needs to be defined as well. If time domain results are requested, the lower limit frequency should be set to 0 Hz.

LS-DYNA has provided many kinds of databases for the post-processing of the acoustic analysis results, including binary files for fringe plot and ASC2 files for xyplot. All these database files can be accessed by LS-PrePost. Ref. [3] gives a detailed review of these databases.

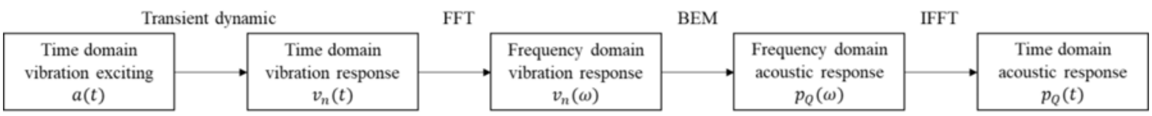

Figure 2. Typical transient acoustic calculation process flow chart in LS-DYNA

2. Numerical Applications

2.1 Cantilever Beam

A cantilever beam model is adopted here to illustrate the transient acoustic simulation. For simplicity, the cantilever beam is assumed to be clamped at one end and subjected to a uniform normal force at the other end. As shown in Fig. 3, the length of the cantilever model $l = 43.72mm$, the thickness of the beam $h = 1mm$. For a clamped-free cantilever, its natural frequency can be calculated from following equation:

$$f_n = \frac{\Omega^2}{2\pi}\sqrt{\frac{E}{12\rho}}\cdot\frac{h}{l^2} \tag{6}$$

where Ω is the frequency parameter of a beam, for a clamped-free cantilever, $\Omega = 1.875$. For the cantilever model describe here, its natural frequency can be calculated from Eq. (6), results as $f_n = 440Hz$. In order to excite at least one mode of the cantilever model, the force loading has to be an impulse-like force, the duration of the force is 12.5ms. A field point is set 500mm away above the cantilever beam.

*DAMPING_GLOBAL is added to define a mass weighted nodal damping, in this card, VALDMP defines the constant damping factor, which can be obtained from damping ratio η by

$$\mathrm{VLADMP} = 4 * \pi * f_{m1} * \eta \tag{7}$$

where f_{m1} is the natural frequency of the lowest mode of interest. For a 0.5% damping ratio of this cantilever beam model, the value of VLADMP equals to about 27.6.

As discussed in Sec. 1.2, *FREQUECNY_DOMAIN_ACOUSTIC_BEM is added to activate the boundary element method in frequency domain. The value of the parameters in the cantilever beam model are shown as in Fig. 4. The acoustic medium is the air. Considering the frequency range of human hearing, the upper limit frequency is set to 20000Hz. Time domain results are requested so the lower limit frequency is set to 0 Hz. The inverse of DTOUT denotes the sampling rate, which should be larger than two times of the frequency upper limit according to Shannon's sampling theorem. PREF defines the reference pressure for the computation of dB result, which equals to $20\mu\mathrm{Pa}$ for air. An FFT algorithm is used to transform time domain velocity or acceleration data into frequency domain. In order to overcome the FFT leakage problem, a window function can be used. In this model the vibration of the cantilever beam has been fully attenuated by the end of the transient analysis, so no window function is need and the FFTWIN can be set to 0, otherwise a window function is always needed. TRSLT is set to 2 for outputting the time domain results, DBA is set to 1 for outputting the A-weighting dB result. The other parameters left in blank are just using the default value.

Fig. 5 shows the radiated sound pressure at the field point, a very typical oscillation attenuation process can be seen. It can be saved as a .**wav** audio files by the plot tool provided in LS-PrePost, to generate a playable sound file. Fig. 6 shows the spectrum of the radiated sound pressure in dB(A), several obvious peak can be seen, the peak frequencies consistent with the natural frequencies of out-of-plane modes of the cantilever mode, as shown in Fig. 7.

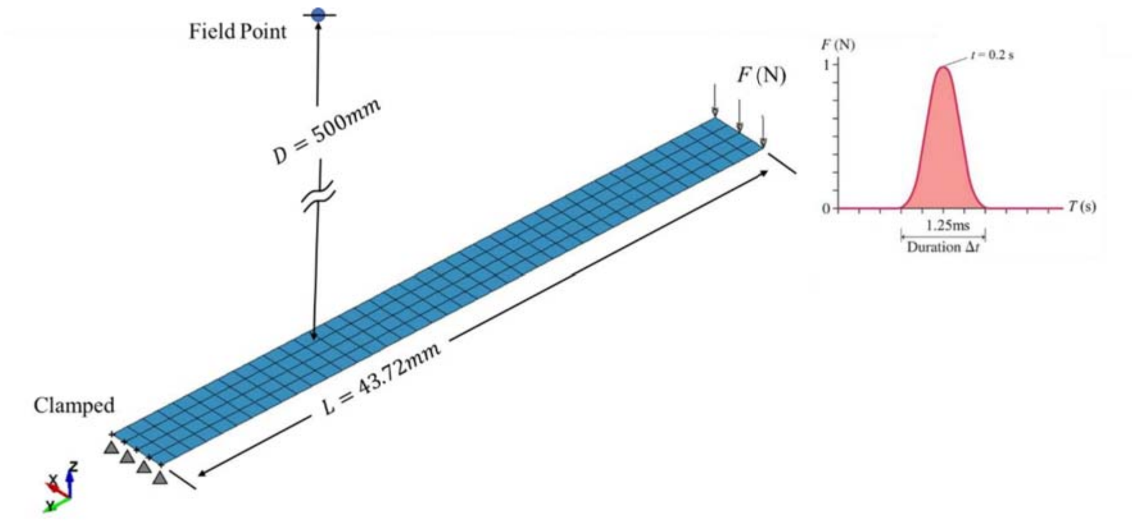

Figure 3. A cantilever beam model

Card 1	RO	C	FMIN	FMAX	NFREQ	DTOUT	TSTART	PREF
	1.21	343	0	20000	4000	2.5e-5	0	2e-5
Card 2	NSIDEXT	TYPEXT	NSIDINT	TYPINT	FFTWIN	TRSLT	IPFILE	IUNITS
	3	1			0	2		
Card 3	METHOD	MAXIT	TOLITR	NDD	TOLLR	TOLFCT	IBDIM	NPG
	2							
Card 4		NBC	RESTRT	IEDGE	NOEL	NFRUP	VELOUT	DBA
		1						1
Card 5	SSID	SSTYPE	NORM	BEMTYPE	LC1	LC2		
	1	2						

Figure 4. Values of parameter used in BEM control card

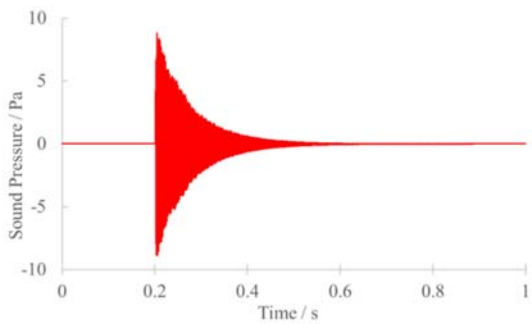

Figure 5. Time history of sound pressure at field point

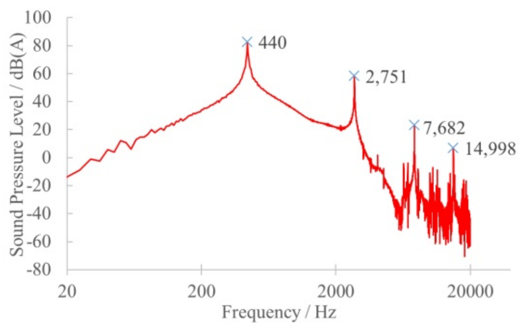

Figure 6. Spectrum of sound pressure level at filed point

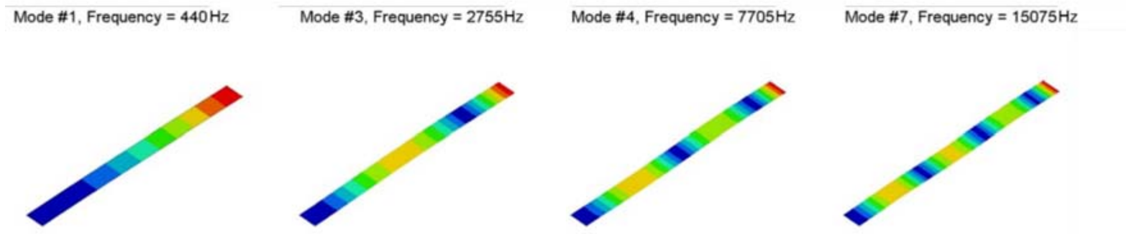

Figure 7. Out of plane mode frequencies and mode shapes of the cantilever model

2.2 Conical Springs

Spring component is widely used in many products, sometimes due to a suddenly lose of contact, it can be excited and will vibrate and radiated "drone" like noise. This problem can be simply improved by replacing the current spring component by another spring component with a higher natural frequency whose rebounding noise would decline faster and easier to insulate.

Fig. 8 shows a finite element model of a conical spring which is clamped on the bottom surface. The spring has a static stiffness of $20\mathrm{N/m}$ with bending mode natural frequency being 153Hz. In order to simulate a suddenly contact lose condition on the top surfaces, a displacement loading is applied in the dynamic relaxation process, then a rebounding process is solved by the explicit solver, the damping ratio is set to 0.5%. A field point is defined 50mm from the central axial of the conical spring, *FREQUECNY_DOMAIN_ACOUSTIC_BEM is then added to calculate the sound radiation, parameters are the same with those as shown in Fig. 4. The radiated sound pressure at the field point is also plotted in Fig. 8, a very typical oscillation attenuation process of the sound pressure can be seen as well.

Fig. 9 shows another conical spring model after optimization. Compare to the original conical spring, the height and the static stiffness are the same. But the natural frequency has raised up to 212Hz by reducing the spring circular number and the diameter of spring wire. Under the same boundary condition, damping, loading and field point position with the previous spring, the radiated sound pressure from the optimized spring is calculated and compared with the original spring in Fig. 9. It is very clear that, due to the higher natural frequency of the optimized conical spring, the sound radiation has the faster decline speed. Besides, higher frequency noise is easier to insulate by surround covers. This makes the optimized conical spring have much better noise behavior than the original one.

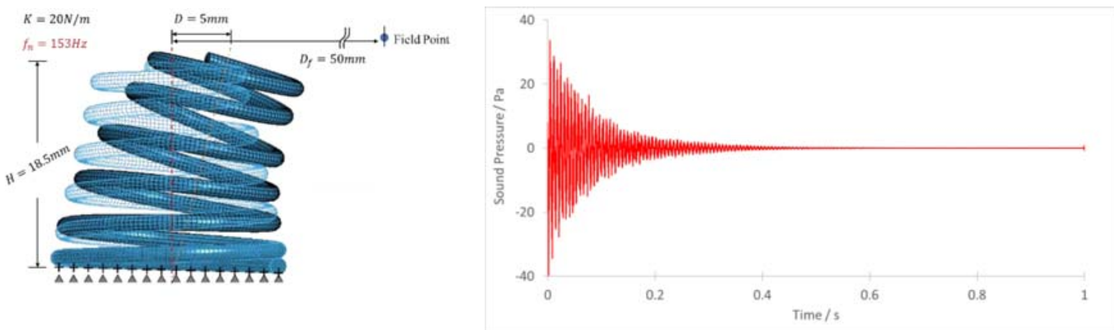

Figure 8. Original conical spring rebound model description and its sound radiation

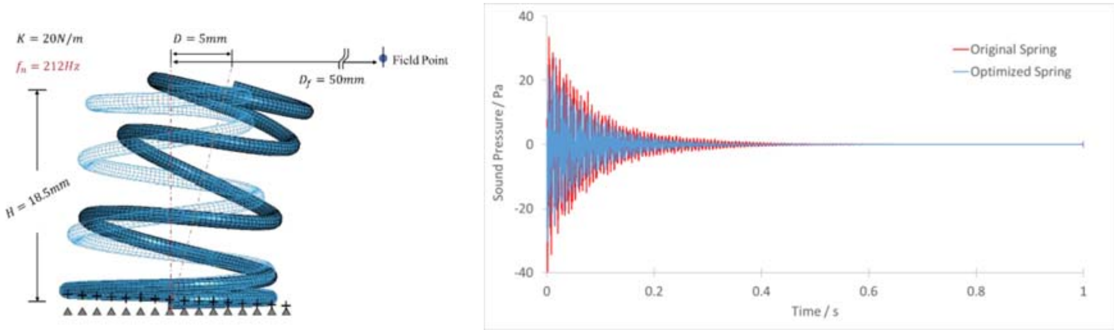

Figure 9. Optimized conical spring rebound model description and sound radiation comparison

2.3 Car Sensor Components in Seatbelt Retractor

Fig. 10 shows the model of a vibrating seatbelt retractor, which consist a car sensor (CS) assembly. The CS assembly is designed for gravity direction detection. In order to achieve this function, a ball that can move freely without any restraint within a housing is used. When the retractor is vibrating, the ball would hit back and forth within the housing and lead to the rattle of the lever as well.

The retractor is excited by a broadband 40 to 100 Hz white noise displacement loading, the other part except for the CS assembly is built into a linear model. Contact conditions are defined only within the parts of CS assembly, as shown in Fig. 11. The lever, housing and ball surfaces are considered as the radiation surfaces and the filed point is set 300mm away from the CS assembly, as shown in Fig. 12.

Fig. 13 shows the spectrum of radiated sound at the field point. Fig. 14 shows the time-history of sound pressure, which can be used to generate the auditable **.wav** file. This shows the feasibility of calculating highly non-linear rattle sound by LS-DYNA. Furthermore, the model parameters of the CS assembly, such as the gap between lever and steering disc, the size of the lever or the material properties of the housing, can be studied for noise radiation reducing design improvement.

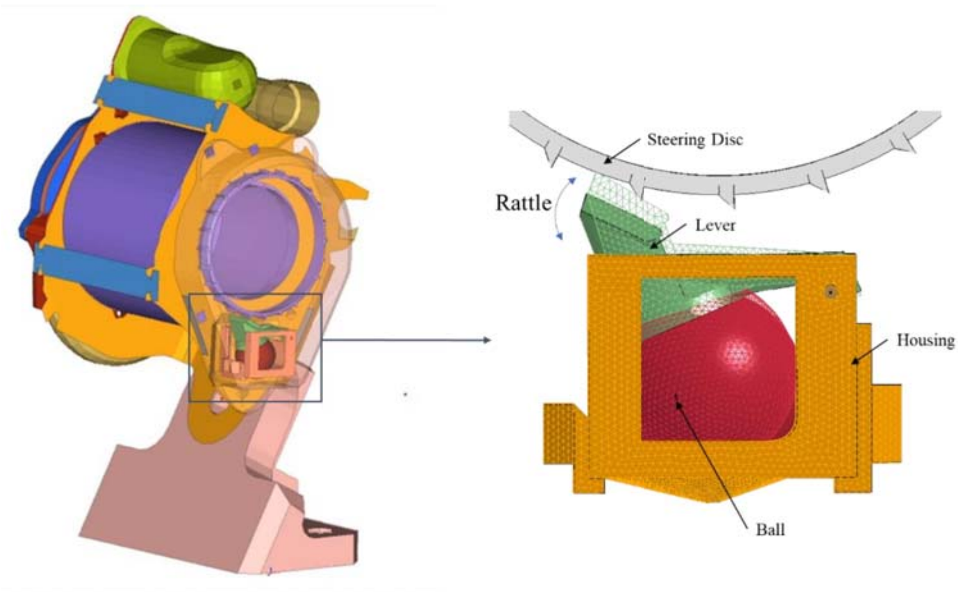

Figure 10. Vibrating seatbelt retractor model and rattling car sensor assembly

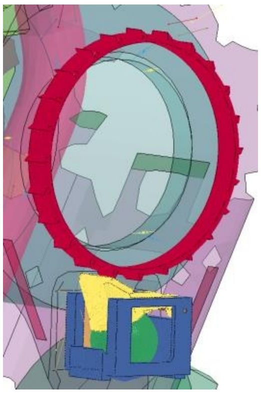

Figure 11. Contact surfaces of CS assembly

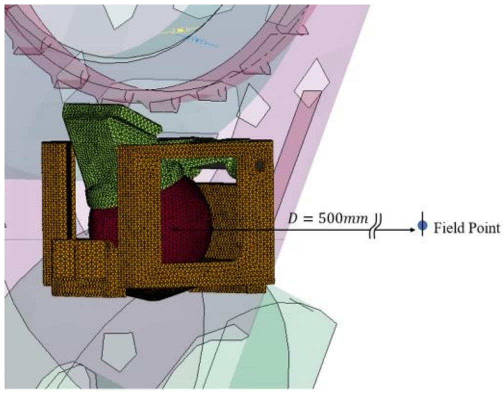

Figure 12. Radiation surfaces of CS assembly

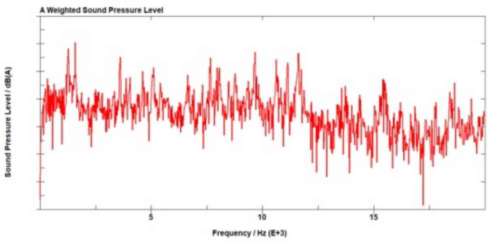

Figure 13. Radiated sound pressure spectrum at field point

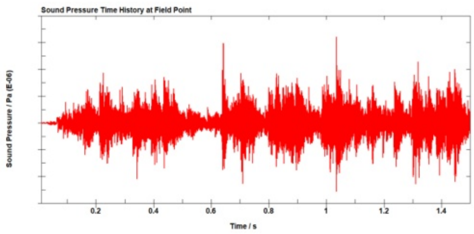

Figure 14. Sound pressure time-history curve at field point

3. Conclusions

In this paper, a typical transient acoustic problem calculation process by LS-DYNA is introduced. Firstly, the time domain structure velocity or acceleration results is obtained through the transient dynamic analysis solved by an explicit solver. Then an FFT algorithm is used to transform time domain velocity or acceleration data into frequency domain, which can be directly used or mapped to boundary element model. At last, controlling by the *FREQUECNY_DOMAIN_ACOUSTIC_BEM card, the boundary element method is used for acoustic calculation. Based on “one model, one code” concept, LS-DYNA has integrated the vibration calculation and acoustic calculation in one code, which has not only simplified the model establishment and avoided the possible errors as well. Several examples are presented, the feasibility and superiority of this method is demonstrated.

4. Reference

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