Sound Radiation Analysis of Tire under The Action of Moving Line Forces

Byoung-Sam Kim*
*Dept of Mechanical and Automotive Engineering, Wonkwang University
e-mail:anvkbs@wonkwang.ac.kr

이동분포하중을 받는 타이어의 음향방사 해석

김병삼*
*원광대학교 공과대학 기계자동차공학부

Abstract

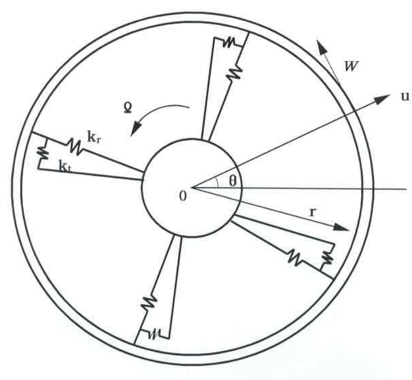
A theoretical model has been studied to describe the sound radiation analysis for structure vibration noise of vehicle tires under the action of random moving line forces. When a tire is analyzed, it had been modeled as curved beams with distributed springs and dash pots that represent the radial, tangential stiffness and damping of tire, respectively. The reaction due to fluid loading on the vibratory response of the curved beam is taken into account. The curved beam is assumed to occupy the plane y=0 and to be axially infinite. The expression for sound power is integrated numerically and the results examined as a function of Mach number, wave-number ratio and stiffness factor. The experimental investigation for structure vibration noise of vehicle tire under the action of random moving line forces has been made. Based on the Spatial Transformation of Sound Field techniques, the sound power and sound radiation are measured. Results strongly suggest that operation condition in the tire material properties and design factors of the tire govern the sound power and sound radiation characteristics.

1. Introduction

The tire noise in automobiles is one of the major sources of noise pollution. Structural vibration in a tire exerts a great influence on automobile noise. Consider the aftermath of a survey about the correlation between tire noise source and its contributing factors. In this paper, the tire is assumed to be curved beam that is considered by basis hardness, basis damping, and tension. Relative sound power that radiates from the curved beam is interpreted when random moving distribution load acts to the curved beam. It is the purpose of this paper to examine sound radiation characteristics from structural vibration in tires

2. Theoretical Background

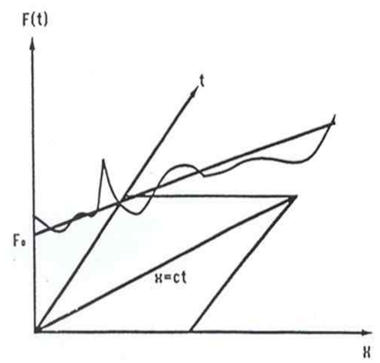
Fig. 1 shows a curved beam or ring model of the tire. The tire is assumed to be a thin elastic ring, or the curved beam that sustains a tension of elastics base.



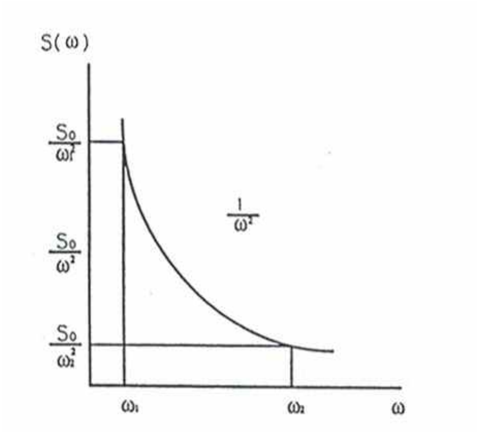
[Fig. 1] Model for a pneumatic tire

The tire model is considered carcass's steady-state characteristic on the soft surface. This study assumes that the vibration of the tire excited by road surface, generates from plane, and assumes plane vibration to the vibration of a circular beam. Applying damping and tension force to the equation of motion by the rotation and centrifugal in the radial direction. The problem of sound radiation is studied by using statistically distributed force in the surface of the tire. The statistically distributed forces are expressed as the irregular random functions in the surface coordination on the tire.

Assuming an input function transmitted from road surface to stationary ergodic function, the input function can be processed as a deterministic function in the frequency band. The quality of input function inside concerned frequency band is road surface that is determined by roughness and the number of vibrations on irregular road surfaces.



[Fig. 2] The motion of a random force F(t) along the tire



[Fig. 3] Spectral density as a function of frequency

By integrating the intensity in the surface over the whole tire, the mean of the sound power is obtained like equation (1).

$$\overline{E}[W] = \mathrm{Re}\left[\int_{-\infty}^{\infty}\int_{-\infty}^{\infty}\lim_{T\to 0}\frac{1}{T}P_T(x,\omega)V_T^*(x,\omega)d\omega\right] \quad (1)$$

The time mean of sound power radiation about power spectrum density is equal to equation (2).

$$\overline{E}[W] = \frac{\rho_o}{\pi}\int_0^{\infty}\int_{-\frac{K_o}{(1+M)}}^{\frac{K_o}{(1-M)}}\frac{(\omega+\xi V)^3}{\sqrt{(K_o+\xi M)^2-\xi^2}}\frac{S(\omega)}{|Z_m+Z_a|^2}\left[\frac{\sin(\xi L)}{\xi L}\right]^2 d\xi d\omega \quad (2)$$

According to the road surface's property of road surface, power spectrum density of random moving distribution load forced tire was assumed to be in inverse proportion to the square of the frequency. It is shown in Fig. 3.

$$S(\omega) = \frac{S_o}{\omega^2} \quad (3)$$

$$\overline{E}[W] = \frac{\rho_o S_o}{\pi(\rho_s A)^2}\int_0^{\infty}\int_{-\frac{K_o}{(1+M)}}^{\frac{K_o}{(1-M)}}\frac{1}{\omega^2}\cdot\frac{(\rho_s A)^2(\omega+\xi V)^3}{\sqrt{(K_o+\xi M)^2-\xi^2}}\left[\frac{Sin(\xi L)}{\xi L}\right]^2 \cdot\frac{1}{|Z_m+Z_a|^2}d\xi d\omega \quad (4)$$

For doing non-dimensional structure impedance in the wave number field in the equation (4). It can be written as equation (5).

$$Z_m = \rho_s A\omega\left[\left\{\omega^2\gamma^2\zeta^2-(1+M\zeta)\omega_c\gamma^2+2\omega_n\gamma^2 N+\frac{\omega_n}{\omega_c\gamma^2}\right\}-j\{2\beta\omega_n-\eta\omega_c\gamma^6\zeta^4\}\right] \quad (5)$$

Equation(4) would become non-dimensional, it could be as equation (6).

$$\Pi = \frac{\pi(\rho_s A)^2}{\rho_o S_o}E\left[\overline{W}\right] \quad (6)$$

so,

$$\Pi = \frac{\int_{r_1}^{r_2}\int_{-\frac{1}{(1-M)}}^{\frac{1}{(1-M)}}\frac{\alpha^3}{\sqrt{\alpha^2-\zeta^2}}\frac{1}{\gamma^5\omega_c^2}\left\{\frac{Sin(K_{cr}L\gamma^2\zeta)}{K_{cr}L\gamma^2\zeta}\right\}^2}{\left[\gamma^4\zeta^4-(\alpha^2-2T_1\gamma_n\zeta^2)+\left(\frac{\gamma_n}{\gamma}\right)^4\right]^2+\left[2\beta\left(\frac{\gamma_n}{\gamma}\right)^2\alpha-\eta\gamma^4\zeta^4+\alpha_o\frac{\alpha^2}{\sqrt{\alpha^2-\zeta^2}}\right]^{-1}}\cdot d\zeta d\gamma$$

3. Numerical Analysis

The ranges of the frequency of excitement from both tire radius and road surface, the length of random moving line force, and the wave length, are equal to following when the vehicle is driving at 40 ~150 Km/h.

In the case of low frequency ($\gamma << 1$)

$$\Pi \sim \int_{\zeta_1}^{\zeta_2} \frac{\alpha^3 \sqrt{\alpha^2 - \zeta^2} \left| \frac{Sin(K_o \zeta L)}{K_o \zeta L} \right|^2 d\zeta}{\left[\left[(-\alpha^2 + \psi^2) \sqrt{\alpha^2 - \zeta^2} \right]^2 + \left[2\beta\psi\alpha \sqrt{\alpha^2 - \zeta^2} + \frac{\alpha_o \alpha^2}{\gamma^2} \right]^2 \right]}$$

(1) In the case of the tire under light fluid loading ($\alpha_o / \gamma^2 << 1$)

$$\Pi \sim \int_{\zeta_1}^{\zeta_2} \frac{\alpha^3 \sqrt{\alpha^2 - \zeta^2} \left| \frac{Sin(K_o \zeta L)}{K_o \zeta L} \right|^2 d\zeta}{\left[\left[(-\alpha^2 + \psi^2) \sqrt{\alpha^2 - \zeta^2} \right]^2 + \left[2\beta\psi\alpha \sqrt{\alpha^2 - \zeta^2} \right]^2 \right]}$$

(2) In the case of the tire under heavy fluid loading($\alpha_o / \gamma^2 >> 1$) after assuming to $\beta, \psi << 1$

$$\Pi \sim \int_{\zeta_1}^{\zeta_2} \frac{\alpha^3 \sqrt{\alpha^2 - \zeta^2} \left| \frac{Sin(K_o \zeta L)}{K_o \zeta L} \right|^2 d\zeta}{\left[\left[-\alpha^2 \sqrt{\alpha^2 - \zeta^2} \right]^2 + \left[\frac{\alpha_o \alpha^2}{\gamma^2} \right]^2 \right]}$$

4. Experiment

The tire used in the experiment for structural vibration noise measurement consists of both a radial tire that is actually installed on a car and an experimental tire that changes design factor in limited extent. In addition, the changed design factors in that the tire used in the experiment is a 195/65R14. Fig.4 is the schematic diagram of a tire structural vibration noise test. Establishing load condition after attaching the tire loading equipment, we can control the travel speed of the tire using a chassis dynamometer. The preliminary operation is practiced to satisfy a fixed experimental condition for 10minutes at 80 Km/h. Then, the microphone is installed at the point, 25 ㎝over the surface of land as well as at horizontal distances, 1m from the tread center to measure sound pressure that radiates from tire sidewall. In the case of using the STSF Technique, a microphone is installed at the point, 60㎝, horizontal distance from the sidewall, 20㎝ over the surface of land, and its interval distance is 20㎝.

5. Conclusions

The results of the study on structure vibration noise control of tire forced random moving distribution load, and sound radiation analysis for low noise tire design shows the following. If non-dimension tension coefficient involved with tire inside air pressure is increased, relative acoustic power increased slightly. And according to increase critical acoustic length related to forced load of tire, relative acoustic power was decreased. But in the low frequency field, if the wave-number ratio increases, relative acoustic power increases as well. In the area of wave number ratio corresponding to forced frequency involved with the running velocity of tire, the relative acoustic power for resonance radiation of tire noise energy was increased significantly. Also wave number ratio produced resonance radiation moved when wave number ratio decreased. If the non-dimensional damping coefficient involved with tire of physical characteristic is increased, relatively acoustic power decreased significantly. It affected the tire bending tension and quantity of matter.

Acknowledgements

This work was supported by grant No.(2010-0023299) from the Basic Research Program of National Research Foundation of Korea

References

[1] A. C. Eberhardt,"Investigation of the Truck Tire Vibration Sound Mechanism", Tire Noise Conference 1979, Stockholm, pp.153~168, 1979.
[2] A. C. Eberhardt, "Truck Tire Vibration Sound", Inter-Noise Conference, Florida, pp.281~ 288, 1980.
[3] W. F. Reiter and A. C. Eberhardt, "The Relation ship Between Truck Tire Vibration and Near and Far Field Sound levels", SAE Paper 762021, 1976.