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Sound Radiation Property of Tribo-System

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Frictional sound is observed in great many practical systems, but its generation mechanism is still unknown. Model systems are best suited for research on the fundamental mechanisms, but results cannot be easily applied to real systems, because each system has different sound radiation properties. At present, there is no easy method for evaluation of these properties. We propose to describe the sound radiation property of a tribo-system by the relationship between friction-induced sound power and the friction-induced vibration velocity of the contact element. It was found that the sound power of a tribo-system is linearly proportional to the mean-square velocity of the sliding element by a constant coefficient having the dimension of mass flow rate (kg/s).

Keywords : Friction-induced vibration, Sound power, Sound radiation property

1. INTRODUCTION

Whenever two dry surfaces slide against each other sound is generated. One practical problem of frictional sound is the brake squeal. It has been extensively studied in the past [1], but still there is no commonly accepted method to avoid it. It is because of the lack of fundamental understanding of squeal generation mechanism. Such fundamental understanding could be obtained by studies on model systems. This approach was taken by Yokoi and Nakai [2]. However, measured sound pressure in the study could not be directly related to real systems, because of the difference of sound radiation properties between experimental and real systems.

The purpose of this paper is to investigate the relationship between the friction-induced vibration of the contact element and the radiated sound, and to propose a parameter to describe the sound radiation property of a tribo-system.

2. EXPERIMENTAL APPARATUS AND METHOD

Experiments were conducted on a reciprocating tester (Fig.1), in which frictional sound was generated by the contact of a disk on a flat bar, attached to a moving stage. To reduce background noise levels, the motor was placed in a separate sound insulated compartment and magnetic screws were used to move the stage. The test chamber was covered with sound-absorbing material on the inside to eliminate sound reflections and create free field conditions. Normal load is applied by the elastic deformation of a leaf spring when the XZ-stage is lowered down. Upper specimen vibration was measured in tangential and normal direction by two accelerometers mounted onto the upper specimen holder. The sound was measured by a free-field microphone, placed at about 8.5 cm from the center of the upper specimen holder. The signals are acquired into a computer and acceleration is numerically integrated to velocity.

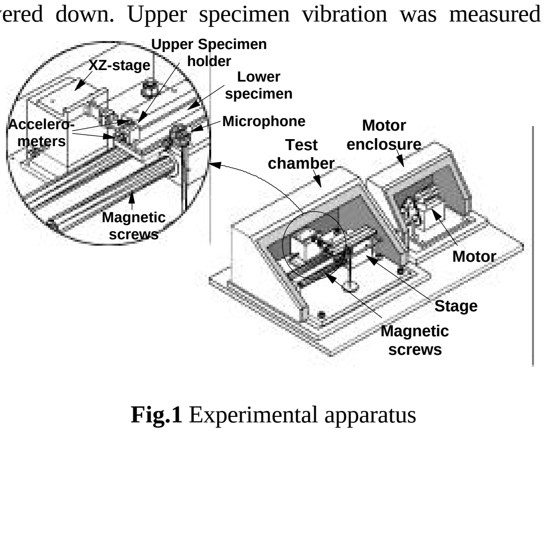

Fig.1 Experimental apparatus

The specimens used in the test were made of stainless steel (JIS – SUS303), upper disk specimen RMS roughness was Rq = 1.05 μm, lower bar specimen RMS roughness - Rq = 0.68 μm. A series of tests were carried out for sliding speeds from 20 to 100 mm/s at load setting of 1 N, 2 N and 4 N. Another series of tests was carried in the same speed range at load of 1 N, but with three different fixing methods of the bar to the stage (Fig. 2).

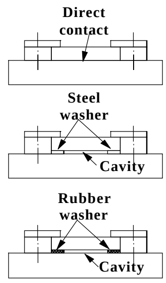

Fig. 2 Fixing method

By comparison of the frequency spectra of frictional sound against the background noise, it was found that in the frequency band from 0.5 to 5 kHz the signal-to-noise ratio was highest. All further analysis was carried in this frequency band only.

3. RESULTS AND ANALYSIS

Relationship between the measured sound pressure p and vibration velocities V_n and V_t of the upper specimen are shown in Fig. 3. The general trend is that sound pressure increases with the increase of vibration velocity and the rate of change is different for the normal and tangential vibrations.

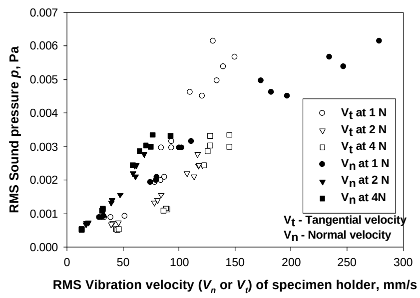

Fig.3 Sound pressure vs. vibration velocity (V_n or V_t) for varied load and sliding speed

It is impossible, however, to explain the different rate of change with different sensitivity of sound pressure to normal and tangential vibrations, because these vibrations are not independent. Although in Fig.3 sound pressure was used, further in this paper sound power will be introduced, because unlike sound pressure, power is independent of the distance to the source. This is advantageous for deeper understanding of the relationship between friction induced sound and vibration.

Beside the experimental approach to establishing the sound-vibration velocity relationship, there is the theoretical approach of solving the wave equations for sound propagation if the surface geometry of the source and the velocity distribution on this surface are completely described. For real tribo-systems such complete description is often impossible.

An alternative is to use a simple model of sound radiation and find the unknown parameters experimentally. One such simple model is the source of plane sound waves. The sound power P_{pl} radiated from such a source is given by [3]:

$$P_{pl} = rcV^2S_r \qquad (1).$$

In eq.(1) rc is the characteristic acoustic impedance of the medium (r – density, c – speed of sound in that medium; for air at 20°C rc = 415 kg/(m^2.s)); V^2 is the mean square velocity of the vibrating plane surface with area Sr .

Eq.(1) is not directly applicable to bodies which have vibration velocity components in three coordinates. We may use it, however, to calculate the power from a plane wave source, which is equivalent to the body in question in the sense that they have the same mean-square velocities. The mean-square value of the velocity vector with three components is the sum of the mean square velocities on each axes. If a contact element of a tribo-system has mean-square velocity in normal direction V_n^2 and in tangential - V_t^2, using eq.(1) we can calculate the sound power P_{eq} from such an equivalent plane wave source as:

$$P_{eq} = rc(V_n^2+V_t^2)S_r \qquad (2).$$

The radiation area S_r and the other constant coefficients can be combined in a new coefficient α, which has the dimension of mass flow rate, kg/s:

$$\alpha = rcS_r \qquad (3).$$

This coefficient can be found if we use the measured airborne sound power P_a instead of P_{eq} in eq.(2):

$$P_a = a(V_n^2+V_t^2) \qquad (4).$$

The coefficient α can be called "characteristic mass flow rate", and it characterises the relationship between the mean-square vibration velocity and the airborne sound power.

In order to use estimate the airborne sound power needed in eq. (4) we make the justified simplifying assumptions that: sound is radiated only from the upper specimen holder as from a simple point source placed on a reflective surface (stage top). Simple point source assumption is valid because the largest dimension of specimen holder is 2 cm - much smaller than the shortest wavelength of interest – about 6.5 cm for a sound wave at 5 kHz. For such a source the sound power can be calculated from the measured RMS sound pressure p and the distance R between the microphone and the center of the source by [3]:

$$P_a = p^2 2pR/(rc) \qquad (5).$$

P_a values calculated by eq.(5) from pressure in Fig.3 are shown in Fig.4. The solid line is obtained by the assumption

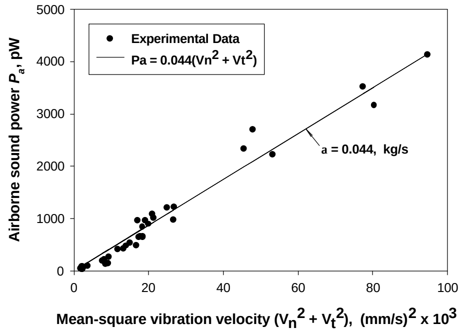

Fig. 4 Sound power vs. vibration velocity for varied load and speed

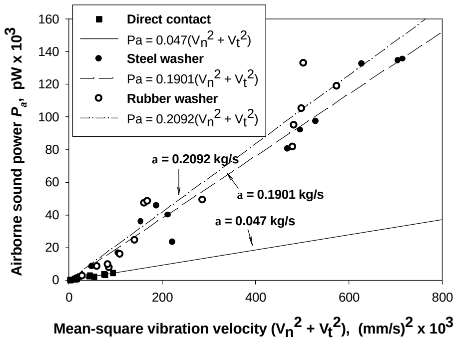

Fig. 5 Sound power vs. vibration velocity for varied speed and fixing method

of eq.(4) and experimental value of α is obtained as α = 0.044 kg/s. Similar results for different fixing methods of the bar specimen at constant load of 1 N and varied speed are shown in Fig.5. The change of slope of α value is caused by the introduction of a cavity and a washer as shown in Fig.2. The change of stiffness by replacing the steel washers with rubber ones did not change significantly the characteristic flow rate.

4. CONCLUSIONS

The sound radiation property of a tribo-system can be described by the relation between the friction-induced vibration of contact element and the sound power. We conclude:

1. The sound power of a tribo-system is linearly proportional to the mean-square velocity of the sliding element by a constant coefficient having the dimension of kg/s. This coefficient characterises the sound radiation property of the tribo-system and we name it "characteristic mass flow rate".

2. The characteristic mass flow rate increased 4 times when a cavity was formed between the bar specimen and the stage top surface, compared to direct contact without cavity.

3. When cavity was present, the characteristic mass flow rate did not change by changing the washers inserted under the bar specimen from steel to rubber.

5. REFERENCES

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