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SOUND RADIATION OF STRUCTURAL METALS AT NORMAL AND ELEVATED TEMPERATURES

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ABSTRACT

The sound radiation efficiency of the tested medium alloy steel, forged iron and cast-iron cylindrical samples (4"dia.x 9'') have been characterized by the attenuation rate of the sound. It has been shown analytically that the attenuation rate of the sound affected by combination of such mechanical and physical parameters of a metal as internal friction $\Theta^{-1}$, yield stress $\sigma_y$, modulus of elasticity $E$, metal density $\rho$, Poisson's ratio $\mu$. The correspondence between measured attenuation rate and those analytically predicted from tabulated metal properties is reasonable good, indicating that some sound radiation parameters can be tailored and predicted to a certain degree through proper selection of metals and metallurgical processing.

Performed spectral analysis of the sound radiation of the medium alloy steel cylinder at elevated temperatures (100°C - 250°C) shows that the resonance frequency peaks have been shifted up to 300Hz with increase of the temperature.

INTRODUCTION

When a compressor component or a structure is acted on by a fluctuating force, whether this force be harmonic, impulsive or random with respect to time, some or all components will vibrate and radiate sound. Despite of wide use of plastic and composite materials for a compressor structural components, metal still makes up a large portion of the average compressor, especially in such parts as crankcase, crankshaft, housing and where combination of high strength at both "normal" and elevated temperatures are required.

Analysis of sound radiation from solid bodies appears in works of Morse and Ingard [7], Skudrzyk [8],Cremer and Heckl [2]. Further studies of this were conducted by Koss and Alfredson [6], Dreiman [3], Endo [4]. The literature survey shows that the sound radiation parameters of a metal in relation to chemical composition, heat treatment and method of manufacturing have not been investigated, even though such study would make it possible to determine physical and mechanical properties of the construction metal satisfying the prescribed sound parameters of the compressor or another mechanical system.

Analysis of vibration and sound which can be generated by a compressor is based very often on the physical and mechanical properties of metal used for manufacturing the compressor parts. As a rule, the mechanical properties of a metal applied to the tests under so-called "normal" conditions, i.e. temperature 20°C and relatively small rates of loads and pressure. It is clear that the mechanical properties of the metals and corresponding sound radiation of the compressor at operating conditions will be affected by variation of the temperature and loads.

Many sound and vibration reduction techniques are known, but the need for an effective means to reduce vibration and sound radiation becomes more difficult to achieve as the surrounding temperature or temperature of the vibrating part increases.

The application of the polymers is limited not only by the temperature barrier but also by the presence of the aggressive chemicals, especially in combination with elevated temperatures.
ANALYTICAL DEVELOPMENT

As a rule, the theoretical development of impact and/or friction is concerned with the transfer of mechanical energy through a structure and neglects the sound produced during the process itself. This omission is usually justified since only a small fraction of the mechanical energy is lost as the radiated sound. The disturbance generated by contact propagates into the structure with finite velocity. Its subsequent reflections from the bounding surfaces then produce vibrations within the structure. It has been shown [2] that the ratio of vibrational energy \( e \) to the initial kinetic energy \( e_0 \) of impact is very small, approximately

\[
e / e_0 = 1/50 \left( \frac{V_0}{C_0} \right),
\]

where \( V_0 \) is the relative velocity of impact and \( C_0 \) is the dilatational wave velocity. Although the ratio is considerably less than unit for most practical examples, this free elastic vibrational energy is responsible for most of the sound associated with the structure. The sound radiation characteristics of a solid may be described in terms of the radiation efficiency, power conversion efficiency, which is defined as the ratio of the power radiated into a medium to the vibratory power supplied to the solid, and in terms of the radiation loss factor which indicates the extent to which the vibration of a system are damped due to the radiation of power from the solid to the ambient medium [2]. The sound radiation efficiency of a solid may also be characterized by attenuation rate \( d \) of the sound when the excitation of the system is removed, so that energy is no longer supplied to the system. In this case the rate of change in the sound pressure level per unit of time will be \( d = \Delta L / t + B \), dB/s, where \( \Delta L \) is change in sound level in dB, \( t \) is time in seconds, and \( B \) is a constant accounting for losses due to the environmental scattering, transmission to neighboring elements, and absorption at interfaces. Consider a system that vibrates at the circular frequency \( \omega \) and which at time \( t = 0 \) has a reversible mechanical energy \( E_0 \). If the system is disconnected from all external sources, part of this energy is changed into heat and reversible energy \( E_R \) that is left in the system at this time \( E_R = E_0 e^{-\eta t} \) where \( \eta \) is a loss factor. It is evident that energy decay may be described by the function \( e^{-\eta t} \). In practice one can measure the reverberation time within which the energy of the vibration is reduced to the one millionth of the initial value and determine the loss factor \( \eta \) from the relation

\[
\eta = \ln 10^6 / f_n t_{60}
\]

The very important source of loss factor is damping properties of the structure. Principally, the sources of damping are the internal friction \( Q \) of the metal, external (Coulomb) friction due to interfacial slip at joints, hydrodynamic (viscous) damping. When a solid vibrates in a gaseous medium and mechanical interfaces are eliminated, the predominating component causing the loss at audio-frequencies is internal friction \( Q^{-1} \). Thus, a duration of the sound radiation is equal to the reverberation time

\[
t_{60} = 2.199 / f_n Q^{-1}
\]

Elastic vibration and consequently sound radiation results from the collision and rubbing of components of a mechanical system joined in kinematic pairs. The dynamic response of the system components to impact-type forces can be considered as forced (acceleration noise) and free vibration (ringing noise). The sudden motion of the end surfaces during contact produces a
sound pulse followed by the ringing noise. Duration of this sound pulse is equal to the impact force duration \( \tau \), and total duration of sound radiation from a vibrating component is equal to the sum of forced and free vibration time. Sound radiation due to the free vibration of the component following impact is very often dominant. In many analytical and experimental works the dynamic analysis of collision and its associated acoustic radiation was based on a scheme suggested by Hertz. However, strict application of the Hertz law to the causes of contact of metallic bodies is very often limited. The simplest and most successful static relation known as the Mayer law may be described by the relation [5]:

\[
F = \pi \sigma_0 a^2 = 2 \pi \sigma_0 r_s \alpha = m \left( \frac{d^2 \alpha}{d t^2} \right)
\]

or

\[
\left( \frac{d^2 \alpha}{d t^2} \right) + \left( 2 \pi r_s \sigma_0 \right) / m = 0
\]  

(3)

where \( F \) is the applied load, \( a \) is the contact diameter, \( m \) is the mass of the sphere, \( r_s \) is the radius of the sphere, \( \sigma_0 \) is the yield stress, and \( \alpha \) is a compression. The maximum compression occurs at the time \( \tau = \left( \pi / 2 \right) \left( m / 2 \pi r_s \sigma_0 \right)^{1/2} \) which represents the entire duration of the contact.

In the present paper, longitudinal impact of a steel sphere on a free circular elastic cylinder has been considered as a typical impact model. In the case of longitudinal impact of a sphere on a finite elastic cylinder, the longitudinal natural frequency \( f_n \) of the cylinder in \( n \)-th vibrational mode is

\[
f_n = n C_L / 2 L ,
\]

(4)

where \( C_L = \left( E_1 / \rho_c \right)^{1/2} / \left( 1 + n^2 \Psi^2 \right) \) denotes dilatational wave velocity corrected for dispersion due to radial inertia, \( \Psi = \pi \mu r_c / 2 L \) is the Love radial dispersion correction factor, \( \mu \) is Poisson's ration, \( r_c \) is the cylinder radius, \( L \) is the cylinder length, \( n = 1, 2, 3, ... \) is the mode number for longitudinal vibrations, \( E_1 \) is the modulus of elasticity, \( \rho_c \) is the material density.

Taking into account total sound radiation time (\( t_{60} - \tau \)), and natural frequency of the cylinder we can estimate the attenuation rate \( d_e \) of the cylinder's sound radiation from the equation below:

\[
d_e = \frac{60 n \eta \left( \sigma_0 E_1 \right)^{1/2}}{2.199 \sigma_0 \rho_0 \left( 4 L^2 + \pi^2 n^2 \mu^2 r_c \right)^{1/2} - \mu \eta \left( 2 \pi r_s \rho_c E_1 \right)^{1/2}} + B
\]

(5)

EXPERIMENTATION AND RESULTS.

A cylindrical specimen of diameter \( D = 4 \) in. (0.102 m) and length \( L = 9 \) in. (0.229 m) was horizontally suspended in the anechoic chamber by steel cables to reduce losses of the vibration energy at contact points. Sound radiation was induced by the impact of the steel sphere of 0.94 in. (23.8 mm) dia. on the center of the end face of the cylinder. The position of the specimens and drop height \( h \) of the steel sphere were kept identical throughout the tests. Hence, the impact velocity (\( V_0 = 2gh \)) and amplitude were assumed to have remained constant through the tests. The Hewlett-Packard Structural Dynamic analyzer 5423A has been
used for the data collection and analysis. The chemical composition of the specimens is shown in the table below.

<table>
<thead>
<tr>
<th>Sample No.</th>
<th>Material</th>
<th>Chemical composition, %</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>C</td>
</tr>
<tr>
<td>1</td>
<td>Medium Alloy Steel</td>
<td>0.2</td>
</tr>
<tr>
<td>2</td>
<td>Forged Iron</td>
<td>3.5</td>
</tr>
<tr>
<td>3</td>
<td>Cast Iron</td>
<td>3.5</td>
</tr>
</tbody>
</table>

Figure 1 shows the attenuation rate of sound for medium alloy steel, forged-iron and cast iron samples. As you can see the choice of material can produce significant differences in the sound attenuation rate. The correspondence between measured rate and those predicted from tabulated properties of the metals is reasonable good, indicating that attenuation rate can be tailored and predicted to a certain degree through proper selection of metal and its metallurgical processing. It is clear that the mechanical properties of the metals will be affected by variation of the temperature caused by change of the compressor operating conditions. The medium alloy steel cylinder (Sample 1) have been heated in the Thermolyne F-6025 furnace for study of the temperature effect on the radiated sound. A temperature stabilization time of approximately 16h have been allowed for the specimen between each temperature setting and following sound radiation test. All measurements were carried out at various temperatures ranging from room temperature to 250°C (480°F).

The spectral analysis of the sound radiation of the specimen at elevated temperature indicates highly dominant components at frequencies in the range of 7.6 kHz to 7.9 kHz. The typical distribution of the peaks in the frequency range 6 kHz to 8.5 kHz for the specimen heated to 115°C (239°F), 121.1°C (250°F), 164.4°C (328°F), 185°C (365°F), 235.6°C (456°F), and, in comparison, the spectrum at room temperature 18°C (64.4°F) is shown in Fig. 2. The shift of the resonance frequency for the investigated range of temperatures is shown in Fig. 3. The peak of resonance frequency was shifted 287 Hz (from 7887Hz to 7600Hz) when the temperature increased from 18°C (64.4°F) to 235.6°C (456°F).

The process responsible for the properties and structural changes are initiated by the activating energy ΔQ by which thermal agitation excites at an efficient rate when temperature of metal rise, a phenomena formally simulated by the Arrhenius law:

\[
\text{rate} \propto \exp \left(- \frac{\Delta Q}{RT} \right)
\]  

where \( R \) is Bolzman constant, and \( T \) - absolute temperature. The shift of the resonance frequency with the temperature change could be calculated using a similar expression:

\[
f(T) = \gamma f_n \cdot \frac{\Delta Q}{RT}
\]
The temperature dependence of the attenuation rate are shown in Fig 4. Compared to its room temperature value (14.51 dB/s), attenuation rate increases to a maximum - 27.16 dB/s at 115°C (239°F), decreases suddenly to minimum (10.33 dB/s) at 161°C (323°F). At higher temperatures of the specimen the sound attenuation rate are increased to 23.81 dB/s.

CONCLUSIONS

1. The attenuation of the sound radiated by the metal has been shown to be affected by the geometric dimensions of the sound radiation system, physical and mechanical properties of the metal. The physical and mechanical properties can be modified by controlling the chemical composition, heat treatment and mechanical working of the metal.

2. The experimental data show that elevated temperature caused resonance frequencies shift and also affected the value of the attenuation rate.

3. Recorded high attenuation of the sound in the temperature range 100°C-130°C is a result of internal friction peak which associated with the presence of interstitial carbon or nitrogen impurity atoms in steel (Snoek effect) [9].

REFERENCES

Fig. 2. SPECTRAL DISTRIBUTION OF THE SOUND PEAKS AT ELEVATED TEMPERATURES

Fig. 3. SHIFT OF THE FREQUENCY WITH THE TEMPERATURE

Fig. 4. TEMPERATURE DEPENDENCE OF THE ATTENUATION RATE