General model of a structure-borne sound source and its application to shock vibration

Yuri Bobrovnitskii and Tatiana Tomilina

Mechanical Engineering Research Institute, 4, M. Kharitonievky Str., 101990 Moscow, Russian Federation
yuri@imash.ac.ru

Three models of structure-borne sound sources that generalize the existing source models are proposed. One model employs the blocked forces, the second one uses the free velocities, and the third model is based on a comparison with reference receiving structure. The models can be used for prediction of the vibration levels on the receiver as well as on the source structures. It is shown that the models are valid for a wide class of industrial sources (machines and mechanisms) that contain physical generators of forces and kinematic excitation. Some results of application of the models to shock vibration sources (shock testing machines) are presented.

1 Introduction

A structure- borne sound source is always a part (subsystem) of a more complex vibrating system generally consisting of a carrying elastic master- structure and surrounding medium. Prediction of vibration and acoustic fields of such a system is a challenging problem solved only in some particular cases. In machinery acoustics, a simplified system consisting of two subsystems – a source of vibration (machine) and elastic structure (receiving structure) is usually considered. Though its vibrations have been analyzed by many authors (e.g. in [1-7]), there are still problems in modeling and characterization of sources and in prediction of the system vibration fields. One of such problems is addressed in the present paper.

The most promising approach to analyzing the vibrations of a source/receiver system is the impedance or mobility method - see, e.g. [1,4,6,7]. In this method, a receiving structure is described by a matrix of the input impedances (mobilities) defined at the source/receiver interface. A source of vibrations is characterized by a similar matrix of the input impedances of the idle source defined also at the interface together with the so called « blocked forces » or «free vibration velocities».

The blocked forces, when applied to the interface between the idle source and receiver, excite the vibration field on the interface that is identical to that excited by the actual physical sound sources inside the machine. So, these model parameters are sufficient for correct prediction the vibration levels at the interface, for obtaining the vibration power flow into the receiving structure, and for computing the vibration field in the receiving structure using the interface values and transmission characteristics of the receiver.

At the same time, as will be shown below, the vibration levels of the source structure, e.g. of the machine case, due to the blocked forces action, are not equal to the levels measured on the real machine. The source model characterized by the mobility matrix & blocked forces is, hence, invalid with respect to the source structure vibrations. As these vibrations are often of the same interest as vibrations of the receiving structure, this model as well as the model characterized by the mobility matrix & free velocities needs to be reformulated and improved.

This is the main objective of the present paper to remove this shortcoming of the existing models via more general consideration of the problem and by introduction of additional model parameters. In fact, three improved models of a structure-borne sound source are presented in this paper. The first and the second models are generalization of the two known models – characterized by the mobility matrix& blocked forces and the mobility matrix & free velocities. The third model proposed here and characterized by a reference receiving structure is, to the authors' knowledge, new. Some results on shock vibrations obtained using the models are also presented.

2 General relations

The starting point of the theory of this paper as well as of most literature is the assumption that a structure-borne sound source may be regarded as a linear mechanical system with the force excitation; more exactly, it can be described by a finite set of linear ordinary differential equations with generalized forces at the right side. Physically this means that a real sound source, e.g. machine, may contain active elements of the force excitation as well as kinematic excitation. Examples of the force excitation are the fuel explosions in an internal combustion engine, electromagnetic interaction forces in electric machines, impacts and others. Kinematic excitation means that prescribed are relative displacements (velocities) of adjacent DOFs. Examples are cam mechanisms, roller and ball bearings, gears, unbalanced rotors, etc. One of the results of the present work is a rigorous proof (not presented here) that any kinematic excitation is equivalent to a certain force excitation.

Consider now a mechanical system, composed of two subsystems – a source of vibration and a receiving structure connected through an interface *A* (Fig.1). No sound sources are assumed present in the receiver structure which is also regarded as a linear mechanical system. Starting from this and the assumption made above concerning the source one can write the following equations that describe the system vibrations in the frequency domain:

$$\begin{aligned} v_S &= \sum Y_{Aj} F_j + Y_S f_S \\ v_x &= \sum Y_{xj} F_j - Y_{xA} f_S, \\ v_R &= Y_R f_R \,, \\ v_y &= Y_{yA} f_R, \\ v_S &= v_R, \\ f_S &+ f_R = 0. \end{aligned} \qquad (1)$$

In these equations, F_j are generalized forces inside the source, f_S and f_R are vectors of the reaction forces acting on the source and receiver at the interface DOFs, v_S and v_R are vectors of the corresponding velocities; *x* and *y* are the observation points on the source and receiver structures and v_x and v_y are the velocity amplitudes at these points; Y_{Aj} and Y_{xj} are the transfer mobility matrices from the internal sources F_j to the interface *A* and observation points *x*; Y_{xA} and Y_{yA} are the transfer mobility matrices from the interface to the observation points *x* and *y*.

Eqs.(1) can be rewritten in the form

$$v_R = v_S = Y_R(Y_R+Y_S)^{-1}\sum Y_{Aj}F_j$$

$$f_R = -f_S = (Y_R+Y_S)^{-1}\sum Y_{Aj}F_j \qquad (2)$$

$$v_x = \sum Y_{xj}F_j - Y_{xA}(Y_R+Y_S)^{-1}\sum Y_{Aj}F_j$$

$$v_y = Y_{yA}(Y_R+Y_S)^{-1}\sum Y_{Aj}F_j$$

Here, the matrices of the input mobilities of the source and receiver with respect to their interface, Y_S and Y_R, as well as the transfer mobilities from the interface to the observation points, Y_{xA} and Y_{yA}, can in principle be measured and are considered therefore as known. At the same time, the internal generalized forces F_j and the transfer mobility from the points of their application can not be measured by existing means and therefore should be excluded from the consideration. In the next section they will be replaced by other model parameters that are measurable at least in principle.

3. Source models

3.1 Model with blocked forces

Suppose that the vibration source under study is separated along the interface A from the receiver and its DOFs at A

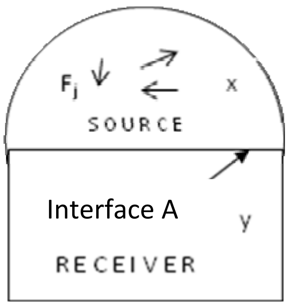

Fig.1. Schematic of a source/receiver system

are blocked with the help of external forces f_{Sb} that are called as «the blocked forces». In this blocked configuration (marked by the lower index «b») the receiving structure is at rest while on the vibrating structure one can measure the velocity response v_{xb} as well as the blocked forces:

$$v_{Rb} = v_{yb} = 0, \quad f_{Rb} = 0; \qquad (3)$$

$v_{Sb} = 0$, v_{xb} and f_{Sb} are measured.

Using these parameters, one can find the unknowns

$$\sum Y_{Aj}F_j = -Y_S f_{Sb}$$

$$\sum Y_{xj}F_j = v_{xb} - Y_{xA} f_{Sb}$$

and, after substituting them into Eqs.(2), express through them the solution of the initial problem

$$v_R = v_S = -(Z_R+Z_S)^{-1} f_{Sb}$$

$$f_R = -f_S = Z_R(Z_R+Z_S)^{-1} f_{Sb} \qquad (4)$$

$$v_x = v_{xb} - Y_{xA}Z_S(Z_R+Z_S)^{-1} f_{Sb}$$

$$v_y = -Y_{yA}Z_R(Z_R+Z_S)^{-1} f_{Sb}$$

Here matrices of the input impedances of the source and receiver, Z_S and Z_R, are the inverse of the corresponding mobility matrices. Note, that measurement of the blocked forces is a problem in general case. But if the system in Fig.1 contains resilient mounts, these mounts can be regarded as a part of the source structure and the blocked forces can then be picked up by ordinary means, e.g. by measuring the relative displacement at the mount ports.

To reveal the structure and physical sense of the solution (4), it is appropriate to consider the following auxiliary problem: find the vibration field of the idle source/ receiver system under the action of external forces f_1 applied to the interface A. In this case, the internal forces F_j in Eqs.(1) are zeros and the last equation (1) is replaced by $f_1 = f_{R1} + f_{S1}$. Using other Eqs.(1), one obtains the solution of this auxiliary problem in the form (all quantities relating to this problem have the additional index 1):

$$v_{R1} = v_{S1} = (Z_R+Z_S)^{-1} f_1$$

$$f_{R1} = Z_R(Z_R+Z_S)^{-1} f_1 \qquad (5)$$

$$f_{S1} = Z_S(Z_R+Z_S)^{-1} f_1$$

$$v_{x1} = Y_{xA}Z_S(Z_R+Z_S)^{-1} f_1$$

$$v_{y1} = Y_{yA}Z_R(Z_R+Z_S)^{-1} f_1$$

Comparing this solution with Eqs.(4), one can verify that the solution of the initial problem in the form of equations (4) is the sum of two components.

$$v = v_b + v_1, \quad f = f_b + f_1 \qquad (6)$$

The first component (index b) corresponds to the case of the blocked operating source (machine), i.e. to the quantities (3). The second component (index 1) is the solution (5) to the forced vibrations of the idle system under the action of the external forces which are equal to the minus blocked forces

$$f_1 = -f_{Sb}\,. \qquad (7)$$

The representation (6) is valid for all the field quantities of the source/receiver system. At the same time the known source model consisting of the blocked forces and matrix of the input impedances Z_S or mobilities Y_S describes correctly only the second component in Eq.(6) and, hence, is valid only for description of the vibration field on the receiving structure that has the zero first component. To describe accurately also the field in the source structure, this structure-borne source model should be complemented by the response data v_{xb} measured in the blocked configuration together with the blocked forces.

3.2 Model with free velocities

Another system configuration where the unknown quantities of the general solution (2) can be expressed through measurable characteristics is the freely suspended source. When the operating source is separated along the interface A from the receiver and freely suspended, the receiver does not vibrate, the interface is free of tension, and the velocity response at point x and A on the source structure, v_{xf} and v_{Sf}, can be easily measured. One has is this configuration (index f):

$$v_{Rf} = v_{yf} = 0,\ f_{Rf} = 0; \qquad (8)$$

$f_{Sf} = 0,\ \ v_{xf}$ and $= v_{Sf}$; are measured.

The unknowns is this case are equal to

$$\Sigma Y_{Aj} F_j = v_{Sf}\ ,\ \Sigma Y_{xj} F_j = v_{xf}\ .$$

Substituting these into Eq. (2), one obtain the solution of the initial problem expressed through the known or measured characteristics:

$$\begin{aligned} v_R &= v_S = Y_R (Y_R + Y_S)^{-1} v_{Sf} \\ f_R &= -f_S = (Y_R + Y_S)^{-1} v_{Sf} \qquad (9) \\ v_x &= v_{xf} - Y_{xA} (Y_R + Y_S)^{-1} v_{Sf} \\ v_y &= -Y_{yA} (Y_R + Y_S)^{-1} v_{Sf} \end{aligned}$$

The structure of this solution and its physical interpretation is the following. It is composed of two components

$$v = v_f + v_2,\ f = f_f + f_2\ . \qquad (10)$$

The first component corresponds to the freely suspended source (index *f*), i.e. to the field quantities (8). The second component (index 2) represents the forced vibrations of the idle source/receiver system under the action of the special external kinematic excitation v_k applied to interface *A*. In our case this excitation means that two subsystems (source and receiver) are separated along *A* and driven by forces of equal amplitudes and opposite direction such that the relative vibration velocities of the subsystems are equal to the kinematic excitation v_k, v_{S2}-v_{R2}=v_k. The solution to this auxiliary problem is

$$\begin{aligned} f_{S2} &= -f_{R2} = (Y_R + Y_S)^{-1} v_k, \\ v_{S2} &= Y_S (Y_R + Y_S)^{-1} v_k \\ v_{R2} &= -Y_R (Y_R + Y_S)^{-1} v_k \qquad (11) \\ v_{x2} &= Y_{xA} (Y_R + Y_S)^{-1} v_k \\ v_{y2} &= +Y_{yA} (Y_R + Y_S)^{-1} v_k \end{aligned}$$

Comparing it to solution (9) one can see that the velocity of the kinematic excitation in Eqs.(11) should be equal to the minus free velocity

$$v_k = -v_{Sf}$$

Thus, the representation (10) that is valid for all field quantities of the system, accurately expresses the solution to the initial problem through the known mobility characteristics and the velocity amplitudes measured in the free suspended machine. The complete model of the structure-borne source must contain here the matrix of the source input mobilities Y_S, the velocity response of the source structure v_{xf} in the free suspension configuration and the free velocities v_{Sf} as a strength of kinematic excitation.

3.3 Model with a reference receiver

One more and the most general configuration where the unknowns of the solution (2) can be replaced by measured quantities uses a reference receiving structure. If the operating source is installed on a such receiver with a known matrix Z_{R0} of its input impedances, all the field characteristics, i.e. the solution (2) to the initial prediction problem with the matrix Z_R, can then be expressed through the velocity responses v_{x0}, v_{S0}=v_{R0} or/and the interaction forces f_{S0} at the interface *A* measured on the operating source/reference receiver system (index 0). It can be shown in a similar manner as above that the solution of the initial problem is represented here as a sum of two components

$$v = v_0 + v_3,\qquad f = f_0 + f_3$$

The first component with index 0 corresponds to the system configuration with a reference receiving structure. The second component (index 3) describes the forced vibrations of the idle system under the action either of the external forces

$$f_3 = -(Z_R - Z_{R0}) v_{S0}\ ,$$

or of the kinematic excitation

$$v_3 = -(Y_R - Y_{R0}) f_{S0}\ ,$$

applied to the interface *A*. The complete source model, valid for prediction vibration field everywhere in the system, consists in this case of the matrix Z_S (or Y_S) of the input impedances and the velocity responses (or reacting forces) of the system in the reference receiver configuration. Note that two models considered above are the particular cases of this model with the absolutely rigid and soft reference receiver.

4. Application to shock vibrations

The models described above are applicable to wide class of source of structure-borne sound. They also can be useful in shock study. The main restriction here is the requirement of system linearity. It is well known [8] that many shock processes (e.g. body collision) are nonlinear. Besides, impacts generate impulse-type forces and stresses at contact points of very high amplitudes resulting in plastic deformations. Nevertheless, in many shock problems the linearity assumption is quite acceptable. Such is e.g., the analysis of shock vibrations of equipment on drop tables or other shock testing machines.

The authors' objective in this study was to reveal, theoretically and experimentally, the potentials of some available shock testing machines to meet the shock requirements to scientific space instruments. Usually, such requirements are formulated in terms of the SRS (Shock Response Spectrum [8]).

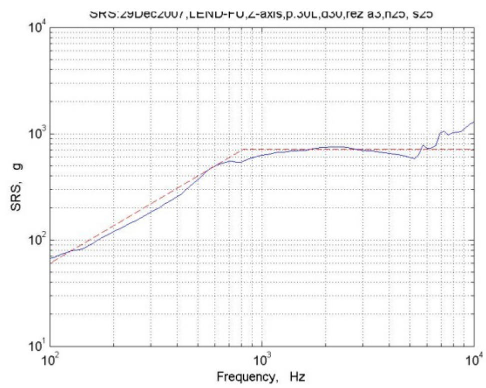

Fig.2. Example of the shock requirements (dashed line) and results of testing a space instrument on a drop table machine.

A typical requirement curve is presented in Fig.2 by dashed line. It consists of two regions. In the first low frequency region, the curve is raising and in the second high-frequency region (above a certain knee frequency) the curve is constant. While the SRS values in the second region is determined by the maximum amplitude of the shock impulse and the knee frequency is depended mostly on the impulse time duration, the slope of the raising region and the minimum value at 100Hz depend on the form of the impulse acceleration time history (on changes of the impulse sign in particular) and are the characteristic of each shock testing rig. It was shown, using one of the source model described above, that the slope of the low frequency SRS curves obtained on shock testing machines of the drop table type, the hammer blow type and with pyro-shock devices may lie between 6 and 12 dB/oct. To provide higher slopes a very special devices are needed.

5. Conclusion

Three models of structure-borne sound sources that generalize the existing models are presented. They can be used for predicting the vibration levels not only on the receiving structure but on the source structure as well. The models are shown to be valid for wide class of the engineering sources in which sound is generated by physical mechanisms that produce the force excitation as well as kinematic excitation. The model with the blocked forces is applied to shock vibrations. Several types of shock testing machines were studied and peculiarities of the typical shock response spectra are revealed using the model.

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