Use of empirical models in the prediction of acoustic noise and vibration induced by turbulent flow

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Abstract

Use of empirical models is described in the prediction of acoustic noise generated by turbulent flow and the induced vibration at the location of a sensitive element. The principle of the models and the result of their application are presented.

Introduction

In the early development stages of a system, it was required to predict the acoustic noise and vibration at the location of a component susceptible to these loads. A schematic description of the system appears in Fig. 1. Two major sources of acoustic noise and vibration exist in the system, a hot gas ejection system (E) and a water cooling system (C). Both sources affect the sensitive element (S). Vibration is induced at the sensitive element due to the acoustic noise generated by the ejected gas and the acoustic noise and vibration generated by the flow of cooling fluid. The paper will relate to the first source and will describe the use of empirical models in the prediction of the acoustic noise and of the related vibrations, at the location of the sensitive element. After the presentation of the acoustic noise model and of the results of its application to evaluate the acoustic field, the vibration prediction model is presented, followed by the results of its use in the vibration prediction. Several concluding remarks summarize the paper.

The acoustic noise model

Acoustic noise generated by turbulent flow is the result of partial conversion of the flow mechanical power W_m into acoustic power W_a according to the relation [1]:

$$W_a=\eta W_m \qquad (1)$$

with η the acoustic efficiency coefficient. The mechanical power can be determined from the relation:

$$W_m=\rho U_e^3 \pi d^2/8 \text{ (Watt)} \qquad (2)$$

with U_e the speed of the gas flow; ρ the gas density and d the diameter of one nozzle. For a multiple case of n nozzles, an effective diameter is used, calculated from the relation [2]:

$$d_{eff}=\sqrt{n}d \qquad (3)$$

The acoustic efficiency is determined by the relation:

$$\eta=\left(\frac{T_g}{T_a}\right)\left(\frac{\rho_g}{\rho_a}\right)K_a M^5 \qquad (4)$$

with K_a the coefficient of acoustic power ($\approx 10^{-5}$) [1] and M the stream Mach number relative to the ambient gas. According to [2] another model for the acoustic power is expressed by the relation:

$$W_a=0.005*nt_iU_e \qquad (5)$$

with t_i the thrust of a single nozzle. The power level in dB is calculated from the acoustic power using the relation :

$$L_W=10\log_{10}(W_a)+120 \qquad (6)$$

and based on L_w, the Overall Sound Pressure Level (OSPL) at a distance r (m) from the nozzle and at an angle θ relative to the stream's axis (Fig. 2) can be determined using the relation:

$$L_p=L_W+DI-10\log_{10}(4\pi r)^2 \qquad (7)$$

DI is the directivity factor, values of which are presented in Table 11.12 of [1]. The spectrum of the acoustic noise was calculated using data from Fig. 11.2 of [1]. The figure presents for each octave frequency band the values to be substracted from L_p, to obtain the sound pressure level in the given band.

Acoustic power values

The acoustic power was calculated from (1) and (5). In the implementation of (1), calculations were performed for several values of the acoustic efficiency coefficient η. Several values of η were tested. The results of the calculations from the two models (described by (1) and (5)) are presented in Table 1.

Table 1: Acoustic power values determined from models (1) and (5)

Model No	Acoustic power (Watt)	Acoustic power dB
1	0.64	118
1	2900	155
1	5800	157
5	2800	154

Overall Sound Pressure Levels

The Overall Sound Pressure Level was calculated from the values of the acoustic power presented in Table 1 along the system's axis for several distances from the nozzles. The calculations were for two directions, downstream (0 deg.) and upstream (180 deg.). The results appear in Table 2.

Table 2: Values of the Overall Sound Pressure Levels (dB)

L_w dB	θ	DI	Distance (m)			
			0.05	1	2	3.7
			OSPL(dB)			
157	0	0	159			
155	0	0	157			
118	0	0	120			
157	180	-17		130	126	123
155	180	-17		127	124	121
118	180	-17		90	87	84

The calculated results were evaluated relative to some previous measurements for a similar flow regime. The OSPL measured at the location above the nozzle, in a closed space was 160 dB . The closest values to this result are 159 and 157 dB in Table 2. It must be taken into account that for an acoustic noise measurement performed in an enclosed space, the measured level is higher by upto 6 dB relative to the measured value under free field conditions. Considering this fact, the source level at 0.05 m downstream in the case treated here was taken as 154 dB for free field conditions. The calculated OSPL in Table 2 of 159 dB and 157 dB are higher but close to the measured level.

The spectral content of the acoustic noise along the system

Under the above assumptions, the spectral content of the acoustic noise at 0.05m, 1m and 3.7 m from the nozzle, upstream of the system (180 degrees relative to the flow axis) is as appears in Fig. 3. These results were obtained under the assumption that the entire system is under free field conditions. In reality the sensitive element, at the 3.7 m location is in an enclosed space. Due to reverberation considerations a value of 6 dB was added to the level at that loction resulting in a total of 122 dB.

Vibration regime at sensitive elements

An empirical model [3] was implemented to evaluate the vibration regime induced by the acoustic noise at the sensitive elements. According to this model the power spectral density (PSD) of the acceleration at frequency f on a structural element, resulting from an acoustic excitation is given by:

$$G(f)=\beta^2Q^2(A/W)^2*P(f) \qquad (8)$$

with

β- an acousto-mechanical conversion factor
Q-an amplification factor
A-the structure's surface area
W-the structure's weight
P(f)- PSD of acoustic noise

Based on [3] values of β=2.5 and Q=5 were used. The PSD of the acceleration and derived displacement on the sensitive element were calculated, using the PSD of the acoustic noise, predicted for the distance of 3.7 and corrected for reverberation effects. The results are presented in Fig. 4

Summary

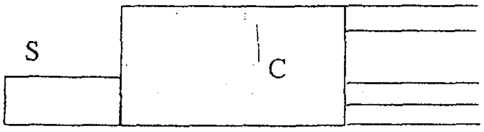
Acoustic noise and vibration were evaluated using empirical models along a system, exposed to acoustic noise generated by a turbulent flow of gas ejected from the system. The results obtained by calculation for the level of the acoustic noise were calibrated relative to measured data and the calibrated results were used for the vibration prediction. The use of the empirical models backed by measurements enabled a quick and preliminary evaluation of the expected vibration at the stage where neither measurements nor analytical evaluations could be performed .

References

1. D. A. Bies et al., "Engineering Noise Control Theory and Practice", London Unwin Hyman, 1988

2. 2."Acoustic Loads generated by the Propulsion System", NASA SP8072

3. 3. F. Spann et al. , "Component Vibration Environment Predictor", The Journal of Environmental Sciences, September /October, 1984



S-sensitive element; C-cooling pipe; E-ejector

Fig. 1: Schematic description of the system

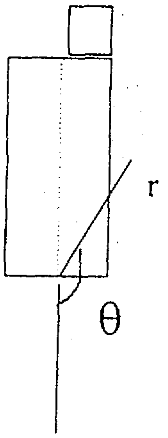


Fig. 2: Schematic description of the directivity elements of the source. (Downstream direction- 0 deg. to the flow axis is down the page)

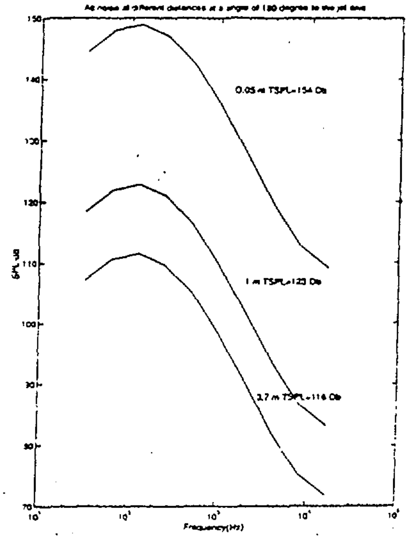


Fig. 3: Spectra of acoustic noise at several locations along the system.

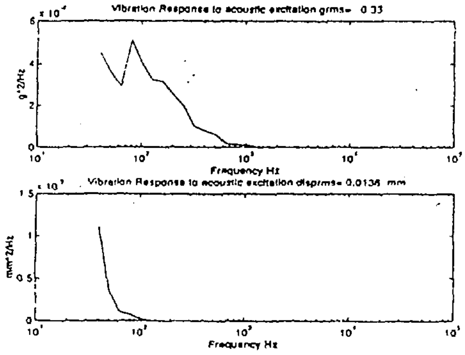


Fig. 4: The PSD of the acceleration and displacement of the excited sensitive element