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MECHANICAL-AERODYNAMIC FEEDBACK IN THE PROCESS OF SOUND GENERATION

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For many years phenomena of aerodynamic sound production have been known in the physical aspect, while a mathematical description of it is still insufficient. Therefore, investigations which have the increase of theoretical knowledge of this problem in view, have been performed. These investigations are concerned mainly with the interactions between aerodynamic and acoustic phenomena. However, in some cases of aerodynamic sound production, mechanical vibrations are of importance.

The aim of this work is investigation of the set-up in which mechanical vibrations play an important part in aerodynamic sound production. An attempt is made to explain the mutual aero-vibroacoustic interactions by means of feedback systems. Laboratory tests were carried out in order to perform the preliminary verification.

1. Introduction

Development of aircraft industry caused the main interest in aeroacoustics to be directed recently to the noise generated by supersonic flows. On the other hand, the noise of subsonic flow has been treated as a well-known phenomenon which belongs to the classic science. However, the conditions of sound generation by subsonic flows are known in terms of quality for simple arrangements only. In reality, more complicated systems occur. Therefore, the above problem is again an object of interest as can be seen in works [4] and [5].

Aerodynamic sounds containing the discrete frequencies frequently occur [1, 6, 7]. The systems producing the discrete frequency sounds depend on a good many parameters. The generation of such sounds is often found in fluid flow machines or installations, and it is still an insufficiently examined question.

The aim of this work is investigation of one example of the discrete frequency sound generated aerodynamically, and an analysis of influence of several factors on the generation conditions. It is a continuation of works [2, 3].

2. Generation conditions of definite frequency signals

The generation conditions of definite frequency signals may occur in feedback systems. An influence of the output signal with adequate amplitude and phase on the input of the system is characteristic for this kind of systems. A selection of adequate values of amplitude and phase, thus making the generation possible, leads to two basic conditions: phase condition and amplitude condition [8]. Acording to [8]: "phase condition for the oscillator lies in this that a sum of phase shifts in a system is a multiple of the round angle. Since the oscillation frequency is established when the phase condition is satisfied, phase condition can be called the frequency condition. The amplitude condition for the oscillator is a state, in which a loss of energy is less than the energy delivered to the system or equals the delivered energy. The equality occurs when a steady state of oscillations exists".

The edge tone production, described among others by POWELL, is an example of aerodynamically generated sound. Powell explains this phenomenon by means of aeroacoustic feedback as follows: disturbances (vortices) arise, when an air jet ($Re < 3000$) is issuing from a nozzle in the shape of long slit. When an air jet is issuing from a nozzle in the shape of the long slit, disturbances at the nozzle lip are arising and traveling downstream. As a result of the jet hitting the sharp edge downstream from the slit, a sound wave is emitted and propagates also upstream to the nozzle lip. The sound wave, reaching the slit, initiates a new disturbance propagating downstream. Thus, the influence of the acoustic field initiated by the flow again on the flow, makes a feedback loopc

This phenomenon may be also explained on the basis of the hydrodynamit theory, where it is assumed that the vortices occurring around the wedge affec. the changes of flow field at the nozzle lip. In more complicated cases, mechanical vibrations of the system accompany the generation of the aerodynamic sound. Thus, the above mentioned amplitude and phase conditions involve additionally the impact of the mechanical system on the aeroacoustic field. It has an effect on the enlargement of the system of feedback loops.

3. Block diagram of the test arrangement

The set-up presented in Fig. 1 was taken into consideration. An air jet issuing from a circular orifice hits a flat bar, whose lower part is inside the acoustic resonator. The discrete frequency sound generated in this set-up has some similar features to the edge tone, but there is also a difference in the range of velocity ($Re > 30\,000$), as well as in the shape of the nozzle. Moreover, the process of generation is more complicated due to mechanical vibrations of the bar. In order to draw a distinction between the two types of generation, the sound produced in the test arrangement was called the quasi-edge tone. Let us take into consideration the arrangement shown in Fig. 1, but without the acoustic resonator. Since the conditions required for the classical edge tone generation are not satisfied, a broad-band noise without the discrete frequencies is obtained. The amplitude condition is not satisfied due to the weak feedback between acoustic and aerodynamic fields. If the acoustic resonator (in the shape of the cylinder) is present, as shown in Fig. 1, an acoustic wave occurs at the edge of the bar and is reflected by the walls of the resonator. It results in reinforcement of some frequencies corresponding to the resonant frequencies of the resonator. If the walls of the acoustic resonator are rigid, it is a half-wave resonator, which for the basic tone is characterized by an antinode of acoustic velocity. Therefore, the bar is set in motion (transverse vibrations) corresponding to the resonant frequency of the resonator. The vibrations of the bar reinforce the acoustic wave, which then reinforces the intensity of vortices if the phase condition is fulfilled.

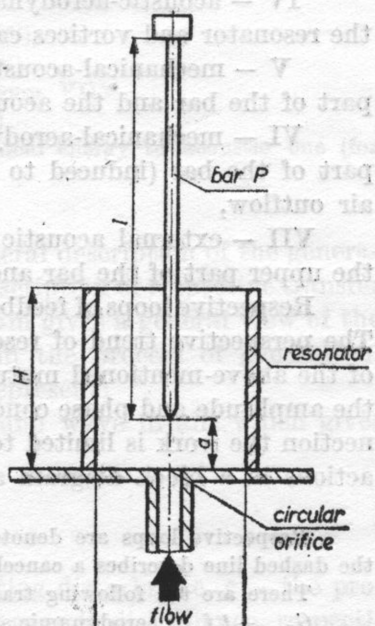

Fig. 1. Diagram of the test arrangement

Futhermore, the resonant phenomena of the vibrations of the bar can play a part in this process. Additional reinforcement of the vibrations of the system appears when the length of the bar is well chosen viz. one of the resonant frequencies of the bar corresponds to the frequency of the signal produced. This physical process may be described by means of a block diagram, whose elements correspond to the respective physical phenomena. The mutual interactions of aero-vibroacoustic phenomena lead to the feedback system.

There are two following types of feedback loops in the investigated arrangement:

1. Resonant loops, in which the output and input signals are of the same type. The transfer functions of these elements are characterized by a dominant frequency.

2. Non-resonant loops, in which the output and input signals are not of the same type.

The following feedback loops can occur:

resonant loops

I – aerodynamic feedback connected with disturbances of the jet at the outflow from the circular orifice,

II – acoustic feedbackcon nected with proper vibrations of the half-wave acoustic resonator,

III – mechanical feedback connected with transverse vibrations of the bar;

non-resonant loops

IV – acoustic-aerodynamic feedback between the acoustic wave inside the resonator and vortices cast off from the edge of the bar,

V – mechanical-acoustic feedback between the vibrations of the lower part of the bar and the acoustic wave in the resonator,

VI – mechanical-aerodynamic feedback between the vibrations of the lower part of the bar (induced to resonant vibrations) and the disturbances of the air outflow,

VII – external acoustic feedback between the acoustic wave emitted by the upper part of the bar and the acoustic wave inside the resonator.

Respective loops of feedback consist of blocks of adequate transfer functions. The perspective trend of research in this domain is a quantitative description of the above-mentioned mutual interactions, in order to determine analytically the amplitude and phase conditions. At this stage it is too difficult. In this connection the work is limited to the proposal of the formulation of mutual interactions in a block diagram as presented in Fig. 2.

Respective loops are denoted in Fig. 2 by Roman numerals (I-VII). Additionally, the dashed line describes a cancellation loop which is discussed further on.

There are the following transfer functions (*t.f.*) in Fig. 2:

G_v – *t.f.* of aerodynamic system,

G_{vv} – *t.f.* of aerodynamic feedback (loop I),

G_{va} – *t.f.* of system converting from aerodynamic energy to acoustic one,

G_a – *t.f.* of acoustic system (resonator),

G_{aa} – *t.f.* of acoustic feedback (loop II),

G_{md} – *t.f.* of mechanical system (lower part of the bar),

G_{mg} – *t.f.* of mechanical system (upper part of the bar),

G_{mm} – *t.f.* of mechanical feedback (loop III),

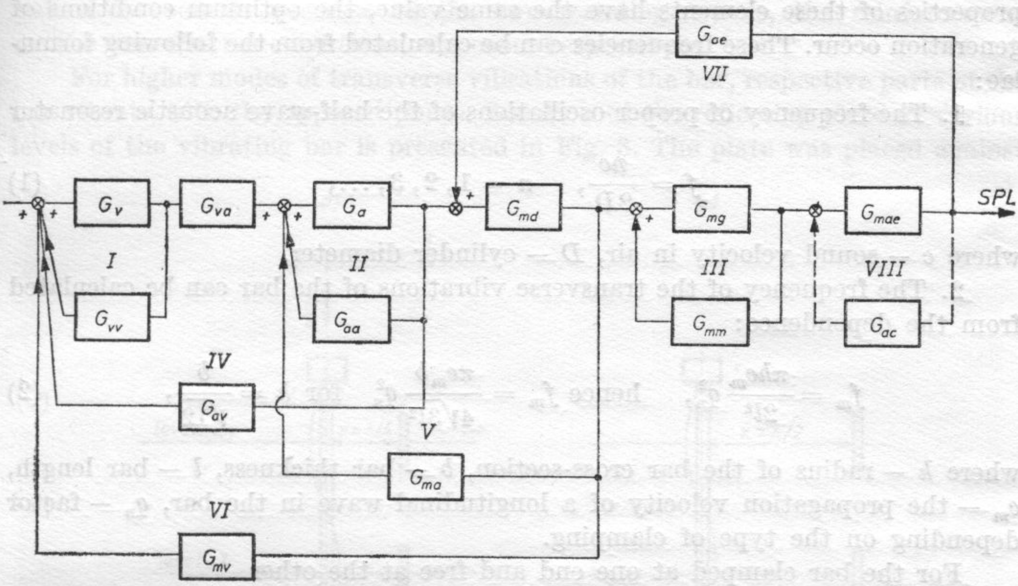

Fig. 2. Block diagram of the arrangement

G_{av} – *t.f.* of acoustic-aerodynamic feedback (loop IV),
G_{ma} – *t.f.* of mechanical-acoustical feedback (loop V),
G_{mv} – *t.f.* of mechanical-aerodynamic feedback (loop VI),
G_{ae} – *t.f.* of external acoustic feedback (loop VII),
G_{mae} – *t.f.* of system converting from the mechanical energy to acoustic one (for upper part of the bar),
G_{ac} – *t.f.* of cancellation system (loop VIII).

This block diagram is a first attempt at a general description of the generation phenomena. At the next stage of research a description of respective transfer functions is provided. Nevertheless, a block diagram gives a general view of the nature of phenomena which may be involved in the process of generation.

The transfer functions shown in Fig. 2 can represent:

1) a delay resulting from propagation of sound wave in air, which gives a phase shift

$$\varphi = \frac{\omega x}{c},$$

where x – the length of the path of the propagating disturbance, c – the propagation velocity of acoustic, mechanical or aerodynamic disturbance, respectively (it is connected with the phase condition);

2) resonant properties of oscillatory elements, which is connected with the amplitude condition.

There are three resonant elements in the test arrangement: the acoustic resonator, the bar and the flow of gas. If the frequencies describing the resonant properties of these elements have the same value, the optimum conditions of generation occur. These frequencies can be calculated from the following formulae:

1. The frequency of proper oscillations of the half-wave acoustic resonator

$$f = \frac{nc}{2D}, \quad n = 1, 2, 3, \ldots, \tag{1}$$

where c — sound velocity in air, D — cylinder diameter.

2. The frequency of the transverse vibrations of the bar can be calculated from the dependence:

$$f_m = \frac{\pi h c_m}{2l^2} q_n^2, \quad \text{hence } f_m = \frac{\pi c_m b}{4\sqrt{3}l^2} q_n^2 \quad \text{for } h = \frac{b}{\sqrt{12}}, \tag{2}$$

where h — radius of the bar cross-section, b — bar thickness, l — bar length, c_m — the propagation velocity of a longitudinal wave in the bar, q_n — factor depending on the type of clamping.

For the bar clamped at one end and free at the other

$$q_n = \frac{2n-1}{2}, \quad \text{for } n = 1, 2, 3, \ldots$$

3. STROUHAL frequency f_{sh}, which describes the maximum value of noise spectrum arising during a subsonic outflow at the velocity u of an air jet from a circular nozzle of diameter d.

It can be calculated using the formula

$$f_{sh} = \frac{Shu}{d}, \tag{3}$$

where Sh — Strouhal's number.

Influence of the external acoustic field on the generation conditions. It is evident from the analysis of the performance of the test arrangement that the acoustic resonator is of importance in the generation of the quasi-edge tone. We call it a basic resonator and the acoustic field produced by this resonator is called the internal field. As a result of multiple reflections between the walls of the resonator and a vibrating bar, a standing wave occurs. The standing wave causes the reinforcement of vibrations for resonant frequencies of the resonator. In consideration of the strong influence of the acoustic resonator on the generation conditions, the question arises: to what degree the external acoustic field can affect the condition of the resonator performance, i.e. on the process of generation. The external acoustic field can be obtained, if we create a resonator at the upper part of the vibrating bar as a reflecting plate against the bar. The additional standing wave can be created on the various levels of the bar (Fig. 3). According to the phase agreement or disagreement of the instantaneous values of acoustic pressure of the two standing waves, the reinforcement or the cancellation conditions, can occur respectively.

For higher modes of transverse vibrations of the bar, respective parts of the bar vibrate with the opposed phase. Location of the reflecting plate at various levels of the vibrating bar is presented in Fig. 3. The plate was placed against

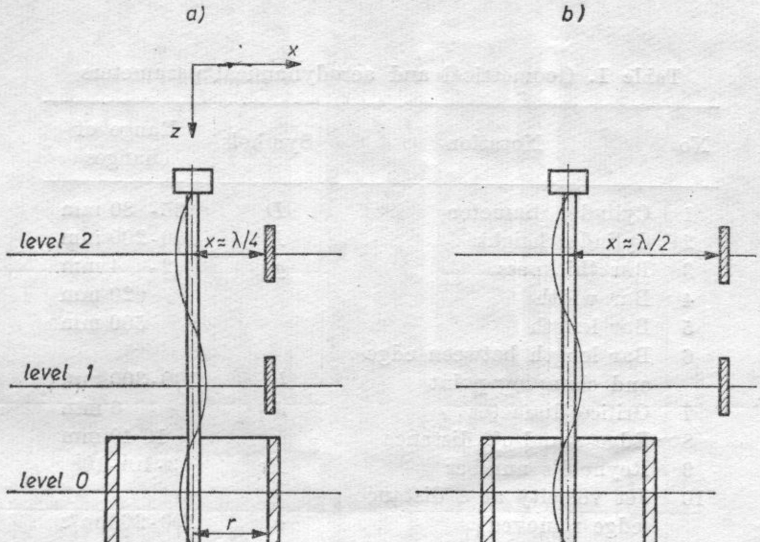

Fig. 3. Arrangement with an external acoustic field

a) cancellation on level 1 and reinforcement on level 2 (for $x = \lambda/4$); b) cancellation on level 1 and reinforcement on level 2 (for $x = \lambda/2$)

both parts of the vibrating bar. Levels 1 and 2 shown in Fig. 3 correspond to two parts of the bar with opposed phases.

On the basis of the theory of image sources [10] it can be proved that the conditions of reinforcement or cancellation for levels 1 and 2 are reversed if the reflecting plate is moved additionally by $\lambda/4$, Fig. 3b. These conditions are presented in the block diagram in Fig. 2 as the eighth feedback loop (VIII).

4. Experiments

The experimental arrangement is shown in Fig. 4. An air jet issuing from the circular orifice hits the sharp edge of the flat aluminium bar placed in the axis of the flow. The distance between the edge and the orifice can be changed stepfree. A cylindrical reflecting surface, resting upon the plate with the outflow orifice, surrounds the lower part of the bar. The range of dimensions and parameter changes being investigated is presented in Table 1.

The following magnitudes were measured: the sound pressure level, and the spectrum of the generated signal, the mechanical vibrations of the bar, and the stagnation pressure of the flow.

Table 1. Geometrical and aerodynamical parameters

No	Notation	Symbols	Range of changes
1	Cylinder diameter	D	35- 80 mm
2	Cylinder height	H	100-200 mm
3	Bar thickness	g	2- 4 mm
4	Bar width		20 mm
5	Bar length		500 mm
6	Bar length between edge and clamping point	l	100-300 mm
7	Orifice diameter	d	6 mm
8	Edge stand-off distance	a	10-40 mm
9	Reynold's number	Re	$3\cdot10^4$-10^5
10	Jet velocity at a distance (edge removed)	u	100-260 m/s

Acoustical measurements were carried out using the typical Brüel and Kjaer equipment containing a 1/2" condenser microphone with a preamplifier, the 2107 analyser and the 2304 level recorder. The set-up was tested in a semi-anechoic room. Acoustic pressure was picked-up by a microphone placed in the free progressive wave in order to avoid interference with a reverberant field.

Mechanical measurements were carried out using a 8307 type Brüel and Kjaer miniature accelerometer. The distributions of acceleration along the bar were measured for different fixing points of the bar. Aerodynamic measurements were carried out using a Pitot tube and a manometer. The stagnation pressure was measured at the jet axis in a jet core and at the distance $a = 30$ mm from the orifice. From these results, the velocity of air jet was calculated.

The following tests were carried out:

a) Dependence of frequency and amplitude of a discrete tone on the parameters of the test system.

b) The distribution of accelerations along the bar and the level of the generated discrete tone for different lengths of the bar.

c) Possibility and conditions of discrete tone cancellation by the acoustic feedback.

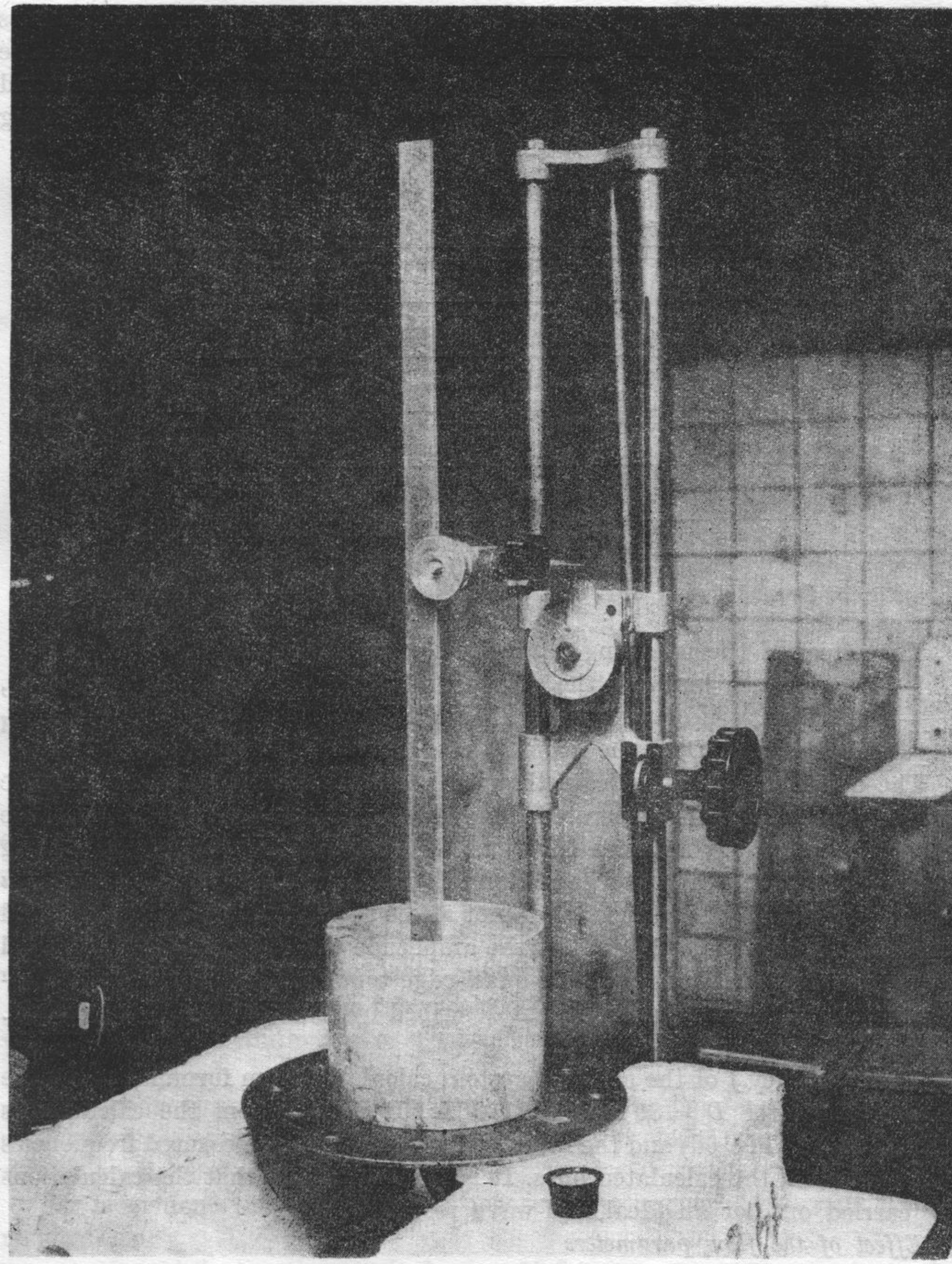

Fig. 4. Test arrangement

5. Results of measurements and calculations

a) Amplitude and frequency of a discrete tone. The results of measurements are presented in Fig. 5 and 6. The spectrum of the investigated quasi-edge tone is shown in Fig. 5 for the jet velocity $u = 220$ m/s. It is evident that a sound pressure level for $f_d = 2500$ Hz (SPL_{2500}) is about 120 dB, and the remaining part of the spectrum is lower by (20-30) dB.

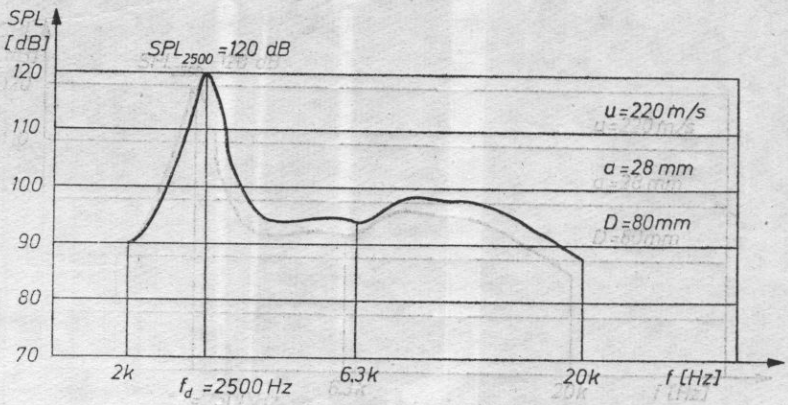

Fig. 5. Dependence of SPL on frequency for quasi-edge tone generation

The effect of the following parameters was tested:
— the distance between the orifice and the edge of the bar (Fig. 6a),
— the velocity u of the air jet (Fig. 6b),
— the diameter D of the acoustic resonator (Fig. 6c, 6d).

The obtained results indicate that the resonant frequency of the acoustic resonator determines the generated dominant frequency. It suggests that the feedback described by transfer functions G_{aa} and G_{ma} (Fig. 2) is very strong and determines the generation conditions. It was found that the jet velocity u and the edge stand-off distance (a) do not affect the frequency of the generated tone (Fig. 6a, 6b), while they affect the amplitude of the signal. It is expected that the greatest amplitudes of the quasi-edge tone should occur when the resonant frequencies of the arrangement described by relations (1)-(3) are similar.

Effect of the cylinder diameter

The frequency f of the acoustic system calculated from formula (1) for the resonator diameter $D = 80$ mm is 2150 Hz. It results from the calculations (dashed curve in Fig. 6d) and the measured results that the measured frequencies are greater than the calculated ones. It is due to the fact that the calculations were carried out for an ideal half-wave resonator.

Effect of the flow parameters

On the basis of analysis of the results shown in Fig. 6 it is evident that for the orifice diameter of 6 mm the best generation conditions occur when the jet velocity u at the edge is 220 m/s.

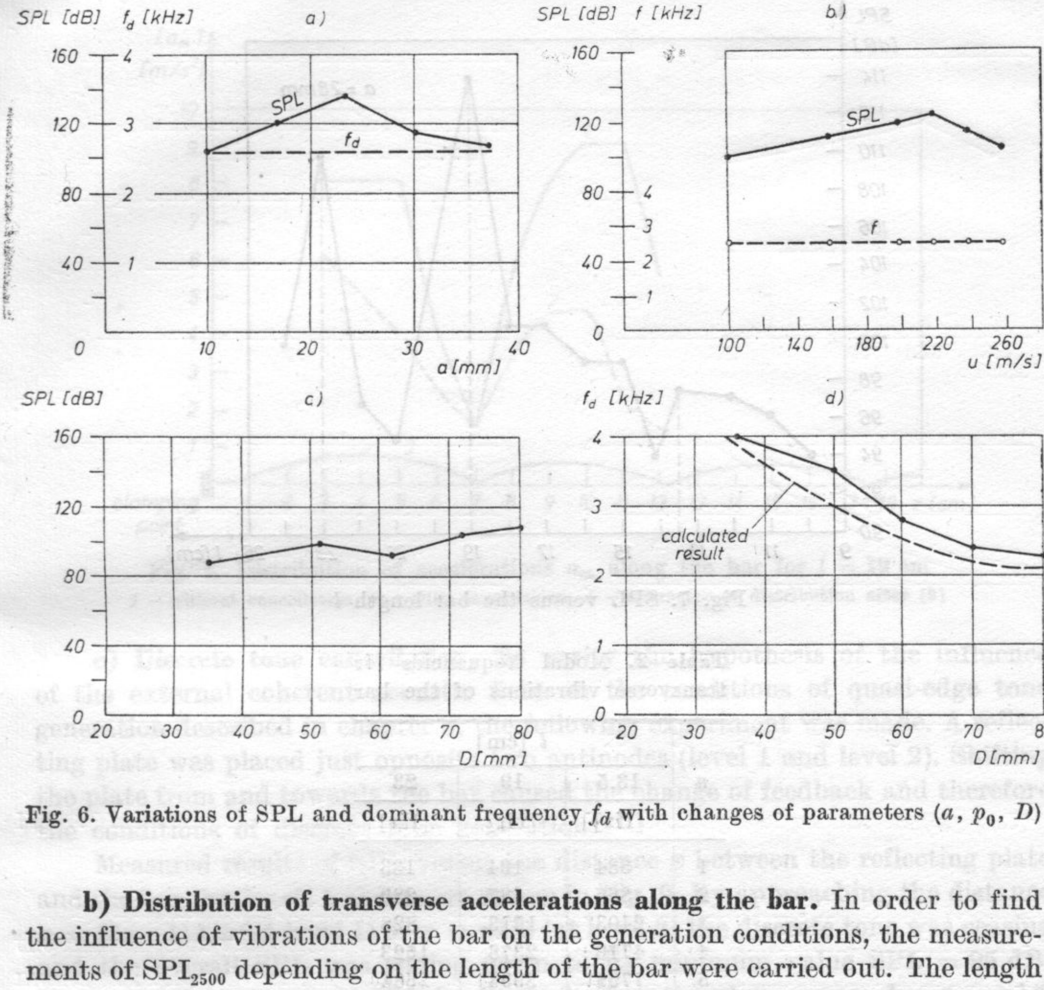

Fig. 6. Variations of SPL and dominant frequency f_d with changes of parameters (a, p_0, D)

b) Distribution of transverse accelerations along the bar. In order to find the influence of vibrations of the bar on the generation conditions, the measurements of SPI_{2500} depending on the length of the bar were carried out. The length of the bar l is the distance between the free end of the bar and the clamping point. The measurement results shown in Fig. 7 prove that the optimum generation conditions arise when l is 19 cm (the best conditions), 23 and 13.5 cm.

In order to calculate the frequency of transvere vibrations of the bar, the velocity of the longitudinal wave c_m was measured. It was found that the velocity $c_m = 5180$ m/s.

Inserting that result into formula (3) for the bar lengths of maximum generation, one can obtain the following values of modal frequencies for transverse vibrations of the bar (Table 2).

It is evident that the value of measured frequency ($f_d = 2500$ Hz) corresponds to:

— the third vibration mode for $l = 13.5$ cm,
— the fourth vibration mode for $l = 19$ cm,
— the fifth vibrations mode for $l = 23$ cm,

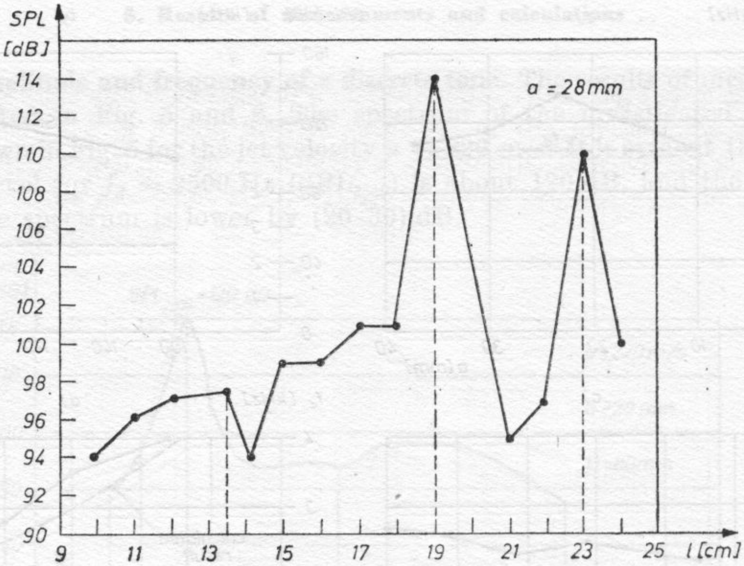

Fig. 7. SPL versus the bar length l

Table 2. Modal frequencies for transverse vibrations of the bar

n	l [cm] 13.5	19	23
	[Hz]	[Hz]	[Hz]
1	384	194	133
2	865	437	289
3	2403	1213	828
4	4710	2378	1623
5	7791	3934	2684

n – mode number, l – length of the bar

Acceleration was measured for the bar length $l = 19$ cm in order to check agreement with the calculated results. The results shown in Fig. 8 – curve *1*, indicate that the nodes of the standing wave in the bar correspond to distances of 7 and 12 cm from the fixing point, i.e. $0.368l$ and $0.631l$, respectively.

A chart of the vibration distribution for the clamped bars (9) indicates that the nodes of the standing wave for the fourth vibration mode correspond to distances of $0.355l$; $0.644l$; $0.906l$ (Fig. 8 – curve *3*). Thus, these values coincide well with the ones obtained in checking measurements.

The analysis performed proves that the maximum of SPL_{2500} corresponds to the resonant frequencies of the bar, which confirms the assumption that the vibrations of the bar are of substantial importance in the process of quasi-edge tone generation.

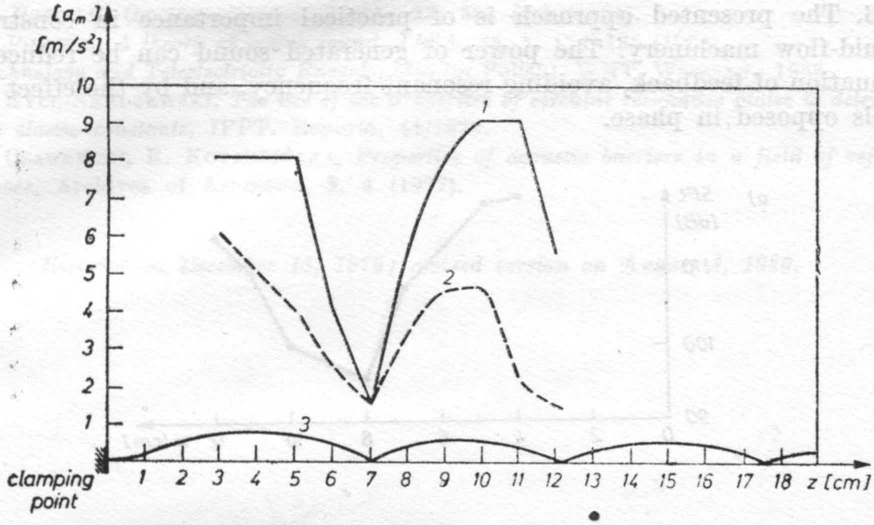

Fig. 8. Distribution of accelerations a_m along the bar for $l = 19$ cm
1 – without cancellation, *2* – with cancellation, *3* – character of distribution after [9]

c) Discrete tone cancellation. To verify the hypothesis of the influence of the external coherent acoustic field on the conditions of quasi-edge tone generation described in chapter 3, the following experiment was made. A reflecting plate was placed just opposite two antinodes (level 1 and level 2). Shifting the plate from and towards the bar caused the change of feedback and therefore the conditions of discrete tone generation.

Measured results of SPL versus the distance x between the reflecting plate and the bar, for levels 1 and 2, are shown in Fig. 9. By approaching the distance $x = (2n-1)\lambda/4$ (at level 1) or $x = n\lambda/2$ (at level 2) the discrete tone was ceasing and the overall SPL was sliding down to its minimum value SPL = 95 dB. Moreover, the increase of SPL is obtained when the plate approaches $x = n\lambda/2$ at level 1, or $x = (2n-1)\lambda/4$ at level 2. This corresponds to the considerations presented in chapter 3. The external coherent acoustic field has much the same influence on the vibration of the bar. The comparison of the vibration distribution without and with feedback due to the plate is shown in Fig. 8. One can see that for a negative feedback (the cancellation) vibration is much smaller (Fig. 8 – curve *2*).

Conclusions concerning discrete tone cancellation

1. A general hypothesis of the feedback between acoustic waves and bar vibration is proper.

2. Insertion of the reflecting plate against the bar enables the cancellation of the quasi-edge tone.

3. The presented approach is of practical importance in construction of fluid-flow machinery. The power of generated sound can be reduced by: attenuation of feedback, avoiding resonant frequency, and by the effect of the signals opposed in phase.

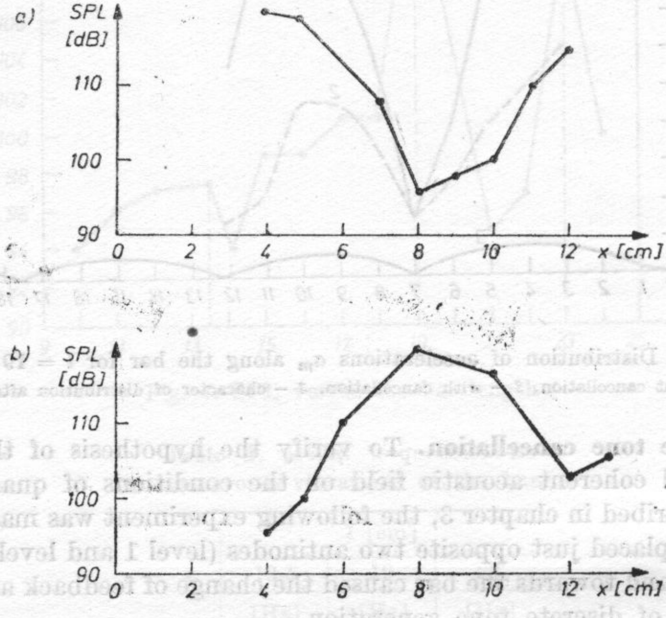

Fig. 9. SPL versus distance x between the plate and the bar
a) plate shifted on level 2, b) plate shifted on level 1

It is assumed that these phenomena are more complicated in reality. This work is only a preliminary attempt at a total description of the feedback between aerodynamic, acoustic and mechanical systems.

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