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THE "WHISTLER NOZZLE" AND HORN AS AERO-ACOUSTIC SOUND SOURCES IN PIPE SYSTEMS

A Hirschberg (1), J C Bruggeman (2), A P J Wijnands (1) and M Morgenstern (1)

(1) Eindhoven University of Technology, The Netherlands
(2) TNO Institute of Applied Physics, Delft, The Netherlands

SUMMARY

The production of self sustained pulsations in a "whistler nozzle" is analysed in terms of periodic shedding of vortices. It is shown that very high acoustic velocities can be obtained by combining the nozzle with an upstream closed side branch. Acoustic velocities of half the main flow velocity have been observed in a nozzle with rounded edges. In a similar way, in combination with a closed side branch a horn appears to be a powerful sound source.

INTRODUCTION

In the course of a systematic study of low frequency aero-acoustic sound sources in pipe systems we became interested in the "whistler nozzle". A whistler nozzle is a pipe termination with a collar (Fig.1). For critical combinations of collar geometry and mean flow velocity U_0, high amplitude self sustained acoustic oscillations were reported by Hill and Greene [1] at low main flow Mach number. Hussain and Hasan [2] extensively studied the phenomenon. The sound production was attributed to"impingement of the shear layer"on the sharp edge at the end of the collar. The shear layer is formed as a result of flow separation at the end of the main pipe, where the pipe diameter increases abruptly from D to D_c. In view of the large amplitudes of the acoustic velocity u' at the entrance of the collar ($u'_{rms}/U_0 = 0.12$) it is likely that the shear layer behaviour is best described in terms of periodic shedding of discrete vortices. Vortex formation was indeed observed by Hussain and Hasan [2] in a whistler nozzle and by Wilson et.al. [3] in a very similar configuration. In such cases the theory of"vortex sound" of Powell [4] and Howe [5] yields a powerful basis to analyse the problem. Inspired by the study of the flute by Howe [6], Bruggeman [7,8] recently used the concept of vortex sound to analyse the aero-acoustic behaviour of closed side branches along a pipe. Flow visualization and Laser Doppler velocity measurements confirmed the validity of flow modeling assuming the shedding of one single line vortex each period. Such models tend to overestimate the acoustic power produced or absorbed by the vortex, but they yield excellent insight in the kind of problems considered here [7,8,9,10].

In the experiments of Hill and Greene [1] the acoustic resonator, which determines the frequency f of oscillation and the acoustic field distribution, is the pipe segment delimited downstream by the whistler nozzle and upstream by the settling chamber (Fig.1).

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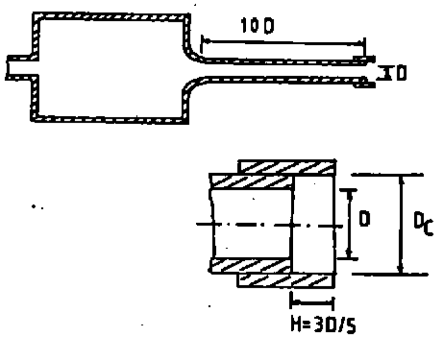

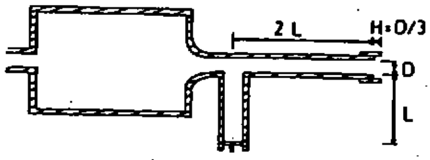

Fig.1 Original whistler nozzle setup. Fig.2 Modified setup with closed side branch.

We placed a closed side branch with adjustable length L, just downstream of the settling chamber (Fig. 2). A closed side branch placed along a pipe is an almost perfect reflector for acoustic waves with a wave length of four times L. When the downstream pipe segment had twice the side branch length (2L) and by matching the collar geometry we obtained a pulsation level $u'_{rms}/U_0 \doteq 0.32$. This is shown in Fig. 3 where the measured pulsation level is given as a function of the Strouhal number $Sr_D = fD/U_0$. The setup used is described by Bruggeman [7]. The pressure pulsations were measured with piezo electrical gauges. Acoustic velocities were deduced from the pressure measurements by assuming the fundamental acoustic mode of the pipe system to be dominant or by using a two microphone method.

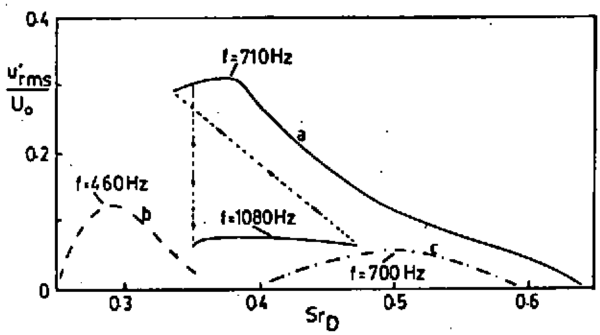

Fig.3 Influence of main flow velocity U_0 on the amplitude of the acoustic velocity u' at the main pipe termination.(a:______ modified setup, see Fig.2, b:- - - original setup, see Fig.1, c: _._._._ side branch with unflanged pipe termination). $D_c/D = (38/30)$, D=0.03 m.

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The mean flow velocity was calculated from volume flow measurements with a turbine meter. The amplitude observed is more than twice the level obtained with the original setup (curve b in Fig. 3) or with the modified setup with an unflanged pipe termination, i.e. without nozzle (curve c in Fig. 3). As the acoustic field upstream of the side branch was very weak we expect that the device will also work without the settling chamber.

We will analyse the sound production in the T-junction between the main pipe and a closed side branch and in the whistler nozzle. We then will use the results of this analysis to obtain an improved "whistler nozzle" geometry. Finally the interaction between the two sound sources will be discussed.

For the purpose of the discussion we define the range of low amplitude level as the range for which linear theory applies to the shear layer behaviour. If linear theory correctly describes the interaction of the acoustic field with the shear layer at the separation point but the shear layer does not behave linearly we say that the pulsation level is moderate. When the acoustic velocity strongly influence the behaviour of the shear layer at the separation point we say that we have a high amplitude level. The exact definition of these ranges depend on the value of the radius of curvature r of the wall at the separation point.

Roughly we can state that ($u'/U_0 < 10^{-3}$) is the range of low amplitude levels, ($10^{-3} < u'/U_0 < 10^{-1}$) is the range of moderate amplitude levels and ($u'/U_0 > 10^{-1}$) is the high amplitude range.

SOUND PRODUCTION BY VORTEX SHEDDING IN A CLOSED SIDE BRANCH

Following Howe [5] in a low Mach number M and high Reynolds number Re isentropic flow the instantaneous power P generated by a nonstationary vorticity distribution w in an acoustic field u' is given by:

$$P = -\rho \int_V \mathbf{u}' . (\mathbf{w} \times \mathbf{v})\, dy \qquad (1)$$

Where ρ is the average fluid density, v is the local velocity including the stationary U and nonstationary u' potential flow velocities, w = rot v and the integral is carried out over the volume V of non-vanishing w. Vorticity is injected into the main flow by boundary layer separation at edges. The resulting shear layers are unstable. Small flow perturbations at the separation point caused by interaction with the acoustic field induce a roll up of the shear layer into regions of concentrated vorticity. These coherent structures can qualitatively be described as line vortices. For moderate acoustic amplitudes ($u'_{rms}/U_0 < 0.1$) Laser Doppler measurements show [7] that the rate of vorticity shedding at the upstream edge of the T-junction is still close to its value for an undisturbed stationary shear layer ($U_0^2/2$). Hence the vortex strength will grow linearly in time during a period of pulsation and further remains constant. The vortex path is also almost independent of the acoustic velocity amplitude and is in first approximation a straight line from the upstream edge to the downstream edge of the T-junction. The convective velocity U_c of the vortex is close to $U_0/2$. It is determined by the main stationary potential flow U and by the vortices present in the T-junction and their images in the walls. At low frequencies and moderate amplitudes U_c appears to be about 0.4 U_0, the magic velocity predicted by the linear theory of a free shear layer as the "most instable" frequency [11]. Linear theory does however not apply in our case [7]. At high amplitudes such as $u'_{rms}/U_0 = 0.3$, we observed an increasing effect of the acoustic velocity on the strength and path of the vortex. The amount of shed vorticity significantly increases [5]. Flow visualisation shows that the path of the vortex is bent into the side branch. This results in a decrease in the convective velocity of the vortex (lower local value of U).

In the fundamental acoustic mode of the pipe configuration considered, there are no acoustic waves upstream of the closed side branch. For this particular acoustic velocity distribution at the T-junction, the vortex will absorb energy in the first half period of pulsation after the moment at which it has been shed. Vortex shedding occurs at the moment when the acoustic velocity turns from outwards to inwards in the side branch, i.e. when the pressure at the end of the side branch reaches a minimum. At the T-junction the initial absorption can be compensated for by sound production in the second half period of oscillation because the acoustic velocity changes sign. When the travel time of the vortex across the T-junction is about one period there is a maximum net gain of acoustic energy. This leads for a given U_0 to an optimal frequency f or Strouhal number $Sr_0 = fD/U_0$. At moderate amplitudes we find $Sr_0=0.5$ while at high pulsation amplitudes we find $Sr_0=0.32$ [7,8]. This is a direct consequence of the decrease in convective velocity U_c of the vortex with increasing acoustic amplitude.

SOUND PRODUCTION IN THE WHISTLER NOZZLE AND HORN

The simple model of a single line vortex convected along a straight line parallel to the pipe axis will now be applied to obtain some insight into the production of sound in a whistler nozzle. For this purpose we compare in Fig.4 the behaviour of an unflanged open pipe termination with that of a whistler nozzle.

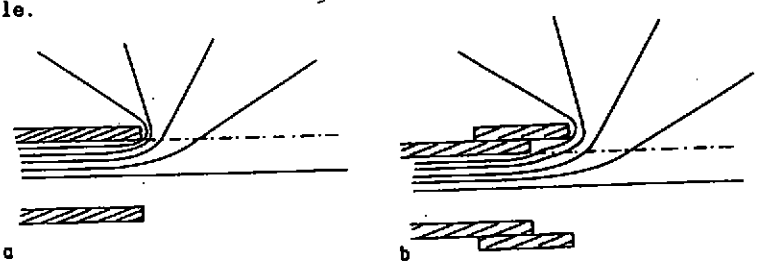

Fig.4 Qualitative comparison of acoustic stream lines (_____) and vortex path (-·-·-·-) at an unflanged pipe termination (a) and in the whistler nozzle (b).

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At the main pipe termination vortex shedding occurs when the local acoustic velocity u' turns towards the exit of the pipe. In the case of an unflanged pipe, vortex shedding results in a strong initial absorption in the first half period of oscillation. This is due to the locally high acoustic velocity amplitude at the sharp edge and because the path of the vortex is almost perpendicular to the acoustic streamlines. This makes $u' \cdot (w \times v)$ in formula (1) very large. The roughly linear growth of the vortex strength with time is not sufficient to compensate for this initial absorption by production in the second half period of oscillation because the amplitude of u' decreases very rapidly as the vortex travels away from the pipe exit. Vortex shedding at an unflanged pipe termination results in strong sound absorption as shown in Fig.5.

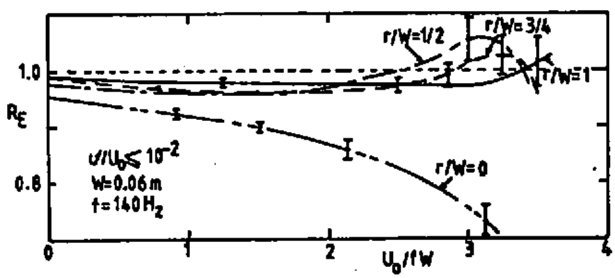

Fig.5 Energy reflection coefficients of an unflanged pipe termination. (r/W=0) and horns (r/W=1/2,3/4 or 1) measured at low excitation levels $u'/U_0 = 10^{-2}$. Square pipe cross section, width:W=0.06 m.

The reflection coefficient was measured with a two microphone method [7]. In the case of a whistler nozzle the initial singularity of the acoustic field at the separation point is less pronounced and the acoustic streamlines are bent by the collar towards the direction of the vortex path. Both effects reduce considerably the initial absorption by the vortex in the first half period of oscillation after its birth. At the end of the collar the acoustic field starts diverging, the vortex path cuts the acoustic streamlines. The acoustic velocity amplitude is still large. Therefore if the travel time of the vortex over the length H of the collar roughly matches the period of oscillation acoustic power can be produced by vortex shedding. This means that the condition for maximum pulsation level has to be expressed in terms of a Strouhal number $Sr_H=fH/U_0$ based on the length H of the collar.

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At moderate amplitudes we find an optimal pulsation level at a value $Sr_H = 0.2$ (curve b in Fig.3). This is much smaller than the value $Sr_D = 0.5$ found at the maximum pulsation level for a setup with a side branch but without nozzle (curve c in Fig.3). In the latter case the optimum Sr is determined by the convective velocity $U_c = 0.4\ U_0$ of the vortices at the T-junction. The low value $Sr_H=0.2$ at maximum pulsation level indicates a lower convective velocity of the vortices in the collar of the whistler nozzle. This can be explained by the strong effect of images due to the vicinity of the collar walls. This influence of the images on the convective velocity U_c also explains the observation of Hussain and Hasan [2] that the optimal collar length H increases with increasing collar expansion ratio D_c/D. Indeed a larger distance of the vortices to the walls implies an increase in U_c. The optimal collar length is also dependent on the amplitude . At moderate amplitudes the optimal value is $H=3D/5$ (Fig.1) while at high amplitudes it is $H=D/3$ (Fig.2). We expect that the increase in the amount of shed vorticity with increasing amplitude is responsible for this effect.

Another non-linear effect typical for high amplitudes is the hysteresis observed (curve a in Fig.3) when the acoustic field jumps from the fundamental mode (f=710 Hz) to a higher mode (f=1080 Hz) as the main flow velocity is increased. Similar hysteresis was observed at high amplitudes in a double side branch system by Bruggeman [7].

As stated by Howe [5] vortex damping at sharp edges is a very efficient way of reducing pulsations. This was confirmed in a spectacular way by our experiments in a double side branch system [7,8]. Our analysis of the sound production in a whistler nozzle indicates that sharp edges are not essential to sound production in a whistler nozzle. Indeed experiments show that rounding the edges at the exit of the collar (radius of curvature $r=D/10$) results in a significant increase in pulsation level from $u'_{rms}/U_0=0.32$ to 0.37. This is the result of a reduction of the absorption of sound by the "parasitic" vortex shedding at the end of the collar. Pushing this idea further, in order to get rid of this parasitic vortex shedding, we decided to replace the whistler nozzle by a horn. In the original setup (Fig.1) a horn did not produce flow pulsations, but in combination with a side branch (Fig.2) a horn with radius of curvature r in between W/2 and W produced pulsation levels up to $u'_{rms}/U_0=0.45$!! (Fig.6).

Please note that the data of Fig.5 and 6 were obtained in a setup with pipes with square cross section, width W. To make comparison with the data obtained in a setup with circular pipes (Fig.3) easier we use the rule proposed by Bruggeman [7] : $Sr_D= 4fW/\pi\ U_0$. In this way the Strouhal condition for maximum pulsation level of a setup with a closed side branch with square pipe cross section becomes identical to that with circular pipe cross section.
From the data shown in Fig.5 and 6 we see that the reflection coefficient R_E of a horn can become larger than unity. The flow velocity U_0 at the maximum of R_E appears to be a function of r.

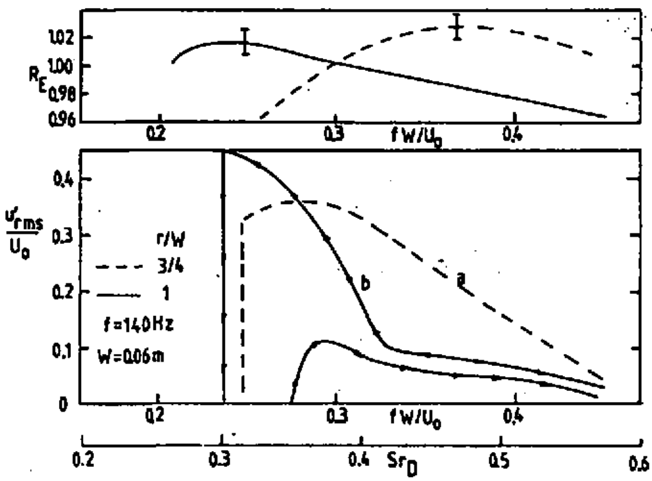

Fig.6 Pulsation level and reflection coefficient R_E of the horn in a pipe system with side branch and pipe termination with horn.(a: r=3W/4 and b: r=W; square pipe cross section, width W=0.06 m , D_{eff}=4W/π).

Mounting the horn in a baffle did not influence the pulsation level indicating that sharp edges are not essential at all for sound production. This conclusion has already been drawn by Wilson e.a. [3] in their beautiful study on the mechanism of whistling. The baffled horn is a common element in pipe systems since junctions of small pipes with main pipes or settling chambers are often designed with rounded edges. Some authors even advice the use of rounded edges to suppress pulsations [12].

INTERACTION BETWEEN SOUND SOURCES

When a closed side branch L is placed 2L upstream of a pipe exit with a horn an almost perfectly closed acoustic resonator is formed at the resonance frequency of the side branch. At high Reynolds numbers such as considered here (10^6<Re<10^8) friction may be negligible so that for rigid pipes the pulsation amplitude is determined by the non linear behaviour of the sources [5,7,8]. When two aero-acoustic sources are present such as in the case of the setup shown in Fig.2 or in a double side branch system, the acoustic coupling between the sources will induce a lock in of the sources. The coupling of vortex shedding at both sources by a common acoustic field may lead to a conflict between the sources if their optimal Strouhal conditions for resonance are different. This occurs in the case of a setup with side branch and a horn. In the case r=W (curve b in Fig. 6) the maximum of the reflection coefficient R_E of the horn is found at Sr_D=0.3. The optimum for the side branch is at moderate pulsation level Sr_D=0.5 and shifts with increasing pulsation level to $Sr_0=0.32$ [7]. We see from Fig.6 that at high Strouhal numbers ($Sr_0 > 0.4$) the side branch controls the pulsation behaviour. When the velocity is gradually increased, there is at $Sr_0=0.4$ a sudden increase in pulsation amplitude indicating that both sources are now cooperating. Below $Sr_0=0.3$ the pulsations stop abruptly. The velocity has to be decreased until $Sr_0=0.35$ is reached, before pulsations start again. This indicates that the behaviour of the horn is strongly influenced by the acoustic field generated by the side branch.

The dependence of vortex shedding on the presence of an acoustic field, makes such sources very attractive as targets for anti-sound. Using a very primitive feedback system and a loudspeaker placed outside the pipe, close to the pipe exit, we could easily reduce the pulsation level by 40 dB.

CONCLUSIONS

The concept of vortex sound yields considerable insight into the mechanism of sound production in pipe systems. Sharp edges appear not to be essential for sound production. On the contrary, when placed at critical positions sharp edges may considerably reduce self sustained pulsations. Combining a whistler nozzle or a horn with a closed side branch of variable length we obtained a powerfull, compact and versatile device for modulation of flow through a pipe. The sound production is due to a lock in of the sound source at the T-junction of the side branch with the sound source in the nozzle.

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