INFORMATION REPORT INFORMATION REPORT

CENTRAL INTELLIGENCE AGENCY

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50X1-HUM

CONFIDENTIAL

COUNTRY	Poland	REPORT	[redacted]
SUBJECT	Acoustics in Poland/Static Siren Developed by Lesniak and Maczewski-Rowinski/Assessment by Expert in Ultrasonics	DATE DISTR.	6 May 63
		NO. PAGES	2
		REFERENCES	[redacted]
DATE OF INFO.	[redacted]		
PLACE & DATE ACQ.			

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[redacted]

[redacted] the following: 50X1-HUM

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[redacted] A 19 page English translation of [redacted] "A Static Siren by the Central Institute of Labor Safety - Poland, by B. Lesniak and B. Maczewski-Rowinski".

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2. The paper discusses the theoretical considerations for the development of a "Static Siren" (Multi-Whistle) using six convergent-divergent DeLaval Nozzles. The choice of the DeLaval Nozzles (diverging throat) was based on the steady increase in intensity proportional to pressure increase found by B Lesniak. This is opposed to the maximum plateau generally observed with Hartmann (converging throat) generators.

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There is also a theoretical discussion of the Hartmann Whistle as well as design considerations for the DeLaval Nozzle, secondary resonance chamber, exponential and conical horns.

3. There is shown experimental work using a single generator in a conical horn, with a flat secondary resonance chamber bottom connected thereto. Briefly, the results for the four different DeLaval Nozzles are:

Power output:	20.6 to 106 watts
Air consumption:	10.5 to 57.6 m^3/hr. (6.2 to 34 CFM)
Pressure:	5 to 7 atm (71.4 to 100 psi)

4. A new static siren will be built using six nozzles with a "theoretical" free field power output reaching 635 watts. A novel feature of the design will be ports built into the horn to remove (by compressor suction) most of the air used for sound generation.

5. From the data given in this well documented paper, [redacted] the single whistle developed by the author do not differ to a great extent from the Boucher Monowhistle or from the Russian whistle of W P Kurkin. However, the design of the Multi Whistle with air suction within the horn seems a very interesting improvement copied from the Levavasseur siren. The air consumption of the new Multi Whistle is relatively high for the power output. The more original feature seems to be the design of several single whistles which allow a nearly continuous frequency shift between 14.6 and 29.3 KC. 50X1-HUM

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ROUTING AND CONTROL RECORD

DATE
30 April 1963

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TITLE

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[redacted]

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A STATIC SIREN BY THE CENTRAL INSTITUTE OF LABOR SAFETY

- POLAND -

Following is a translation of an article by B.Lesniak and B.Maczewski-Rowinski in the Polish-language periodical Ochrona Pracy (Labor Safety), No 11, November 1962, pages 19-25.

Introduction.

Among various methods of generating a strong acoustic field, dynamic air generators also commonly called ultrasonic sirens have become most prominent. It should be noted, however, that this nomenclature is incorrect in most cases, because audible (audioacoustic) frequencies are used for practical purposes, especially for coagulation of aerosols, drying and emulsification. For this reason a more general term, namely "acoustic sirens", is used to describe them and more specifically "audio-acoustic" or "ultra-acoustic" whichever the case may be.

Interest has been shown in recent years for the method of generating acoustic waves, which utilizes hydrodynamic flow generators known already for a few decades. Special among them is the whistle invented by the Danish physicist HARTMANN. Such whistles are placed at the focus of a parabolic reflector and thus a source of unidirectional sound radiation is obtained.

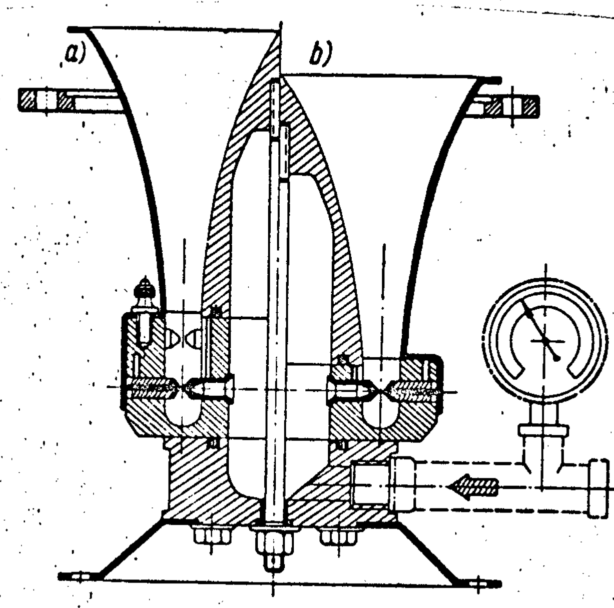

Fig.1 Static ... of whistles
b)with

Static Sirens by Boucher (a) with two rows of whistles
(b) with single row of whistles.

The power generated by one such source was usually insufficient for practical purposes; therefore, the whistles by HARTMANN were for a long time used only in laboratory research.

Only BOUCHER's application of an acoustical horn to the HARTMANN whistles has resulted in a new type of siren, called "static" (fig.1). Such name is justified fully by the fact, that there are no moving parts whatsoever in this device. The siren contains several HARTMANN whistles located on the periphery of a ring chamber, which is also equipped with an acoustic horn. Other types of static sirens, namely those by SZKOLNIKOWA, KURKIN and TARTAKOWSKI were built in a similar manner.

In Poland, the development of a static siren was for the first time undertaken by the Central Institute of Labor Safety (Centralny Instytut Ochrony Pracy) with the cooperation of the Institute of Principal Problems of Technology at the Polish Academy of Sciences (IPPT-PAN). This siren is designed with a new type of whistle having a DE LAVAL nozzle.

Another innovation in the developed siren is a foreseen possibilty of generating "pure" waves without an acoustic whiff. This is done by means of properly designed channels which suck away the air blown into the horn by the whistle.

The advantage of static sirens lies in the easy and inexpensive construction, the absence of any rotating parts and its compactness.

Its disadvantages include the difficulty of generating waves at frequencies below five kilocycles/second. This is unfavorable in many cases of practical application. (Lately the CIOP and IPPT-PAN have undertaken the development of a new type of static generator for audio-acoustic waves of greater power and lower frequency).

The Construction and Operating Principle of Acoustic Flow Generators (Whistles).

Acoustic flow generators utilize phenomena which appear when gases leave a round nozzle at critical or at above critical velocity. When gas flows out of the nozzle, rarification waves and oblique impact waves are generated in the gas stream, but a steady gas flow is established with periodic pressure variations in the direction of the main stream. This is due to the multiple reflection of the waves from the boundary surface of the stream. BUSEMANN has found, that the stream of a gas flowing out of a nozzle is divided into fields between the ends of its edges. Same pressures and velocities prevail in each successive field. The passages of compression or rarification waves from one field to another occur more or less simultaneously.

Fig.2 shows the pressure profile in a gas stream as measured at the axis of symmetry by means of a PITOT tube.

The pressure profile becomes wave shaped at higher pressure differences as a result of generated shock waves in which supersonic velocity is converted into subsonic velocity as the entropy increases. At the same time, pressure losses result: they are largest in the fields of highest velocity, that is where the magnitude of pressure is minimum; they are smallest in the fields of lowest velocity or maximum pressure.

In order to obtain acoustical vibrations, HARTMANN placed a resonator tube from x_1 to x_2 or from x_3 to x_4 (fig.2). This resonator was periodically (in a pulsating manner) filled by and emptied of the air stream. HARTMANN worked out on this basis an approximate formula defining the frequency of generated waves:

$$f = \frac{c}{4(h_r + 0{,}3\, d_r)} \qquad [1]$$

where c- is the velocity of sound (meters/second) in the gas under the condition when the diameter d_r (cm) of the resonator is equal to its depth h_r (cm).

For air the generated frequency was from a few kilocycles /second up to 120 kilocycles/second. At an incoming air pressure of two atmospheres gage and at a frequency f= 28.2 kilocycles/second the acoustic power amounted to 13.4 Watts and several times more at higher pressures and lower frequencies. This relates to the size of the resonator diameter and to the amount of expended air. (The emitted power increases with higher air expenditure). The efficiency of the whistles was η = 4 to 5%.

HARTMANN, BOUCHER, SAVORY and others were of the opinion, that in order to obtain acoustical vibrations, the stream velocity of the gas flowing out of a nozzle must exceed the velocity of sound. However, as is well known, they used converging nozzles which could not satisfy this condition. Other authors thought that the nozzle ought to have such shape as to enable the pressure head at its exit to reach the proper magnitude.

In Poland, B.LESNIAK has in recent years compared the action of converging nozzles and DE LAVAL nozzles by examining the effect of the velocity and exit pressure head of the gas on the emitted acoustic power.

He conducted comparative tests on the change of acoustic field intensity with three converging and three DE LAVAL nozzles. The exit cross-sections of both types of nozzles were the same. Each nozzle was in turn operating with the same resonator at identical frequencies and at maximum emitted power.

The pressure of the entering air was maintained within the limits between 2 and 6.4 atmospheres gage.

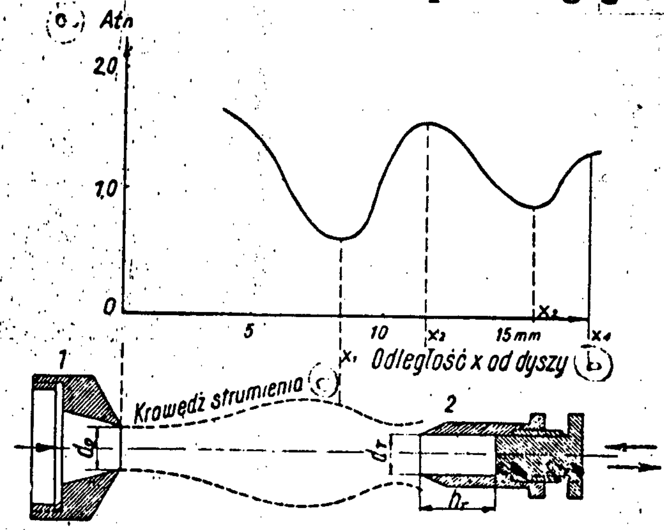

Fig.2 Pressure profile in a gas stream flowing out of a converging nozzle (HARTMANN): 1- nozzle, 2 - resonator, d_1- exit diameter of nozzle, d_r-inside diameter of resonator, h_r- depth of resonator.
a) atmospheres gage
b) distance x from nozzle
c) edge of stream

Fig.3 shows the characteristics of the discussed nozzles in the form of graphs.

Investigations have shown that for a MACH number M=1.4 and M=1.5 the maximum intensity of the acoustic field was obtained with converging nozzles at a gas pressure of about 4.5 to 5.25 atmospheres gage. On the other hand, no maximum intensity was observed with DE LAVAL nozzles - the intensity as function of pressure was increasing greatly, steadily and almost proportionally. It was also found, that the expenditure of air supplying DE LAVAL nozzles was lower than that for the converging nozzles. This leads to the conclusion, that the acoustical efficiency of DE LAVAL nozzles is greater than the efficiency of converging nozzles.

Within the project with the cooperation of IPPT-PAN, the entire set of whistles of the new type was utilized in the experimental single-whistle siren. The latter served for the preliminary establishment of basic parameters absolutely necessary for the development of the multi-whistle static siren.

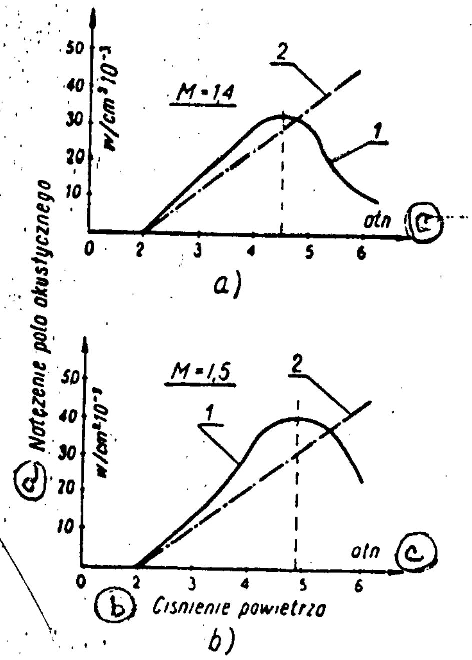

Fig.3 Comparison between converging nozzle and DE LAVAL nozzle characteristics: a) for a MACH number M=1.4; b) for a MACH number M=1.5; 1 - for a converging nozzle, 2 - for a DE LAVAL nozzle.
Measurements were made at a distance of 40 cm from the sound source.
(a) Intensity of the acoustic field
(b) Air pressure
(c) Atmospheres gage

A set consisting of a whistle and a resonator placed inside a secondary resonating chamber is suitable for interchanging nozzles, regulation of the resonator depth h (fig. 2), varying the distance from nozzle to resonator and also varying the depth H of the secondary chamber (fig. 4). The generating set is equipped with an acoustic horn, thus forming a single-whistle siren shown in fig. 4.

For the purpose of calculating the parameters of the acoustic horn, the basic frequency f= 2000 cycles/second and d_0=15 millimeters as diameter of the horn entrance were assumed.

The calculation was made starting out with the equation for the effective horn cross-section:

$$\frac{\partial^2 y}{\partial x^2} - \left(\frac{\omega}{c}\right)^2 \cdot (1 - \tau^2)\, y = 0 \qquad [2]$$

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With the proper values of constants, we obtain

$$y = y_0 \left[\cosh\left(\frac{x}{h}\right) + T\sinh\left(\frac{x}{h}\right)\right] \qquad [3]$$

$$S = S_0 \left[\cosh\left(\frac{x}{h}\right) + T\sinh\left(\frac{x}{h}\right)\right]^2 \qquad [4]$$

where

S - cross-section area of the horn at the distance x, cm^2

S - cross-section area of the horn at the entrance cm^2

T - shape factor of the horn

y - radius of the horn cross-section at the distance x, cm

x - distance of the given cross-section from the apex of the horn cone, cm

c - velocity of sound, cm/sec (c=34,400 cm/sec)

ω - angular velocity

τ - transmission coefficient

$h = \frac{c}{2\pi f}$ contraction factor of the horn

f - damping frequency, cycles/second

With the values T=1 and $y = y_0 e^{x/h}$ the horn has an exponential profile; with T=0 the profile becomes catenary; when $T = h/x_0$ and h goes to infinity, then the horn becomes conical with an angle

$$2\vartheta = 2\tan^{-1}\left(\frac{y_0}{x_0}\right)$$

First, the dimensions of the exponential horn were calculated according to the formula $y = y_0 e^{x/h}$ for the assumed parameter values f= 2000 cycles/second and $y_0 = d_0/2 = 0.75$cm $1/h = 0.365$

It must be pointed out that the horn with an exponential profile has an advantage; namely, its resistance at the entrance just above the critical frequency f_0 is independent of the frequency and is equal to the wave resistance of air. In horns with other profile this characteristic appears at much higher frequencies.

In the particular case here, however, the exponential horn proved out to be too short in view of structural considerations; therefore, a conical horn was designed and used.

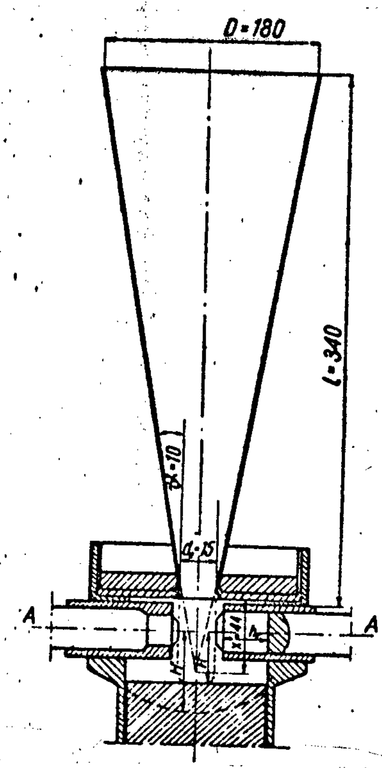

Fig.4 Sketch of a static single-whistle siren with a conical horn and a secondary resonance chamber.
H, H' - distance of the flat and of the concave bottom of the chamber from the axis A-A, d_o- entrance diameter of the horn, D - exit diameter of the horn, x - distance of given cross-section from the apex of the horn cone, h - depth of the resonator, ϑ- apex angle of the horn.

The shape factor of the conical horn is defined by the equation

$$T = \left(\frac{x}{x_o}\right) \tag{5}$$

with h going to infinity and with the apex angle $\vartheta = \frac{\gamma}{2}$, or

$$\vartheta = \tan^{-1}(y_o/x_o) \tag{6}$$

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An angle 2ϑ from 10° to 20° is often used. When the angle ϑ is too small, then the horn has a low acoustic efficiency. The exit diameter D of the horn must be greater than the emitted wavelength λ, in order to avoid the reflection of the wave back into the horn. The size of the exit diameter of the conical horn depends on its length l.

In the case under consideration, with f=2000 cycles/sec and λ= 170 millimeters, D should be greater than 170 mm; for 2ϑ=20° the length of the horn is 340 mm (fig. 4).

The damping frequency was checked according to MORSE:

$$f=\frac{c(a-b)}{2\,lb} \qquad (7)$$

where

c - velocity of sound (34,400 cm/sec)
a - radius of the exit cross-section of the horn (9 cm)
b - radius of the entrance cross-section (0.75cm)
l - length of the horn (34 cm)

upon substitution of these values

$f= 1770 < 2000$ cycles/second was obtained.

Secondary Resonance Chamber

The secondary resonance chamber plays an important role in static sirens. A properly shaped chamber increases the efficiency of the siren - as tests conducted by BOUCHER have shown. However, the problem of the resonance chamber has not yet been fully solved, nor is an exact method of design and calculation known.

In the design of a secondary cylindrical resonance chamber one may use calculations similar to those for cylindrical fifes. Therefore, the authors of this article have adopted the general BERNUOILLI equation used everywhere for the calculation of cylindrical fifes:

$$N_n = \frac{2n-1}{4H} \cdot e \qquad (8)$$

where

H - depth of the fife

n = 1,2,3 ...

N - frequency of consecutive harmonics

but for the fundamental frequency:

$$N_1 = \frac{c}{4H} \tag{9}$$

This method can be applied to the case when the chamber diameter is small compared to the emitted wavelength.

The design of the secondary resonance chamber, carried out by the authors, is based on the modified BERNUOLLI equation:

$$H = n\frac{\lambda}{4} \tag{10}$$

from which follows for the case considered here:

$$\lambda = \frac{c}{f} = \frac{34400}{2000} = 17.20 \text{ cm}$$

$$\frac{\lambda}{4} = 17.20/4 = 4.30 \text{ cm} = 43 \text{ mm}$$

but at higher frequencies, for example f= 23400 cycles/sec

$$\frac{\lambda}{4} = \frac{34400}{4x23400} = 3.675 \text{ mm}$$

$H_{min} = \frac{\lambda}{4} = 43$ mm was decided on with due consideration of structural factors and of the feasibility to generate a few harmonics. Besides, the adjustable bottom of the chamber makes it possible to regulate the depth within wide limits.

As to the effect of the bottom shape, only the results of BOUCHER's experiments are known; they indicate, that at lower pressures (f.e. 4 atmospheres gage) a flat bottom is more advantageous. In view of this, a flat bottom was provided for the secondary resonance chamber of the experimental single-whistle siren.

As a result of the above analysis, it was possible to develop a simplified single-whistle siren (fig. 5). The flat bottom in this siren can be replaced by a bottom of a more suitable shape to yield maximum acoustic efficiency, in case other pressures are applied or different frequencies are generated.

Tests have shown, that:

the average intensity of the acoustic field along the siren axis amounted to	0.1 Watt/cm^2
intensity of sound	150 decibels
total power radiated	approximately 15 Watts
diameter of the nozzle and the resonator	3.5 mm

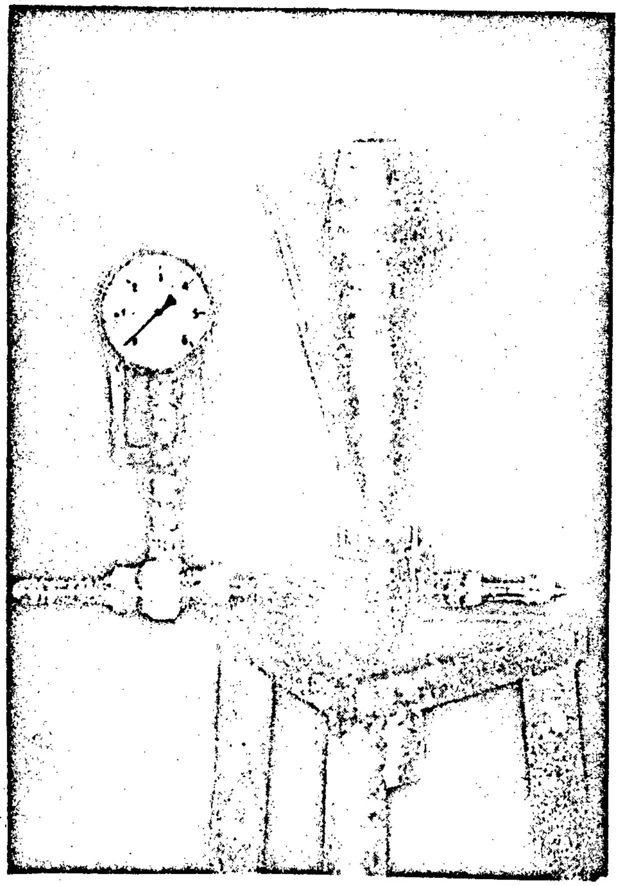

Fig. 5 Single-whistle static siren by CIOP-IPPT-PAN.

Development of a Multi-Whistle Siren.

In a system of several whistles the operation of each one is affected by the others. The performance of whistles in a system changes in reference to the performance of a single sound source , mainly as a function of their mutual distances. The field generated by a multi-source system is the result of interaction between several sources and acoustic interference phenomena. This problem has not yet been worked out. The majority of designs (not many in existence) of acoustical static sirens is probably made on experimental basis.

In order to arrive at an approximate evaluation of the radiation characteristics of a whistle system, the authors of this work attempted to utilize the results of analysis well known in acoustics: namely, the analysis of radiation from membranes and other systems of sources.

One of the most essential factors which determines the mode of radiation from a given source of sound is the direction coefficient R. Its value depends on the shape and on the size of the source.

In the case of a system of n radiating sources spaced uniformly around the circumference of a circle of radius a, this coefficient can represented in the form of an infinite BESSEL series:

$$R = J_o(ka \sin\gamma) + 2\sum_{p=1}^{\infty} j^{pn} \cdot J_{pn}(ka \sin\gamma) \cdot \cos pn\varphi \qquad [11]$$

where

γ, φ are defined by the coordinates system

J_o, J_p are symbols of the function

k - is the wave number ($k = \frac{\omega}{c} = \frac{2\pi f}{c}$)

j - is the symbol of phase rotation ($j = \sqrt{-1}$, translator's note)

BESSEL functions converge fast for small arguments; therefore, for practical purposes one may consider the first terms of the sum only

$$R = J_0 \cdot (ka \sin\gamma) + 2\,j^n J_n \cdot (ka \sin\gamma) \cos n\varphi + {} + 2\,j^{2n} J_{2n} (ka \sin\gamma) \cos 2\,n\varphi + \ldots$$

The first term expresses the direction coefficient of a source system densely distributed around the circumference of a circle; the second term gives the correction due to the fact that the number of sources is finite.

If the distance between sources along the circumference is smaller than the emitted wavelength, then the second term can be neglected; the third term can be disregarded already for $n \geq 3$.

By analogy, considering a multi-whistle siren, one may state: starting out with a system of n whistles the distances between them being close to the emitted wavelength, a further increase of the number of whistles will not influence fundamentally the radiation mode of the entire sytem.

Calculations and Their Results for the Multi-Whistle Static Siren by the CIOP.

Following assumptions were made: 1)whistles with the DE LAVAL nozzle will be used; 2)number of whistles - six; 3)whistles will be placed in separate resonance chambers; 4)a simplified horn will be made with an exponential profile; 5)the siren will be able to emit "pure" acoustic waves, within the limits of practicality; to accomplish this, proper channels will be built into its body by means of which the return air can be sucked out of the horn.

The diameter of the secondary resonance chamber and the entrance diameter of the horn was made 0.8 cm; this diameter should be greater than 1.4 d_3(d_3-exit diameter of nozzle), following the recommendations by KURKIN and TARTAKOWSKI. The horn dimensions were calculated on the basis of successive cross-sections, from formula (4) and the following others:

$S = S_o e^{\beta x}$ (S_o - entrance area of the horn, 0.5 cm^2 at x=0; x - distance from the ordinate axis, cm; β - exponent of the curve, 1/cm;)

$\beta = \frac{4\pi}{\lambda_n}$ = 0.4 or 0.3 1/cm where λ_n is the maximum acoustic wavelength, cm.

Thus:

$$= \frac{4\pi}{\beta} = \frac{4\pi}{0.4} = 31.4 \text{ cm}$$

while the frequency of damping

$$f_{min} = \frac{c}{\lambda_n} = \frac{34400}{31.4} = 1095 \text{ cycles/second}$$

Fig. 6 shows the profile of the horn exponentially curved for β=0.4/cm and for β=0.3/cm, while λ=41.8 cm and f_{min} = 822 cycles/second.

The second of these curves was adopted for the construction of the horn; its exponent is close to the one used by SZKOLNIKOWA. According her investigations, this horn shape is suitable for the frequencies applied here.

In order to simplify the construction, it was assumed that the external generatrix of the horn is a straight line and coincides with the axis of abscissae according to equation $S_x = S_o e^{\beta x}$. The coordinates system was rotated by α = 10° from the vertical position. A similar horn is also used by other authors.

Four sets of nozzles (fig. 7) were designed; their diameters are d_2= 2, 2.5, 3 and 4 mm at the smallest cross-section. The gas pressure is to be p_1= 5 and 7 atmospheres absolute.

In the nozzle with a diameter d_2= 2 mm, the critical pressure p_2 was

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$$p_2 = p_1 \left(\frac{2}{\varkappa + 1} \right)^{\frac{\varkappa}{\varkappa - 1}} \qquad (12)$$

$$p_2 = 5 \left(\frac{2}{2.41} \right)^{\frac{1.41}{0.41}} = 2.65 \text{ atmospheres absolute}$$

with pressure p_1 = 5 atmospheres absolute

The specific volume of air V_1 at p_1= 5atm.abs and t= 20°C is

$$V_1 = \frac{RT}{p_1} = \frac{29.27 \times 293}{5 \times 10^4} = 0.173 \text{ m}^3/\text{kilogram}$$

where

R - gas constant for air

T - absolute temperature

The critical velocity is

$$c_2 = \sqrt{2g \frac{\varkappa}{\varkappa + 1} \cdot p_1 V_1} = 316 \text{ m/sek} \qquad [13]$$

The exit velocity is

$$c_3 = \sqrt{2g \frac{\varkappa}{\varkappa - 1} \cdot p_1 V_1 \cdot \left[1 - \left(\frac{p_2}{p_1} \right)^{\frac{\varkappa - 1}{\varkappa}} \right]} = 466 \text{ m/sek} \qquad [14]$$

For p_1= 7 atm.abs. and t=20°C, the results were respectively:
V_1= 0.123 m^3/kilogram c_2= 314 m/sec. c_3=500 m/sec.

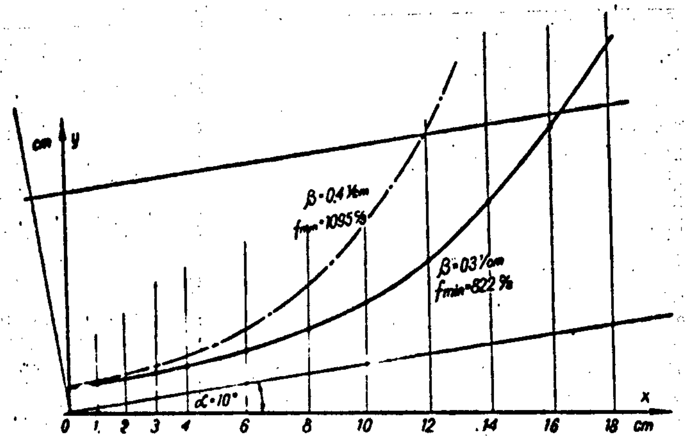

Fig. 6 Profile of a simplified horn (exponentially curved) for the multi-whistle static siren by CIOP, with the exponent β_1=0.4 and β_2=0.3 1/cm

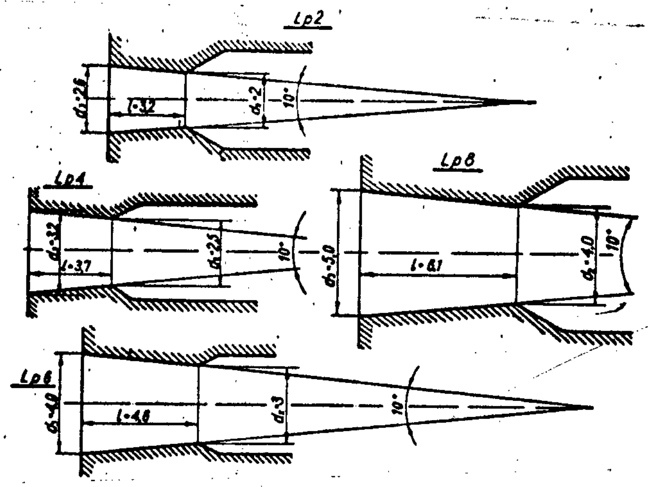

Fig. 7 Contour outlines of several DE LAVAL type nozzles used for the whistles in the CIOP static siren for a pressure $p_1 = 7$ atm.abs. (see table 1).

The diameter d_3 was calculated from the ratio of cross-sections s_2/s_3 and from the equation

$$\frac{s_2}{s_3} = \frac{s_2}{s_3} = \left(\frac{p_1}{p_3}\right)^{\frac{1}{\varkappa}} \cdot \left(\frac{p_3}{p_1}\right)^{\frac{1}{\varkappa}} \cdot \sqrt{\frac{\varkappa+1}{\varkappa-1}\left[1-\left(\frac{p_3}{p_1}\right)^{\frac{\varkappa-1}{\varkappa}}\right]} \qquad (15)$$

After substituting appropriate values and further transformation, following results were obtained:

for $d_2 = 2$mm and $p_1 = 5$ atm.abs.

$$s_3 = \frac{s_2}{0.725} = \frac{3.14}{0.725} = 4.33 \text{ mm}^2 \quad \text{and} \quad d_3 = 2.35 \text{ mm}$$

for $d_2 = 2$mm and $p_1 = 7$ atm.abs.

$$s_3 = 5.02 \text{ mm}^2 \text{ and } d_3 = 2.57 \text{ mm}$$

The expenditure of air Q_2 at $p_1 = 5$ atm.abs. is

$$Q_2 = s_3 \cdot c_3 \cdot 3600 = \frac{4.33}{10^4}(466)(3600) = 7.3 \text{ m}^3/\text{hour}$$

and 9 m^3/hour at $p_1 = 7$ atm.abs.

In the final design of nozzles, the HARTMANN formula was used for simplicity's sake; this is an equation for the nozzle output per cm^2 of nozzle (m^3/minute-cm^2) as a function of pressure p_1, namely:

$$\frac{q}{d^2} = 0.852\,(p_1 + 1.033) \qquad (16)$$

where

q - per unit output, m^3/minute

d - diameter of the nozzle exit, cm^2

p_1- pressure, kilograms/cm^2

The ratio of cross-sections s_2/s_3 is always the same for the assumed parameters (p_1= 5 and 7 atm. abs; p_2= 1 atm. abs.)

Table 1. Characteristic Data of Whistles and Siren.

1) order number
2) pressure
3) diameter of nozzle
4) narrowest
5) exit
6) angle between generatrices
7) in angular degrees
8) length of nozzle
9) resonator
10) aperture diameter
11) depth
12) ratio
13) distance from nozzle to resonator
14) velocity of air
15) critical
16) exit
17) air output of whistle *)
18) air output of entire siren
19) calculated fundamental frequency
20) calculated power of one whistle
21) theoretical acoustic power of a six-whistle siren
22) remarks
23) whistles emit many harmonics simultaneously
24) nozzles no. 2,4,6,8 (see fig. 7) were chosen for construction
25)*)calculated according to HARTMANN

Table 1.

Characteristic Data of Whistles and Siren

1 Lp.	2 Ciśnienie p_1		3 Średnica dyszy		7 Kąt rozwarcia tworzących	8 Długość dyszy	9 Rezonator		12 Stosunek	13 Odległość dyszy od rezonatora	14 Prędkość powietrza		17 Wydatek powietrza gwizdka *)	18 Wydatek powietrza całej syreny	19 Obliczona podstawowa częstotliwość	20 Obliczona moc 1 gwizdka	21 [illegible] moc akustyczna syreny 6 gwizdkowej	22 Uwagi
			4 najwęższa	5 wylotowa			10 średnica otworu	11 głębokość			15 krytyczna	16 wylotowa						
	atn	ata	d_z mm	d_e mm	w stopniach	l mm	d_r mm	h mm	$d_r : d_z$	l_r mm	c_1 m/sek	c_2 m/sek	Q_1 m³/godz	Q_s m³/godz	f c/s	W	W	
1	4	5	2	2,30	10	2,00	3,5	3,5	1,5	3,5	316	466	10,5	69,0	29300	20,6	125	Gwizdki emitują równocześnie liczne harmoniczne 23
2	6	7	2	2,60	10	3,20	3,8	3,8	1,5	3,8	314	500	12,0	72,0	29300	26,6	158	
3	4	5	2,5	2,90	10	2,46	4,4	4,4	1,5	4,4	316	466	16,0	96,0	23400	32,2	195	
4	6	7	2,5	3,20	10	3,75	4,7	4,7	1,5	4,7	314	500	22,4	134,0	23400	41,5	248	
5	4	5	3	3,60	10	3,03	5,3	5,3	1,5	5,3	316	466	23,1	138,0	19500	46,5	279	
6	6	7	3	4,10	10	4,60	5,7	5,7	1,5	5,7	314	500	32,4	194,0	19500	60,0	360	
7	4	5	4	4,70	10	4,12	7.1	7,1	1,5	7,1	316	466	41,2	247,0	14600	82,5	485	
8	6	7	4	5,10	10	6,10	7,6	7,6	1,5	7,6	314	500	57,6	345,0	14600	106,0	635	

Do wykonania przyjęto dysze Lp. 2, 4, 6 i 8 (patrz rys. 7) 24   *) Obliczone wg Hartmanna 25

The following formula by HARTMANN was used for the approximate calculation of the per unit acoustic power emitted by the whistles

$$\frac{J}{d^2} = 295 \sqrt{p - 0.93} \quad \text{Watt/cm}^2 \qquad (17)$$

where d is in centimeters and p is in kilograms/cm^2

The contour sketches of several nozzles used for the static siren by the CIOP is shown in fig. 7.

The results of calculation of the main siren parameters are presented in table 1.

The ratio d_r: h of the diameter to the depth of the resonating chamber was chosen 1.5 on the basis of the authors' own tests as well as those by SZKOLNIKOWA and BOUCHER. This ratio differs from the one recommended by HARTMANN (d_r:h=1).

The building of the nozzles was planned without sharply cut edges at the exit, because they erode quickly during operation. This is probably caused by cavitation. SZKOLNIKOWA used flat edges with sleeves of hard thermo-setting material, which gave good results. In the design of the siren discussed here, a similar structural solution of the problem was planned using, however, other materials.

Should it be necessary to operate with only a minimum of acoustic whiff, the air blown through the nozzles will be sucked out through six vent holes (4 in fig.8) to the blower and then carried into suction duct of the compressor. The arrows in fig. 8 show the path of air circulation. The siren is now being built in accordance with the design discussed here and should be soon ready for laboratory tests.

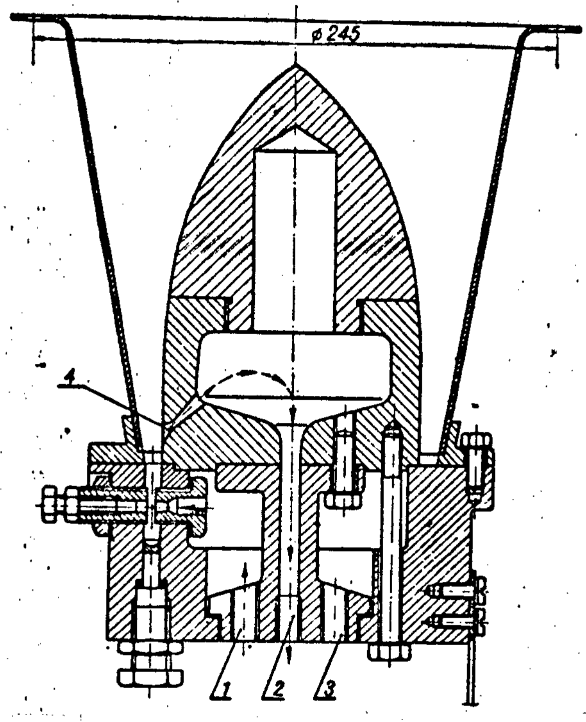

Fig. 8. Sketch of the multi-whistle static siren by CIOP with DE LAVAL whistles (type IPPT-PAN): 1 - incoming compressed air, 2 - suction of the air to the compressor, 3 - location of the manometer, 4 - vent for sucking away the air.

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