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EXPERIMENTAL EVALUATION OF ACOUSTIC
AGGLOMERATORS AS AN AIR CLEANING SYSTEM
CONCEPT FOR EMERGENCY USE IN LMFBR PLANTS

FINAL REPORT

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EXECUTIVE SUMMARY

- Acoustic Agglomeration can be carried out in two operating modes - traveling-wave and standing-wave. From the specific energy consumption viewpoint (hp/1000 SCFM) the standing-wave operation is more effective than the traveling-wave operation by a factor of 2 to 4.
- Three agglomeration mechanisms should be considered in the theoretical modeling: orthokinetic, hydrodynamic and turbulent interactions. The orthokinetic acoustic agglomeration is highly frequency dependent, while the hydrodynamic and turbulent acoustic agglomeration are relatively insensitive to frequency in their operation.
- Based on measurements of particle size distribution function as a function of time under intensive acoustic waves, it is found that low frequency agglomerators are more suitable than high frequency ones (larger than 1 kHz) because of lower acoustic attenuation in the aerosol and the higher acoustic generation efficiency.
- For incontainment direct acoustic agglomeration application only traveling wave operation is feasible. The problem of protecting personnel and equipment either during the routine testing of acoustic systems or the accidental startup of the acoustic system is difficult. For this reason the recirculating acoustic system plus cyclone is preferable. In this case the acoustic system can be isolated by inlet and outlet acoustic insulators outside the reactor containment building.

- Narrow band acoustic waves (monotone sound) should be used for acoustic agglomeration to reduce the acoustic insulation cost. For single frequency sound, very effective insulation can be achieved by conventionally designed acoustic mufflers. The power required for acoustic operation is about 160 dB. The acoustic agglomeration rate increases according to the square root of intensity for LMFBR. Low acoustic frequency (400-700 Hz) is recommended.
- A movie which was presented to DOE, was made of high intensity acoustic agglomeration under standing and traveling-waves. It shows the repeated agglomeration of NaCl aerosol in a transparent agglomeration chamber powered by a high intensity (160 dB) vibrating piston sound generator.
- Among the three high-power generators - the mechanically vibrating piston, the dynamic siren, and the static whistle - the vibrating piston seems to be the best in terms of experimental setup, but it is also the most expensive one. The whistles are the cheapest, but the difficulty of expulsion of the spent air is a complex and unresolved problem.

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I. INTRODUCTION

In the development of the Liquid Metal Fast Breeder Reactor, a major safety problem concerns the suppression of the sodium oxides aerosol particles produced during a hypothetical core disruptive accident.

In a recent report entitled, "Evaluation of Air Cleaning System Concepts for Emergency Use in LMFBR Plants"[1], twenty-four different air cleaning systems were investigated. Among these, direct in-containment acoustic agglomeration is rated to be number 1 in terms of effectiveness and number 4 in terms of fabrication. These suggest that if research and development effort can improve the credibility, reliability and compatibility of this concept, it may be one of the top candidates for such emergency application. The present paper presents some theoretical calculations concerning sonic agglomeration rates of such a system in terms of acoustic frequency, intensity and particles. The information is useful in providing some guidelines for the evaluation and design of sonic systems for LMFBR emergency applications. A review on aerosol generation in emergency situations in LMFBR plants can be found in reference [2].

Sonic agglomeration has been an interesting subject of many investigations ever since the first experiment by Patterson and Cawood in 1931 [3] who discovered that aerosol particles are quickly agglomerated in a standing-wave sound tube. Shortly afterward independent acoustic

agglomeration experiments were carried out by Brandt, Freund, and Heidemann [4] in Germany; Andrade and his co-workers in Great Britain [5]; and St. Clair at the United States Bureau of Mines [6]. Their results generated great enthusiasms for acoustic agglomeration and eventually large scale industrial testing of its effectiveness. During the second World War, serious efforts were made by the United States Navy and Army to use acoustic dissipation of fogs for the improvement of visibility for landing at airports and on aircraft carriers. Although the field testings gave a positive result in calm weather, this technique was proved to be ineffective under moderately high-wind conditions and the project was abandoned [7]. During the post-war years several commercial models of high power sirens with an acoustic efficiency of the order of 40-60% were available and this spurred a widespread interest in the application of sound waves for the control of particulate emissions from power plants, steel mills, and other processing and chemical industries. The work in this period was summarized by Mednikov [8], Shirokova [9] and more recently by Shaw [10].

II. TRAVELING-WAVE OPERATION

II.1 Acoustic Agglomeration Constants

The objective here is to make direct experimental measurements on the acoustic agglomeration constants. Two different experimental setups are used in the present measurements: One uses a high power electromagnetic speaker operated in the relatively low frequency range (1 to 3 kHz), and the other uses an aerodynamic siren which can generate high-frequency sound (10 to 20 kHz).

Figure 1 shows schematically a flow-through aerosol test tube (7' long x 6" dia.) with an EM speaker used as the sound source. The aerosol particles are injected from below and flow upward at a constant speed. A sonic frequency analyzer (Bruel and Kjaer Model 2121) with a 1/2" condenser microphone is used to determine the sonic frequency and intensity. An inverted cone of foam rubber is placed on the bottom plate to prevent the reflection of the sound wave and to ensure a relatively homogeneous traveling-wave field inside the tube.

Moderately monodisperse aerosols are generated by either nebulizing polystyrene latex (PSL) particles or by nucleation of dioctyl phthalate (DOP). In the former case the aerosol is generated from the PSL suspensions (supplied by the Dow Chemical Co.) using an atomizer type generator. An impactor is used for the elimination of the large droplets to reduce the number of multiple aggregates. A Krypton-85 charge neutralizer is used to eliminate the possible residual electrostatic charges. Then the PSL particles flow through a drier where the moisture from evaporating the nebulized droplets is thoroughly removed. In the case of DOP particles, particles are produced by nebulizing DOP liquid into droplets. These droplets are completely

FIG. 1

evaporated in a heating section and the DOP vapor is condensed in a condensing section in which moderately monodisperse droplets (geometric standard deviation = 1.4) are produced.

The aerosol sampling position can be changed along this tube. The distance between the aerosol inlet and the sampling port, together with the aerosol flow speed inside the tube, controls the acoustic agglomeration time. The aerosol size distribution function is measured by the combined use of an Electrical Aerosol Analyzer (Thermo-Systems Ind. Model 3030), an optical counter (Bausch and Lomb Model 40-1) and a Phoenix-Sinclair Aerosol Photometer (Model JM-4000). Table 1 summarizes all experiments (#1-10) with the EM high-power speaker as the sound source. Typical size distribution measurements are shown in Fig. 2.

Three experimental runs have been made with the aerodynamic siren operated at high frequencies. The siren in our laboratory is shown in Fig. 3. The siren has been tested in our environmental simulation chamber which consists of two mutually perpendicular cylinders as shown in Fig. 4. A 3-mil mylar sheet is used to separate the auxiliary chamber, Fig. 5, from the main one so that the compressed air used to drive the siren does not mix with the testing aerosol in the main chamber. The siren is driven by a 3 hp motor to spin the rotor. The compressed air required for the siren operation is supplied from a compressed air tank (6' dia. and 24' long) which can be pumped up to 170 psi.

The experimental results are plotted in Fig. 6 together with the theoretical predictions. The Brownian coagulation constant is also included in the figure for reference purposes. Three PSL particle radii are used: 0.085, 0.5 and 1.0 μm with geometric deviations ranging from

TABLE 1.

Exp. No.	No. of Runs	Aerosol			Acoustic Field			Acoustic K_a [cm ³ /sec]			Brownian K_a [cm ³ /sec]	(K_a) acoustic / (K_a) Brownian
		Type	Radius (μm)	Standard Deviation	Generator	Frequency	Intensity db	Mean	Data-spread	Uncertainty		
1	2	PSL	0.085	1.3	EM	1 KHz	145	7×10^{-9}	$\pm 3.0 \times 10^{-9}$	$\pm 1.5 \times 10^{-9}$	1.2×10^{-9}	5.8
2	3	PSL	0.085	1.3	EM	3 KHz	145	7×10^{-9}	$\pm 3.0 \times 10^{-9}$	$\pm 1.5 \times 10^{-9}$	1.2×10^{-9}	5.8
3	3	DOP	0.12	1.4	EM	1 KHz	145	6×10^{-9}	$\pm 2.5 \times 10^{-9}$	$\pm 1.5 \times 10^{-9}$	1.0×10^{-9}	6.0
4	3	DOP	0.12	1.4	EM	3 KHz	145	6×10^{-9}	$\pm 2.5 \times 10^{-9}$	$\pm 1.5 \times 10^{-9}$	1.0×10^{-9}	6.0
5	1	DOP	0.17	1.4	EM	1 KHz	145	5×10^{-9}	$\pm 1.5 \times 10^{-9}$	$\pm 1.5 \times 10^{-9}$	0.9×10^{-9}	5.5
6	2	DOP	0.17	1.4	EM	3 KHz	145	5×10^{-9}	$\pm 1.5 \times 10^{-9}$	$\pm 1.5 \times 10^{-9}$	0.9×10^{-9}	5.5
7	3	PSL	0.5	1.25	EM	1 KHz	145	8×10^{-7}	$\pm 2 \times 10^{-7}$	$\pm 2 \times 10^{-7}$	1.1×10^{-9}	720
8	3	PSL	0.5	1.25	EM	3 KHz	145	1.5×10^{-6}	$\pm 2 \times 10^{-7}$	$\pm 2 \times 10^{-7}$	1.1×10^{-9}	1360
9	4	PSL	1.0	1.25	EM	1 KHz	145	2×10^{-6}	$\pm 1 \times 10^{-6}$	$\pm 2 \times 10^{-6}$	2.3×10^{-9}	800
10	4	PSL	1.0	1.25	EM	3 KHz	145	1.2×10^{-5}	$\pm 1 \times 10^{-6}$	$\pm 2 \times 10^{-6}$	2.3×10^{-9}	4800
11	3	PSL	1.0	1.25	Siren	10 KHz	145	3.0×10^{-7}	$\pm 1.5 \times 10^{-7}$	$\pm 1.5 \times 10^{-7}$	2.5×10^{-9}	120

Figure 2

Measured Size Distribution functions (Experimental runs 7 & 8)

- Without acoustic treatment
- · - · - 3 KHz acoustic wave applied ($\Omega = 145$ db)
- - - 1 KHz acoustic wave applied ($\Omega = 145$ db)

Fig. 3 Photograph of the high-power siren mouth.

FIG. 4

Fig. 5. The Siren Chamber separated from the Aerosol Chamber
by a mylar sheet.

Fig. 6 Comparison of experimental measurements and theoretical predictions of the acoustic agglomeration constant. ----- theoretical prediction of the acoustic agglomeration constant, — coagulation constant without acoustic field, experimental data are taken at three frequencies (0-1 KHz, Δ -3 KHz, $\square$ - 10 KHz) detailed data tabulated in Table I.

1.25 to 1.30 as shown in Table I. Two DOP particle radii ($a = 0.12$ and $0.17 \mu\text{m}$) have a standard deviation of 1.4. To compare these data with theoretical calculation the spread in size distribution must be taken into consideration. Thus, the value of the theoretical acoustic agglomeration constant has to be expressed as an integrated average value given by

$$\bar{K}_a = \int_0^{\infty} K_a n(a) da / \int_0^{\infty} n(a) da \quad (1)$$

where $n(a)$ is assumed to be a lognormal distribution function.

To obtain the experimental data on acoustic agglomeration, the following equation is used.

$$-\frac{dn}{dt} = K_a n^2 + \lambda_s n \quad (2)$$

where λ_s is a rate constant for surface losses. Assuming that K_a and λ_s do not vary with time, integration yields.

$$\frac{1}{n} = \left(\frac{1}{n_0} + \frac{K_a}{\lambda_s} \right) \exp(\lambda_s t) - \frac{K_a}{\lambda_s} \quad (3)$$

The value of λ_s is experimentally determined in the present paper by placing microscopic grids on the testing chamber wall in several strategic positions and the rate of wall deposition is determined by direct counting under either an optical or an electron microscope. In all cases, the wall-loss term $\lambda_s n$ is less than 15% of the agglomeration term $K_a n^2$. Since λ_s

is known and n is measured as a function of time with an electrical aerosol analyzer, a plot of $1/n$ as function $\exp(\lambda_s t)$ yields a straight line whose slope is $\frac{1}{n_0} + \frac{K_a}{\lambda_s}$. When $t = 0$ we have $n = n_0$. Thus the straight line intercepts abscissa at $1/n_0$. Given the values of n_0 and λ_s , K_a is experimentally determined.

In Fig. 6 both theoretical and experimental curves show that the acoustic agglomeration is relatively independent of acoustic frequency for particles with radius longer than $1 \mu\text{m}$. This is because for larger particles, agglomeration is controlled by hydrodynamic interactions which are insensitive to frequency. This very important point will be discussed further in a later section.

The very large data spread for particles with radius equal to $1 \mu\text{m}$ is partially due to the difficulty in comparing data obtained from two different experimental systems. In general, the data obtained in the flowing system (Fig. 1) are considered to be more reliable. When it is compared to the siren system (Fig. 4), the acoustic field in the flowing system is found to be more uniform, the effect of low turbulence is smaller and the sonic field is more stable. Another reason for the higher uncertainty at larger particle size is the fact that the particle number concentration generally decreases as the particle size increases. When n decreases, the relative importance of the surface losses $\lambda_s n$ increases, giving an increasingly larger uncertainty as shown in Table. 1.

II.2 The Effects of Acoustic Intensity and Particle Size on Agglomeration Rate Experimental Testing

Based on the experimental results discussed in Sec. II.1, a theoretical model has been developed for the prediction of the acoustic agglomeration rate and the time-dependent size distribution function under an intense acoustic field. The model will be used later in the report for the determination of the optimum frequency and intensity for LMFBR applications. The details of this theoretical model can be found in reference [11].

Figures 7 and 8 show the comparison of the theoretical and experimental curves as plotted in normalized number concentrations. It is seen that the theoretical curves are invariably higher than the experimental data for particles with diameter less than $0.1 \mu\text{m}$. These discrepancies become larger as the acoustic intensity increases. This point can also be observed in Fig. 9 in which the total normalized number concentration is plotted against time for several acoustical intensities. While the agreement of the theory with the experiment is exceedingly good at 135 dB, the error amounts to about 10% when the intensity is 150 dB.

One of the reasons for the discrepancy may come from the underestimation of the wall plating. As the acoustic intensity increases, a convective flow pattern may be developed inside the aerosol chamber due to acoustic streaming. In our system, this means a general downward flow near the center of the chamber and a return upward flow along the chamber wall. Such convective flow would reduce the diffusional boundary layer thickness and thus increase the diffusional deposition of small particles. Although extensive theoretical analyses have been reported on acoustic streaming, mostly around spheres and cylinders, very little experimental

Fig. 7. Comparison of theoretical and experimental results on the decaying of the total normalized number concentration with respect to time at various values of acoustic intensity.
Acoustic frequency $f = 1.5\text{KHz}$, ——— experimental, - - - theoretical

Fig. 8 Comparison of theoretical and experimental normalized size distribution function at various times after the acoustic source is turned on. — experimental, - - - theoretical $N = 4.65 \times 10^4 \text{ cm}^{-3}$, acoustic intensity $\Omega = 135 \text{ db}$, and acoustic frequency $f = 1.5 \text{ KHz}$.

Fig. 9 Comparison of theoretical and experimental normalized size distribution function at various times after the acoustic source is turned on. $\overline{N} = 6.37 \times 10^4 \text{ cm}^{-3}$ experimental, - - - theoretical $N = 6.37 \times 10^4 \text{ cm}^{-3}$, acoustic intensity $\Omega = 145 \text{ db}$, and acoustic frequency $f = 1.5 \text{ KHz}$.

data is available on the acoustic streaming inside a cylinder under traveling-wave condition. Further improvement of the present theoretical model requires a better understanding of streaming velocity under various acoustic conditions.

II.3 The Prediction of the Effects of Acoustic Intensity, Particle Size and Mass-Loading on Agglomeration Rate

The good agreements between the theory and the experimental data as shown in Fig. 6 give us reasonable confidence on the validity of our theoretical model. In this section we will attempt to make some predictions concerning the agglomeration rate and the dwell time based on this model which has been verified experimentally with the acoustic intensity ranging from 135 to 150 dB.

We can define a half-life decay time $t_{1/2}$ as the time when the aerosol concentration reduces to half of its original value

$$\frac{N(a_2)}{N_o(a_2)} = \frac{1}{2} = \exp(-\tilde{K}_a t_{1/2}) \quad (4)$$

or

$$t_{1/2} = \ln 2 / \tilde{K}_a \quad (5)$$

where $N(a_2) = \int_0^{a_2} n(x) dx$ is the particle number concentration with radius less than a_2 , $N_o(a_2)$ is the value of $N(a_2)$ at $t = t_o$ and $\tilde{K}_a$ is the effective acoustic agglomeration constant for a log-normally distributed aerosol.

Based on the experimental values of K_a , the variation of the half-life decay time $t_{1/2}$ can be calculated once the particle mass loading ($\mu\text{g}/\text{cm}^3$), the log-normal standard deviation σ , the geometric mean radius a_g (μm), the acoustic intensity Ω (Watt/cm^2) and acoustic frequency f are given.

Figure 10 shows some theoretical predictions based on this model. Here, the fraction of the fine particles ($d < 2 \mu\text{m}$) remained airborne after the acoustic field ($f = 1 \text{ kHz}$) is turned on at $t = 0$. The geometric mean diameter of the particles, weighed with respect to number concentration, is fixed at $0.15 \mu\text{m}$. The size distribution function is assumed to be log-normal with the total number concentration N and the geometric standard deviation σ_g as variables. The initial values of N and σ_g are $1.2 \times 10^5 \text{ cm}^{-3}$ and 3.45 respectively (curves 1, 2 and 3). As shown, the $t_{1/2}$ for curve 1 ($\Omega = 0.01 \text{ W}/\text{cm}^2$), curve 2 ($\Omega = 0.1 \text{ W}/\text{cm}^2$) and curve 3 ($\Omega = 1 \text{ W}/\text{cm}^2$) are respectively 95, 30 and 9.5 sec. This is consistent with the theoretical model since U_g can be expressed as

$$U_g = (2\Omega/c \rho_g)^{1/2} \quad (6)$$

one can state that

$$t_{1/2} \sim \Omega^{-1/2} \quad (7)$$

The effect of reducing the standard deviation value σ_g is demonstrated by curve 4. Here σ_g is reduced from 3.45 (for curves 1, 2 and 3) to 1.6. Such a relatively monodisperse aerosol has a very small value of acoustic agglomeration constant $\tilde{K}_a$ and thus a very large value of $t_{1/2}$.

FIG. 10 - Fraction of fine particles (radius $a \leq 1.0 \mu\text{m}$) remaining airborne as a function of time.

Curve 1	$N = 1.2 \times 10^5 \text{ cm}^{-3}$, $d_g = 0.15 \mu\text{m}$, $\sigma_g = 3.45$, $f = 1 \text{ KHz}$,	$\Omega = 0.01 \text{ W/cm}^2$ $\Omega = 0.1 \text{ W/cm}^2$ $\Omega = 1.0 \text{ W/cm}^2$
Curve 2		
Curve 3		
Curve 4	"	"
Curve 5	$N = 6 \times 10^5 \text{ cm}^{-3}$	"
Curve 6	"	"

The agglomeration becomes ineffective because the number of large particles (which act as the collecting centers of small particles) are drastically reduced when σ_g is decreased. In this case, very little can be observed as indicated on curve 4.

The fact that the number of available large particles has a dominant effect on agglomeration rate suggests that for practical applications it may be advantageous to introduce some sort of spray system in the agglomeration chamber. The mean size of the spray droplets should be large in comparison with the fine-particle size to be controlled, but should be kept small enough so that the droplets remain airborne for a relatively long period of time. Also the spray liquid must be chemically inert so that no additional air-pollution problem is introduced.

The effect of increasing particle concentration is demonstrated by curves 5 and 6 of Fig. 10. For example, for the case when the initial conditions are those shown in the figure, and $\Omega = 0.1W/cm^2$, an increase of concentration by a factor of 5 reduces the value of $t_{1/2}$ from 30 sec (curve 2) to 6 sec (curve 6). In fact, it can be shown that

$$\tilde{K}_a \sim \frac{1}{N_1} \quad (8)$$

where N_1 is the total number of large-collecting particles.

III. STANDING-WAVE OPERATION

Acoustic agglomeration under the standing-wave condition has been investigated by many authors in the past. St. Clair[11] investigated the problem theoretically by considering the rate at which particles are pushed toward the vibrational loop by the radiation drift. However, this result is contrary to experimental observations which show that particles are drifted toward the node points. Dukhin[12] showed that such a drift is caused by asymmetric vibration of particles in a standing-wave. Several early papers have reported on the streaming flow pattern and radiation pressure in an acoustic "Kundt: tube. However, few of these provide any data on acoustic agglomeration of particles in the tube.

Dianov, Merkulov and Nikitenko [13] investigated the acoustic agglomeration in a standing-wave chamber. Based on the light transmission measurement, the settling time, defined as the time for the light transmission to be nearly 100%, was found to vary inversely with the sound intensity Ω in the high intensity region ($\Omega > 160$ db at the

node point). Furthermore, it was shown that the aerosol settling rate is relatively independent of the acoustic frequency. This is shown in Fig.11 in the variation of the nephelometer current per unit time plotted against the frequency for a given mass loading and acoustic intensity. As the acoustic frequency varies from 0.2 kHz to 20 kHz, the value of dK/dt (which is proportional to the acoustic agglomeration constant) reduces less than 20%.

III.1 The Stationary System

Figure 11 shows schematically the testing chamber for standing-wave acoustic agglomeration experiments. The aerosol concentration is monitored by light transmission method. If I_0 is light transmission (100%) after the tube is filled up by the aerosol the sound is switched on, the transmission of light through an aerosol is given by

$$\ln \frac{I}{I_0} = -(b_{abs} + b_{scat}) L \quad (9)$$

where I_0 and I are the beam intensity of the incident light and the beam intensity after traversing a distance L , respectively. b_{abs} and b_{scat} are the particle absorption and scattering coefficient, respectively. For most aerosol work, $b_{abs} \ll b_{scat}$ [14]. The scattering coefficient b_{scat} has been shown to correlate directly to the aerosol mass [15]. Thus, for two aerosol mass concentrations m_1 and m_2 we can write

$$\frac{m_1}{m_2} = \frac{\ln(I_0/I_1)}{\ln(I_0/I_2)} \quad (10)$$

Fig. 11

The variation in nephelometer current per unit dI/dt versus frequency taken from ref. 64. Acoustic pressure = 3500 bar, aerosol concentration = 5 g/m^3 .

where m_1 and m_2 are the mass concentrations corresponding to the light intensities I_1 and I_2 , respectively.

The half-life decay time can be determined from Eq.10 with subscript 2 referring to the aerosol before the sound is applied and 1 after the sound is applied. If m_1 is one-half of m_2 , the time it takes for I_1 to satisfy

$$\frac{1}{2} = \frac{\ln(I_0/I_1)}{\ln(I_0/I_2)}$$

is the half-life decay time. This simplifies to

$$I_1 = \sqrt{I_0 I_2} \quad (11)$$

This determination is shown on a typical test run in Fig.12.

The attenuation and the dispersion of the sound field by aerosol particles are tested for two frequencies, 1070 Hz and 640 Hz. The standing wave is first established in the test chamber without aerosol particles and the resonant frequency and intensity at the nodal point are recorded as shown in Table 1. When aerosol is introduced, the resonant test chamber is detuned and adjustments must be made on the acoustic frequency to reestablish the standing wave condition. The changes in the resonant frequency and the nodal point intensity are shown in Table 1.

TABLE 1

Tube condition	Resonant frequency (Hz)	Intensity (nodal) (db)
I. Empty	1070	155.5
Aerosol	1060	149.5
II. Empty	640	154.0
Aerosol	635	152.0

Fig. 12 Variations of the light transmission signal as a function of time in standing-wave acoustic agglomeration.

It is seen that the resonant frequency reduction due to the introduction of aerosol is 10 Hz at 1070 Hz and 5 Hz at 640 Hz; while the nodal point intensity reduces by 6 db and 2 db at 1070 Hz and 640 Hz, respectively. Thus, the higher the frequency, the higher the attenuation and the dispersion.

The half-life decay time, as determined by Eq.(11) has been measured for acoustic frequencies ranging from 640 Hz to 1.7 KHz under both the stationary condition and the flow-through condition. For the stationary system, Fig. 13 shows that the lower the resonant frequency, the shorter the half-life decay time $t_{1/2}$. The reduction in the nodal point intensity shown in Table 1 is responsible for about half of this reduction. In other words, the value of $t_{1/2}$ is higher at high frequencies because of the higher sound attenuation at such frequencies. But the attenuation alone cannot explain the observed effect of acoustic frequency on $t_{1/2}$. This slight improvement in effectiveness by switching to lower frequencies is not very well understood at the present time.

III.2 The Flow-Through System

For the flow-through system, the aerosol residence time in the test chamber can be controlled by varying the aerosol flow rate. Figure 14 shows the fraction of particle mass remaining at the exit as functions of the residence time for both the traveling-wave and standing-wave conditions.

It is interesting to compare the effectiveness of acoustic agglomeration of a flow-through system to that of a stationary system. To do this, however, a new parameter must be defined since the use of half-life decay time as shown in Fig.13 for the stationary system is not applicable for the flow-through system.

Fig.13 Half-life decay time vs. intensity for three frequencies in vertical position with drive at the bottom.

□ 640 Hz, △ 850 Hz, ○ 1070 Hz

Fig. 14 Comparison of the fraction of particle mass remaining at the exit in a flowing system with traveling-wave and standing-wave for different residence times in min. at intensity of 110 dB(open air) and frequency of 1070 Hz. Vertical position with the sonic source at the bottom.

The reduction in mass by acoustic precipitation as a function of time can be expressed as

$$m(t) = m_0 \exp (- k_m t) \quad (12)$$

where m_0 is the value of m at $t = 0$. k_m is an effective particle removal rate constant and can be related to $t_{1/2}$ by considering the case when $m(t) = m_0/2$. From Eq. (12) we have

$$k_m = \ln 2 / t_{1/2} \quad (13)$$

Thus the value of k_m can be determined for the flow-through system based on Eq. 12 by measuring the time decay of the light extinction signal. m_0 is determined by the Sinclair-Phoenix photometer. By plotting $m(t)/m_0$ vs. time on a semi-log paper, the slope gives k_m . These values of k_m are then compared with those determined from Eq. 13 based on half-life decay time measurement. Figure 15 shows that in the low intensity region (nodal point intensity < 160 db) a small flow velocity greatly reduces the values of the mass removal rate constant k_m . The magnitude of this reduction decreases rapidly as the intensity is increased. When the nodal point intensity is 170 db there is no observed difference between the values of k_m obtained from the flow-through system and the stationary system. The reduction of k_m in the flow-through system is attributed to the reduction in acoustic turbulence. Based on these results it seems that in the standing-wave acoustic agglomerator turbulence plays an important role in particle agglomeration. No theory exists at the present time on the mechanisms responsible for particle agglomeration in a standing acoustic wave.

Fig. 15 Removal rate constant K vs. intensity for different frequencies for stationary system and for flowing system.

Stationary System

- 640 Hz
- ⊕ 850 Hz
- 1070 Hz
- ▲ 1700 Hz

Flowing System 1070 Hz

- △ 2 lit/min
- ⊙ 1 lit/min
- ⊗ 3 lit/min

IV. HIGH-INTENSITY ACOUSTIC GENERATORS

The Laboratory for Power and Environmental Studies of SUNY/Buffalo is probably the best equipped laboratory in the world for experimental investigation of acoustic agglomeration of aerosol particles with diameter ranging from 0.2 to 20 μ . Several high-intensity sound sources are available in the laboratory. These include a dynamic siren, several Hartmann shock-jet whistles, high-power electromagnetic speakers and a mechanical-piston Braxton system. Their performance characteristics will be discussed in this section.

All high-intensity acoustic generators have been tested in one of our two acoustically insulated enclosures: one is a horizontal rectangle (6' x 6' x 20') with a cupola at one end. The other has dimensions approximately 20' x 20' x 18'. Both enclosures are capable acoustic sources with intensity up to 170 dB. Two high power speakers (University Sound Model B-12P) are available in our laboratory, both are capable of generating an acoustic field between the range of 500 to 3,000 kc/sec with a total sonic power output of about 25 W under traveling-wave conditions in a tube 4" in diameter. The speaker consists of 12 small speakers (University Sound LD-60 or ID-75) connected together to provide high output power. The operation of the speaker is very stable and the output frequency is continuously tunable.

IV.1 The Dynamic Sirens

Siren-type pneumatic sonic generators have been employed on an industrial scale for acoustic agglomeration. They produce uniform coverage of

a wide frequency range, and have a very high efficiency in converting the energy of compressed air into sound waves with practically no limit on acoustic power output. In the late 1950's several companies were in the business of building large industrial dynamic sirens. To the best of our knowledge, all of them went out of business about ten years ago. The siren in our laboratory is an axial type (Fig. 3) which consists of three essential components: a source of compressed air, a rotor which interrupts the air flow, and a stator with ports through which the air escapes. The fundamental frequency f of a siren is given by

$$f = Zn/60 \quad (14)$$

where Z is the number of ports on the stator, and n is the rpm of the rotor. For our siren $Z = 100$ and the motor speed varies from 7,200 rpm to 13,800 rpm, which gives an output frequency in the range of 12 kc/s to 23 kc/s.

The output acoustic power P_o generated by the siren can be determined by

$$P_o = k QW_{ad}\eta(\text{kW}) \quad (15)$$

where k is a constant with a value between 7×10^{-3} to 1.4×10^{-2} , Q is the compressed air flow through the ports, W_{ad} is the adiabatic work done in the expansion of LM^3 of the compressed air, and η is the acoustic efficiency of the siren. The siren yields an output power in the range of 100 to 200 W depending on the output frequency. The siren has been tested and run in our environmental simulation chamber which consists of two mutually perpendicular cylinders as shown in Fig. 4. The main chamber is 5 ft in diameter and 7 ft in length with two optical windows at both ends for continuous

optical monitoring of mass concentration variations. An auxiliary cylinder (3.5' diameter x 4' long) will be used to house the main chamber. The siren is mounted on its test stand by means of six bolts through the flange holes and aligned, in a manner that the rotor shaft is parallel to the floor of the sound chamber. The siren is driven with a 3 hp motor.

The compressed air required for the siren operation is supplied from a compressed air tank (6' diameter and 24' long) which can be pumped up to 170 psi. The tank was originally designed for a large (test section cross-section $\sim 1 \text{ ft}^2$) supersonic wind tunnel and located right next to the SUNY/Buffalo Environmental Simulation Chamber shown in Fig. 4.

IV.2 The Mechanically Vibrating Piston Generator

For industrial applications utilizing sinusoidal waves two additional types of sound generators are available: the vibrating piston and the whistle. The vibrating piston in our laboratory is a Braxton system which can generate an acoustic field at 440 Hz and 170 dB (Figs. 16 and 17). The aerosol testing chamber is about 2' diameter x 10' long. The system is probably the largest mechanically vibrating sonic generator in the world. The system is designed to allow the admission of steam and/or water droplets. The steam can be added in the acoustical chamber or added in the inlet duct about 50 feet before the chamber. This allows the dust and steam to interact, thus increasing the cohesive properties of the dust.

Braxton 250 cfm Sonic Agglomeration System

Fig. 17

IV.3 The Whistles

The whistle shown in Fig. 18 is a slightly modified version of the traditional one in that small holes are drilled on the bottom of the piston cup for cooling purposes. Figure 19 shows a schematic diagram of the device and the pressure distribution along the stream. At position X_1 , the pressure drops to a minimum which causes a pile-up of the gas beyond this point and a shock wave is formed. This effect repeats itself in regular intervals X_1, X_2 , etc. The pile-up regions are unstable and can be used to maintain oscillations of a resonant cavity. This pulsating frequency may be estimated by

$$f = c/4(L + 0.3 d) \quad (16)$$

where c is the speed of sound in gas, d and L are indicated in Fig. 19.

The acoustic power output P_o

$$P_o = 3d^2(P_e - 0.9)^{1/2} \text{ Watts} \quad (17)$$

where P_e is the upstream pressure. Generally speaking, the acoustic conversion efficiency of Hartmann whistles is much worse than that of static sirens (4 to 6% for whistles as compared to 10 to 20% for sirens). The major advantage of whistles is the isotropic nature of the acoustic output. In other words, the acoustic intensity is almost uniform in value for all θ values.

Side view, nozzle and resonator

End view, nozzle and resonator

Fig.18 a and b. Hartmann whistle.

Fig. 19

Schematic Diagram showing the Production of Acoustic Waves with a
Hartmann Whistle

Several power whistles are in operation in the laboratory with frequencies ranging from 0.7 to 15 kHz.

IV.4 Expulsion of Spent Air

The sirens (Fig. 3) and whistles (Figs. 18 and 19.) have one shortcoming in common, namely, the difficulty to expel the spent air. The problem is even more serious for the whistles because of the large volume of compressed air required to drive the device. There are three possible methods to resolve this problem: (1) to use the dirty air to drive the generator directly, (2) to separate the acoustic generator from the aerosol chamber by a thin membrane, and (3) to expel the spent air properly without a membrane.

The feasibility of method (1) for whistles has been demonstrated for laboratory-type experiment. The problem comes primarily from the deposition of dust particles on the whistle itself. Typically, the efficiency of the whistle deteriorates and sometimes clogging develops in the nozzle, especially for high-frequency operation. Sirens require clean air, thus cannot be run by dusty air which will clog up all stationary ports in the stator.

For laboratory scale testing the simplest method for expulsion of spent air is the use of a divider membrane. Figure 5 shows the picture of the siren chamber (the smallest cylinder in Fig. 4) separated from the aerosol chamber (the larger cylinder in Fig. 4). Under laboratory conditions the membrane method has indisputable advantage over other methods insofar as it provides an easy way to get rid of the spent air. However, for large-scale testings, the method has serious drawbacks. The principle ones are, (1) the loss of sonic intensity across the membrane, and (2) the structural endurance. These are caused by the conflicting requirements.

that the membrane must be strong enough to withstand the compressed air pressure and at the same time must transmit sound with minimum attenuation of sonic intensity. To achieve the latter, the mass reactance of the membrane must be smaller than the acoustic impedance, i.e., $\rho_g c \gg 2\pi\rho_m Lf$ (ρ_g and ρ_m are respectively the density of gas and membrane material, c is the sound velocity, f the frequency, and L the thickness of the membrane). Thus thin sheets of material with low density are used. Such membranes typically are made of plastic material (for example, mylar) with a thickness of 10 μm and higher.*

Attempts have been made to build a resonance membrane which has a natural oscillation frequency the same as the acoustic frequency. This can be achieved by properly choosing the membrane thickness and weight and by adjusting the membrane tension. Unfortunately, such resonant membranes are difficult to build for large-scale systems. First of all, large membranes made of plastic material such as mylar sheets are difficult to stretch out evenly to generate the desired single-frequency oscillation. Even if a single frequency is produced after careful adjustment of tension on the membrane, it is virtually impossible to maintain the tension uniformity because the plastic material ages quickly with time. Metal foils have been tried but they break after one or two hours caused by fatigue from repetitive oscillation. Thus, although membranes are frequently used in laboratory operation in which their short lifetimes are not considered to be critical, their application to large-scale testing is more difficult.

* For air under normal conditions, let $\rho_m = 1 \text{ g/cm}^3$, $fL = 7(\text{kHz})(\mu_m)$. If $f = 1 \text{ kHz}$, $L = 7 \mu\text{m}$ for a nearly 100% transmission.

Further on large-scale devices are necessary before full assessment of this technique is possible.

Finally, the technique based on the principle of expulsion of spent air has been tested as shown in Fig. 20. The spent air is transported along the collecting tube (1) mounted coaxially with the resonator (2). It is found that the effectiveness of the collecting tube depends strongly on the design of the resonator, for example, the sharpness of the inlet angle (4), the distance between the resonator and the nozzle (5), etc. For our present system, about 70% of the air is estimated to have been collected. To prevent the rest of the spent air from going into the aerosol chamber a thin membrane is used (3). Since only a small fraction of the compressed-air pressure is applied to the membrane, the structural requirement is less stringent. This means a thinner membrane and less acoustic attenuation.

Fig. 20 Hartmann whistle with a collecting tube for the expulsion of spent air.

1) Collecting tube	3) Divider membrane
2) Resonator cavity	4) Inlet wedge
5) Nozzle	

V. CONCLUSIONS AND RECOMMENDATIONS

In view of the results discussed in this report, the following conclusions may be made on the use of acoustic agglomerators as an emergency system for the suppression of Na-fire aerosol in LMFBR plants:

1. Comparing the traveling-wave and standing-wave agglomerators, the agglomeration is much more effective for the standing-wave operation. The agglomeration rate is inversely proportional to the mass concentration, thus making it attractive to use the agglomerator as a preconditioner. More conventional devices, such as cyclones and HEPA filters should be used after the acoustic agglomerator to remove the agglomerated large particles.
2. Based on our preliminary data the acoustic agglomeration is relatively independent of frequency. This makes it possible to operate at a frequency of a few hundred Hz. It should be recalled that three agglomeration mechanisms have been considered in the present study - orthokinetic, hydrodynamic, and turbulent interactions. Old theory has identified the orthokinetic interaction, which is frequency dependent, as the predominant mechanism of acoustic agglomeration. New results show that the orthokinetic interaction is important only for very small particles (diameter $< 1 \mu\text{m}$). For larger particles, hydrodynamic and turbulent interactions become important and they are relatively frequency independent.
3. Low-frequency agglomerators are more desirable than the high-frequency ones ($f > 1 \text{ kHz}$) because of the lower acoustic attenuation in the aerosol and across the divider membranes. Acoustic agglomerators also have a higher sound generation efficiency at low frequencies. Furthermore, lower

frequencies allows the use of larger diameters for the agglomerator. For standing-wave operation it is desirable to have the tube-diameter less than one-half of the acoustic wavelength. For example, for frequencies ranging from 300 to 800 Hz, the tube diameter ranges from 50 to 20 cm, respectively. If the diameter is larger than one-half the wavelength, standing-wave in the radial direction will greatly reduce the effectiveness of agglomeration, thus should be avoided.

4. Among the three high-power generators - the mechanically vibrating piston, the dynamic siren, and the static whistle - the vibrating piston seems to be the best in terms of experimental setup, but it is also the most expensive one. The whistles are the cheapest, but the difficulty of expulsion of the spent air is a complex and unresolved problem.

5. The power requirement for an agglomerator operated at 160 dB ($\approx 1 \text{ watt/cm}^2$) is estimated to be 0.2 to 1 kWhr/1000 m³, which is an order of magnitude the same as the venturi scrubbers, but higher than those for the cyclones, fabric filters and electrostatic precipitators.

1. R.K. Hilliard, J.D. McCormack, A.K. Postma, "Evaluation of Air Cleaning System Concepts for Emergency Use in LMFBR Plants" Report No. HEDL-TME-76-41, December 1976.
2. R.E. MacPherson, "Sodium Burning and Aerosol Release - an Evaluation of State-Of-Art," ORNL-TM, 1937, 1968.
3. H.S. Patterson and W. Cawood, Nature, 127, 667 (1931).
4. O. Brandt and E. Hidemann, Trans. Faraday Soc., 32 (184):1101 (1936).
5. E.N. da C. Andrade, Trans. Faraday Soc., 32 (184)30 (1936).
6. H.W. St. Clair, Ind. & Eng. Chem., 41, 2434 (1949).
7. D. Sinclair, "Coagulation by Sonic and Supersonic Vibrations," Handbook of Aerosols, Washington, D.C. 1950.
8. E.P. Mednikov, Acoustic Coagulation and Precipitation of Aerosols, translated from Russian by C.V. Larrick, Consultant Bureau, New York 1965.
9. N.L. Shirokova, Physical Principles of Ultrasonic Technology Vol 2. ed. L.D. Rozenberg, translated from Russian by J.S. Wood, Plenum Press, New York, p. 477, 1970.
10. D.T. Shaw, Recent Developments in Aerosol Science, ed. D.T. Shaw, Wiley Interscience, New York, 1978. Chap. 13.
11. H.W. St. Clair, Ind. & Eng. Chem., 41, 2434 (1949).
12. S.S. Dukhin, Koll Zh., 22 (English translation) 128 (1960).
13. D.B. Dianov, L.G. Merkulov, and V.I.N. Kitenko, Soviet Phys. Acoustics, 8, 44 (1962).
14. W.E. Middleton, Vision Through the Atmosphere, University of Toronto Press, Toronto, Canada, 1968.
15. A.P. Waggoner and R.J. Charlson, "Measurements of Aerosol Optical Parameters," from Measurements of Fine Particles, Ed., B. Liu, Academic Press, 1976.