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AIR FLOW AND HEAT REMOVAL IN THE CLINTON PILE

M. C. Leverett

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AIR FLOW AND HEAT REMOVAL IN THE CLINTON PILE

This report is a pre-printing of Chapter XI, Volume V, "Graphite Production Piles" of the Plutonium Project Record.

[redacted]

Chapter XI - Air Flow and Heat Removal

In this chapter we shall give in some detail the data on air flow in the Clinton Pile cooling system, and heat and air flow in the pile itself.

1. The Cooling System

The Clinton pile is cooled by air which is sucked by fans through a system of duct work and the pile. The air is sucked in from out-of-doors first through a filter. The air then passes downward, is divided into two streams and then united again after a series of baffles and turns, the object of which is to effect the deposition of any liquid water which may have been formed by the cooling which results when the air passes through the filter. This cooling is due to the expansion of the air in going from atmospheric pressure at point of intake to sub-atmospheric downstream from the filter. The air then passes through a duct which is below floor level in the pile building. This duct emerges at a lower corner of the inlet plenum chamber, which is approximately 3' wide 24' long and 37' high. The plenum chamber is bridged, in line with each graphite channel, by a steel pipe or bridge tube. This tube is 1¼" IPS galvanized steel pipe, and it is slotted with a slot ½" wide and 4" long. The tube projects slightly (about 2") into the 1 3/4" x 1 3/4" square channel in the graphite, leaving a space between the outside of the tube and the graphite. Air enters the channel through both this opening and the slot in the bridge tube. The slugs, having a total OD of about 1.17" lie in the 1.75" square channel; since the axes of the square are vertical the channel appears diamond shaped, with the slugs lying in corner of the diamond. The air flows through the remaining space. The channel is 24' long, but normally is not completely full of slugs; the number of slugs in any one channel is usually 30 to 68. The slugs have an overall length, including the jacket, of about 4.1" each.

The air emerges from the pile channels into the discharge plenum chamber which had a width of approximately 5½' and is 30' wide and roughly 24' high. This chamber is unobstructed by any bridge tubes or other objects. The air leaves it to enter another sub-floor duct 4' x 5' in cross section. This duct leads to the fan building, and in the straight portion of the duct a Venturi meter is located. This is simply a constricted throat, 2-3/4' in diameter, which is tapered in the standard way to connect with the larger rectangular portions of the duct. The air then passes into a large manifold (fan inlet manifold), off of which lead 3 smaller ducts; each of these leads the air through a pit and under a hanging wall in the pit; these pits may be filled with water in order to seal off any fan from the inlet manifold. The air next passes through the fans of which one is a small steam engine driven fan with a capacity of about 5000 cfm. The other two fans are Buffalo Forge Co. double suction, 3500 rpm centrifugals with a rated capacity of 60,000 cfm of air at 55" water suction and 150°F (air density 0.0555 lbs./ft^3). The small fan is for emergency use only and usually is not running. From the fans the air passes through another set of seal pits into the fan discharge manifold and thence to the discharge stack, which is 200' high and 5' in diameter inside. The two large fans usually are operated together, exceptions occurring only when one of them is shut down for repairs. They are driven by electric motors capable of developing 700 horsepower each. Each fan is equipped with a shutter-type damper which makes it possible to regulate the air flow when necessary. This damper also prevents sucking back of air through the small fan when the large ones are running.

2. Pressure Drops in Flow System

The rather complicated geometry of the duct work and flow system generally makes it difficult to calculate its resistance to air flow. One would suppose, therefore, that the most reliable estimates could be got from experimental measurements of the pressures throughout the system during actual operation. Actually this is probably true, but the experimental measurements themselves are somewhat subject to question since it is sometimes hard to separate the "static head" at a point from the "velocity head" because of the poor locations of existing holes for pressure measurement. New holes cannot easily be made since the duct walls are concrete, 1 or 2 feet thick. In spite of the uncertainties mentioned the experimental values are considered better than the calculated ones. The following tabulation is taken from a smoothing of experimental data. The figures apply for an inlet air density of 0.070 lbs/ft^3, which is approximately the average air density in the system preceding the pile with one fan running, pile power 2000 kw and outdoor air at 30° C, and outlet air density of 0.058 lbs/ft^3 which represents one fan running, power 2000 kw, and exit air temperature near 85° C. Other data, typical of more recent practice, appear in Section 15 of this chapter.

TABLE 1

Pressure Drops in Pile Cooling System
(Values Smoothed from Experimental Data)

Exit Air Flow lbs/sec	ΔP_1	ΔP_2	ΔP_3	ΔP_4	ΔP_5	ΔP_6
	Inches of Water					
40	1.13	0.54	2.25	1.60	4.8	9.4
80	4.0	1.90	8.0	5.6	16.5	31
120	8.2	4.0	16.5	12.0	35.0	60

ΔP_1 - Atmosphere to Pitot tube near end of inlet duct
ΔP_2 - Pitot to pile inlet plenum chamber
ΔP_3 - Pile inlet plenum chamber to pile outlet plenum chamber
ΔP_4 - Pile outlet plenum chamber to Venturi
ΔP_5 - Venturi to stack discharge
ΔP_6 - Total pressure rise across fan.

P_3 evidently depends markedly on the loading of the pile and is included in the above table for illustrative purposes only. The values given refer to a loading of 44 slugs per channel in 713 channels, 75 channels open and empty and the remainder of the 1252 channels plugged up. The above table also assumes air in-leakage of 15%. The available data can be expressed in slightly more general form as a series of the form

$$\Delta P = \alpha W^{1.8}$$

where ΔP is the pressure drop for the section under consideration, in inches of water, W is the air flow rate in the section expressed in pounds per second, ρ is the average air density in the section in pounds per cubic foot, and α is a proportionality factor which also contains the necessary numerical conversion factors. Values of the factor α are tabulated below:

Table 2

Air Resistance Factors in Pile Cooling System

	ΔP_1	ΔP_2	ΔP_3*	ΔP_4	ΔP_5
α	1.41×10^{-4}	$.69 \times 10^{-4}$	1.91×10^{-4}	1.22×10^{-4}	3.57×10^{-4}

*For conditions of Table 1.

3. Pressure Drop through Pile Proper

The pressure drop through the pile is made up of

1. Entrance losses where air enters bridge tube and graphite.
2. Contraction loss where air enters metal-filled section of channel.
3. Expansion loss where air leaves metal-filled section.
4. Expansion loss where air leaves graphite.
5. Friction losses in empty sections of graphite channel.
6. Friction loss in metal-filled section of channel.

The expansion and contraction losses may be calculated approximately using standard formulas (See Perry, Chemical Engineer's Handbook, 2nd edition, p. 821) and sum up to

$$\Delta P = \frac{23.3\, w^2}{\rho}$$

where

P = pressure losses due to expansion and contraction, in. of water
w = lbs. air/sec flowing through channel
ρ = air density (average), lbs/ft^3

The frictional losses in the empty sections of the channels plus that in the metal-filled portions may be computed from the Fanning equation, using the concept of equivalent hydraulic radius. This being done these losses are found to be calculable by the equation

$$\Delta P = \frac{w^{1.8}}{\rho}\,(2.9L + 16.2)$$

where L is the length of the metal filling in feet and the other symbols are as before. In each case the constant contains the properties of air, the channel and/or slug dimensions and other numerical factors peculiar to this system only. The validity of the above equations was tested experimentally by Rupp and Bornwasser who produced the following comparison:

Table 3

	Pressure Drop	
	Measured	Calculated
Open Graphite tube, inches water per ft.	0.12	0.15
Metal filled section, inches water per ft.	0.64	0.71
Expansion and contraction, inches water	0.37	0.38

The formulas given above were modified to give results concordant with the data of Bornwasser and Rupp; the formulas were then combined to give

$$P = \frac{w^{1.8}}{\rho} (2.67L + 13.0 + 23.3\, w^{0.2})$$

Actual measurements of the pressure drop across the pile fail to check this result; for example, the actual value was found to be 7.0" water when it was calculated to be 9.8" water. Several possible explanations for this discrepancy have been advanced:

a) The graphite channels may not be exactly 1.75" square, but may be larger. They would have to be about 1.83" square to explain the results, however, and it does not seem likely that such a large deviation from construction specifications occurred.

b) Some air may leak around the metal-filled channels either across the top of the pile or through channels in the graphite presumed to be plugged up. Leakage across the top of the pile seems unlikely in view of the fact that the sealing-baffle originally installed has been replaced with one of better construction. The plugs used may, it is true, be allowing some air leakage but extensive laboratory testing of the plugs before insertion does not support this view.

c) Possibly the most likely explanation is that there is a little leakage past the plugs, but that the major portion of the effect is caused by transfer of air, through cracks between graphite stringers, from unplugged channels to plugged ones back of the plugs. This hypothesis is strengthened by the fact that there are many transverse stringers and openings in the pile which would facilitate cross flow of the air.

4. Stack Draft

Continued operation of the pile causes heating of the ducts and the walls of the concrete stack. When the pile is shut down, cold air passing through the graphite, ducts, fans, and stack becomes heated, and a thermal syphon effect ensues because of the decrease in density. A single observation on this effect showed it initially to be capable of drawing roughly 5,000 cfm of air through the system with no power supplied to the fans. After several hours the magnitude of the effect would be much decreased.

5. Fan Characteristics

The fans originally installed may be briefly described as follows:

#1 5,000 cfm capacity
#2 30,000 cfm capacity, max. pressure rise 15" water @ 15° C.
#3 50,000 cfm capacity, max. pressure rise 27" water @ 15° C.

Late in 1943 it became desirable to increase the power of the pile; in part this was done by substituting new fans in the place of the old #2 and #3 fans, above. These new fans are identical; they were rated by the manufacturer (Buffalo Forge Co.) to deliver 60,000 cfm each at 150° F (air density - 0.0555 lbs/ft^3) with a pressure rise of 55" water. All calculations of performance to be expected (see, for example, report CS-1407) were based on the manufacturer's curves. Once the fans were installed, however, direct determination of their characteristics became possible and a discrepancy between prediction and actual characteristics appeared. The conditions for these curves are inlet air density 0.0555 lbs/ft^3, and no throttling.

The two curves are reproduced in Fig. 1. As it turned out, the operating region is not far from the point at which the predicted and actual curves cross, so that the discrepancy between them has had no practical consequences.

A more useful plot of the fan characteristics is shown in Fig. 2. This set of curves shows the pressure rise across the fan as a function of the mass rate of air it is permitted to handle; as a parameter the power of the pile is shown. Also shown on this plot are the "system characteristics", for a particular pile loading, i.e., the pressure drop which occurs when air is passed through the cooling system at a given mass rate; here again the parameter is pile power. It is evident that the intersections of the fan characteristics with the system characteristics are operating points. Given a fixed inlet air temperature, each such operating point is characterized by a definite air temperature, slug temperature, and fan horsepower requirement. Fig. 3 shows a fan horsepower curve. Air and slug temperatures are considered in a later section.

6. Measurement of Air Flow

The cooling system is provided with two devices for measuring air flow; one, located in the inlet duct, is a type of Pitot tube; the other, located between the pile and the fan intake manifold, is a Venturimeter. The Pitot tube was calibrated by the manufacturer before installation. The Venturimeter was cast of concrete to detailed and exacting dimensional specifications. In the course of an experiment to determine the heat per fission and the product/power ratio it was necessary to re-calibrate both the Pitot and the Venturi. This was done by introducing NH_3 of known purity into the ingoing airstream, and determining its concentration in the stream at various points in the cooling system. The NH_3 was introduced from ordinary steel cylinders, and the amount introduced was determined by weighing the cylinders before and after the experiment. By this means it was established that the flow rates are given by

$$Q = 7100 \sqrt{\Delta P/\rho} \quad \text{(Venturi)}$$

$$Q = 19{,}500 \sqrt{\Delta P/\rho} \quad \text{(Pitot)}$$

MonN-157
Drawing No. 2046

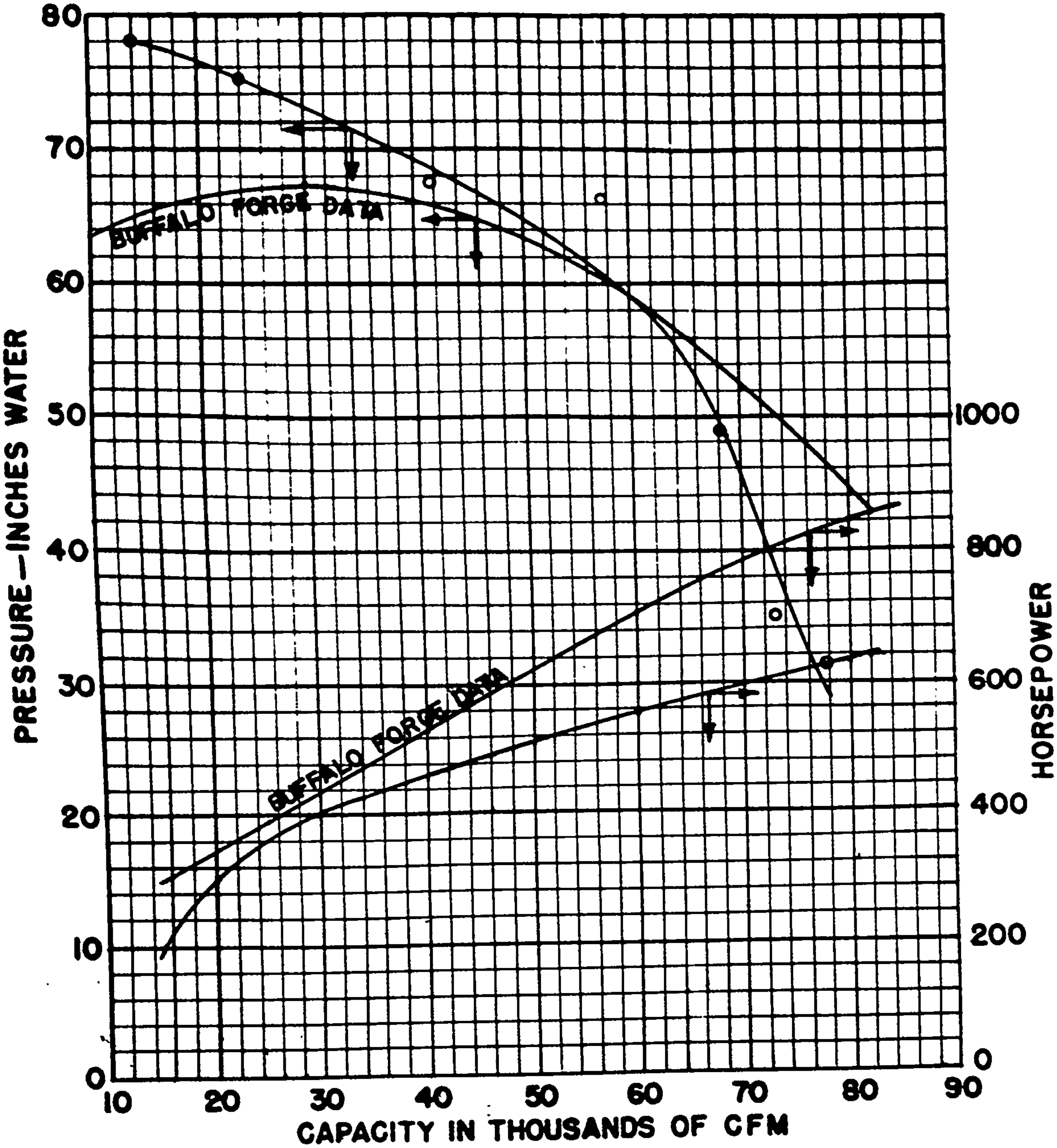

FIG. I. PERFORMANCE CURVES OF 70" S.S.P. FAN COMPARISON OF BUFFALO FORGE DATA WITH MEASURED DATA

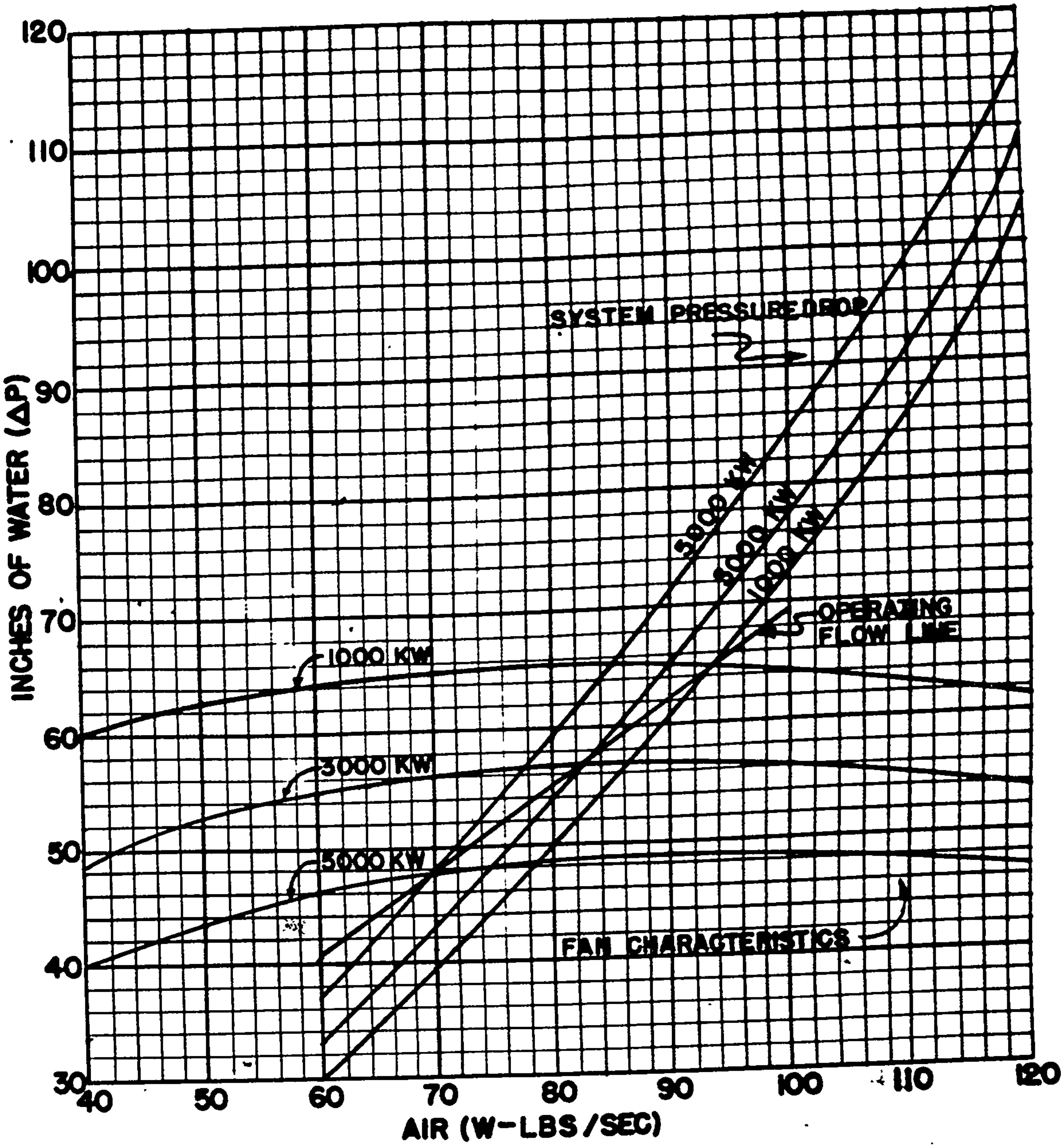

FIG. 2. OPERATING FLOWS FOR 9.0' X 22.3' PILE, 2 FANS OPERATING

MonN-157
Drawing No. 2048

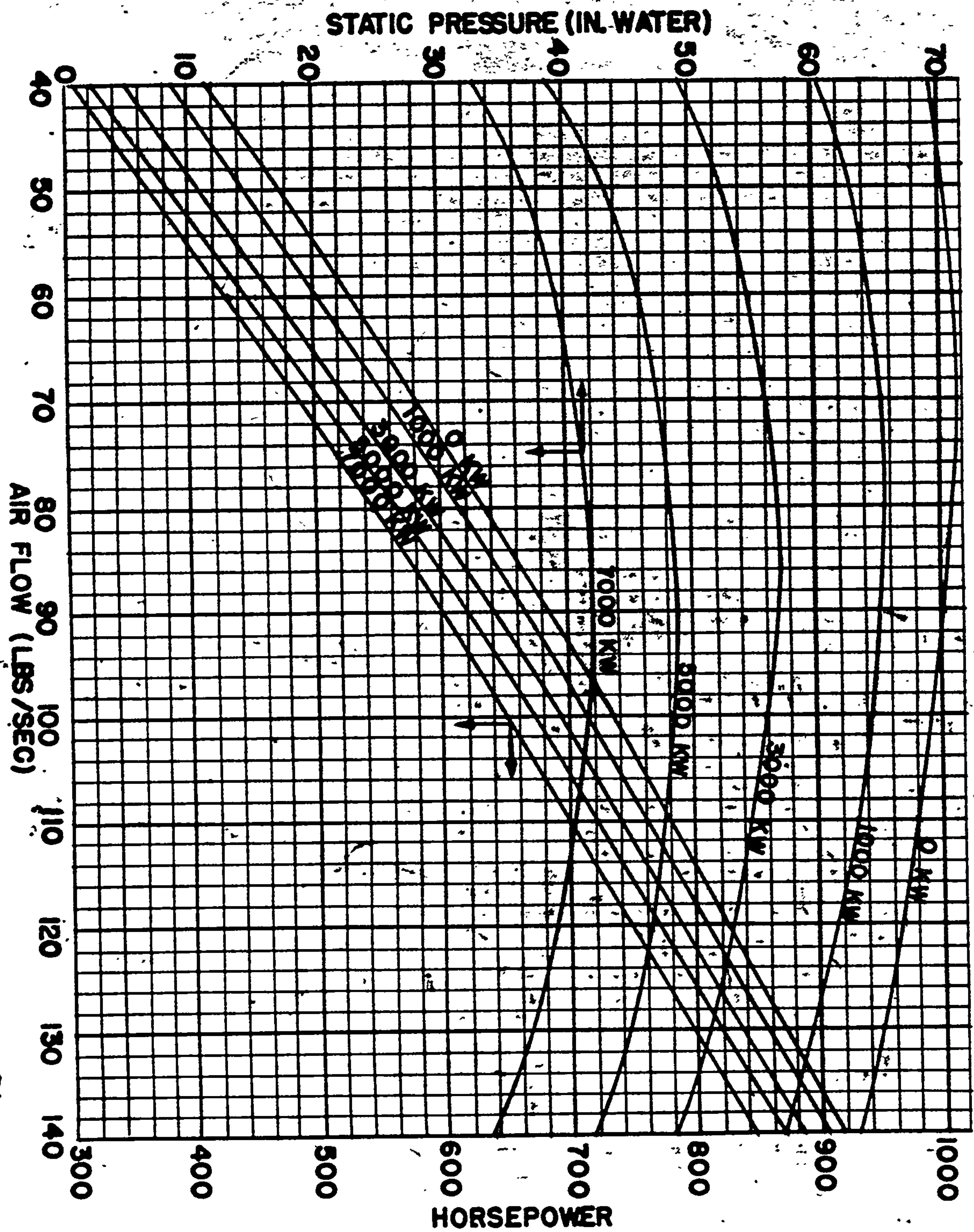

FIG. 3. OPERATING CHARACTERISTICS OF TWO 50000 CFM FANS IN PARALLEL

where Q is in cubic feet per minute measured at local conditions, ΔP is the pressure differential reading of the device, in inches of water, and ρ is the local air density in pounds per cubic foot. The discharge coefficient of the Venturi meter was found to be 0.976 ± 0.005 as compared to the usual value of 0.98 given in engineering handbooks.

7. Air Leakage into Pile

The calibration experiments of the foregoing section resulted in the discovery that 8% to 17% of the air passing through the fans does not go through the pile but leaks in around the pile structure. The normal value seems to be near 15%. Part of this air is introduced intentionally in the space back of the Cellamite false wall which covers the discharge plenum chamber on all sides except the east, occupied by the pile graphite, and the south. Nominally this cooling air is 1000 cfm; actually it may be somewhat more, but in any case it does not account for all the air in-leakage or even a majority of it. Most of the leakage enters through the various experimental holes. By sealing up most of these openings the leakage was reduced to about 8%; some, however, could not be sealed, and most of them cannot be kept sealed without inconvenience to experimenters on the pile.

8. Heat Transfer from Slugs to Air

The principal mode of heat removal from the pile is by transfer of heat from slugs to air. The pile is controlled from observations of the temperature of the hottest, or almost hottest, slug in the pile; this temperature usually must not exceed 250° C and is measured by a thermocouple attached to the slug. The heat transfer coefficient was calculated from the standard Dittus and Boelter equation, using an "effective hydraulic radius". Because of the position of the slug in the channel, at least 1/4 of its surface is not in close contact with the air stream. It was therefore assumed that the effective heat transfer area from slug to air was 3/4 that of the cylindrical slug surface. The calculation of h, the film heat transfer coefficient, requires a knowledge of the air velocity in the channel. However, the air velocity in the channel is not uniform over the pile because the air in the center channels receives more heat, hence becomes less dense than in other channels. Less air thus passes the center channel than the average one. The relationship is approximately

$$\text{Air thru central channel} = \frac{\text{total air through pile}}{\text{Number channels}}\sqrt{\frac{\text{Density in central channel}}{\text{Density in average channel}}}$$

The radical was found to be as small as 0.95, averaging about 0.97, which value was used throughout as an adequate approximation.

The heat transfer coefficient, h, was calculated from the well known equation of Dittus and Boelter[9]

$$h = 0.023\,\frac{k}{d_e}\left(\frac{C_p\mu}{k}\right)^{0.4}\left(\frac{d_eG}{\mu}\right)^{0.8}$$

in which

k = thermal conductivity of the gas
d_e = equivalent hydraulic diameter = 4x cross section/perimeter
C_p = heat capacity at constant pressure
μ = viscosity
G = mass flow rate, lbs/(ft²) (sec).

For air at 58°C (an "average" air temperature in the pile) and channels and slugs as described above d_e = .0762 ft. (free area for flow = 1.99 sq. in.) and the above equation reduces to

$$h = 4.59\, G^{0.8}$$

or

$$h = 142\, W^{0.8}$$

the heat transfer coefficient decreases as the air temperature rises, falling about 0.095% per degree C. air temperature rise.

9. Heat Transfer from Slug to Graphite

A considerable portion of the heat from the slug leaves it to enter the graphite rather than the air. From the graphite it is eventually transferred to air, either in the same channel or another one. This fact somewhat complicates the calculation of the slug temperature so that the following assumptions were made:

1) All the heat generated in a given channel is carried away by the air passing through the channel. There is no substantial conduction through the graphite to other channels.

2) Longitudinal (parallel to air flow) heat conduction in the graphite is negligible compared to the heat produced in any given channel.

Then one may define K, the overall heat transfer coefficient per unit length of metal-filled channel, by the equation

$$q = K\,(t_m - t_a)$$

where

q = heat generation per unit length of channel

t_m = metal surface temperature

t_a = air temperature

Furthermore, one may show that

$$K = h\left[A_m + A_g\left(1 - \frac{1}{1 + \mathcal{H}/hA_g}\right)\right]$$

where

h = film heat transfer coefficient, assumed to be the same for both metal and graphite

A_m = effective metal surface taking part in heat transfer to air per foot of tube - 0.228 ft^2/ft.

A_g = corresponding graphite surface, 0.486 ft^2/ft

$\mathcal{H}$ = contact heat transfer coefficient, PCU/(hr)(°C)(ft length of metal)

This equation follows from the set of equations

$$q = q_m + q_g$$

$$q_m = h\,A_m\,(t_m - t_a)$$

$$q_g = \mathcal{H}(t_m - t_g)$$

$$q_g = h\,A_g\,(t_g - t_a)$$

where

q_m = heat transferred direct from metal to air, PCU/(ft)(hr)

q_g = heat transferred from metal to graphite to air, PCU/(ft)(hr)

t_g = graphite temperature, °C

The contact coefficient $\mathcal{H}$ was evaluated experimentally. A replica of the graphite channel was set up, and a slug bearing a thermocouple welded to its jacket was heated in an oven to 150 - 200° C. The hot slug was then placed in the graphite channel, protected from air currents, and the fall of its temperature as a function of time was followed. The rate of cooling may be assumed proportional to the difference in temperature between the slug and the initial temperature of the graphite since the latter is a good conductor and represents a total heat capacity large compared to that of the slug. The rate of temperature fall is hence

$$\frac{dt}{d\theta} = -\frac{\mathcal{H}t}{C}\ ;\ \mathcal{H} = C/\theta_{rel.}$$

where t is the elevation of slug temperature above graphite temperature, θ is time and C is the heat capacity per foot of the slugs. $\theta_{rel.}$ is the relaxation time of t. $\theta_{rel.}$ proved to be about 3.5 minutes, whence $\mathcal{H}$ = 4 PCU/(ft)(hr)(°C). The relative importance of the various factors going to make up K may be appreciated from the statement that in a rather typical case (air velocity about 150 fps. in metal filled part of pile)

$$K = 27.8\left[0.228 + 0.486\left(1 - \frac{1}{1 + 4/27.8 \times .486}\right)\right]$$

$$K = 27.8\,(0.228 + 0.111)$$

$$K = 9.4$$

In this case it is seen that about 1/3 the heat is transferred by contact between metal and graphite. At higher air velocities this contribution becomes less important, at lower ones more so.

10. Calculation of Maximum Slug Temperatures

The hottest slug evidently will lie in the channel whose radial position is such that this channel is the one of maxiumum average neutron density. For a pile of roughly cylindrical outline the distribution of relative neutron density n/n_o along any channel is given by

$$n/n_o = \cos\frac{\pi x}{r_0}$$

where

n = local neutron density

n_o = neutron density at center of channel (i.e., the maximum density in any given channel)

x = distance along the channel, measured from its center.

L' = effective length of pile. $L' = L + \Delta L$, where L = actual length of metal in channel, ΔL = effective augmentation due to reflector effect, if the reflector is L_r feet thick, approximately

$$\Delta L = 1.6 \tanh \frac{L_r}{1.6} \quad \text{in feet}$$

1.6' is the relaxation length in the graphite.

If one denotes the total pile power by Q, the number of active channels by N_t, and the ratio

$$\frac{\text{heat in given channel}}{\text{heat in average channel}} = \alpha$$

the film temperature difference between metal and air is

$$\frac{\dfrac{\alpha Q}{KN_tL} \cos \dfrac{\pi x}{L'}}{\dfrac{2}{L}\displaystyle\int_{-L/2}^{L/2} \cos \dfrac{\pi x}{L'}\, dx} = \frac{\alpha Q \pi}{2KN_tL'} \, \frac{\cos \dfrac{\pi x}{L'}}{\sin \dfrac{\pi L}{2L'}}$$

The air temperature rise is given by,

$$\frac{1}{wC_p} \, \frac{\alpha Q \pi}{2N_tL'} \, \frac{1}{\sin \pi L/2L'} \int_{-L/2}^{x} \cos \frac{\pi x}{L'}\, dx$$

where

w = weight rate of flow through channel

C_p = specific heat of air

$$= \frac{\alpha Q}{2wC_pN_t} \left[\frac{\sin \dfrac{\pi x}{L'}}{\sin \dfrac{\pi L}{2L'}} + 1 \right]$$

The metal temperature is however the sum

$$t_m = t_o + t_f + t_r$$

where

t_o = incoming air temperature

t_f = film temperature drop

t_r = air temperature rise

whence

$$t_m = t_o + \frac{\alpha Q}{2N_t}\left[\frac{\pi}{L'K}\frac{\cos\frac{\pi x}{L'}}{\sin\frac{\pi L}{2L'}} + \frac{1}{wC_p}\left(\frac{\sin \pi x/L'}{\sin \pi L/2L'} + 1\right)\right]$$

t_m will be greatest in any given channel when its derivative is zero, i.e., when

$$\frac{\pi}{L'}\sin \pi x/L' = \frac{1}{wC_p}\cos \pi x/L'$$

$$\tan\frac{\pi x}{L'} = \frac{L'K}{\pi wC_p}$$

For a typical set of conditions the maximum temperature is calculated by this formula to be at x = 2.4'. The conditions assumed for this calculation were

800 channels
44 slugs/channel
air rate = 120,000 cfm (ρ = 0.058) at fans
G = 8.7 lbs/(ft^2)(sec)
w = 430 lbs./(channel)(hour)
K = 8.4 PCU/(ft)(°C)(hr)
L' = 18'

The calculation of α, the relative heat per channel, depends to a large extent on the location of poison in the pile. This changes from time to time in accordance with the changing purposes for which the pile is used, and although α may be calculated from theory, the complex poison configurations in the pile make such a calculation approximate only. Further, the pile is not always cylindrical, but usually has been somewhat bowed in shape so that the central channels contain fewer slugs than the outer ones. The calculation of α, again, is possible for such configurations, but will not be considered further here. Usually it is in the neighborhood of 1.5. Likewise, the ratio of maximum to average neutron density in the pile is variable but usually in the vicinity of 2.0. Where the exact value of the neutron flux must be known, it is customary for the experimenter to determine it directly, for example, by the use of a silver wire the activation of which is measured.

11. Accuracy of Slug Temperature Predictions

In spite of the various approximations (more will be mentioned subsequently) which are necessary in predicting the slug temperature under given conditions, this prediction proves to be of satisfactory accuracy. For example, Lane and Webster predicted that at 3088 kw, the maximum slug temperature would be 182°C above inlet air. Several months later, when the specified loading was achieved, the slug temperature was found to be 183°C above inlet air temperature.

12. Heat Flow in Graphite

The calculation of slug temperature is made on the assumptions of no heat flow through the graphite between channels nor along a given channel. Since graphite is a fairly good heat conductor, these assumptions evidently require examination.

The first requisite is knowledge of the heat conductivity of the graphite pile. This is less than that of graphite because the joints between blocks act as partial barriers to heat flow. The conductivity of the graphite was determined in two ways:

1. Steady state experiments on a 3' x 3' x 20' pile of graphite erected for the purpose. Only the radial conductivity was determined in these experiments, performed by R. B. Briggs.
2. Unsteady state experiments on the X pile, made by W. R. Kanne. The data were analyzed by J. A. Lane.

In Briggs' experiments an aluminum tube was inserted horizontally through the center of the graphite pile which was 20' long, 3' high, and 3' wide. Numerous thermocouples were buried in the graphite at various distances from the tube, and the tube was fed with dry saturated steam. The condensate was measured, giving the rate of heat conduction away from the tube, and the graphite temperature gradient was determined from the thermocouple readings.

In Kanne's experiment the pile was allowed to come to operating equilibrium. In this condition the center sections of the pile are, of course, hotter than the outside ones. The power was suddenly shut off, and the air flow stopped, and the readings of a number of thermocouples embedded in the graphite followed for several hours. The method of analysis was to imagine the pile as divided into a number of cylinders. Starting with the hottest cylinder, the quantity of heat leaving it in any arbitrary short time interval is calculable from its average temperature decrease, volume, and heat capacity. This heat flows into the adjacent cylinders, part of it flowing radial and part longitudinally, and the rate of flow in each case is proportional to the mean temperature gradient and the thermal conductivity in that direction. Since the temperature gradient can be approximated from Kanne's measurements, one can calculate both the longitudinal and radial conductivities if one assumes the ratio between them. The process may now be repeated on those cylinders adjacent to the first one considered, and the conductivities again are calculated, assuming, as before, a definite ratio between them, and also taking into account the heat flowing into them from the first cylinder. This process can be repeated until all such cylinders in the pile have been considered; if the conductivities thus successively calculated check each other the ratio assumed between them is correct; if not new ratios must be assumed until checks are obtained.

In this way it was determined that the radial conductivity was about 12 PCU/(hr)(ft^2)(^{o}C/ft) in the unsteady state experiments, as compared to about 9.8 in Briggs' experiments; the longitudinal conductivity was found to be 36 PCU/(hr)(ft^2)(^{o}C/ft) from Kanne's data.

With these data one can estimate the radial and longitudinal heat flow in the pile under operating conditions. The method used was to assume first that the graphite temperature was as calculated from the equations of Section 9 of this chapter.

This gave a first approximation to the temperature gradients in the graphite and by starting in the center of the pile and working outward, both radially and longitudinally, one could calculate the second approximation. Actually the Clinton pile is centrally poisoned so that an almost flat radial temperature distribution prevails in the neighborhood of the hottest slugs. Hence an insignificant effect on maximum metal temperature will result.

The longitudinal effect also can be ignored according to calculations by Webster.

13. <u>Heat Flow and Generation in Shield</u>

The shield of the X pile is built without expansion joints; it is therefore necessary carefully to limit temperature gradients in it so as to avoid setting up stresses which would cause it to crack. Practically this amounts to the following, according to the du Pont Design Division:

1. The temperature to which the concrete is exposed locally must not exceed 180° F (82° C).

2. The temperature gradient across any wall must average 30° F (17° C) or less.

When an increase in pile power was proposed, it was therefore necessary to calculate the shield temperatures which would result.

The temperature of the shield rises above cooling air temperature for two reasons.

1. It is in thermal communication with the outside of the graphite pile which in turn is heated by the drainage from its hotter interior.

2. Absorption of neutrons and gamma rays in the shield causes it to heat internally.

The temperature of the graphite was estimated in the manner outlined in the foregoing section. In this case, however, it was found that there was a considerable effect of conduction through the graphite. Thus, it was calculated that, at 3150 kw, an outside channel would average 24° C above incoming air temperature when graphite conduction was taken into account, whereas it would be calculated to average only 16° C above incoming air temperature if graphite conduction were not taken into account. In the former case, of course, the temperature rise exceeds the specifications, and some channels (20) were recommended to be left open so that the graphite would be cooled adequately. This diversion of air flow decreased the pile power about 1%.

The heat generated in the shield by the absorption of neutrons is given approximately by the product

F fissions per second in pile

ν neutrons formed per fission

E energy liberated by capture of 1 neutron in sheild (estimated at 6 mev)

f thermal utilization of neutrons in pile

$k-1$ neutron leakage per generation.

At 3150 kw Lane calculated this heat to be not greater than 10 kw. The temperature rise in the shield was calculated to be about 4° C, and the temperature drop across the asbestos layer also was about 4° C. Summing up the various calculations the final result got by Lane, assuming operation at 3150 kw, with 20 open channels to aid in cooling the graphite, was

Temperature inlet air, average	15° C
Temperature rise of graphite above inlet air Temperature.	16° C
Temperature rise of shield due to neutron absorption	4° C
Temperature drop across asbestos	4° C
Average temperature of room air	20° C
Average temperature difference across shield	19° C

This compares with the prescribed value of 17° C.

Actually the pile has been operated at a power level in excess of that proposed by Lane and Webster for some time, and more than 20 cooling channels have been opened up. The shield temperature difference, meanwhile, has been followed by means of thermocouples in the experimental holes. The above calculations take no account of the fact that some heat leaks outward through the shield. Probably this leakage amounts to about 1/3 kw per side, i.e., about 1/5 the heat of neutron absorption. It is therefore negligible.

14. Methods of Increasing Pile Power

It is advantageous to operate the pile at a power level as high as possible, and for this reason numerous expedients for increasing the power have been proposed and investigated. Some of these will be enumerated and briefly discussed.

The limitation that the shield shall not exceed, locally, a temperature of 82° C applies, in practice, only to the south panel of the pile exit plenum chamber; all other surfaces in the chamber are either free of this limitation (bottom and east side) or are provided with a Cellamite false wall (west, north, and top surfaces) behind which cooling air is introduced. It may reasonably be assumed that the south panel attains the temperature of the air in contact with it; but this temperature is not easily calculated because there is mixing in the plenum chamber. The temperatures in this region have, therefore, been determined experimentally.

While operating at about 1600 kw (before installation of new fans) a maximum temperature of the air in this region was found to be 71° C and the minimum 40° C. The average was 55° C. Higher temperatures have probably been experienced since installation of the new fans and increasing the pile power, but no ill effects have been observed. Another limitation is that placed on maximum slug temperature. It has been found that at high temperatures the cans develop leaks which permit air to oxidize the slugs. The expansion resulting from the formation of oxide then ruptures the can, spilling active dust into the air stream and eventually blocking the flow of air in the channel. This effect is found not to be serious below 240°- 250° C.

The simplest method of increasing the pile power if the maximum slug temperature is not being attained is to allow a higher slug temperature. The increase thus attained is not quite linear in temperature, however, because the decreased density of the hotter air leads to increased air resistance in the pile and heated ducts, and also to decrease pressure rise and decreased mass rate of air through the fans. The heat transfer coefficient also decreases slightly. In the region of interest an increase of about 10 kw per degree of maximum slug temperature increase can be expected.

Another simple method of increasing the power is to "monitor" the air supply, i.e., to distribute it among the channels in proportion to the energy generation in each. It has been estimated that the power could be increased by several per cent if this were done. However, it is quickly seen that this procedure would result in over-heating the shield and the south panel of the discharge chamber. Gains from this procedure are therefore rather limited and have not been considered worthwhile.

The power can also be increased by cooling the pile inlet air, for example by a humidifying tower. The effectiveness of this procedure depends upon the spread between the atmospheric dry bulb and wet bulb temperatures. This spread is considerable only during a portion of each summer day, and it has been estimated that the attainable increase would be less than 5% averaged over the year.

Cooling the air after its discharge from the pile but before passage through the fans would increase the air density at the fans, increase the pressure rise across them and increase their mass handling capacity. Probably the method of cooling would be a tubular heat exchanger, and this would result in increased pressure drop in the cooling system. There would also be a considerable amount of new construction required for installation of such a cooler. It has been proposed that the cooling be accomplished by a spray of cold water directly introduced. However, this might lead to entrainment of droplets which would be carried through the fans, possibly causing damage there. A rough estimate of the power increase attainable by after-cooling of the air gives ~25%.

A somewhat different proposal has been made that water could be introduced into the air as a fine mist or fog, thus effectively increasing its average specific heat. This has been approached in a limited number of experiments which indicated a possible gain of ~40%. However, numerous practical difficulties were encountered, and the proposal was considered not attractive.

15. Typical Operating Data

The following table presents operating data typical of June 1946.

Table 4

Typical Operating Conditions, June 1946

Power, average while operating		3400 kw.
% of time not operating		9%
Metal in pile		49.09 tons
Pile loading		
24 uranium metal slugs/ channel	1 channel	
36 uranium metal slugs/channel	59 channels	
40 uranium metal slugs/channel	26 channels	
44 uranium metal slugs/channel	306 channels	
50 uranium metal slugs/channel	438 channels	
2 uranium metal doughnuts	1 channel	
4 thorium metal slugs	1 channel	
40 thorium carbonate slugs/channel	2 channels	
14 thorium carbonate slugs	1 channel	
30 molybdenum metal slugs	1 channel	
65 calcium nitrate slugs/channel	100 channels	
Empty channels	316	
Air inlet temperature		25° C
Air discharge temperature		90° C
Air into pile		6500#/ min.
Air into fans		7200#/ min.
Static pressures in system		
atmospheric		740 mm. of mercury
at Pitot		722
across Pile		36
at Venturi		664
at fan inlet		662
at fan exit		753

Maximum metal temperature	245° C
Average metal temperature	~120° C
Maximum graphite temperature	150° C
Average graphite temperature	~90° C

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