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**EVALUATION OF GAS-GRAPHITE SUSPENSIONS
AS NUCLEAR REACTOR COOLANTS**

by

G. P. Wachtell
J. P. Waggener
W. H. Steigelmänn

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August 1961

Prepared for

U. S. ATOMIC ENERGY COMMISSION

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THE FRANKLIN INSTITUTE
LABORATORIES FOR RESEARCH AND DEVELOPMENT
PHILADELPHIA PENNSYLVANIA

Review

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ABSTRACT

A survey was made of experimental and theoretical work pertaining to the use of a suspension of graphite particles in a gas as a nuclear reactor coolant. Following a general review of the state-of-the-art and a detailed study of recent experimental data, it was concluded that present theory and experimental data are encouraging but inadequate as a basis for reactor design. Especially lacking are both theoretical and experimental work defining the hydrodynamic and thermal requirements for preventing the plating-out of the solids from the suspension. Future work is recommended in which the stability of suspensions receives primary attention. Supporting parallel studies of basic theory, erosion, effects of nuclear radiations, and compatibility of materials, as well as development of specialized instrumentation and plant components, are also recommended. These would then be followed by experimental heat transfer and pressure drop studies to obtain correlations for reactor design applications.

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LIST OF SYMBOLS

SUBSCRIPTS AND SUPERSSCRIPTS:

★	Designates that the quantity is based upon a unit volume of mixed gas-solids suspension
AB	Between heater sections A and B
BC	Between heater sections B and C
B	Black wall
cd	Conduction
cv	Convection
G	Refers to properties of a purely gaseous coolant
g	Refers to properties of the gaseous phase of the suspension
geom	Geometrical value
hi	Heater inlet
ho	Heater outlet
i	Generalized subscript
L	Left
P	Refers to properties of the particles in the suspension
R	Right
rd	Radiation
S	Refers to properties of the suspension
w	Heater tube wall (inner surface)
0	Initial value or value at $x = 0$
1,2	Designations of points or walls

NOTE: A bar above quantity denotes the average value of that quantity.

NOMENCLATURE:

A	Area
A'	Projected area
a,b,c	Variable exponents

LIST OF SYMBOLS (Cont.)

C	Specific heat capacity at constant pressure
D	Characteristic dimension or diameter of a channel
d	Boundary layer thickness
F_i	Fraction of black body radiation energy in the i^{th} wavelength range
f	Friction factor
g	Gravitational acceleration
H	Total rate of heat transfer
h	Heat transfer film coefficient (the absence of a subscript designates that the total coefficient is intended)
I	Intensity of infra-red radiation
k	Thermal conductivity
L	Length of a channel
λ	Exponent of the Reynolds number in the friction factor correlation
m	Exponent of the Reynolds number in the heat transfer film coefficient correlation
N	Number density of particles
Nu	Nusselt number $(\frac{hD}{k})$
n	Exponent of the Prandtl number in the heat transfer film coefficient correlation
P	Pumping power
Pe	Peclet number $(\frac{\rho VDC}{k})$
Pr	Prandtl number $(\mu C/k)$
p	Pressure
Δp	Pressure difference
Q	Rate of energy absorption
q	Heat flux
R	Radius of a sphere enclosing a volume element
Re	Reynolds number $(\frac{\rho VD}{\mu})$

LIST OF SYMBOLS (Cont.)

r	Radius of a spherical particle
T	Temperature
$\hat{T}$	Temperature (absolute scale)
ΔT	Coolant temperature rise during flow through a channel
δT	Average temperature difference between the coolant and a channel wall
t	Thickness of a flake-like particle
V	Flow velocity
v	Volume
W	Mass flow rate
X, Y, Z	Exponents in coolant comparison expressions
x	Distance Variable
α	Numerical constant in the heat transfer film coefficient correlation
β	Numerical constant in the friction factor correlation
γ	Absorptivity
θ	Loading ratio of suspension (weight of solids per unit weight of gas)
λ	Wavelength of radiation
μ	Viscosity
ρ	Density
σ	Stefan-Boltzmann constant
σ_t	Microscopic cross section for absorption and scattering of infra-red radiation
τ	Temperature gradient
$\bar{\Phi}$	Volumetric flow rate
ϕ	Angle between incoming radiation and particle surface normal
ζ	Mean free path of infra-red radiation

I. INTRODUCTION

In the selection of coolants for nuclear power reactors, the use of a gas offers several advantages; especially for high temperature applications. Hydrogen, helium, nitrogen, carbon dioxide, and argon (and mixtures of these) have all been considered as potential reactor coolants. The most attractive characteristic of gaseous coolants is their general inertness with respect to chemical, metallurgical, and nuclear reactions. Gases, however, due to low volumetric heat capacities and low film heat transfer coefficients, require large and consequently relatively expensive equipment. A means of increasing the volumetric heat capacity and heat transfer of gases without the offsetting expense of high pressure equipment will enable a direct decrease in the capital costs of nuclear power. With this in mind, the concept of using a two-component fluid comprised of a gas and small solid particles has been seriously considered. Volumetric heat transport is obviously increased by using suspended solids. However, the effect of using suspended particles in a gas stream upon pressure drop and heat transfer are not so obvious. Likewise, the means of circulating and measuring the flow of a mixture of solids in a gas require study. Questions regarding erosion of equipment and behavior of the mixture over an extended time in a reactor environment must be resolved.

Work recently completed by the Babcock and Wilcox Company represents the most extensive effort to date towards answering such questions. A series of reports (the last issued in October 1960) describes this work, and it is recommended in the final report that the research be extended. With these reports as a focal point, the Atomic Energy Commission requested The Franklin Institute to review the status of the gas-suspension technology and recommend a future course of action. Accordingly, work at The Franklin Institute was initiated on January 9, 1961.

II. SURVEY OF THE STATE-OF-THE-ART

A. THEORETICAL WORK

The most complete analytical investigation of heat transfer to a flowing two-phase system in a closed conduit was published two years ago. This work (1), "Transport Processes in Two Phase Turbulent Flow" by Chang-Lin Tien was a doctoral dissertation completed at Princeton University in June 1951. Tien's work is limited by a number of assumptions in order to render the problem more tractable. Most limiting among these is that the particles are sufficiently far apart so that the velocity distribution of the carrier fluid remains unaffected by the presence of the particles. The author states that this assumption is invalid for solid loadings ratios greater than 1 and calculated curves are presented which compare well with low temperature air-solids data reported in Reference (11) at loading ratios up to 1. In other calculations the effect of radiation is included for the same loading condition. No data are available as yet for checking radiation heat transfer. It is noted that Tien predicts only a 5% increase in Nusselt Number as the loading ratio increase from 0 to 1. For graphite, a loading ratio of 1 corresponds to about 0.1% solids volume fraction.

In his introductory remarks, Tien outlines previous studies on the related topics of the statistical theory of turbulence, particle motion, and momentum transfer. His bibliography includes 47 items covering the years from 1888 through 1958, and should be consulted for any related study.

At the present there is no theoretical basis for predicting heat transfer at higher solids volume fractions. Apparently the next theoretical advance will depend on relating to heat transfer the effects of the particles on the velocity distribution of the fluid (both the solid and gaseous phases), and on the structure of the turbulence.

Hanratty and Kada (2) studied the effects of the presence of solids on the diffusion of a salt solution injected in a vertical stream. Peskin (3) treated the interactions of particles in a stream. Soo and Tien (4) in 1959 considered the effect of the wall on two-phase turbulent motion. The inclusion of these effects on heat transfer, however, has not appeared.

The work of these and others referenced are applied directly or indirectly to the problem of predicting the heat transfer from a surface to a two-component flow. As particles are added to a gas in order to increase the heat transmission and heat transport thereof, the density of the mixture will increase and, for the same volumetric flow rate, the power required to pump the fluid will also correspondingly increase. Another phase, therefore, of the problem associated with selecting a good nuclear coolant is concerned with evaluating the gain in heat transport versus the cost of circulating the coolant.

B. EXPERIMENTAL WORK

Experiments studying heat transfer to mixtures of solids in gases and solids in liquids have been reported over the past decade. Some of the earliest work was done at the University of West Virginia in a series of four theses starting in 1950. Air was used to transport small particles of various materials including soft coal. References (5) and (6) summarize this work, which is more exploratory than definitive. At about the same time, Bonilla (7) studied heat transfer to chalk and water mixtures. Orr (8) continued work with solids in water at Georgia Institute of Technology under the direction of Prof. J. M. DallaValle and completed an excellent study there in 1952. He concluded that the Sieder and Tate (18) relationship derived for homogeneous fluids also applied to mixtures of liquids and solids. In 1954 Peskin (61) was granted a U.S. Patent claiming the use of solid particles in gases in a heat transfer system. White (9) reported additional studies with water in 1955. Further work of an exploratory nature was reported by Schluderberg (10) working with

solids and ceramic particles. Farbar and Morley (11) at the University of California published results in 1957 of studies of heat transfer to air-solids mixtures. This was used by Tien (1) for comparing his theory to experiment. Farbar and Morley reported a sharp increase in heat transfer after solids loading was increased to over a one-to-one ratio. They concluded that the Nusselt number varied with Reynolds number to the 0.6 power and loading ratio to the 0.45 power. Work sponsored by the United States Atomic Energy Commission was recently completed by the Babcock and Wilcox Company and is reported in references (12) through (17). These studies of heat transfer and pressure drop included nitrogen, helium, and freon as the gaseous phase mixed with graphite particles in several tube diameters and over a wide range of temperature. Numerous parallel studies were conducted by Babcock and Wilcox to determine the feasibility of utilizing a gas-graphite mixture as a nuclear reactor coolant. An analysis of some of the experimental data from this work is presented in Appendix A. The Bureau of Mines, under its Process Heat Reactor Program, is currently studying problems associated with circulating gas-graphite mixtures at about 1200 degrees Fahrenheit (22, 23, 24). Heat transfer studies in this program are presently limited by instrumentation developments. Wen and Miller (25) recently presented a correlation of many of the references mentioned above including the Babcock and Wilcox work. They concluded that the governing factors were solids loading and particle size. The indicated scatter from the correlation is in some conditions over 100%. Some of the studies do lie quite close to the generalized correlation, and it is interesting to see all of them as well as some fluidized bed work correlated on the same basis. This correlation is unique in that it includes the terminal velocity of the particle in the carrier fluid.

Numerous studies have been made of the pressure drop in transporting solids by means of gases and liquids. Typical of these are some theses from the Georgia Institute of Technology (26, 27, 28). Another study by Muschelknautt in 1959 is available in German (29). An interesting

study by Wen and Simons (30) describes forms of very dense transport in horizontal pipes. Orr (8) concluded that the conventional smooth tube data applied to water-solids mixtures if the viscosity of the water and density of the mixture were used to calculate Reynolds number.

The problem of particles "plating-out" of suspension may be a severe limitation on the design of suspension-type heat transfer systems. Aitken (31) studied the tendency of particles to collect on the colder of two surfaces defining a thermal gradient across a dusty space. Figure 6 of Reference (24) is a dramatic example of what may be this same phenomenon occurring in a finned tube heat exchanger at the Bureau of Mines loop. Friedlander and Johnstone (32) in 1957 described studies of particle deposition from a turbulent gas stream. Work on the homogeneous reactor concept at Oak Ridge National Laboratory included studies (33, 34) of the problem of thorium oxide particles plating-out of water systems. No systematic study of either a theoretical or experimental nature has been made to define the hydrodynamic and thermal conditions required to maintain particles in a gas suspension over extended periods of continuous operation. Korsmeyer, following conversations with personnel of the Bureau of Mines and the Babcock and Wilcox Company, recommended that in-pile facilities at the Oak Ridge National Laboratory should not be used for proposed gas-graphite suspension tests because of the lack of knowledge regarding prevention of plating-out (35). If high Reynolds numbers are required to maintain suspensions in the presence of large thermal gradients, it should be noted that in gases, both density and viscosity vary adversely with increasing temperature, thus decreasing the Reynolds number. Reference (35) noted that the shape of particles makes a difference in the caking problem associated with operating with water-uranium dioxide particles. Spherical particles are preferred to flat ones.

No experiments have been reported which define the degree to which thermal radiation will increase heat transfer to gas-solids mixtures. References (36), (37), and (38) pertain to the theory of the

problem. Tien (1) predicted that thermal radiation will be important at wall temperatures greater than approximately 1000°F at light loadings. Appendix B of this report concludes that this may also be true for heavily-loaded suspensions.

Erosion has not been observed at either the Babcock and Wilcox tests or the Bureau of Mines tests. The effect of prolonged loop operation on erosion has not been investigated.

Early work by D.C. Schluderberg(10) and current Bureau of Mines work have shown that centrifugal impellers of standard design are adequate to circulate gas-graphite mixtures as if they were homogeneous fluids. No experimental study of the effect of graphite on pump efficiency has been reported.

C. REVIEW OF BABCOCK AND WILCOX WORK

The Babcock and Wilcox work is presented in seven volumes dated from August 1959 to October 1960. Within the broad scope of activities covered in these reports, of primary importance are the experimental results relating to heat transfer and pressure drop of a gas-graphite mixture. Representatives of the Atomic Energy Commission selected some of the work done in the latter part of the project as representative of the most accurate and reliable data acquired. Of primary interest in this work was heat transfer data taken in plain round tubes because they enable the correlation expression for the data of the experiments to be directly compared to a pure gas heat transfer correlation.

Starting with essentially raw data consisting of thermocouple voltages, currents and voltages to the heater sections, fluid density and pressure readings, Franklin Institute personnel recalculated heat transfer film factors and other correlation parameters. The data used and the heat transfer film factors calculated therefrom are reported in Appendix A. The resultant data are plotted in Figure 1. The solid line in Figure 1 is a correlation (39) of local heat transfer coefficients for fully developed flow in smooth, round tubes. When $\theta=0$, the

FIG 1 CORRELATION OF THE BABCOCK AND WILCOX EXPERIMENTAL HEAT TRANSFER DATA
(Suspension is Being Heated)

correlation expression, which is presented on the figure, reduces to that given in Reference (39) for a pure gas. Significant to the comparison is the loading ratio term. At equal Reynolds numbers and Prandtl numbers the mixture has a smaller heat transfer coefficient than pure gas. At equal velocities however, the addition of graphite increases the transfer of heat in a similar manner but to a lesser degree as would an equal increase in density due to a higher pressure in the case of pure gas.

The evaluation of a pressure drop correlation was not possible because of the scatter of the data. As stated in Appendix A, the experimental data appear to indicate that the friction factor may be smaller than that for a pure fluid, but the data must be considered inconclusive at this time.

III. COMPARISON OF REACTOR COOLANTS

The only valid method of assessing the total "quality" of a proposed coolant for a nuclear power reactor is to determine the cost of producing power in a reactor which was optimized in terms of its design and operating conditions for that coolant. (This viewpoint is also expressed in Reference (40).) Since such an exhaustive calculation was beyond the scope of this study, and because the engineering properties of graphite-gas suspensions are still largely unknown, we have attempted to derive some simple relationships which indicate, under certain specified conditions, how a graphite-suspension coolant compares to a purely gaseous coolant in regard to a few of the quantities which have a relatively strong influence on power reactor economics.

A. PRELIMINARY CONSIDERATIONS

Lacking an adequate heat transfer theory for estimating heat transfer to a heterogeneous mixture, it is interesting to apply a correlation expression used in predicting the heat transfer to the flow of pure fluids. This was done by assuming that the addition of particles would change the heat transfer rate simply by changing the values of viscosity, heat

conductivity, specific heat, and density, which were calculated and estimated for the mixtures at various solids loadings. The resulting estimates of film coefficients of suspensions compared to the pure gas values are shown in Figure 2.

The ordinate of Figure 2 is the ratio of the calculated film coefficients of a mixture having ρ_p^* pounds of graphite per cubic foot to the corresponding film coefficient for pure gas flowing at the same velocity. The construction of Figure 2 points up a property of the suspension which may be important for maximum heat transfer results. Curves A and B are both for helium-graphite suspensions. Curve C is a plot of the essentially identical points corresponding to both nitrogen and argon suspensions. Curve A is based on spherical graphite particles while curve B is based on flake-like particles. In both helium cases, the variation of fluid viscosity was estimated from viscosity tests made of various particle shapes in water discussed in Reference (8), and the thermal conductivity was also calculated from data in Reference (8). Whether these data are applicable to gas is yet to be shown. The end point of both curves is at a density indicative of the void fraction available for solids before clogging occurs. The use of spherical particles may be advantageous on three counts: (a) If graphite is allowed to settle out of two equal volumes of equal density gas suspension, the height of a chip-like suspension will be four or five times greater than the height of a suspension of spherical particles. It would, therefore, be expected that more mass in the shape of spheres could be circulated than in the shape of flakes. (b) At a given mass density of the mixture, a mixture containing spherical particles is expected to have a smaller viscosity and therefore a smaller pressure drop than one containing the flake-like particles. (c) For a given mass density, thermal radiation heat transfer may be increased.

The probable relative effect of using different gases as the carrying medium for graphite is shown in Figure 3. The expected superiority of helium suspensions over nitrogen and argon suspensions is

FIG 2 ESTIMATED EFFECT OF GRAPHITE LOADING ON RELATIVE
HEAT TRANSFER TO GAS SUSPENSIONS

$$h = 0.027 \frac{k}{D} \left(\frac{VD\rho}{\mu} \right)^{0.8} \left(\frac{\mu C}{k} \right)^{0.33} \quad (\text{REF 8})$$

V, D Constant

k, μ, ρ, C Properties of gas or suspension
at 300psia, 572°F

FIG. 3. ESTIMATED EFFECT OF GRAPHITE LOADING ON RELATIVE HEAT
TRANSFER TO GAS SUSPENSIONS - FLAT PARTICLES

$$h = 0.027 \frac{k}{D} \left(\frac{VD\rho}{\mu} \right)^{0.8} \left(\frac{\mu C}{k} \right)^{0.33} \quad (\text{REF 8})$$

V, D Constant

k, μ, ρ, C Properties of suspension
at 300 psia, 572°F

clearly shown here. This result might be expected on the basis of the conductivity comparison between the pure gases involved, helium having a conductivity five and seven times that of nitrogen and argon, respectively, at typical conditions.

Figure 4 shows the influence of the graphite loading ratio on the specific heat of a helium-graphite suspension, both on volumetric and on mass bases. It is noted that the effect of the specific heat of the gas is of minor importance for moderate suspension densities. In all cases the specific heat of the graphite is the paramount factor. From considerations such as these, one is led to suspect that the use of graphite in helium might promise favorable heat transfer coefficients.

An important criterion in the selection of one coolant over another is the pumping power ratio; the ratio of the power extracted by the gas compared to the power required to pump it, under a given set of operating conditions. Given a reactor configuration with a fixed output power rating, a fixed temperature rise of the coolant, and assuming that pressure drop varies as the square of the flow, that the circulator efficiencies are equal, and that friction coefficients are equal, it can be shown that the pumping power ratio varies inversely as $\rho^2 c^3$ of the circulating fluid. A comparison of a helium-graphite suspension compared to pure helium on this basis is shown in Figure 5. From the figure, it is seen that at a fluid density of 2 lbs/cu ft the pumping power ratio is only 10% of that of pure helium. It must be noted that in this comparison, the heat transfer characteristics of the suspension are neglected. The assumption is made that the fuel surface temperature will reach whatever value it required to insure adequate heat transmission to raise the coolant temperature by the required amount as it passes through the core. Since, for a given power output, the flow rate will decrease as the volumetric heat capacity increases, the resulting film factor at low flows may be prohibitively small because it results in an excessively large fuel temperature.

FIG. 5. EFFECT OF GRAPHITE ON PUMPING POWER RATIO WHERE $\frac{\text{PUMP POWER}}{\text{REACTOR POWER}} \propto \frac{1}{C^3 \rho^2}$

$$\rho_{He} = 0.106, \quad \rho_S = 106, \quad (C)_{He} = 1.24, \quad C_S = 0.325$$

B. DETAILED ANALYSIS

In the cases considered here, we will compare the performance of the two coolants in reactors having identical geometries, or, which is the same thing, in the same reactor. For the purpose of this analysis, consider the coolant flowing in a round channel of diameter D through a homogeneous reactor core of length L . The total rate of heat transfer to this channel is H , the power required to circulate the coolant through the channel is P . The total temperature rise of the coolant from inlet to exit is ΔT , and the average temperature difference between the coolant and the channel wall is δT . The following relationships are assumed to apply, and to each factor and exponent in these relationships either the subscript S or G (for suspension- and pure gas-type coolant, respectively) may be applied.

$$H = \pi D L h \delta T \quad (1)$$

$$P = \dot{V} \Delta p = \left(\frac{\pi}{4} D^2 V \right) \left(f \frac{L}{D} \frac{\rho V^2}{2g} \right) = \frac{\pi}{8g} f D L \rho V^3 \quad (2)$$

$$\Delta T = \frac{H}{\frac{\pi}{4} D^2 \rho V C} \quad (3)$$

$$f = \beta \left(\frac{\rho V D}{\mu} \right)^\ell = \beta \rho^\ell V^\ell D^\ell \mu^{-\ell} \quad (4)$$

$$h = \alpha \frac{k}{D} \left(\frac{\rho V D}{\mu} \right)^m \left(\frac{C \mu}{k} \right)^n = \alpha k^{1-n} \rho^m V^m D^{m-1} C^n \mu^{n-m} \quad (5)$$

These equations may be combined and factored as shown below:

$$\begin{aligned} \frac{1}{\pi} \frac{H}{\delta T} &= (DL) \left(\alpha k^{1-n} \rho^m V^m D^{m-1} C^n \mu^{n-m} \right) \\ &= \alpha k^{1-n} \rho^m V^m D^m L C^n \mu^{n-m} \end{aligned} \quad (6)$$

$$\begin{aligned} \frac{8g}{\pi} P &= (DL^3 \rho V^3) (\beta \rho^\ell V^\ell D^\ell \mu^{-\ell}) \\ &= \beta \rho^{\ell+1} V^{3+\ell} D^{\ell+1} L \mu^{-\ell} \end{aligned} \quad (7)$$

$$\frac{4}{\pi} \frac{H}{\Delta T} = D^2 \rho V C \quad (8)$$

Now, forming the product of equation (6) raised to the ath power, equation (7) raised to the bth power, and equation (8) raised to the cth power yields

$$\begin{aligned} \left(\frac{1}{\pi} \frac{H}{\delta T} \right)^a \left(\frac{8g}{\pi} P \right)^b \left(\frac{4}{\pi} \frac{H}{\Delta T} \right)^c = \\ \alpha^a \beta^b \rho^{am+b(\ell+1)+c} V^{am+b(3+\ell)+c} D^{am+b(\ell+1)+2c} L^{a+b} C^{an+c} k^{a(1-n)} \mu^{a(n-m)-b\ell} \end{aligned} \quad (9)$$

We may now eliminate certain of the parameters by selecting appropriate values for a, b and c. The resulting equation in each case can be solved for one or more parameters which are to be compared.

The first case to be considered investigates the pumping power and the average wall-to-fluid temperature drop. Selecting the values $a = -1$, $b = m/(3+\ell)$, and $c = 0$ eliminates V and ΔT as parameters. The resulting equation is

$$\delta T P^X = \frac{\pi^{X-1}}{(8g)^X} H \alpha^{-1} \beta^X \rho^{-2X} D^{-2X} L^{X-1} C^{-n} k^{-(1-n)} \mu^{3X-n} \quad (10)$$

where $X = m/(3+\ell)$.

Let us now assume that the identical reactors postulated earlier (one operating with a suspension-type coolant and the other with a pure gas) are both operating at the same power level, so that H is the same for both. Introducing suitable subscripts, we may determine a ratio using equation (10) for a suspension-type coolant and for a purely gaseous coolant. This ratio is:

$$\frac{(\delta T)_S P_S^{X_S}}{(\delta T)_G P_G^{X_G}} = \left(\frac{8g D^2}{\pi L} \right)^{X_G - X_S} \frac{a_G^{X_S} \beta_S^{2X_G} \rho_G^{n_G} C_G^{1-n_G} \mu_S^{3X_S - n_S}}{a_S^{X_G} \beta_G^{2X_S} \rho_S^{n_S} C_S^{1-n_S} \mu_G^{3X_G - n_G}} \quad (11)$$

In attempting to substitute numerical values for the various quantities appearing in equation (11) a number of complications arise. The quantities μ, ρ, k , and C are dependent upon the temperature and pressure conditions assumed, and, for the suspension, they are dependent upon the weight of graphite per unit volume of fluid, ρ_p^* . In addition, the values of a, β, ℓ, m , and n are not accurately known for the suspension. In this comparison, we shall give the gaseous coolants the benefit of a higher pressure. Since there are still a number of significant questions as yet unanswered pertaining to circulating a suspension, the economic favor (capital charges) obtained by operation at lower pressures may, at least in part, be cancelled by the increase costs for non-standard pumping and coolant handling equipment.

The numerical comparison will be based on helium as the purely gaseous coolant, and on argon-, nitrogen-, and helium-graphite mixtures as the suspension coolant. A range of values for the weight of graphite per unit volume of fluid will be investigated in each case. Because of the lack of useful data to the contrary, we shall assume that the thermal conductivity and viscosity of the suspension are equal to the values for the gas component in the suspension. Also, because of the lack of conclusive data, we shall assume that the relationship between friction factor and Reynolds number is the same as for purely gaseous coolants.

The numerical values for the heat transfer correlation for the suspension and for the gas will be based upon the formula given in Figure 1.

Tabulations of the numerical values used are presented below.

Parameter	Gaseous Coolant		Suspension Coolant	
T_f , °F	1000		1000	
p_f , psia	300		200	
λ	-0.2		-0.2	
β	0.19		0.19	
m	0.8		0.8	
n	0.4		0.4	
α	$0.0205 \left(\frac{\hat{T}}{\hat{T} + \frac{ST}{2}} \right)^{0.8}$		$0.0205 \left(\frac{\hat{T}}{\hat{T} + \frac{ST}{2}} \right)^{0.8} (\theta+1)^{-0.4}$	
Material Property	Graphite	Argon	Nitrogen	Helium
ρ , lb/ft ³	120	0.51	0.36	0.051 at 200 psia 0.077 at 300 psia
C , Btu/lb-°F	0.40	0.126	0.27	1.24
k , Btu/hr-ft-°F	--	0.022	0.033	0.15
μ , lb/ft-hr	--	0.115	0.088	0.094

The results of the numerical comparison appear in Figure 6. The dependence of the numerical value for α on the wall-to-fluid temperature drop has been neglected since, for most cases, this dependence is not significant. It can be seen from the curves in this figure that, under the conditions and assumptions cited previously: (a) an argon suspension apparently does not offer any advantages over either of the other types of suspension or a purely gaseous coolant, (b) a helium suspension is superior (in the sense that it enables a lower fuel temperature for a given coolant outlet temperature and a given pumping power) to both an argon and a nitrogen suspension, and (c) in all three instances, the more heavily loaded suspension is better. None of these cases, however, show a marked superiority over pure helium.

The next case to be considered will investigate the heat transport capabilities of suspension-type coolants. In particular, we will be concerned with the relationship between power removal and the film temperature drop on the wall as a function of the coolant through the reactor and the fraction of

FIG. 6. FILM TEMPERATURE DROP RATIO VERSUS GRAPHITE LOADING
(L, D, P, and H Identical and Constant)

the total power used for circulating the coolant are both constant. This investigation is similar to that reported by the Babcock and Wilcox company on pages 38 through 44 of Reference (17). We shall first derive an expression for the ratio of the power transferred to the suspension to that transferred to the purely gaseous coolant, and then derive an expression for the ratio of the film temperature drops for the two fluids.

The exponents a , b , and c in equation 9 are first chosen as 0, $-1/(2+\ell)$, and $(3+\ell)/(2+\ell)$, respectively, to eliminate the parameters V and δT and to obtain the ratio P/H which is to be held constant for the two fluids. The equation which results from this substitution and some rearrangement is

$$H = \frac{\pi}{4} \Delta T^{Y+1} \left(\frac{2g}{L} \frac{P}{H} \right)^Y D^{Y(5+\ell)} \beta^{-Y} \rho^{2Y} C^{Y(3+\ell)} \mu^Y \ell \quad (12)$$

where $Y = (2+\ell)^{-1}$. Again introducing the subscripts "S" and "G", we may take the ratio of equation (12) for the suspension and for the pure gas. There results:

$$\frac{H_S}{H_G} = \left(\frac{2g}{L} \frac{\Delta T}{H} \frac{P}{H} \right)^{Y_S - Y_G} D^{Y_S(5+\ell_S) - Y_G(5+\ell_G)} \frac{\beta_G^{Y_G} \rho_S^{2Y_S} C_S^{Y_S(3+\ell_S)} \mu_S^{Y_S \ell_S}}{\beta_S^{Y_S} \rho_G^{2Y_G} C_G^{Y_G(3+\ell_G)} \mu_G^{Y_G \ell_G}} \quad (13)$$

To obtain the corresponding expression for the ratio of the film temperature drops, the values $a = 1$, $b = -(1-m)/(2+\ell)$, and $c = (3+\ell-m)/(2+\ell)$ are selected. Substituting these values into equation 9 yields

$$\delta T = \left(2g \frac{P}{H} \right)^Z \left(\frac{\Delta T}{L} \right)^{Z+1} D^{3Z+1} a^{-1} \beta^{-Z} \rho^{2Z} C^{Z-n+1} k^{-(1-n)} \mu^{m-n+Z} \ell \quad (14)$$

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where $Z = (1-m) / (2+\ell)$. The ratio of the film temperature drops is found to be

$$\frac{(\delta T)_S}{(\delta T)_G} = \left(2g D^3 \frac{\Delta T}{L} \frac{P}{H} \right)^{Z_S - Z_G} \frac{\alpha_G^{Z_G} \beta_G^{2Z_S} \rho_S^{Z_S - n_S + 1} k_G^{1 - n_G} \mu_S^{m_S - n_S + Z_S \ell}}{\alpha_S^{Z_S} \beta_S^{2Z_G} \rho_G^{Z_G - n_G + 1} k_S^{1 - n_S} \mu_G^{m_G - n_G + Z_G \ell}}$$

The curves appearing in Figures 7A and 7B result from using the numerical values which were presented in the first case examined. The first of these figures shows the power removal ratio versus graphite loading, and was computed using equation (13). The curves in Figure 7B show the corresponding film temperature drop ratio (from equation (15)). The data presented in these figures can be combined as a plot of the temperature drop ratio versus the power removal ratio, as appears in Figure 8. From this latter figure, it can be seen that, in the same or in identical reactors, it is possible to increase the power generation while maintaining the coolant inlet and outlet temperatures and the fractional pumping power constant by using a suspension-type coolant. The film temperature drop increases as the power increases, but at a much smaller rate. Again, a helium-graphite suspension appears to be superior.

In order to investigate the significance of the $(\theta+1)^{-0.4}$ term in the correlation equation, we next assume that the pure gas correlation applies equally well to suspensions, as Orr (8) found with water suspensions when the fluid properties are properties of the mixture. The calculations are therefore repeated with $\alpha_S = \alpha_G$. The results of these repeat calculations for the first case considered appear in Figure 9, and for the second case in Figure 10. It can be seen that a significant improvement in the core temperature conditions or in the core power density would be achieved if further experimental work should verify that Orr's results apply for gaseous suspensions as well as for liquids. The reason for the peculiar shape in the helium-graphite curve in Figure 10 is the fact that the density of the gas in the suspension is less than that of the purely gaseous coolant.

FIG. 8. FILM TEMPERATURE DROP RATIO VERSUS POWER REMOVAL RATIO FOR
THREE TYPES OF GRAPHITE SUSPENSIONS
 (L, D, P/H, and ΔT Identical and Constant)

FIG. 9. FILM TEMPERATURE DROP RATIO VERSUS GRAPHITE LOADING ASSUMING THAT THE GAS AND SUSPENSION HEAT TRANSFER CORRELATIONS ARE IDENTICAL
(L, D, P, and H Identical and Constant)

FIG 10 FILM TEMPERATURE DROP RATIO VERSUS POWER REMOVAL RATIO ASSUMING
 THAT THE GAS AND SUSPENSION HEAT TRANSFER CORRELATIONS ARE IDENTICAL
 (L, D, P/H, and ΔT Identical and Constant)

It should be noted that no consideration has been given in this analysis to the possible variations in the efficiency of the fluid circulator with graphite loading, since no information on this factor is available. In principle, though, this factor should be considered. Because of the uncertainties in the constants appearing in the heat transfer and friction factor correlations and because of the other assumptions, such as ignoring the effect of adding solid particles to a gas stream on the viscosity of the mixture, these results can only be regarded as preliminary, and subject to change as better experimental data become available.

An analysis similar to that illustrated here can be performed to compare two coolants on many other bases, but these will be only partial answers since, as stated in the beginning of this section, the true quantitative "quality" of a coolant can only be ascertained in terms of the cost of the power which it helps to produce. Since it is very probable that a reactor optimized for operation with one coolant will be different (in terms of dimensions, power, temperature, pressure, etc.) from one optimized for another coolant, the simple comparisons presented here in which it is postulated that certain quantities are identical in the two systems will not be a valid quantitative comparison. It does, however, indicate a qualitative comparison of the two fluids as reactor coolants.

IV. DISCUSSION

As a result of detailed analysis (Appendix A) of Babcock and Wilcox data for heat transfer from round tubes to a flowing mixture of nitrogen and graphite, it was possible to derive a correlation of the data which is directly comparable to a pure gas correlation. This is presented in Figure 1. The term $(\theta+1)^{-0.4}$ defines the basic difference in film coefficient between the two fluids. The correlation is such that the pure gas case can be considered as a suspension of zero loading.

The Babcock and Wilcox calculations and discussions of the advantages in equipment size presented in Reference (16), Figure 51, are based on the assumption stated on page 50 of the reference that a pure gas correlation is sufficiently correct for suspensions. According to data analyzed in Appendix A a pure gas correlation is not applicable. The recognition of this difference is essential in evaluation of the Babcock and Wilcox discussion of the advantages in equipment size of the addition of graphite. If a pure gas correlation were applicable, the favorable equipment sizes presented in Figure 51 would probably be attainable. The last example of coolant comparison in the previous section indicated that if the correlation expression for the suspension is identical with the conventional expression for a pure gas, a much lower suspension film drop is achieved for equal pumping powers. In contrast, when the actual correlation based upon Appendix A was used, graphite mixtures were about equal to pure gases as coolants, as shown in Figure 6.

The calculations using the latter correlation also showed that a very high film drop would accompany the use of the heat transport capabilities of the suspension. That is, if the coolant temperature rise and the fraction of the output power used for coolant circulation are the same for the suspension as for the purely gaseous coolant, a significant increase in power removal can be achieved, but only at the penalty of a relatively large film temperature drop. A similar trend is depicted in Figures 24 through 27 of Reference (17). However, if the correlation expression for the suspension were identically equal to that for the pure gas, a very large increase in the power removal will entail but a modest increase in film drop, as shown in Figure 10 of this report. Since the beneficial effects of additional heat transfer due to thermal radiation have been shown in Appendix B to be theoretically probable, high temperature experiments may show that the total heat transfer film coefficient (convection plus radiation) is significantly improved over that shown by the relatively low temperature experiments performed to date.

The four most important aspects of the evaluation problem are: (a) definition of the conditions necessary to avoid plating-out of the suspension in thermal gradients over long periods of time; (b) the heat transfer gains associated with thermal radiation at high temperatures; (c) the degree to which convective heat transfer approaches that of a pure fluid; and (d) the pressure drop (friction factor) characteristics of the suspension. If plating-out can be avoided at reasonable Reynolds numbers and if radiation can add substantially to heat transfer at high temperatures, the gas-graphite coolant will enable comparatively high volumetric heat capacity to be achieved at economic pressures. If the combined effects of thermal and convective heat transfer are shown to be nearer to the pure gas correlation than present data indicates, corresponding savings in heat transfer area can also be achieved.

V. CONCLUSIONS

1. Study of gas-graphite coolants should be continued.
2. The most critical problem area is that of maintaining graphite in suspension. This problem may be attributed to several causes, the most serious of which appears to be the influence of a thermal gradient. This is expected to become more serious as reactor exit temperatures are increased.
3. The Babcock and Wilcox conclusion that the use of gas-graphite mixtures will result in smaller heat transfer areas was based on equivalent heat transfer correlations for both the pure gas and the gas-graphite mixtures. On the basis of the data examined the correlations are not the same; heat transfer area for low temperature gas-graphite coolants therefore must be nearly equal to the area required for pure gases. The conclusions of References (16) and (17) regarding heat transfer and pumping power must therefore be considered as tentative until further test experience is evaluated.
4. The correlation expression for heat transfer at high temperature should include the effect of thermal radiation. The probable beneficial effects of thermal radiation from the wall to the graphite is the

main reason for expecting better heat transfer film coefficients from future test programs.

5. Principle advantages probably to be gained by using a gas-graphite coolant are those associated with the low pressure, high volumetric heat capacity. Smaller circulators and relatively thin-walled primary loop and heat exchanger equipment are examples of such gains. Smaller heat exchanger areas will probably be possible due to radiant heat exchange at high temperatures.

6. Study of Babcock and Wilcox data showed that no conclusion can be drawn regarding the pressure drop of nitrogen and graphite flowing isothermally in a plain round tube.

7. No theoretical prediction has been derived in the literature for heat transfer to a flowing gas-graphite mixture having a solid volume fraction of a practical amount.

8. Helium is probably the superior carrier for gas-graphite coolants applied to nuclear power.

9. The use of spherical graphite particles will permit higher graphite loadings than the conventional flaky form used, and will probably minimize the pressure drop resulting from the circulation of gas-graphite mixtures. New graphite processing developments may be required to produce spherical particles with the size required. An alternate possibility is to use spherical carbon particles if they can be shown to be sufficiently non-erosive.

VI. RECOMMENDATIONS

The use of a gas-graphite coolant for nuclear reactor applications has been shown in this study to merit further investigation.

Among the multitude of activities necessary for a final evaluation, a judicious determination of relative priority must be established at the onset of an evaluation program. Furthermore, the governing philosophy for establishing priority will necessarily change as the program progresses. At the present stage all further development depends on one characteristic of the proposed fluid--the solid particles have a tendency

to accumulate on surfaces confining the mixture. Until this tendency can be controlled, all other development may prove to be superfluous. With this in mind, the governing philosophy at this time should be to engage in activities tending to define conditions necessary to maintain a stable suspension for an indefinite period in environments characteristic of proposed service. The scope of this phase of the development warrants a background of theoretical development prior to extensive additional loop testing programs. Many aspects of a complete program may be required to complete this phase. Adequate theoretical and experimental background will probably require most of the equipment and instrument development work contemplated for heat transfer and pressure drop studies. The governing factor, however, throughout all activity must remain the same - the establishment of those conditions necessary to prevent loss of solids from the gas-solids mixture. If it can be shown that a suspension can be maintained for an indefinite period at desirable heat fluxes and at high temperatures, the program can be pursued further with some confidence that the end result will warrant the effort expended.

The governing philosophy of the second phase should be to engage in those activities tending to define the economic importance of operating with gas-solids mixtures as compared to other coolants.

A primary function of these studies will be the development of heat transfer and pressure drop correlations for gas suspensions. The initial program should probably be concerned with comparing a gas suspension to a pure gas in a round heat transfer tube in fully developed flow. This presumes that if a gas suspension does not appear advantageous in the simplest configuration, it will likewise be of little advantage over other fluids in the more complex configurations of conventional heat exchanger designs.

Such work requires the determination, if not done in the first phase, of the physical properties of the heat transfer medium as well as accurate test loop instrumentation. The physical characteristics of the

fluid of interest here are thermal conductivity, specific heat, viscosity, and the transmission and absorption of thermal radiation. Instrumentation is needed to accurately determine the density, mass flow rate, pressure, and temperature of the suspension.

The development of heat transfer and pressure drop correlations should be straightforward, with the exception of decisions concerning parameter ranges for various studies. For example, should the viscosity of a gas-graphite suspension be measured at 2500°F or is it reasonable to use extrapolated values from a range of low temperature data? Over what range of temperature and solids loading should conductivity be determined? Is the effective cross-section for thermal radiation absorption independent of temperature at 2500°F? Where in the program should the effect of particle size be introduced as a parameter? For what conditions should a mass flow meter be tested and against what standard can it be compared? Careful decisions based in many cases on theoretical efforts on these and other questions are important to an efficiently operated research program.

An enumeration of basic and supporting activities in both phases has been set forth in the form of the following Tasks.

1. Fluid Stability	7. Heat Exchanger Development
2. Physical Properties	8. Heat Transfer and Pressure Drop
3. Basic Theory	Correlations
4. Instrument Development	9. Compatibility
5. Turbo-machinery Development	10. High Temperature Development
	11. Coolant Comparison
6. Graphite Development	12. Project Control

Within the above group Tasks 1 through 10 are oriented towards the establishment of both a fundamental technology and experimental observation in numerous areas. Task 10 has been included because of the growing interest in very high temperatures (~2500°F) for process heat applications. This information is then organized within Task 11 to place on a firm basis the comparison of gas-graphite suspensions and other reactor coolants. Task 12 has been included to emphasize the need

for careful definition of scope and close control of each task throughout the program. At the conclusion of these tasks it will be possible to consider the feasibility of designing a specific reactor. Further tests on specific designs will be required prior to the development of a reactor prototype.

Each task has been generally described below. It is not intended that sub-divisions of each task necessarily be performed under one task subcontractor in the sequence listed. Rather, it is recommended that work be distributed whenever and wherever possible on the basis of competence and schedule, in whatever manner is deemed appropriate to the completion of the first two phases described above.

TASK 1. FLUID STABILITY

a. Formulate the theory relating the effect of a thermal gradient on the separation (settling-out) of particles mixed in a gas. Direct a number of experiments leading to correlation expressions for defining conditions necessary to avoid particle deposition due to thermal gradients. Include the effect of particle size and shape, and of volume fraction of the particles.

b. Formulate the theory describing the prerequisites for maintaining a homogeneous gas-particle mixture in isothermal flow in a round tube.

c. Apply the results of a and b to heat exchanger design.

d. Investigate the importance of electrostatic phenomena in test loops and prototypes using gas-graphite suspensions.

TASK 2. PHYSICAL PROPERTIES

a. Viscosity - Measure the effect of particle size, shape and volume fraction of solids on the viscosity of gas-graphite suspensions. Obtain correlation equations and compare to theory.

b. Thermal Conductivity - Measure the thermal conductivity of gas-graphite suspensions. Obtain correlation equations and compare to theory.

c. Thermal Radiation Characteristics - Measure the absorption of thermal radiation in a gas-graphite suspension. Vary particle size, shape, and volume fraction of solids. Develop the theory of the effect of particle size of non-spherical particles on the absorption coefficient for thermal radiation and compare to correlation equations.

d. Density - Measure the density of the graphite in the particles obtained in Task 5. Determine the effect of particle size on particle density.

e. Specific Heat - Prepare analytical or graphical standard values for the specific heat of gas-graphite suspensions for general project use.

TASK 3. BASIC THEORY

a. Derive a theory of heat transfer from a heated circulator duct to a flowing suspension of solid particles in a gas. Include the effect of particle size on absorption coefficient as developed in Task 2. Relate the theory to experimental parameters. Consult on design of experiments and correlation of data.

b. Repeat a for the pressure drop and flow aspects of the investigations.

c. Derive a theory for the viscosity of a gas-graphite mixture.

TASK 4. INSTRUMENT DEVELOPMENT

a. From qualified manufacturing concerns, procure instrumentation suitable for determining the density, the mass flow rate, and the pressure of gas-graphite mixtures. (It is recommended that instrumentation for determining temperature be the responsibility of those conducting the heat transfer tests.)

b. Construct a calibration facility suitable for calibration of instrumentation required for heat transfer and pressure drop experiments on gas-graphite mixtures.

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c. Install instrumentation in above facility for evaluation and calibration over the range of anticipated test loop parameters.

TASK 5. TURBO-MACHINERY DEVELOPMENT

a. Select a hydrodynamic pump configuration most suitable as a circulator for a gas-graphite suspension.

b. Select a smaller pump of similar specific speed suitable for gas-graphite test loop requirements.

c. Supply a modified standard pump selected in b and supply with it a calibrated motor for gas-graphite test loop service and pump tests.

d. Supervise a pump test program at a test loop to determine the effects of graphite loading on the pump efficiency, the characteristic curve of the pump, and its expected life.

TASK 6. GRAPHITE DEVELOPMENT

a. Determine the feasibility of producing small spherical graphite particles. Consider effect of graphitizing furnace black (Ref. (41), p. 20), condensed vaporized graphite, and crushed, previously melted graphite (Ref. (41), pp. 20, 39).

b. Investigate methods of collecting graphite particles of similar size, and shape. Supply samples of such graded particles to the various experiments.

c. Determine the loop circulation characteristics of irradiated graphite. Include studies of particle size stability, equipment erosion, particle agglomeration, and radioactivity of particles exterior to the reactor.

TASK 7. HEAT EXCHANGER DEVELOPMENT

Design and fabricate a gas-graphite to water heat exchanger suitable for a heat transfer test loop. Incorporate design parameters developed in Task 1 to preclude separation of particles from the suspension.

TASK 8. HEAT TRANSFER AND PRESSURE DROP CORRELATIONS

Obtain correlation equations for heat transfer and pressure drop in round tubes for fully developed flow of gas-graphite mixtures. Insofar as is practical, it is desirable for the Atomic Energy Commission to supply information, equipment, and material in the following areas, and to specify the ranges of the various parameters to be investigated.

- a. Pump and motor suitable for pump performance tests for circulation of test fluid.
- b. Instrumentation for the measurement of pressure drop, fluid density, and mass flow rate.
- c. Graphite particles
- d. Physical properties of gas-graphite suspensions.
- e. Heat exchangers, optimized for suspension applications, for maintaining steady state loop temperatures.
- f. Equipment for injecting and removing graphite from the test loop.

TASK 9. COMPATIBILITY

- a. Investigate the feasibility of circulating graphite in available in-pile loops.
- b. Design and procure special equipment required to modify existing in-pile loop facilities for compatibility with gas-graphite suspensions.
- c. Design and procure special equipment required to measure the effects of circulating a gas-graphite suspension through an in-pile loop.
- d. Formulate test procedures for in-pile loop experiments. Include tests for erosion, particle separation, particle agglomeration, stability of particle size and shape, and the radioactivity of the suspension (including its contaminants) exterior to the reactor.

TASK 10. HIGH TEMPERATURE DEVELOPMENT

- a. Determine the probable reactor fuel element configuration for 2500°F outlet fluid temperature.
- b. Determine the feasibility of constructing and operating a 2500°F heat transfer test loop directly applicable to the fuel element configuration selected above.
- c. Determine the feasibility of measuring physical properties at 2500°F.
- d. Determine the feasibility of bench testing the effect of 2500°F on particle separation due to thermal gradients.
- e. Determine the probable heat exchanger configuration for 2500°F reactor outlet fluid temperature in a reactor power installation.
- f. Determine the feasibility of installing a 2500°F heat exchanger in a moderate temperature heat transfer test loop.

TASK 11. COOLANT COMPARISON

Utilize conclusions regarding heat transfer, pressure drop, fluid stability, erosion, radiation effects, instrumentation, fluid circulation equipment, and high temperature operation to compare gas-graphite suspensions to other reactor coolants.

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TASK 12. PROJECT CONTROL

- a. Establish mutually consistent scopes and practical limits to parameter studies in all tasks, in the light of the results of Task 10.
- b. Prepare specifications and evaluate proposals.
- c. Schedule tasks for minimum overall completion time.
- d. Monitor technical competence.
- e. Prepare summary project reports.
- f. Recommend deletion and addition of tasks as work progresses.

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A P P E N D I X A

ANALYSIS OF THE BABCOCK AND WILCOX COMPANY EXPERIMENTAL DATA

THE FRANKLIN INSTITUTE • *Laboratories for Research and Development*

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The Franklin Institute has conducted an independent analysis of a portion of the experimental work by the Babcock and Wilcox Company. These experiments were performed on a specially-constructed loop facility, and had as their primary goal the determination of basic engineering information pertaining to the heat transfer, pumping, and handling properties of this two-phase coolant. Other goals of the investigations included: a study of the compatability of this coolant with other materials, a preliminary investigation of the behavior of the suspension during a long-term application in a nuclear reactor system, and a preliminary evaluation of the net advantages of the solid-gas mixture as a nuclear reactor coolant.

Since accurate information concerning the heat transfer and mass flow properties of this coolant are essential to an economic appraisal of its application as a reactor coolant, a significant portion of the effort in this study was directed towards analyzing the experimental work used to derive this information. The material used in this analysis consisted of: Babcock and Wilcox Reports BAW-1159, BAW-1181, BAW-1194, BAW-1200, BAW-1201, and BAW-1207; copies of the essentially "raw" data obtained from the experiments conducted during the month of July, 1960; copies of the digital computer flow sheet describing the method and form of the data reduction; and copies of digital computer print-out giving values for the various parameters computed. Since the data sheets examined here had also been used for preparing the input data for the digital computer routine, several of the instrumentation readings had been modified in order to eliminate certain readings which were obviously in error. It is not known, however, whether these modifications were necessary because of faulty or insensitive instrumentation, erroneous data-recording, fluctuating readings, or for some other reason. The analysis described herein was restricted to the runs listed in Table I since these were regarded by all concerned as probably being the most reliable, and they contained the significant data consistent with the purposes of the analysis.

Table 1

EXPERIMENTAL RUNS OF INTEREST TO THE ANALYSIS

Run No.	Heater Section Considered	Gas	Turbulence Promoter
3-62	One continuous heater - first half	N ₂	First Half: No Second Half: Yes
3-63	"	"	"
3-64	"	"	"
3-65	"	"	"
3-66	"	"	"
3-67	"	"	"
3-76	A	"	Yes
3-76	B	"	"
3-82	A	He	"
3-82	B	"	"
3-89	A	N ₂	"
3-89	B	"	"
3-92	A	He	"
3-92	B	"	"
3-94	A	N ₂	No
3-94	B	"	"
3-95	A	"	"
3-95	B	"	"

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The specific goals of the experimental work were to obtain accurate correlations for the heat transfer film coefficient and for the friction factor, in terms of conventional dimensionless parameters such as Reynolds number, Nusselt number, and Prandtl number. Basically, the final form of the experimental loop consisted of the following items arranged in series: a heater region composed of three electrical-resistance type heater sections in series (two of these sections had circular flow areas and the third had either a circular or an annular flow area), a cooler region composed of two parallel branches with four coolers (gas-to-water heat exchangers) in each, a "hairpin" loop used to determine the frictional characteristics of the fluid (this was not present in every run), and a pumping system to circulate the fluid through the system. The following sections discuss the experimental procedures used to obtain data for these correlations, and also discuss the work by The Franklin Institute in analyzing the experiments and data. In many cases the information available was not sufficient to enable an accurate estimation of the magnitude of experimental errors. For this reason the tolerance placed on the numerical values for various parameters are somewhat arbitrary, and are intended to indicate only the approximate range of uncertainty in the resulting correlations.

A. Heat Transfer Film Coefficient

The heat transfer portions of the experimental loop were intended to give four different types of information in the form of usable correlations of dimensionless numbers: (a) heat transfer to the fluid in circular channels, (b) heat transfer to the fluid in circular channels containing "turbulence promoters" (a twisted metallic strip along the axis of the channel), (c) heat transfer to the fluid in annular channels, and (d) heat transfer from the fluid in circular channels. Of these four types of information, type (a) is considered to be the most useful at this relatively early stage in the overall research into this new type of coolant since it is more basic and may be readily compared with the extensive work already done using purely gaseous coolants. Type (b) was investigated in

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more restricted detail since it introduces additional parameters in the form of the shape and dimensions of the turbulence promoters. Types (c) and (d) were not investigated because they are more of a "special case" than type (a), and because the experimental set-up and procedures were of poorer quality.

The primary quantities which are required to determine heat transfer film coefficients in the experiments are the heat flux into the fluid, the temperature of the heated tube inner surface, and the temperature of the fluid. In addition, in order to determine a heat transfer correlation, it is necessary to know the loading ratio in the suspension; the mass flow rate, specific heat, thermal conductivity, and viscosity of either the gaseous phase of the suspension or of the suspension itself; and the characteristic dimension (diameter) of the heater. A brief discussion of the work in this analysis pertaining to each of these items is presented below.

a. Heat Flux to the fluid. This quantity is computed from the power input to the fluid and the surface area for heat transfer. The latter quantity is relatively accurately known, but the former contains a high degree of uncertainty for a number of reasons. One of these is the uncertainty in the electrical power input to the heater. Both wattmeter and voltmeter-ammeter instrumentation was provided in most of the runs, but in about 75% of these cases the wattmeter reading exceeded the product of the voltmeter and ammeter readings. Since this condition presents an obvious impossibility, at least one of these pieces of instrumentation is in error; the problem is to determine which and by how much. As an attempt to solve this difficulty recourse was made to investigating the electrical resistivity of the heater. The resistivity was calculated in different ways by using the dimensions of the tube and different combinations of the instrumentation readings, and these various values compared to a standard resistivity versus temperature curve. It was hoped that this method would indicate which of the instrumentation gives the most reliable information. No general correlation of this type could be determined, however, but it

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was possible to select one heater section for each run which gave reasonably consistent results.

Another uncertainty arises from a consideration of heat losses. It is quite possible that appreciable quantities of heat escape from the system by such paths as: conduction through the insulation; conduction in the electrical connection lugs welded to the tubes, with heat being lost by radiation and convection from the lugs and by direct transfer to the water-cooled electrical leads attached to the lugs; and conduction along the piping itself, both at the ends of the heater region ~~and~~ between individual heater sections. Some approximate calculations were made in The Franklin Institute's analysis which showed that the heat losses may not be significant, but it is difficult to draw any definite conclusions in this matter because of the paucity of instrumentation with which to detect losses, and because the mass flow rate - heat balance calculations made as part of this analysis showed, in some of the runs, wide discrepancy between electrical power input and power removed by the fluid.

Due to the difficulties discussed above, the procedure for analysis which evolved in this study consisted of making use of the power input to the fluid, as determined basically by electrical measurements, in at least one of the three heater sections for each run. This power input was then used, with measured fluid temperature at the inlet to and exit from the heater section, in a calculation of the mass flow rate of the suspension based on a heat balance for that heater section. Since the mass flow rate is common to all the heater sections, it is possible to use the value calculated for the one heater (for which the heat input to the fluid is believed to be correct) to compute the heat into the fluid from the other heater sections. In the analysis, the temperature and power readings were modified in various ways which seemed reasonable in order to arrive at the most consistent set of values (for each run) for fluid temperature, power input to the fluid, and mass flow rate.

b. Heater Surface Temperature. The temperature of the heater wall is not constant, but increases in the direction of fluid flow. The

temperature of the outer surface of the wall was measured at five axial stations in each heater section by five sets of three thermocouples located 120° apart. The three readings at each station were averaged, and these values used to plot curves of temperature versus axial distance. In most cases, however, the five points did not determine a smooth curve. It was not possible to determine the specific reasons for the "jaggedness" observed in these plots, but the following possible explanations are suggested: faulty temperature instrumentation, faulty data reading and recording, non-uniform heat losses through the insulation which surrounded the heater, heat losses through the electrical-connection lugs, and some unknown and unanticipated phenomena occurring in the fluid. An attempt was made to correlate the deviations from a smooth plot with location for the various runs. Although some correlation could be observed for some of the runs, the results are inconclusive. For instance, it appears that in runs 62 through 67 the thermocouple readings at positions 6 and 7 have been interchanged. In an attempt to eliminate heat losses through the insulation, shroud heaters were installed around the insulation and instrumented with thermocouples. Sufficient power was applied to these shroud heaters to bring the temperature at the outer surface of the insulation equal to that of the outer wall of the heater, at each axial station.

The temperature of the inner surface of the heater wall was computed from the power input, dimensions, and thermal conductivity of the steel heater tube. In almost every run analyzed, this calculation had a very small effect on the heat transfer film coefficient, so that the uncertainties in the various parameters are, in this case, negligible.

c. Fluid Temperature. Measurements of the temperature of the suspension were made by means of in-stream thermocouples at the entrance to the heater region, the exit from the heaters, the entrance to the first coolers in each of the parallel cooler branches, between the heaters (except for runs 62 through 67 which consisted of one continuous heater tube), between the coolers, and at the exit from the last coolers in each branch.

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A discrepancy was observed between the readings of three thermocouples which, to a first approximation, should agree. These three are those located at the heater output and at the entrance to the first coolers in the parallel cooler branches. It was possible, however, to make a partial correlation of this discrepancy with heater power and suspension flow rate. Generally, when the power and the mass flow rate did not have large values, the temperature of the suspension at the inlet to the coolers was significantly lower than that at the outlet from the heaters. When the power was high, the two sets of temperatures were nearly equal. A possible explanation for this phenomenon is that in the former case the piping is relatively cool and heat is lost from the suspension, while in the latter case heat travels along the pipe from the heater region and therefore essentially no heat is lost from the suspension to the relatively hot piping.

An examination of the data sheets indicates that in virtually every run the temperature at the entrance to one cooler unit was changed so as to agree with that at the entrance to the other unit. These considerations make it difficult to determine a true suspension temperature at the heater outlet which corresponds to the total heat input to the suspension. The modification of these temperatures in order to obtain consistent heat balances was discussed in (a). In general, the modifications to the indicated fluid temperatures were small and usually were within the accuracy of the thermocouples.

In run 95 it appears that an error in the data reduction procedure has occurred. The data sheet omits the reading of the heater inlet temperature, and a value of zero was evidently supplied to the digital computer. Because of this, the value for the density of the gaseous-phase of the suspension was calculated high, and consequently the loading ratio was calculated low. Also, the values for the fluid temperature at each heater station were calculated low, and therefore the heat transfer film coefficients were calculated to be lower than the values which correspond to a more realistic heater inlet temperature.

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As was also done in the case of the wall temperature versus axial distance profile, "most probable", "optimistic", and "pessimistic" fluid temperature versus distance curves were constructed and used in the calculation of the film coefficient.

d. Loading Ratio. The loading ratio (pounds of solids per pound of gas) is computed from the experimentally-measured density, temperature, and pressure of the fluid, and from the density of the solid-phase component (graphite). Various types of instrumentation were attempted by Babcock and Wilcox, but the only device which worked reasonably well was one which measured the absorption of Sr-90 beta particles by the suspension. The Babcock and Wilcox method assumed a simple exponential attenuation when calibrating the device. As part of this analysis, The Franklin Institute has experimentally determined with a Sr-90 source an approximate attenuation curve for the suspension, and have found that the deviations from a simple exponential function may be significant for high loadings. This new conversion curve was employed in analyzing the data, but for many of the runs the difference was small because the loading ratio was small. There may be, however, an appreciable uncertainty in the density readings because of other aspects of the way in which the density meter was calibrated and operated.

The temperature and pressure readings were found to be reasonably accurate (except for run No. 95 noted in (c) above), although some improvements would have been worthwhile. There exists some question as to the actual density of graphite in the form of a fine powder, but even a large uncertainty in this quantity has a small effect on the numerical value for the loading ratio.

e. Mass Flow Rate: As mentioned above in (a), the power input to the fluid, the fluid temperature, and the mass flow rate were all determined by heat balance considerations. Although this method does not give a true value for any of the parameters, it does indicate a set of values which are both reasonable and consistent. In the runs which had

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three individual heaters present, the mass flow rate for each run was computed using four different sets of data (fluid temperatures and power for each heater section individually and these values for the entire heater region as a single unit). The final sets of values yielded a common mass flow rate.

f. Specific Heat. The specific heat of the fluid is calculated from the loading ratio and the specific heats of the constituent materials. No problems were experienced here.

g. Thermal Conductivity and Viscosity. No attempt was made in the experiments to determine values for these parameters. In the course of this analysis, several references to work by others along these lines were found, but it did not appear that sufficient information was available to be useful. Where necessary, it was assumed that the properties of the fluid were equal to those of the gaseous component of the suspension.

h. Tube Diameter. Values for this parameter are given in the Babcock and Wilcox Company reports cited above. No reliable estimate of the accuracy of the figures is given, but the uncertainty is probably small.

The results of the analysis of the runs involving flow in circular channels without turbulence promoters is presented in Table II. This table gives a comparison of the Babcock and Wilcox and The Franklin Institute values for the various experimental parameters which are required to calculate the value of the heat transfer film coefficient and to correlate the data. In general, it can be seen from the table that in most instances the FI values are approximately the same as the B & W values. It must be emphasized, however, that the values obtained in this analysis (and the range of uncertainty on these values) are not necessarily correct in an absolute sense. Rather, they represent the "most probable" values which can be deduced from the available information.

Because of the relatively narrow range of the variables in the runs analyzed, the data will "fit" several different correlations. The correlation equation which appears to be the most useful is

Table II

PLAIN CIRCULAR CHANNEL FLOW HEAT TRANSFER DATA

Parameter (units)	Run No. - Heater								
		62-A	63-A	64-A	65-A	66-A	67-A	94-A&B	95-A&B
Φ_{hi} (lb/ft ²)	B&W	12,990	12,770	13,060	15,510	16,800	19,470	13,800	15,480
	FI	12,990 ±5%	12,770 ±5%	13,060 ±4%	15,510 ±4%	16,800 ±4%	19,470 ±3%	13,800 ±4%	15,480 ±4%
T_{hi} (°F)	C&W	104.7	103.3	106.8	116.9	114.3	169.7	177.3	0
	FI	104 ±4%	103 ±4%	106 ±4%	117 ±4%	114 ±4%	169 ±4%	150 ±20%	150 ±20%
T_{AB} (°F)	B&W	-	-	-	-	-	-	252.9	261.2
	FI	-	-	-	-	-	-	257 ±5%	255 ±5%
T_{EC} (°F)	B&W	-	-	-	-	-	-	352.6	351.3
	FI	-	-	-	-	-	-	357 ±5%	355 ±5%
T_{ho} (°F)	B&W	833	825	802	747	582	547	469.9	459.8
	FI	900 ±2%	912 ±2%	868 ±2%	798 ±2%	687 ±3%	547 ±3%	468 ±4%	460 ±4%
$[\rho_p]_{hi}$ (lb/ft ³)	B&W	?	?	?	?	?	?	?	?
	FI	124 ±20%	124 ±20%	124 ±20%	124 ±20%	124 ±20%	124 ±20%	124 ±20%	124 ±20%
$[\rho_g]_{hi}$ (lb/ft ³)	B&W	?	?	?	?	?	?	?	?
	FI	0.418 ±5%	0.412 ±5%	0.419 ±5%	0.489 ±4%	0.531 ±4%	0.562 ±3%	0.411 ±7%	0.459 ±7%
$[\rho_s]_{hi}$ (lb/ft ³)	B&W	1.0	0.9	0.9	1.0	3.05	6.13	1.85	2.25
	FI	1.0 ±5%	0.9 ±5%	0.9 ±5%	1.0 ±5%	2.92 ±5%	5.42 ±5%	1.83 ±5%	2.20 ±5%
θ (lb _s /lb _g)	B&W	1.396	1.138	1.151	1.059	4.825	10.33	3.748	2.986
	FI	1.40 ±20%	1.20 ±20%	1.15 ±20%	1.06 ±20%	4.61 ±12%	9.04 ±11%	3.51 ±17%	3.87 ±12%
W_s (lb _s /hr)	B&W	174	163	163	220	383	877	1348	1206
	FI	155 ±10%	149 ±10%	153 ±10%	205 ±10%	388 ±10%	913 ±10%	1230 ±10%	1500 ±10%

Table II (Continued)

PLAIN CIRCULAR CHANNEL FLOW HEAT TRANSFER DATA

Parameter (units)		Run No.									
		62-A	63-A	64-A	65-A	66-A	67-A	94-A	94-B	95-A	95-B
T_w (°F)	B&W	747 805	745 795	802 848	726 758	939 963	848 857	652 679	796 860	658 690	815 892
	FI	800 ±30°	820 ±30°	820 ±30°	770 ±30°	970 ±30°	880 ±30°	700 ±30°	870 ±30°	710 ±30°	890 ±30°
T_s (°F)	B&W	294 344	290 340	296 347	287 333	274 316	271 300	251 271	346 364	150 181	288 313
	FI	350 ±30°	390 ±30°	330 ±30°	340 ±30°	320 ±30°	300 ±30°	250 ±15°	350 ±15°	250 ±15°	350 ±15°
δT (°F)	B&W	453 461	455 455	506 501	439 425	665 647	577 557	401 408	450 496	508 509	527 579
	FI	450 ±15%	430 ±15%	490 ±12%	430 ±15%	650 ±9%	580 ±10%	450 ±10%	520 ±9%	460 ±10%	540 ±8%
q (Btu/hr-ft ²)	B&W	56,400 59,100	52,700 49,800	53,300 51,100	64,700 62,000	104,100 99,800	161,200 154,500	57,640 57,640	35,640 35,640	70,570 70,570	42,020 42,020
	FI	54,600 ±10%	51,610 ±10%	50,440 ±10%	60,100 ±10%	99,900 ±10%	150,000 ±10%	52,320 ±15%	34,510 ±15%	62,740 ±15%	41,990 ±15%
h (Btu/hr-ft ² °F)	B&W	118 125	110 116	102 130	146 147	154 156	277 279	144 141	73.3 71.2	139 139	79.7 72.5
	FI	121 ±25%	120 ±25%	103 ±22%	140 ±25%	154 ±19%	259 ±20%	116 ±25%	66.3 ±24%	137 ±25%	77.8 ±23%
D (in.)	B&W	0.313	0.313	0.313	0.313	0.313	0.313	0.532	0.834	0.532	0.834
	FI	0.313 ±2%	0.313 ±2%	0.313 ±2%	0.313 ±2%	0.313 ±2%	0.313 ±2%	0.532 ±2%	0.834 ±1%	0.532 ±2%	0.834 ±1%

NOTE: The two values shown for many of the B&W items are the magnitudes of the respective parameters at the final two stations of the heater indicated.

$$(Nu)_S = 0.0205 (\theta + 1)^{-0.4} (Re)_S^{0.8} \left(\frac{\hat{T}_S}{\hat{T}_S + \frac{ST}{2}} \right)^{0.8} (Pr)_S^{0.4} \quad (1)$$

All the parameters in Eq. (1) are evaluated at the average temperature of the film, $T_S + \frac{ST}{2}$, and the temperatures in the equation are measured relative to absolute zero. It may be noted that Eq. (1) reduces to the correlation given in reference (39) for a pure gas when θ is equal to zero.

Some of the other equations which the data appear to correlate to are:

$$(Nu)_S \left(\theta \frac{C_p}{C_g} + 1 \right)^{0.45} = 0.4 (Pe)_g^{0.5} \quad (2)$$

$$(Nu)_S = (Pe)_S^{0.42} \quad (3)$$

$$(Nu)_S = 0.023 (\theta + 1)^{-0.7} (Re)_S^{0.8} (Pr)_S^{0.4} \quad (4)$$

$$(Nu)_S = 0.015 (\theta + 1)^{-0.4} (Re)_S^{0.8} (Pr)_S^{0.4} \quad (5)$$

The experiments which included the presence of turbulence promoters were investigated in the same manner as the runs without them. The results of the analysis were the following correlations.

For 0.113 inch diameter heater with 4 in. turbulence promoters:

$$(Nu)_S = 0.0168 (Pe)_g^{0.864} \left(\theta \frac{C_p}{C_g} + 1 \right)^{0.575} \quad (6)$$

For 0.834 inch diameter with 6-1/2 in. turbulence promoters:

$$(Nu)_S = 0.0281 (Pe)_g^{0.864} \left(\theta \frac{C_p}{C_g} + 1 \right)^{0.575} \quad (7)$$

For the 0.313 inch diameter heater the results were inconclusive. The data points were clustered closely together so that it was not possible to determine an equation. The points were located very near to the respective points from the Babcock and Wilcox analysis, however, so it might be presumed

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that their equation is reasonable. It should be noted that Eq. (6) is identical with the Babcock and Wilcox equation for the same runs. Equation (7) differs from the corresponding Babcock and Wilcox equation in that the exponent on the Peclet number is smaller in Eq. (7). When plotted, The Franklin Institute correlations are parallel rather than converging as found by Babcock and Wilcox.

B. Friction Factor

The calculation of the friction factor (or frictional pressure loss) has been performed by the Babcock and Wilcox Company for the following cases:

- a. Flow in a special, isothermal "hairpin" loop located in the experimental loop following the coolers,
- b. Flow in the tubular heaters with turbulence promoters, and
- c. Flow in the annular heaters.

In all three of these cases considerable scatter is present in the data. The analysis by The Franklin Institute has concentrated on case (a) for two reasons: this type of flow is the most basic and is readily comparable with the corresponding data for pure fluids, and the results obtained by Babcock and Wilcox show that, for large Reynolds numbers, the friction factor is less than that for pure fluids.

The analysis performed by The Franklin Institute consisted of two parts: "parameter sensitivity evaluation", which showed that the magnitude of the friction factor is particularly sensitive to the values of (and hence errors in) the measured pressure drop, the pressure level at the exit from the hairpin loop, the mass flow rate, and the loading ratio; and a recalculation of the friction factor for several of the runs using the results of the analysis discussed in Section A for the mass flow rate and the loading ratio. This recalculation also indicated that the friction factor is less than that for a pure fluid flowing in a smooth tube at the same Reynolds number. The data are still too scattered and subject to error to be useful in deducing a correlation equation.

One factor which is not included in the analysis is the true viscosity of the fluid. As mentioned previously, the value for the gaseous component of the suspension was used due to the lack of better information.

Working with aqueous slurries of thorium oxide particles, the Oak Ridge National Laboratory (20) has noted a similar trend but to a smaller degree and at lower Reynolds numbers. Evidently certain suspension-type flows exhibit a non-Newtonian behavior, and two consequences of this are smaller friction factors over some range of Reynolds numbers and an extension of the laminar flow range into high Reynolds numbers than for perfect fluids. If particles in the boundary layer tend to move out due to Bernoulli forces, the viscosity of the layer may be less than the average of the flow. This would tend to decrease the friction factor. At this stage, the experimental results pertaining to friction factor must be considered inconclusive.

It should be noted that the magnitude and direction of a heat flux across the fluid boundary layer may significantly affect the friction factor and heat transfer film coefficient. The main body of this report mentions the influence of heat flux on particle concentration in the boundary layer, and it is the particle concentration which apparently influences the heat transfer and frictional flow characteristics of the gas-solids suspension. Therefore, these characteristics may be significantly influenced by whether the suspension is being heated or cooled. Rosenblatt and LaMer*, for example, discuss the effect of a thermal gradient on the motion of small suspended droplets in a stationary gas. Additional theoretical background is presented by Kennard**.

*Philip Rosenblatt and Victor K. LaMer, "Motion of a Particle in a Temperature Gradient; Thermal Repulsion as a Radiometer Phenomenon", Physical Review, Vol. 70, Nos. 5 and 6, September, 1946.

**Kennard, "Kinetic Theory of Gases", McGraw-Hill, 1938.

A P P E N D I X B

ANALYSIS OF THERMAL RADIATION

F-A2392-02

The Babcock and Wilcox studies of heat transfer to suspension of graphite particles in gases yielded data which were correlated by Babcock and Wilcox by means of groups of variables that did not involve radiation. Inasmuch as the radiation from a black surface depends upon T^4 (where T is measured on the absolute scale), whereas ordinary conduction is approximately linear in T , radiation may be expected to be important at high enough temperatures. It is of interest to discuss:

- (a) Whether the Babcock and Wilcox measurements involved high enough temperatures so that radiation effects should have been detected, and hence included in their correlations, and
- (b) Whether thermal radiation effects may be important in practical applications of graphite suspensions.

A. Black Spherical Particles Of One Size

For present purposes, it is sufficient to consider a one-dimensional theory of the propagation of radiation through a suspension of black spherical particles of uniform size.

The cross-sectional area of a particle of radius r is:

$$A = \pi r^2 \quad (1)$$

If there are N such particles per unit volume, consider parallel rays of infra-red radiation crossing a unit area. Upon traveling a small distance, dx , the rays will encounter $N dx$ particles, having a total cross-sectional area $NA dx$. The radiation intensity, I , therefore, obeys the differential equation.

$$dI = -NAI dx \quad (2)$$

which has the solution

$$I = I_0 e^{-x/\zeta} \quad (3)$$

where I_0 is the initial intensity, and ζ is the mean free path for the radiation given by

$$\zeta = 1/(NA) \quad (4)$$

$$= \frac{1}{\pi r^2 N}$$

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In deriving this result, the assumption is made that the intensity is reduced by the same factor in each distance dx . This assumption is valid provided the particles are randomly distributed, and provided also $N dx \gg 1$ for some $dx \ll \zeta$. This is equivalent to $A \ll 1$. Since we are concerned with very small randomly distributed particles, the conditions are satisfied. A further assumption is that geometrical optics are obeyed; this is true if $r \gg \lambda$, where λ is the wavelength of the radiation.

1. One-Dimensional Theory of Effect on Thermal Conductivity

Assume that

$$\hat{T} = \hat{T}_0 + \tau x \quad (5)$$

in a medium that absorbs but does not scatter radiation. Assume that ζ is independent of wavelength. The emissivity of a layer of small thickness dx is equal to its absorptivity, and is given by

$$1 - e^{-\frac{dx}{\zeta}} = \frac{dx}{\zeta} \quad (6)$$

the radiation (heat) flux arriving at $x = 0$ from a layer at $x = -x_1$ is then

$$\left(\sigma \hat{T}^4 \frac{dx}{\zeta} \right) \left(e^{-x/\zeta} \right) \quad \text{where} \quad T = T_0 - \tau x_1 \quad (7)$$

so that the total flux arriving at $x = 0$ from the left is

$$\begin{aligned} q_L &= \int_0^\infty \sigma (\hat{T}_0 - \tau x)^4 \frac{dx}{\zeta} e^{-x/\zeta} \\ &= \sigma \left[\hat{T}_0^4 - 4\hat{T}_0^3 \tau \zeta + 12 \hat{T}_0^2 (\tau \zeta)^2 - 24 \hat{T}_0 (\tau \zeta)^3 + 24 (\tau \zeta)^4 \right] \end{aligned} \quad (8)$$

where σ is the Stefan-Boltzmann constant. A similarly derived expression for the flux arriving from the right, q_R , is identical to this except that all the terms are positive. The net flux from right to left is

$$q = q_R - q_L = \sigma \left[8\hat{T}_0^3 \tau \zeta + 48 \hat{T}_0 (\tau \zeta)^3 \right] \quad (9)$$

For sufficiently small τ , we have $\tau \zeta \ll \hat{T}_0$, and the second term may be neglected in equation (9). Dividing by τ , we find the contribution of radiation to the thermal conductivity is

$$k_{rr} = 8 \sigma \hat{T}_0^3 \zeta \quad (10)$$

if the ordinary conductivity due to molecular motion is k_{cd} , the total effective thermal conductivity is

$$k = k_{cd} + k_{rd} \quad (11)$$

The numerical coefficient, 8, arises from the one-dimensional treatment. Taking all directions of radiation into account would lead to a somewhat different numerical value. Since the derivation involves an integration from $x = 0$ to infinity, it must be assumed that the uniform temperature gradient, τ , extends over a region several times larger than ζ , and that no solid boundaries exist within a distance large compared to ζ .

2. Effect of Direct Radiation from Wall

To estimate the order of magnitude of the effect of direct radiation from the wall, consider a relatively cool suspension, between two parallel hot walls at absolute temperature $\hat{T}$. The walls are opaque, and have absorptivity γ and reflectivity $(1-\gamma)$. We assume that all radiation travels perpendicular to the walls, which are separated by distance D .

If the radiation arriving per unit area at the first wall is q_1 , then the reflected radiation is $(1-\gamma) q_1$, so that the total radiation leaving the first wall is $\gamma \sigma \hat{T}^4 + (1-\gamma) q_1$. This is attenuated by the factor $e^{-D/\zeta}$ by the time it reaches the second wall. By symmetry, the radiation arriving at the second wall, q_2 , is equal to q_1 . Therefore

$$q_2 = q_1 = e^{-D/\zeta} \left[\gamma \sigma \hat{T}^4 + (1-\gamma) q_1 \right] \quad (12)$$

Solving for q_1 , we find

$$q_1 = \frac{\gamma \sigma \hat{T}^4}{e^{D/\zeta} - 1 + \gamma} \quad (13)$$

The radiation leaving the first wall is $e^{D/\zeta} q_1$. The part of this radiation that is absorbed is $(e^{D/\zeta} q_1 - q_1)$. An equal amount is absorbed from the other wall. The mean rate at which energy is absorbed by the suspension per unit volume is found by dividing the sum of D:

$$\bar{Q} = \frac{2}{D} \frac{(e^{D/\zeta} - 1) \gamma \sigma \hat{T}^4}{e^{D/L} - 1 + \gamma} \quad (14)$$

If $\zeta \gg D$, or if $\gamma=0$, then $\bar{Q}=0$, as is physically reasonable, since these are cases of either no absorption by the medium, or no emission by the walls.

If the walls are black, so $\gamma = 1$, then

$$\bar{Q}_B = \frac{2}{D} (1 - e^{-D/\zeta}) \sigma \hat{T}^4 \quad (15)$$

Equations (14) and (15) neglect radiation from one region in the suspension to another and from the suspension back to the walls. Thus, the suspension is assumed to be at a much lower temperature than the wall temperature $\hat{T}$ even at distances much less than ζ from the wall. In other words, if d is the boundary layer thickness, equations (14) and (15) assume

$$d \ll \zeta \quad (16)$$

If $\zeta \ll d$, the heat flux will not be enhanced by $\bar{Q}$, but instead will be enhanced by the effect of the additional conductivity, k_{rd} .

Generally speaking, these results may be summarized as follows: If the wall temperature is $\hat{T}_o$, and the suspension at distance ζ from the wall is at temperature $\hat{T}_s$, then the radiant flux per unit area of black wall is of the order

$$\sigma (\hat{T}_o^4 - \hat{T}_s^4) \quad (17)$$

B. Disk-Like Particles

The particles which are presently available for experimental purposes are not spherical, but rather they are flat disks or "flakes". We now consider the value of ζ for flat particles, assuming as before that the particles are black and that the laws of geometrical optics are obeyed.

Assume that the particles are uniformly and randomly distributed, with N particles per unit volume; that they have area A , thickness t , and density ρ_p ; and are randomly oriented. Random orientation is defined as follows: consider a volume element large enough to contain a large number of particles, and draw a sphere of radius very much greater than the maximum dimension of the volume element, the center of the sphere being inside the volume element. Construct the surface normals of the particles, extending them so that they intersect the surface of the sphere. Random orientation means that the number of intersections per unit area of spherical surface is the same everywhere on the sphere.

We imagine labeling the sides of the particles as "side 1" and "side 2" and construct only the outward surface normal on "side 1" of each particle. If the volume element has volume v , it will contain Nv particles. Half of these will have surface normals making an angle $\phi \leq 90$ degrees with the incoming parallel rays of infra-red radiation and the other half will have surface normals for which $\phi \geq 90$ degrees. Each group of particles will account for half the total projected area on a plane perpendicular to the light rays. We consider the group for which $\phi \leq 90$ degrees.

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For a sphere of radius R , the strip for which the angle between the light rays and the particle surface normals is in the range ϕ , $\phi + d\phi$ is of width $Rd\phi$ and circumference $2\pi R \sin \phi$ and so has area $2\pi R^2 \sin \phi d\phi$. Since the hemisphere has area $2\pi R^2$, the number of particles whose surface normals intersect the strip is

$$\frac{Nv}{2} \frac{2\pi R^2 \sin \phi d\phi}{2\pi R^2} = \frac{Nv}{2} \sin \phi d\phi$$

Each of these particles has projected area $A' = A \cos \phi$. The total projected area for this group of particles in the hemisphere is

$$\int_0^{\pi/2} A \cos \phi \frac{Nv}{2} \sin \phi d\phi = \frac{ANv}{4}$$

If we now include the particles for which $\phi > 90$ degrees, the total projected area is $ANv/2$. The average projected area $\overline{A'}$ is then found by dividing by Nv , so that

$$\overline{A'} = A/2 \quad (18)$$

Let ρ_p equal the mass of graphite per unit volume of suspension.

Then

$$N \rho_p A t = \rho_p \star \quad (19)$$

and the mean free path for the radiation is

$$\begin{aligned} \zeta &= 1/\overline{NA'} = 2/NA \\ \zeta &= 2 \rho_p t / \rho_p \star \end{aligned} \quad (20)$$

C. Wavelength Effects

Stull and Plass¹ have investigated the emissivity of dispersed carbon particles theoretically, calculating the emissivity of spherical carbon particles in the visible and infrared regions of the spectrum.

The total cross-section is the sum of the absorption cross-section and

¹V.R. Stull and G.N. Plass, Jour. Opt. Soc. Am., Vol. 50, No. 2, Feb. 1960, page 121.

scattering cross-section. Stull and Plass found that the emissivity calculated with the scattering cross-section taken into account differed only very slightly from the emissivity calculated without taking scattering into account. We may, therefore, take their calculated values for total cross-section as accurate values for absorption cross-section.

The wavelength effect is such that for wavelengths appreciably larger than the radius of the particle, the cross-section decreases with increasing wavelength. In order to apply their results approximately to the particles of interest to us, we assume that the wavelength effect is the same as that for spherical particles having the same geometrical cross-section. If A is the average area of the particles, then the effective radius, r , of the equivalent spherical particle is found with the help of equation (18):

$$\pi r^2 = \overline{A} = A/2,$$

$$r = \sqrt{A/2\pi}$$

Babcock and Wilcox gives 0.7 microns as an average flake diameter, so that

$$\sqrt{A} = \frac{\sqrt{\pi}}{2} 0.7 \times 10^{-4} \text{ cm} = 0.62 \times 10^{-4} \text{ cm}$$

and

$$r = 0.25 \times 10^{-4} \text{ cm} = 2500 \text{ \AA}$$

The largest radius that Stull and Plass used in their calculations was 987 $\text{\AA}$. In their Figure 3, σ_t is plotted on a logarithmic scale versus wavelength. The curves are very nearly parallel for wavelengths between 2 and 20 microns. Estimating $\log (\sigma_t \times 10^{16})$ at wavelength = 20 microns from their Figure 3, we construct the following table:

$r(\text{\AA})$	$\log r$	$\log (10^{16} \sigma_t)$ (σ_t in cm^2)	$\frac{4.8 + \log (10^{16} \sigma_t)}{\log r}$
47.8	1.68	0.3	3.04
159	2.20	1.9	3.04
446	2.65	3.2	3.02
700	2.84	3.8	3.02
987	2.99	4.3	3.04

The constant, 4.8, in the numerator in the fourth column was selected to make the first and last entries equal. Thus.

$$4.8 + \log (10^{16} \sigma_t) = 3.04 \log r \quad (21)$$

therefore

$$\sigma_t = \text{const} \times r^{3.04} \quad (22)$$

No physical significance is here attached to the value of the exponent in equation (22). However, we may use this equation as an extrapolation formula, in order to obtain approximate values of σ_t for $r = 2500\text{\AA}$.

Thus, if $\sigma_{2500\text{\AA}}$ stands for σ_t for wavelength = 2500 $\text{\AA}$, etc.,

$$\sigma_{2500\text{\AA}} = \left(\frac{2500}{987} \right)^{3.04} \sigma_{987\text{\AA}}, \text{ approximately, or} \\ \sigma_{2500\text{\AA}} = 16.8 \sigma_{987\text{\AA}} \quad (23)$$

For a black body at a temperature of 1500°F, the following table shows the approximate wavelength in microns for which a given percentage of the total radiation is at shorter wavelengths:

<u>Percentage</u>	<u>Wavelength</u>	<u>Percentage</u>	<u>Wavelength</u>
99	21	30	2.9
98	16.3	20	2.47
90	8.7	10	2.02
80	6.3	5	1.74
70	5.15	1	1.34
60	4.4		
50	3.8		
40	3.3		

We may use this table to estimate the mean free path for radiation as follows: suppose F_1 is the fraction of the energy from the black body for which the mean free path is ζ_1 . For temperature gradient τ , the flux due to this fraction of the energy will be, according to equation (10):

$$q_1 = F_1 8\sigma_o^{\wedge} T_o^3 \tau \zeta_1 \quad (24)$$

The total flux due to radiation will be

$$q = \sum_i \dot{q}_i = 8\sigma T_o^3 \tau \sum_i (F_i \zeta_i) \quad (25)$$

Comparing with equation (10) we find the effective mean free path to be

$$\zeta = \sum_i (F_i \zeta_i) \quad (26)$$

This is only an approximation because it is assumed that the spectral energy distribution of the black body radiation is unchanged as it passes through the medium. However, we may use it to estimate an effective value.

For N particles per unit volume,

$$\zeta_i = 1/(N\sigma_{ti}) \quad (27)$$

so that

$$\zeta = \frac{1}{N} \sum_i (F_i / \sigma_{ti}) \quad (28)$$

whence

$$\sigma_t = \frac{1}{\sum_i (F_i / \sigma_{ti})} \quad (29)$$

where F_i is the fraction of the energy in the narrow wavelength range number i , and σ_{ti} is the total cross-section for a typical wavelength within that range.

Using the last table given above, equation (23), and Stull and Plass's Figure 3, we perform the approximate numerical integration shown in the following table:

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F_i	Wavelength (microns)	$10^{12} \sigma_{ti}$ for $r = 987\text{\AA}$ (cm^2)	$10^{12} \sigma_{ti}$ for $r = 2500\text{\AA}$ (cm^2)	$10^4 F_i / 10^{12} \sigma_{ti}$
.02	20	2	33.6	5.95
.08	12	6	101	7.90
.10	7	15	252	3.96
.10	6	18	302	3.32
.10	5	20	336	2.98
.10	4	40	672	1.49
.10	3.5	50	830	1.21
.10	3	70	1180	0.85
.10	2.7	80	1350	0.74
.10	2.2	100	1680	0.60
.05	1.9	150	2520	0.40
.05	1	400	6720	0.15
				Sum = 29.55

$$10^{12} \sigma_t = 10^4 / 29.55 = 340 \text{ cm}^2 \quad (30)$$

$$\sigma_t = 340 \times 10^{-12} \text{ cm}^2$$

Using $r = 2500 \times 10^{-8} \text{ cm}$, we find

$$\frac{\pi r^2}{\sigma_t} = 5.8 \quad (31)$$

Consequently, the wavelength effect is such that the effective mean free path for thermal radiation is of the order of 5.8 times as large as its "geometrical optics" value.

D. Numerical Examples from the Babcock and Wilcox Data

Using the values

$$\rho_p = 106 \text{ lb/ft}^3, t = 0.03 \text{ microns},$$

equation (20) leads to

$$\zeta_{\text{geom}} = \frac{6.36}{\rho_p^*} \text{ microns}, \quad (32)$$

where ρ_p^* is in lb/ft^3 .

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We now consider whether, in the Babcock and Wilcox experiments, radiant heat transfer should have been a large enough fraction of the total heat transfer to have been detected and included in the empirical correlations. The fraction transferred by thermal radiation will generally be larger for higher wall temperatures and for smaller convective film coefficients. In view of the effect of graphite loading in enhancing the convective film coefficient, the experiments in which radiation has the best chance of being important are those which $\rho_p^\star$ is small. The examples selected are therefore not typical, but instead are chosen specifically because of the possibility of appreciable radiant heat transfer. The calculations are based on the assumption that the pipe wall is black.

If $\rho_p^\star$ and $\rho_g^\star$ are the masses of solid and gas per unit volume of suspension, respectively, then $\rho_p^\star/\rho_g^\star = \theta$ is the mass of solid per unit mass of gas. Furthermore, $\rho_p^\star + \rho_g^\star$ is the density of the suspension or fluid mixture, ρ_s .

$$\rho_g^\star = \frac{\rho_p^\star}{\theta} \quad (33)$$

so that

$$\rho_s = \rho_p^\star + \rho_g^\star = \rho_p^\star \left(1 + \frac{1}{\theta}\right) = \rho_p^\star \left(\frac{1 + \theta}{\theta}\right) \quad (34)$$

Finally,

$$\rho_p^\star = \frac{\rho_s \theta}{1 + \theta} \quad (35)$$

For small $\rho_p^\star$, we may take the thermal conductivity of the suspension to equal that of the gas. We also need the following relationships:

$$T_w = T_s + \frac{q}{h} \quad (36)$$

$$d = \text{boundary layer thickness} = \frac{k}{h} \quad (37)$$

where k is thermal conductivity and h is heat transfer film coefficient. The examples are listed in the following table:

Example No.	<u>1</u>	<u>2</u>	<u>3</u>	<u>4</u>
Run No.-Station No.	62-13	69-15	71-15	80-15
θ	1.4	0.304	0.355	0.328
ρ_s (lb/ft ³)	0.39	0.18	0.22	0.17
$T_s - \hat{T}_s$ (°F-°R)	716-1176	669.6-1130	707.8-1168	706.4-1166
q (BTU/hr-ft ²)	5.72×10^4	1.12×10^4	2.45×10^4	1.29×10^4
h (BTU/ft ² -hr-°F)	194.7	28.0	74.9	46.1
k (BTU/ft-hr-°F)	0.026	0.026	0.026	0.026
$\hat{T}_w$ (°R)	1470	1530	1495	1446
d (microns)	40	280	104	170
ρ_p^* (lb/ft ³)	0.228	0.042	0.0576	0.042
$\zeta = 5.8 \times 6.36/\rho_p^*$ (microns)	162	880	640	879
D (in. - microns)	0.313-8000	0.121-3100 [†]	0.121-3100 [†]	0.121-3100 [†]
$\sigma(\hat{T}_s^4 - \hat{T}_s^4)$, $\sigma = 1.712 \times 10^{-9}$ (BTU/ft ² -hr-°R ⁴)	4700	6500	5360	5500
Radiant Flux/Total Flux	0.08	0.58	0.22	0.42

[†]Note: In this case D refers to the annular gap width

In each of these examples, $D \gg \zeta > d$, so that the calculation of radiant flux is reasonably accurate. Since an error by a factor of two in ζ will not alter this inequality, the method of estimating ζ is adequate to support the following conclusion:

Under the conditions available in some of the Babcock and Wilcox experiments, radiant heat transfer from black walls should have constituted a large enough fraction of the total heat transfer to affect some of the empirical correlations. However, it should be noted that the data for the examples on the previous page were all from heater section C, which was not considered in the data analysis described in Appendix A.

With the possibility of much higher temperatures being considered, the relative effect of radiant heat transfer becomes even more important. For example, if $\hat{T}_w = 2500^\circ\text{R}$, the radiant heat flux will be approximately 4.6 times as much as it is when $\hat{T}_w = 1500^\circ\text{R}$, for the same temperature difference, $T_w - T_s$.

The dependence of total cross-section on particle size may not be the same for flat discs as for spheres, and the use of an equivalent spherical particle to represent a flat disc is only an approximation. Further theoretical study may support or undermine this approximation. For the present, however, we may point out that in the case of spherical particles of uniform size, ζ appears to depend mainly on $\rho_p^\star$, and not very much on the size of the particles.

To see the reason for this, we note that an inspection of the numerical integration given above for estimating $1/\sigma_t$ shows that the main contribution comes from terms corresponding to wavelengths longer than about 3 microns, where equation (22) still holds. In equation (22), σ_t is the total cross-section at 20 microns. This equation holds approximately at all wavelengths substantially greater than the particle radius, and consequently continues to hold for the total cross-section for black body radiation, as found by the numerical integration.

It follows that $1/\zeta = N\sigma_t$ is approximately proportional to $N \times \frac{4}{3} \pi r^3 = \rho_p^\star$. This conclusion arises from the effect of wavelength on σ_t .

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(If there were no wavelength effect, so that σ_t were simply the geometrical value, πr^2 , we would find $1/\zeta$ proportional to ρ_p^*/r .)

This result suggests that for small enough particles of any given shape, ζ depends mainly on ρ_p^* , and not on particle size. The particles used by Babcock and Wilcox were small enough for this tentative conclusion to apply.

E. Conclusions

For the conditions of the Babcock and Wilcox experiments, thermal radiation should have contributed a large fraction of the heat transfer under some of the experimental conditions. Consequently, the empirical correlations should have included some terms or factors taking radiation into account.

This conclusion is based on the premise that the mean free path for radiation is small compared to the pipe diameter and large compared to the boundary layer thickness. The estimate of the mean free path is rather crude, being an extrapolated application to flat particles of theoretical results obtained for much smaller spherical particles.

It is also assumed that the pipe wall has an emissivity close to unity. This is a reasonable assumption because the pipe wall is likely to be coated with a layer of graphite in intimate thermal contact. (It should be recognized, however, that if no such layer is formed, the infrared emissivity of a polished metal surface may be of the order of 0.1 to 0.5.) The importance of thermal radiation will be much greater at the higher temperatures under consideration for reactors.

The relation between total cross-section and wavelength for small spherical particles suggests the tentative conclusion that the mass of graphite per unit volume is the main factor in determining the mean free path; the size and number of particles do not have much effect as separate factors, provided the particles are smaller than the wavelengths in which most of the radiation is transported. Figure B1 shows the general behavior

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tentatively concluded for the relation between radiation mean free path and particle radius. For small r , ζ is approximately independent of r , for a given value of r_p^* . For large r , ζ is proportional to r . The exact extent of the transition region and the dependence of ζ on r in the transition region have not been estimated. The effective radius, 0.25 microns, estimated for the particles used by Babcock and Wilcox is certainly in the "small r " range, however, in which ζ is independent of r . On the other hand, the basis for estimating the effective radius may be open to question.

F. Additional Studies Recommended

The tentative nature of the conclusions arises from a lack of specific knowledge, which may be supplied by theoretical and experimental studies such as the following:

1. Extension of the theoretical work of Stull and Plass to particles of the size and shape of interest. The required information is the scattering and total cross-sections, as functions of wavelength, from which absorption calculations may be made.
2. Measurement of infrared transmissivity of samples of suspensions, as a function of wavelength. Transmissivity is not likely to depend very much on temperature, so that the measurements can be made at room temperature with a small set-up, using standard infrared spectrometer equipment. A great deal of useful information would be obtained, such as the actual dependence on cross-section on particle size.
3. Thermal conductivity should be measured at elevated temperatures, to verify the theoretical prediction of an additional term.
4. The emissivities of solid surfaces that are likely to be possible pipe wall surfaces should be measured under temperature and suspension environmental conditions simulating possible conditions of use, in order to settle any question as to the assumption that the emissivity is nearly unity. The importance of radiation at elevated temperatures is great enough to justify efforts to ensure that the emissivity is in fact high. One should not be satisfied with the plausibility

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that a black coating will form, unless one knows enough about the forces of adhesion to be sure that they will operate as expected, under unusual chemical and temperature conditions.

The foregoing recommendations relate specifically to thermal radiation, and are in addition to other recommendations concerning other aspects of graphite suspensions.

FIG. B-1. $\zeta \frac{\rho_p^*}{\rho_p}$ VERSUS PARTICLE RADIUS

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