NATURAL CONVECTION BETWEEN A VERTICAL CYLINDER AND A SURROUNDING ARRAY

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Received by OSTI

AUG 24 1992

ABSTRACT The generic situation considered is natural convection between a single heated, vertical cylinder and a surrounding array of cooler vertical cylinders in a triangular pattern. The ratio of the test section temperature to the cooling tube temperature was varied up to 2.6 by adjusting the electrical power. The Rayleigh number, based on test section diameter and air properties evaluated at cooling tube temperature, ranged from 2.9×10^4 to 4.6×10^5 . Results indicate that the convective heat transfer data could be approximated as $Nu_D (T_b/T_c)^{0.14} = 0.156 Ra_D^{1/3}$ in the apparent turbulent region for $Ra_D > 1.2 \times 10^{11}$.

I. INTRODUCTION

A number of nuclear reactors utilize vertical circular tubes in arrays to serve as components of the reactor core. The tubes are used for a variety of purposes, such as coolant tubes, fuel rods, control rods, safety rods, etc. In the case of water reactors the core may be partially filled with air or another gas in the event of a loss-of-coolant or loss-of-flow incident. The thermal energy evolving in the tubes must then be transferred by a combination of natural convection, thermal radiation and thermal conduction. Data for natural convection in this complex geometry are sparse or non-existent.

The generic situation considered in the present study is turbulent natural convection between a single heated, slender, vertical circular cylinder and a surrounding array of cooler vertical cylinders. This case describes the circumstance when, for one reason or another, one cylindrical component is at a higher temperature than the surrounding components. For example, a control rod or safety rod may be heated by absorption of an incident gamma flux after a reactor is shut down, while water flow through cooling tubes holds the surroundings to a lower temperature.

For a typical nuclear reactor the length-to-diameter ratios of individual tubes fall in the range of 30 to 300; they tend to be slender tubes. If the temperature range is taken to be from room temperature to the melting point of aluminum, the length Rayleigh number - the governing natural convective parameter - will be greater than 10^{10} so that the induced flow can be expected to be mostly turbulent above a half meter.

It is evident that the generic configuration cannot reasonably be considered to be an isolated single tube in infinite surroundings. Essentially, the cylinder and its surrounding array form an enclosure, possibly described as annular, with an irregular and/or extended outer surface consisting of the other tubes.

Directly-related previous work is limited. A number of useful surveys and texts have been published on the general topic of natural convection in the past few decades [Ostrach, 1958; Gebhart, 1971; Raithby and Hollands, 1985; etc.]. Flow over cylinders has been summarized by Morgan [1975] and natural convection in enclosures has been emphasized by

Catton [1978]. The reader is referred to these summaries for a general background on the topic.

A search of the literature, including examination of the reviews mentioned above, revealed no work on the present topic: turbulent natural convection from a vertical cylinder to a surrounding array. No published data were found for turbulent natural convection in an annular enclosure. Almost no results were even available for turbulent natural convection from isolated slender vertical cylinders of aspect ratios pertinent to the present study. Morgan [1975] reports data by Carne [1937] and Eigenson [1940]; Steimke [1991] measured convective parameters at temperatures up to about 250C for an isolated tube in large surroundings.

Lacking turbulent data for the geometry considered, the designer is forced to extrapolate from information for an isolated cylinder or for a rectangular enclosure. Such extensions must be tested by basic experiments before they can be considered reasonable for design application.

The general goal of this research is to provide data on natural convection, in a geometric arrangement comparable to actual nuclear reactors, so that maximum expected temperatures can be predicted reliably over a practical temperature range for individual, cylindrical components surrounded by gas without forced flow.

In order to accomplish this goal, the specific objectives of the experiment were defined to be (1) obtain temperature measurements in geometries simulating the cores of actual reactors at realistic temperature levels, (2) in order to separate natural convective contributions from thermal radiation, measure thermal emissivities of the surfaces involved before and during the experiment, (3) document the experimental configuration and the resulting data and (4) deduce the natural convection behavior as functions of appropriate governing parameters and compare to predictions available.

II. APPARATUS

The experimental configuration was chosen to approximate a vertical cylindrical assembly, which might represent a safety rod or a control rod, and three surrounding cooling tubes or fuel assemblies in near full scale. Figure 1

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Fig. 1. Schematic diagram of experiment

schematically describes the overall apparatus and Figure 2 is a scale drawing of a cross section of the test assembly.

Atmospheric air serves as the primary working fluid, representing the gas in the reactor. Applied thermal boundary conditions approximated uniform wall heat flux for the test section and uniform wall temperature for the cooling tubes. The lower end plate was designed to provide a constant temperature, which might correspond to a gas-liquid interface, at the same temperature as the cooling tubes. The upper end was insulated to approach an adiabatic boundary condition. This apparatus was surrounded radially by an acrylic shroud to serve as a convection shield, avoiding drafts in the bay where it was mounted. The dominant natural convective path was from the test section to the nearby cooling tubes.

As shown in Figure 1, the test section is heated electrically. Overall height of the assembly is about six meters. Direct current is employed and shunts measure the current flow. A voltmeter measures the voltage drop from electrode to electrode in order to check the test section electrical resistance and the total power generation.

Three aluminum cooling tubes surround the test section to remove the thermal energy from the test apparatus so steady state conditions may be approached. Water flows upward through the cooling tubes at flow rates sufficient to keep these tubes nearly isothermal at the water inlet temperature, which is near room temperature.

The surrounding air temperature is measured at three locations by thermocouples suspended in the air between the cooling tubes, the test section, and the surrounding acrylic

Fig. 2. Scale drawing of enclosure formed by test assembly.

(Plexiglas) shroud. Additional thermocouples affixed on the cooling tubes and the inside of the shroud measure their surface temperatures. Small openings in the lower end plate maintain the air pressure inside the enclosure at the local atmospheric pressure which is approximately 0.84 atm at the elevation of the experiment.

The test section is fabricated from seamless stainless steel tubing, type 304, per ASTM 269. The outside diameter of the test section was measured in two directions along its length as the apparatus was assembled vertically. The range was 0.944 to 0.946 inches with 0.9443 inch (2.399 cm) taken as the average. Copper electrodes of approximately the same diameter are brazed at each end. Heated length is about 160 diameters. The test section is supported at the top and allowed to expand freely through the bottom end plate, which serves as a guide.

Inner wall temperatures were measured with standard grade, sheathed Type K (Chromel/Alumel) thermocouples inside the tube at about thirty cm intervals. Near the copper electrodes at the ends of the test section, the spacing is closer. These thermocouples are mounted in an Aremcolox ceramic assembly which fills the test section.

To simulate the cooling of fuel assemblies in a reactor and to cool the apparatus, three cooling tubes surround the test section in a triangular pattern. They are fabricated from Aluminum Alloy 6061-T6. Dimensions are those of 3-1/2 in., schedule 40 pipe with the outside diameter varying from 3.945 to 4.00 in. (10.02 to 10.2 cm). They are located on a circle of 20.5 cm diameter.

The test section, perforated tube and cooling tubes are surrounded by an acrylic shroud which serves as the outer boundary of the apparatus as well as providing a weak simulation of the effects of the next ring of assemblies that would be at a larger radius in a nuclear reactor. This shroud is of about 36 cm outer diameter and is six mm thick.

Approximately three meters above the level where the heated section starts, a fifteen cm viewing port was cut into the plastic shroud in order to provide direct visual access to the test section during a heated experiment.

The primary measurements are the temperatures and test section power, in order to deduce heat transfer coefficients for comparison to correlations or theory and for testing numerical simulations of reactors with comparable geometries. Temperatures are measured via standard grade, Type K thermocouples. The thermocouples are connected to Type K extension wire which runs to a 150 F (65.5 C) reference oven. Total electrical current is measured via a parallel pair of calibrated shunts connected in series to the upper copper electrode.

Measurement of the normal thermal emissivity of the test section for the wave length range from 8 to 14 μm is accomplished manually with an Omega infrared pyrometer (model OS-2104 L) with a measurement spot size of 6 mm.

Further details and a summary of the instrumentation are listed in a more detailed report by McEligot et al. [1992].

III. PROCEDURES

As part of the design calculations the thermal response of the test section and the axial thermal conduction were estimated. The time constant was estimated to be about 1/2 hour, indicating that manual control and data recording would be feasible and that one would expect two to three hours would be required to reach an approximate steady state. The characteristic length for axial conduction along the stainless steel tube was estimated to be about eight cm so end effects would become negligible about thirty cm from either electrode.

Since the heat transfer rate by thermal radiation is typically of the same order of magnitude as that by natural convection to gases, it is desirable to minimize the thermal radiation and to measure the radiative properties in order to deduce the convective heat transfer parameters accurately. As indicated in Section IV, the convective heat transfer rate in an experimental run is deduced by subtraction of the estimated radiation heat transfer rate from the total energy generation rate. Therefore, the experimental uncertainties of the convection results are sensitive to the magnitude of the thermal radiation and to the uncertainties in the radiative properties of the surfaces involved.

O'Brien [1991] developed our technique for deducing the emissivities and conducted measurements before the experiment was operated. Essentially, the Omega infra-red pyrometer is employed to deduce a normal, band emissivity for the range 8 to 14 μm by comparison between the instrument indication and the surface temperature measured by thermocouples. He measured values as functions of temperature for samples cut from the same tubing that was used to fabricate the experiment. Using approaches suggested by Siegel and Howell [1972], these data were used as the bases for estimates of the total, hemispherical emissivities which were required for the data reduction.

Procedures for the experimental runs were straightforward. Actual runs involved mainly setting the electrical current to the desired constant level, then monitoring

appropriate signals until an apparent steady state was reached. Shortly after initiating the transient, we recorded "instantaneous" values of the test section voltage, current and temperature to check that the overall resistance-to-temperature relation had not changed since the previous run. Once it was determined that an apparent steady state condition had been reached, a final scan of the electronic signals was recorded and manual observations were written in the experiment log.

If the radiative properties of the test section changed significantly due to "aging" during the experiment, the total heat transfer rate and apparent convective heat transfer rate would also change. Therefore, the test section emissivity was monitored. After steady state measurements had been recorded, the removable port could be opened and the normal band emissivity of the test section could be checked. Throughout the range of experiments conducted, there was no evidence of a change in the emissivity versus temperature relationship of the stainless steel. (This conclusion was also checked by repeating runs and examining the deduced convective parameters.)

IV. DATA REDUCTION

Results are presented primarily in terms of non-dimensional natural convection parameters. Since no correlations were found for the complicated annular enclosure studied, definitions were selected so that in the limiting case of large outer radii for the enclosure they would correspond to typical correlations for isolated, vertical, slender cylinders. The difference between the temperatures of the outer surface of the cylinder and of the cooling tubes was chosen as the characteristic temperature difference, corresponding to the idealized boundary conditions. In the limiting case this value would approach the temperature difference between the cylinder surface and the ambient air at "infinite" distance.

The local convective heat transfer rate was determined, via an energy balance for steady state, from the difference between the thermal energy generation rate and the net thermal radiation rate. The fraction of the total energy that was transferred as thermal radiation is estimated to have been from 16 to 48 per cent.

Local thermal energy generation rates were calculated from the measured cross section of the stainless steel test section, the local measured temperature and values of the electrical resistivity versus temperature taken from the literature. Data presented by Touloukian [1967] were approximated by a linear relation; for the range from 650 to 800 K this approximation was adjusted for the non-linearity evident above 650 K in the data in his figure. The local energy generation rate is then given as $G' = i^2 \rho_{\text{elec,ts}} / A_{\text{cs,ts}}$. The calculated value of the resistivity was checked approximately by comparing the predicted resistance per unit length to a value deduced from the measurements of current and overall voltage. These comparisons agreed to within 1.7 per cent over the range of data presented herein.

For the bare test section the energy loss by thermal radiation was calculated as for a small grey body in large surroundings [Incropera and deWitt, 1981],

$$q'_r = \epsilon_{t,h} \pi D_{ts} \sigma (T_{ts}^4 - T_{ct}^4)$$

with the surroundings represented by the surface temperature of the cooling tubes. A linear approximation of the measurements of O'Brien [1991] was used to estimate the total hemispherical emissivity as

$$\epsilon_{t,h} = 0.165 + 0.0001667 (T - 300)$$

with T in degrees Kelvin. His measurements were taken on a sample of the same tubing as used for fabricating the test section.

Local natural convection rates were determined from steady state energy balances, in a region where the axial variation of the surface temperature was negligible, to be

$$q'_c = G' - q'_r$$

Since the gas temperature in an enclosure is a dependent variable rather than a boundary condition, the convective heat transfer coefficient is defined in terms of the primary surface temperatures in the convective heat transfer situation,

$$q'_c = h \pi D_{ts} (T_{ts} - T_{ct})$$

The heat flux is evaluated at the outer surface of the test section (i.e., the inner surface of the irregular annular enclosure). With this choice the definition becomes equivalent to that for an isolated cylinder as the distance to the surrounding surfaces becomes very large.

Air properties were evaluated from the work of Hilsenrath et al. [1960]. As a consequence of the large range of temperature differences in the study, the question arises concerning evaluation of gas properties in calculating the non-dimensional parameters. Lacking data on turbulent natural convection in comparable geometries in general, one has no direct guidance on how to account for the effects of temperature-dependent transport properties in this specific situation. For a vertical flat plate of large dimensions (hence high Rayleigh numbers), Siebers, Moffat and Schwind [1985] examined the same question; they concluded that it was better to use the temperature outside the thermal boundary layer to evaluate properties than to use the usual film temperature. We have chosen an approximately equivalent approach: our air properties are evaluated at the cooling tube surface temperature at the same height (and at 0.84 atm). The volumetric coefficient of thermal expansion is calculated in the same manner, $\beta = 1/T_{ct}$.

The primary non-dimensional parameters in the results are the local Nusselt number and the local Rayleigh number, which we define using the test section diameter as the characteristic length as

$$Nu_D = h D_{ts} / k$$

and

$$Ra_D = [\rho^2 g \beta D_{ts}^3 (T_{ts} - T_{ct}) / \mu^2] \cdot Pr$$

Given these explicit definitions and treatment of gas property variations, the reader may convert to definitions he or she prefers (this conversion is eased by approximating the properties in terms of power laws of the absolute temperature [McEligot, 1967]).

The primary contributors to the experimental uncertainties of the results [Kline and McClintock, 1953] reported here are those in the thermocouple measurements and the thermal emissivities. The standard-grade thermocouples with their connectors and reference junctions yielded estimates of uncertainties of temperature differences ranging from 4-1/2 to 5-1/2 degrees C, depending on the test section temperature. Estimates for the emissivity data range from 10 to 14 per cent, for the range of emissivity measurements, and are slightly higher at low temperatures. The uncertainty in thermal energy generation is dominated by a 2-1/2 per cent uncertainty in test section thickness.

In general, the uncertainty in temperature measurement dominates the results at low temperature. Then as the fraction of energy transferred as thermal radiation grows, the uncertainty in q'_r becomes more important. The result of these trends is that the uncertainty in convective heat transfer parameters is greatest at the extremes - lowest and highest power - and is minimized at intermediate levels (see Figure 5 later). Consequent typical estimates of the uncertainties of the deduced Nusselt numbers become about 18, 5 and 13 percent for low, medium and high temperature runs, respectively.

V. RESULTS AND DISCUSSION

Data were obtained for conditions ranging from near room temperature to maximum test section temperatures near 800K (1000 F). The temperature ratio, T_{ts}/T_{ct} , varied from about 1.1 to 2.6 over this range. Local measurements are presented mainly for a height of $x/D = 127$ and at that location the Rayleigh number, Ra_D , varied from about 2.9×10^4 to 4.6×10^5 . The Prandtl number for air could be considered constant at about 0.71 when based on the cooling tube temperatures. Summaries of the data are available in the report by McEligot and coworkers [1992].

Cooling tube surface temperatures were maintained between 286K and 296 K, essentially near room temperature. The cooling water flow kept the non-dimensional surface temperature of the acrylic shroud between 0.06 and 0.1 (based on $T_{ts}-T_{ct}$) except for the lowest temperature run. Average air temperature was not determined since it would require integration over a distribution of a large number of air temperature measurements in the enclosure; however, related data in the same enclosure imply that it would be equal to the cooling tube temperature plus about fifteen per cent of the temperature difference between the test section and the cooling tubes [McEligot et al., 1992].

Eight sets of data were recorded at steady state, with the last run chronologically repeating control power settings for an earlier run to verify reproducibility. Axial temperature profiles for four typical runs are presented in Figure 3; left to right corresponds to bottom to top along the test section. All eight runs show comparable behavior.

At both ends of the test section, there is evidence of the effect of axial conduction reducing the temperature to that of

Fig. 3. Measured axial temperature distributions.

the copper electrodes. It appears to be significant only for 45 to 60 cm, decreasing with temperature as the total heat transfer coefficient increases as expected from fin theory [Incropera and deWitt, 1981]. For all runs it may be seen that axial conduction is negligible at $x/D = 127$, the location of emphasis for the local results.

The appearance of a local temperature maximum at the lower end of the test section is believed to be due to the lower convective heat transfer coefficients of a laminar, and then transitional, boundary layer. From an experiment for a slender, isolated cylinder with low temperature differences, Hama and Recesso [1958] concluded that the transition Rayleigh number, Ra_x , should be approximately 1×10^{10} . For our higher temperature differences, this value corresponds to x/L of about 1/6 to 1/100, decreasing as the temperature increases. Consequently, the peak at low x/D is expected to move closer to the origin as the power level rises, eventually becoming insignificant relative to the effects of axial conduction to the electrode. This trend corresponds to the data observed in Figure 3.

For all conditions studied, the temperature became effectively constant for the upper half (or more) of the tube. Thus, for the near uniform heat flux of resistive heating, the local convective heat transfer coefficient became independent of axial location, an indication of turbulent natural convection. Turbulent correlations usually take the form $Nu_x \sim (Ra_x)^{1/3}$ so that x cancels and hence $h(x)$ is constant under constant property conditions. Accordingly, we presume the boundary layer is turbulent in this region. A few attempts at flow visualization, with cloth tufts inserted through the observation port, also appeared to confirm this claim.

In order to account for the variation in gas properties across the boundary layer, an approach comparable to that of Siebers, Moffat and Schwind [1985] is applied. Properties are

Fig. 4. Experimental results in terms of natural convective parameters for isolated, slender cylinders.

evaluated at the temperature of the cooling tubes and a correction factor is introduced based on the ratio of absolute temperatures, specifically,

$$Nu (T_{is}/T_{ct})^{0.14} \sim f_n(Ra)$$

In the limit of constant properties, the temperature ratio approaches unity and such variable properties correlations become equivalent to the appropriate constant properties correlations.

Experimental results at $x/D = 127$ are plotted for comparison to correlations for isolated slender cylinders in Figure 4. Nagendra, Tirunarayanan and Ramachandran [1970] developed a piecewise continuous correlation based on a natural convective slenderness parameter from their data [Nagendra, Tirunarayanan and Ramachandran, 1969], those of Hama, Recesso and Christiaens [1959] and earlier measurements. In the range of our present results their recommendation is

$$Nu_D = 1.37 [Ra_D (D/L)]^{0.16}$$

Their treatment assumed laminar flow. (Local analytical predictions for laminar boundary layers are also tabulated by Hama, Recesso and Christiaens and are shown to agree with their own data.) The present results yield Nusselt numbers that are about twice those of Nagendra, Tirunarayanan and Ramachandran - as would be expected for turbulent flow.

Data and correlations for turbulent flow from slender cylinders (i.e., tall cylinders) are sparse. Our literature survey found none for annular enclosures with regular or irregular outer surfaces. For an isolated slender cylinder, in the Handbook of Heat Transfer Raithby and Hollands [1985] have suggested a method to extrapolate predictions from the better explored vertical flat plate, but they caution that data do not seem to exist to check their tentative proposal. Recommendations on their page 6-24 can be rearranged to the form

Fig. 5. Data for turbulent natural convection from a smooth tube in an irregular annular enclosure.

$$Nu_D = 1.8 \left[\ln \left\{ 1 + (17.48 / Ra_D^{1/3}) \right\} \right]^{-1}$$

If the cylinder is tilted, the plume that they assume to envelop it would no longer be centered, giving the result that the heat transfer would be higher; the effect is particularly sensitive near vertical [Raithby, 1992]. One would expect that the close proximity of the bounding surfaces in our enclosure would have comparable effects and improve the convective heat transfer. This equation has been plotted also on Figure 5 and it does appear that our measurements are indeed higher (even accounting for the differences in definitions).

Morgan [1975] suggested a relation of the form

$$Nu = 0.148 Ra^{1/3}$$

to correlate the data of Eigenson [1940] for an isolated cylinder. In this case the heat transfer coefficient is defined in terms of the temperature difference between the heated cylinder and the undisturbed air at a distance.

Results of our measurements are presented in Figure 5 in terms of parameters based on test section diameter and cooling tube temperature as defined earlier. Also shown are equations which correspond to the correlations suggested by Raithby and Hollands, by Siebers, Moffat and Schwind [1985] for a flat plate and by Morgan's representation of the Eigenson data. Since these correlations are based on different definitions, it must be emphasized that we are not making comparisons to the correlations directly.

For our irregular annular enclosure with an irregular outer surface formed by the three cooling tubes plus shroud and with our definitions, the data can be approximated by

$$Nu_D (T_{ts}/T_{ct})^{0.14} = 0.156 Ra_D^{1/3}$$

reasonably well for the range $4 \times 10^4 < Ra_D < 5 \times 10^5$.

However, this may be a fortuitous result. In particular, it is not a confirmation that the cylinder can be considered to be an

isolated cylinder. In our technical report [McEligot et al., 1992] it is shown that - when converted to parameters based on "freestream" air temperature and heat transfer coefficient defined in terms of the difference between the test section and air temperatures (instead of cooling tube temperature) - the equivalent correlation is effectively of the order of forty per cent higher than the Eigenson correlation.

The present experimental results correspond to one unique reactor geometry and, probably, would provide conservative estimates of the convective heat transfer coefficients in cases where the spacing to the cooling tubes is closer (in non-dimensional terms). In a safety analysis for a reactor, the thermal emissivities of the actual surfaces must be estimated to predict the contribution by thermal radiation; since we attempted to keep the emissivities low for accuracy in deducing the convective contribution, we would expect more energy transfer by thermal radiation in an actual application than in our experiment.

VI. CONCLUDING REMARKS

The general result of the present work is the observation that available correlations for single vertical cylinders do not predict closely the behavior of turbulent natural convection from a small cylinder to a surrounding array of cylinders. On the other hand, a correlation evolved from the early data of Eigenson [1940] did not fare as poorly as one might expect, considering the significant differences in geometry. Its use would provide a conservative estimate of tube temperature when the heating rate is the controlled variable.

For the present geometry, the correlation

$$Nu_D (T_{ts}/T_{ct})^{0.14} = 0.156 Ra_D^{1/3}$$

is recommended for the ranges

$$4 \times 10^4 < Ra_D < 5 \times 10^5, \quad T_{ts}/T_{ct} < 2.6, \quad Pr = \sim 0.7$$

provided the gas properties are evaluated at the cooling tube temperature, the parameters are calculated as defined above and $Ra_x > \sim 10^{10}$ (turbulent).

Only the heating rate was varied significantly in the experiment, so the effect of property variation has been assumed rather than tested by varying temperature ratio independently from Rayleigh number. Likewise, the geometric ratios have not been varied. However, one probably could reasonably expect that results for larger radius ratios between the test section and the nearest cooling tube surface would be bracketed by the present measurements and the Eigenson correlation for an isolated tube.

NOMENCLATURE

c_p	specific heat at constant pressure
D	tube diameter
G'	thermal energy generation rate per unit length

g	acceleration of gravity
h	convective heat transfer coefficient
i	electrical current
k	thermal conductivity
L	characteristic length, x, D, etc.; length of heated section
q'	heat transfer rate per unit length
Pr	Prandtl number, $c_p \mu / k$
T	absolute temperature
x	axial location, measured upward from start of heating
β	volumetric coefficient of expansion, $1/T$
ϵ	thermal emissivity
μ	absolute viscosity
ρ	density; ρ_{elec} , electrical resistivity

Subscripts

c	convective
ct	cooling tube
r	radiative

ACKNOWLEDGEMENTS

The work reported was supported by the U. S. Department of Energy, Assistant Secretary for Defense Programs under DOE Idaho Field Office, Contract DE-AC07-76ID001570, to whom we are grateful. We appreciate the guidance and interest of Dr. Fred E. Witmer, Ms. Sharon L. Zeigler and Mr. Charles Noble, DOE, who helped as technical program monitors.

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This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

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