AIIM

Association for Information and Image Management

1100 Wayne Avenue, Suite 1100
Silver Spring, Maryland 20910
301/587-8202

Centimeter

Inches

MANUFACTURED TO AIIM STANDARDS
BY APPLIED IMAGE, INC.

1 of 1

CONF-940553--57

EXPERIMENTAL HEAT TRANSFER AND FLUID FLOW OVER DRIFT-EMPLACED CANISTERS

William G. Culbreth
University of Nevada
Department of Civil and Environmental Engineering
P.O. Box 454015
Las Vegas, Nevada 89154-4015
(702) 895-3426

Satish Pattisam
University of Nevada
Department of Civil and Environmental Engineering
P.O. Box 454015
Las Vegas, Nevada 89154-4015
(702) 895-3701

ABSTRACT

Drift-emplaced waste canisters are under consideration for the long-term storage of high-level spent fuel in the proposed underground repository at Yucca Mountain. These canisters will be placed on pedestals above the floor of the drifts and exchange heat with the walls of the drift and with air circulating through the repository. To assess the requirements of the repository ventilation system, values of the dimensionless convective heat transfer coefficient and the pressure drop across individual canisters were measured in a experimental model of a drift. The results were curvefitted as functions of the spacing between the canisters and the Reynolds number of the flow. Both natural and forced convection effects were investigated.

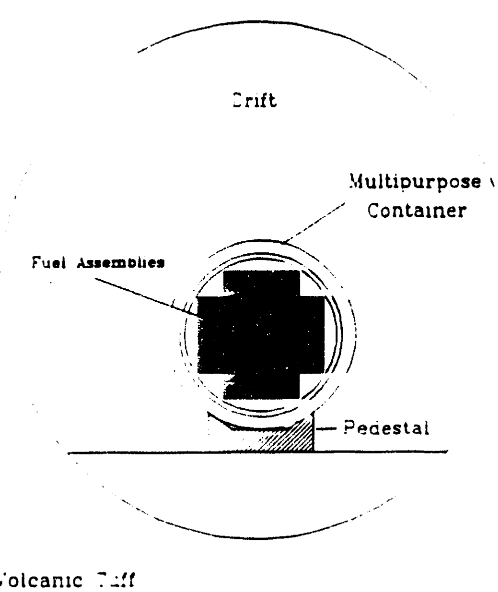

Fig 1. Cross-Sectional View of a Drift-Emplaced Waste Canister

I. INTRODUCTION

The design of canisters to be used for the long-term storage of high-level nuclear waste have progressed from the borehole model indicated in the original site characterization plan to drift-emplaced multi-purpose containers. These canisters may contain 21 or more PWR fuel assemblies[1]. A cross-sectional view of the drift and emplaced waste is shown in figure 1.

Removal of decay heat is important if the Zircaloy degradation temperature of 350°C is not to be exceeded. Above this temperature, the oxide layer on the cladding grows continuously leading to early failure. In the preclosure drift system, air ventilation will provide convection cooling of the canister walls and radiative heat exchange with the drift walls will remove decay heat.

The proper design of the required repository ventilation system must be based on a knowledge of the total pressure drop experienced as air passes through the drifts and over the waste canisters. Knowledge of the convection heat transfer coefficient between the waste canister surface and the flowing air will also be needed by the designer. The convective heat transfer from the canisters and the pressure drops may be determined through appropriate computer models, however, a scale experimental model of the drift/waste canister geometry provides accurate results of dimensionless pressure drops and heat transfer coefficients as functions of air flow rate and ambient temperatures.

In this study, an experimental model of a repository drift was constructed at the University of Nevada, Las

FC08-90NV10872

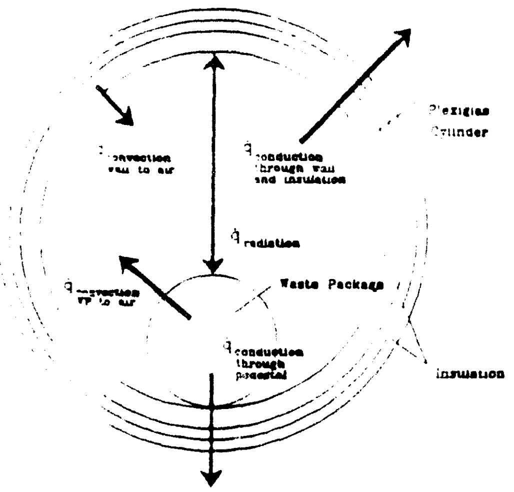

Fig 2. Combined Conduction, Convection, and Radiation Heat Transfer from a Waste Canister

Vegas to investigate fluid flow and thermal transport about waste canisters in underground drifts. The model has provided pressure drop data, visualization of temperature distributions, and heat transfer coefficients. This work complements computational studies on repository temperature distributions.[2]

II. THEORETICAL BACKGROUND

The flow of heat from the experimental model approximated the effect of heat transfer from a waste package in an underground repository. Three heat transfer mechanisms affected the cooling of waste canisters in the repository drifts. Conduction through the pedestal, convection to the air, and radiation heat transfer between the canister surface and the drift wall all contribute to the removal of the maximum 21 kilowatts of heat from the package. Figure 2 summarizes the paths followed by the heat generated by a model of a waste canister into the environment.

An analysis of the total heat lost from a waste canister accounted for all of these mechanisms and allowed the effect of convection heat transfer from the canister to the air flowing through a drift to be computed. The relevant equations necessary to summarize the heat loss will be described.

A. Forced Convection Heat Transfer

Convective heat transfer is quantified through the non-dimensional Nusselt number:

$$Nu = \frac{h\,d}{k_f} \tag{1}$$

where, h = convection coefficient (W/m^2-K)
k = air thermal conductivity (W/m-K)

The heat transfer rate due to convection is given in terms of the temperature difference between the waste canister surface and the ambient air:

$$\dot{q}_{conv} = h A_s (T - T_\infty) \tag{2}$$

Nusselt numbers are functions of the air velocity, kinematic viscosity, drift and canister dimensions, and are often determined experimentally. Curvefits are used to express Nu as a function of the Reynolds number, ($U_\infty d/\nu$), and the Prandtl number (ν/α) of the fluid. As a non-dimensional parameter, Nusselt numbers obtained from a scale model can be directly applied to the full scale prototype. For heat transfer to fluids in a circular pipe, the Dittus-Boelter relation[3] can be used:

$$Nu_d = 0.023\, Re_d^{0.8}\, Pr^{n} \tag{3}$$

where, n = 0.4, for heating of the fluid
n = 0.3, for cooling of the fluid

B. Natural Convection Heat Transfer

In the absence of forced air flowing over the waste canister, heat may be convected from the surface of the canister by natural convection currents set up in the ambient air. Natural convection is expressed in terms of the Grashof number for the flow:

$$Gr = \frac{g\beta(T - T_\infty)\, d^3}{\nu^2} \tag{4}$$

where β = 1/T for an ideal gas. The Rayleigh number is also used, Ra = Gr Pr. As in the case of forced convection, the heat transfer rate is often curvefitted in terms of the Reynolds number in an equation of the form:

$$Nu_d = a\, Ra^{n} \tag{5}$$

where a and n are coefficients of the fit. For the specific case of natural convection from horizontal cylinders, Churchill and Chu[4] suggested the following relationship valid for 10^{-5} < Gr Pr < 10^{12}:

$$\overline{Nu}_d = 0.60 + \left\{ 0.387 \left[\frac{Gr_d Pr_d}{[1+(0.559/Pr)^{9/16}]^{16/9}} \right]^{1/6} \right\}^2 \tag{6}$$

In this work, heat transfer from the model waste canister in stagnant air was measured and expressed in the form of similar curvefits. The results will be compared to the above equation.

C. Conduction Heat Transfer

As shown in figure 2, heat transfer can occur through the pedestal or through the floor of the model and the layers of insulation surrounding the experimental drift. Such heat losses may be accounted for by using Fourier's Law:

$$\dot{q}_{cond} = -kA\frac{dT}{dx} \tag{7}$$

where x is measured through the insulation or the drift wall.

D. Radiation Heat Transfer

The flow of heat from the waste canisters is governed by the emissivity of the waste canister wall and of the drift wall, the temperatures of each surface, and the shape factor between the two surfaces. For one surface completely surrounded by another, the radiation heat transfer may be approximated by:[5]

$$\dot{q}_{rad} = \frac{\sigma A_1 (T_1^4 - T_2^4)}{\frac{1}{\epsilon_1} + \left(\frac{1}{\epsilon_2} - 1\right)\frac{A_1}{A_2}} \tag{8}$$

E. Total Heat Transfer

The total heat transfer from the waste canister shown in figure 2 accounts for the loss of heat by radiant exchange with the drift wall, by conduction through the floor, and by convection to the ambient air. The drift wall in the experimental model losses heat through convection to the internal fluid flow and by conduction through the outer layers of insulation to the environment. The corresponding heat balance equations are:

$$\begin{aligned} \dot{q}_{gen} &= \dot{q}_{cond-1} + \dot{q}_{conv-1} + \dot{q}_{rad} \\ \dot{q}_{rad} &= \dot{q}_{cond-2} + \dot{q}_{conv-2} \\ \dot{q}_{gen} &= \dot{q}_{cond-1} + \dot{q}_{cond-2} + \dot{q}_{conv-1} + \dot{q}_{conv-2} \end{aligned} \tag{9}$$

where subscript 1 indicates the waste canister surface and 2 represents the interior drift wall surface. By combining the first two equations, the radiation heat transfer terms in each equation were equated and the total heat generated from the waste canister model can be expressed in terms of convection and conduction terms from each surface. This is shown in the third equation.

F. Pressure Drops

Minor pressure losses due to obstructions in pipe flows are often expressed in terms of a loss coefficient, K:[6]

$$K = \frac{\Delta p}{\frac{1}{2}\rho_f U_\infty^2} \tag{10}$$

The loss coefficient may be a function of geometry and the Reynolds number of the flow.

III. EXPERIMENT DESCRIPTION

The actual drifts will be approximately 7 meters in diameter and house waste canisters approximately 4 meters long with a diameter of 1.2 m.[2] Crossflowing air will be necessary to remove decay heat from the canisters and to provide an environment cool enough for human operators. The spacing between canisters will depend upon the desired area power density.

A laboratory model of the drift and emplaced wastes was constructed at the University of Nevada, Las Vegas and is shown in figure 3. Clear plexiglas tubing with an inner diameter of 152 mm and a thickness of 12.7 mm was used to model the drift. The total length of the drift model was 8.5 m. A slotted plate and flow straighteners were used to condition the input air flow. An automated pitot tube traverse with position feedback and a pressure transducer was used to measure the volumetric flowrate through the drift model. A computer-based data acquisition system controlled the traverse.

A precision micromanometer was used to calibrate the pitot tube traverse, measure the static pressure of the air inside the drift model, and to determine the pressure drop across the test section.

The 2 meter long test section housed models of the waste packages. The waste canister models were constructed from aluminum to a 1/50 scale. Core heaters were placed in the canister models to simulate the generation of decay heat. The test sections was wrapped in three layers of insulation to prevent heat loss by conduction. The 6.35 mm thick layers of insulation were

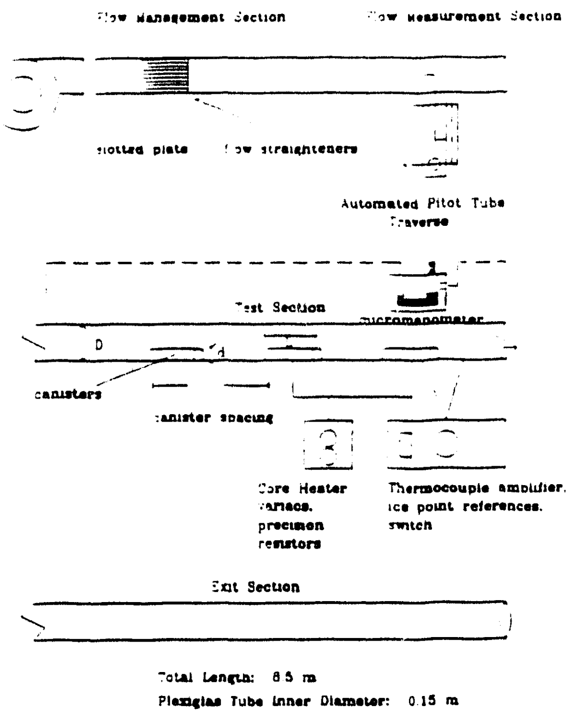

Fig. 3 Experimental Setup for Fluid Flow and Heat Transfer Studies about Drift-Emplaced Waste Canisters

formed from two layers of aluminum sandwiching air-filled plastic bubbles.

A total of five canisters were placed in the drift to provide an established velocity distribution to model a sequence of canisters in a drift. The central canister was heavily instrumented with thermocouples and a core heater. Variacs and power resistors were used to control the heat output and to measure the electrical power into the core heater.

Eighteen type-K thermocouples were used to measure the surface temperature of an instrumented canister, drift wall temperatures, and air inlet and outlet temperatures. The locations of the thermocouples are shown in figure 4. Thermocouples sensed the temperatures on the sides, bottom, top, front, and back of the instrumented waste canister. The thermocouples were connected to an amplifier/ice point reference and an Omega readout device accurate to ± 0.1 K. Thermocouples located on both sides of the plexiglas cylinder and throughout the layers of insulation provided data used to predict the conduction losses through the drift wall. The thermocouples were calibrated versus an NIST-standard mercury thermometer with a resolution of ± 0.001 K. Data from the thermocouples was used to compute Nusselt numbers as functions of both Reynolds and Prandtl numbers in the flow.

One canister was coated with liquid crystals and photographs of the canister showed the temperature distribution as a function of ambient flow rate.

Additional instrumentation available in the laboratory included a Dantec two-component laser Doppler velocimeter and a TSI hot wire anemometer for air velocity measurements. Air density was calculated from atmospheric pressure measured by a National Weather Bureau type mercury barometer.

This flexible experimental setup worked quite well for the measurement of pressure drops and heat transfer coefficients.

IV. RESULTS

A total of 36 experimental runs were made to observe the affect of varying canister spacing and flowrates on heat transfer rates. The power to the core heater was limited to prevent an excessive amount of heat loss from the waste canister by radiation. In a typical run, approximately 0.15% of the total heat generated by the canister was lost by conduction between the canister and the floor of the drift. 3.1% of the available heat was transferred by conduction through the insulation surrounding the drift model and 3.8% was lost by convection between the drift wall and the internal air flow.

In the case of natural convection, approximately 20% of the generated heat was lost through the insulated walls.

A. Pressure Drop Data

Pressure drop data was acquired for a range of Reynolds numbers as shown in figure 5. The spacing

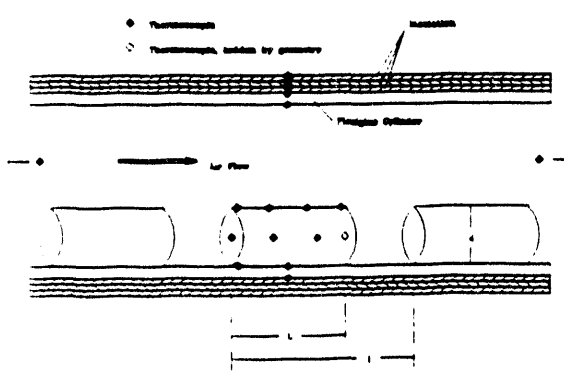

Fig. 4 Location of Thermocouples in the Experimental Setup

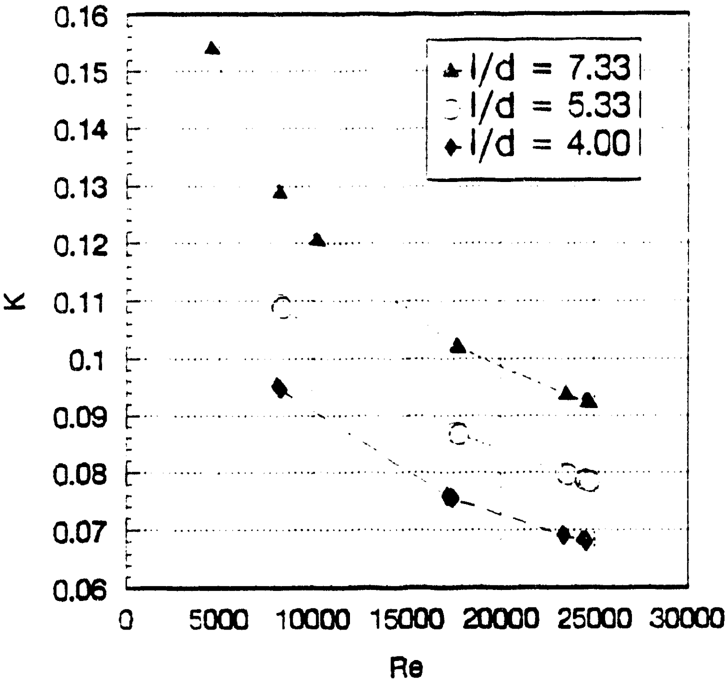

Fig. 5 Loss Coefficients for Canisters in a Drift with 3 Different Spacings

between canisters was varied and the data is valid for d/D = 0.25, l/d = 7.33, and for $4{,}000 < Re_d < 30{,}000$. The data was obtained across a section that included 5 identical model canisters. The total pressure drop was computed based on one container and expressed in terms of the non-dimensional loss coefficient, K, as shown in figure 5.

The loss coefficient decreased as the spacing between canisters decreased. This was likely due to the limited space available for wake formation in the closer spacings and the subsequent loss in wake drag. The data was curvefitted using a regression analysis for the range of $4{,}000 < Re_d < 25{,}000$:

$$K = 0.7314\, Re^{-0.3044} \left(\frac{l}{d}\right)^{0.5055} \tag{11}$$

The standard error for the fit was 0.041.

B. Visualization

Color slides were taken of a single canister coated with liquid crystals for various flowrates. The slides demonstrated that regions of limited flow existed near the bottom leading edge of each canister. These regions appeared as hot spots and may affect the internal temperature of the spent fuel. This effect is observed in figures 6a and 6b. Temperatures measured from the surface of the waste package using thermocouples were non-dimensionalized to express the ratio of the local convective heat loss to the average convective heat loss. This data is presented in the two figures.

$$\frac{\dot{q}}{\dot{q}_{avg}} = \frac{(T - T_\infty)}{(T_{avg} - T_\infty)} \tag{12}$$

Figure 6 was valid for (l/d) = 5.33, d/D = 0.25, and for a total heat generation of 11.4 watts.

In figure 6a, the flow was turned off and natural convection existed from the canister surface. The resulting Nusselt number based on canister diameter was 14.4. The heat flux distribution was fairly uniform across the surface and slightly lower at the two ends. The thermocouples on the bottom of the canister were in contact with the "drift" wall and are not influenced by natural convection. They were included to illustrate the relative temperature, only.

Figure 6b shows the effect of air flow across the surface

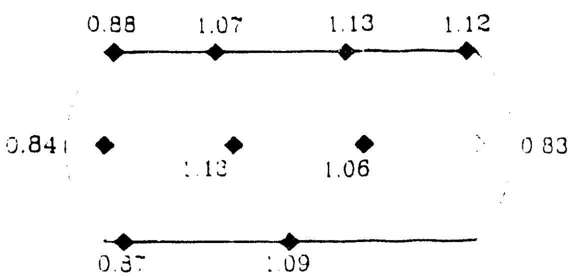

Fig. 6(a) Non-Dimensionalized Temperature Distribution, Natural Convection

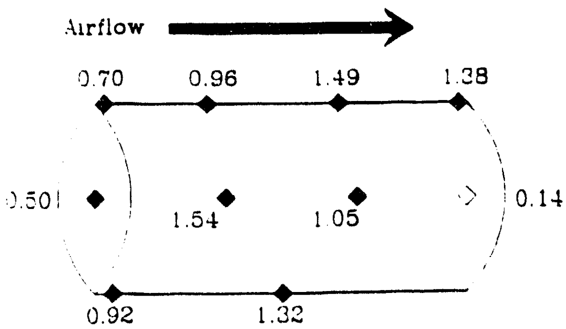

Fig. 6(b) Non-Dimensionlized Temperature Distribution, Re_d=17,580

17,580 and Nu_L = 142. The thermocouples at the front and back of the canister experienced much lower temperatures than those on the sides. A zone of low heat flux existed on the leading edge at the top of the canister and the flux rose rapidly toward the trailing edge. The thermocouples located on the side verified the liquid crystal visualization showing regions of pronounced temperature rise along the sides toward the leading edge of the canister.

C. Nusselt Numbers, Forced Convection

The convective heat transfer from the surface of an instrumented waste canister was accounted for by using equations 9. The resulting data, plotted in figure 7, showed that the canister spacing had very little effect on the convection heat transfer. A regression fit of the data for all 36 runs provided the following equation:

$$Nu_d = 0.0239\, Re_d^{0.865} \left(\frac{l}{d}\right)^{0.0263} \tag{13}$$

The standard error for the fit was 29.1. The Dittus-Boelter equation (3) was used for comparison and demonstrated good agreement between the new data and previous results for the similar geometry of convection from the walls of a circular pipe.

D. Nusselt Numbers, Natural Convection

For each of the 36 experimental runs, thermocouple temperatures were sampled before the flow was turned on. This provided data on the natural convection from a cylindrical canister in a very long pipe. The resulting data is shown in figure 8.

The Churchill-Chu equation (6) is shown as a dashed line on the figure for comparison. Note that the Nusselt number decreases as the spacing increases. A regression fit of the available data yielded:

$$Nu_d = 0.00382\,(Gr_f Pr_f)^{0.664} \left(\frac{l}{d}\right)^{-0.0566} \tag{14}$$

The standard error for the fit was 1.704. The data was valid for 150,000 < $Gr_f\, Pr_f$ < 220,000. The results showed that the heat transfer rates for natural convection from canisters in a cylindrical drift exceeded the case of long cylinders with no enclosure by 12% to 50% over the indicated range.

V. CONCLUSIONS

Proper modeling of the waste canister surface and internal temperature distribution will require a knowledge of the convective heat transfer that occurs in the drift. An experimental setup has been constructed that allowed the measurement of overall Nusselt numbers and pressure drops.

The data provided from these experiments provided plots and curvefits describing the Nusselt numbers for both free and forced convection, and the pressure drops across the canisters, for a waste package to drift diameter ratio of 0.25.

Experiments are now underway to measure similar values for a range of waste package model diameters. To model the actual drift/canister geometry in more detail, the bottom of the drift will be filled to approximate the floor of the drift in an actual repository design.

ACKNOWLEDGMENTS

This work was funded by the U.S. Department of Energy under Cooperative Agreement #FC08-90NV10872. Special thanks are extended to Mark Jones and Robert DeGutes for their excellent work in the laboratory experiments.

NOMENCLATURE

a coefficient
A_s surface area
cond conduction
conv convection
d waste canister diameter
D drift diameter
f fluid
g gravitational acceleration
Gr Grashof number
h convective heat transfer coefficient
k thermal conductivity
K loss coefficient
l spacing between canisters
n coefficient
Nu Nusselt number
Pe Peclet number
Pr Prandtl number
Ra Rayleigh number
rad radiation
T temperature
T_∞ free stream temperature
U_∞ free stream velocity

α thermal diffusivity of the fluid
β 1/T
ε emissivity
ρ density of the fluid
σ Stefan-Boltzmann constant
ν kinematic viscosity of the fluid

REFERENCES

1. U.S. Department of Energy, *Site Characterization Plan, Yucca Mountain Site, Nevada Research and Development Area, Nevada*, DOE/RW-0199, HQO.881201.0002 (1988).

2. Culbreth, W. G., "Computational Heat Transfer Studies of the Proposed Nuclear Waste Repository," Proceedings of the 9th International Conference on Mathematical and Computer Modelling, Berkeley, California, July 1993.

3. Leinhard, J. H., *A Heat Transfer Textbook*, 2nd ed, Prentice-Hall, Inc. (1987).

4. Churchill, S. W., and Chu, H. H. S., "Correlating Equations for Laminar and Turbulent Free Convection from a Horizontal Cylinder," *Int. J. Heat Mass Transfer*, vol 18, p. 1049 (1975).

5. Holman, J. P., *Heat Transfer*, 7th ed, McGraw-Hill (1990).

6. White, F. M., *Fluid Mechanics*, 2nd ed, McGraw-Hill Book Co. (1986)

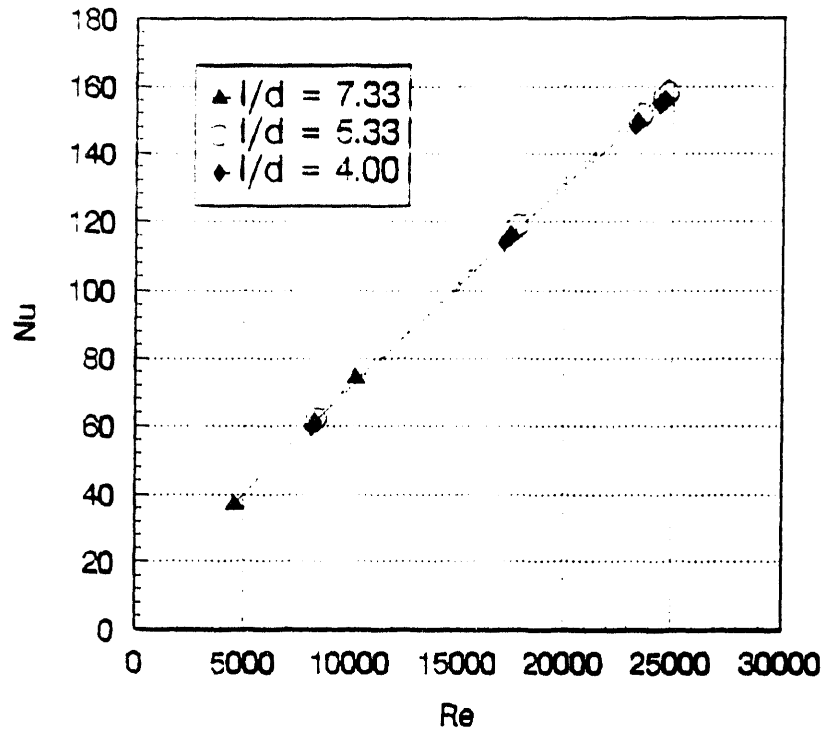

Fig. 7 Nusselt Numbers for Forced Convection across the Canister Surface

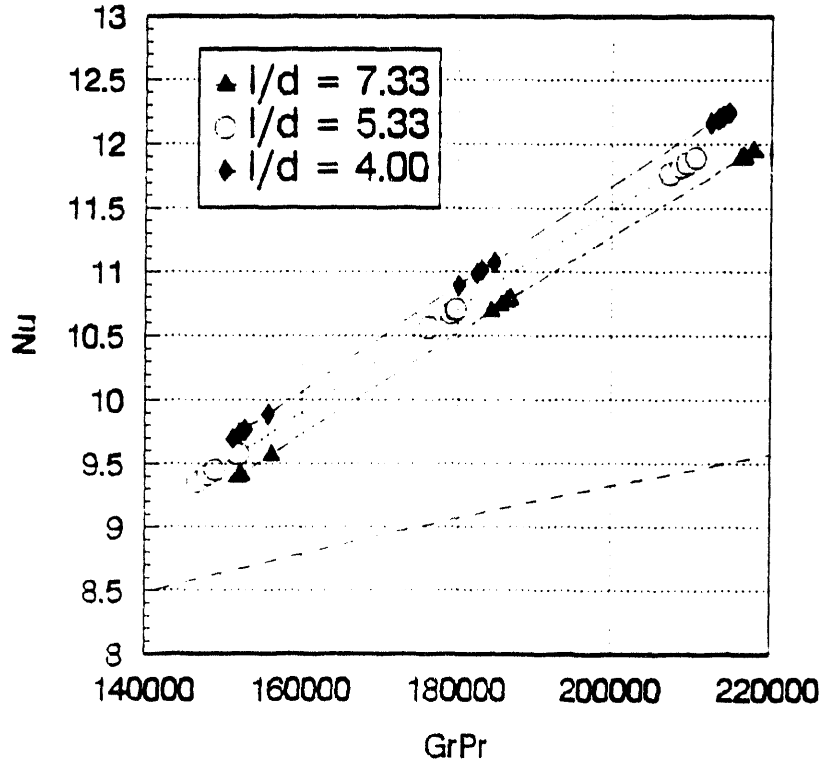

Fig. 8 Nusselt Numbers for Natural Convection from the Canisters

DATE

FILMED

7/25/94

END