EXPERIMENTAL MODELING OF A NUCLEAR WASTE REPOSITORY: DETERMINATION OF CONVECTIVE HEAT TRANSFER COEFFICIENTS AND DRIFT TEMPERATURE PROFILES

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ABSTRACT

An experimental model was developed for a nuclear waste repository storage room. Data were taken over a Reynolds number range of 6 000 to 180 000, covering both the forced and mixed (combined natural and forced) regimes of convection. Data are presented for several circumferential boundary conditions. Results indicate that the natural convection component is significant. The finite difference heat conduction code HEATING5 was used to plot temperatures around the repository room for the different heat transfer coefficients. The plots show that the use of standard forced convection relations can result in over estimating the room temperatures by as much as 70 C.

INTRODUCTION

The nuclear waste disposal plan for the United States calls for high level waste and/or spent fuel to be stored underground in deep geological formations. One of the design constraints on such a repository is that these storage areas must be ventilated. Ventilation is required for for both cooling and air supply for worker safety and comfort. Ventilation may also be required to limit temperatures in both the repository and the waste package.

In order that the ventilation and cooling requirements can be determined in the repository, accurate values of the convective heat transfer coefficients on the interior of the disposal rooms are needed. General flow and heat transfer conditions suggest that the rooms will be operating in the fully turbulent, mixed convection regime[1]. This fact is illustrated in Fig. 1, showing the different flow regimes for horizontal pipe flow[2]. Thus, standard forced convection analyses need to be expanded to include the natural convection component of the heat transfer. Also shown in Fig. 1 is the range of data from the present work.

Several studies, although mostly analytical, have addressed the ventilation in a repository. Svalstad and Brandshaug[3] considered "blast cooling" to cool a repository drift rapidly prior to waste retrieval. The study was analytical and used a finite element code to solve the governing equations. A constant velocity of 1 m/sec was used, which corresponds to a Reynolds number of 280 000 (see DEFINITIONS section). Flows of this type are purely forced convection. Boyd[4] also considered the ventilation required for retrieval. In this study, also analytical, mixed convection was investigated explicitly, using convection coefficients from various workers. Boyd's results indicate mixed convection reduces the total heat transferred, but this effect appears to come from the overall reduction of the convection coefficient, h, not the presence of mixed convection. Danko and Cifka[5] determined the convection heat transfer coefficients of an underground copper mine in situ. Although their data were quite scattered, these workers determined that the effects of the natural convection component were significant.

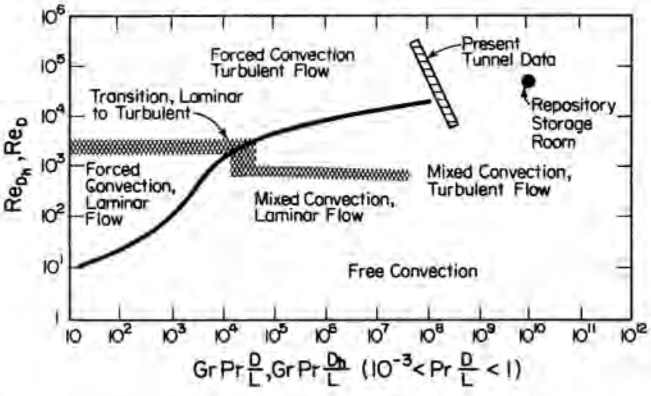

Fig. 1. Flow Regimes in a Horizontal Tube[2] and Estimated Waste REpository Drift Conditions.

The boundary conditions in a repository drift differ significantly from most standard mine ventilation problems. Although many mines, particulary extremely deep mines, experience high heat fluxes from the rock surfaces, the distribution of the fluxes appears to be unique to the nuclear waste repository. The heat sources located in the floor create an extremely skewed heat flux profile that is present for several years after waste emplacement.

The aim of this work was to determine the heat transfer characteristics in an experimental system which models the repository storage rooms, both thermally and hydrodynamically. The waste is modeled to be located in the floor. The results must be non-dimentionalized so that they apply in general to any similarly designed repository.

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EXPERIMENTAL APPARATUS

The experimental apparatus was a flow channel of rectangular cross section with heated walls. The flow channel was constructed of galvanized sheet steel and had a cross section of 0.305 m wide by 0.457 m high. The total available heated length was 4.42 m. These dimensions were chosen so that the aspect ratio of the model was consistent with preliminary designs of repository rooms.

The heat sources were silicon rubber resistance heaters placed behind the sheet steel. The heaters provided a continuous heat flux axially down the tunnel, thus in effect smearing the discrete heat sources that actually exist in the repository. The work of Sisson[6] and Christensen et. al.[7] show that this assumption is reasonable, depending on the waste canister spacing. The heaters were arranged as shown in Fig. 2, with the opposing side wall heater pads connected in parallel to give the same heat output. This arrangement gave a total of five separately controllable sets of heaters around the circumference of the tunnel. Each of these sets of heaters was powered by a continuously variable power supply. Thus a wide variety of circumferential heat flux profiles could be imposed on the tunnel walls.

The current and voltage inputs to the heaters were measured to determine the electrical power produced by the heaters. Thermocouples were placed at each heater to determine the wall surface temperatures. Pairs of thermocouples connected in series were placed on the tunnel's wall insulation, one thermocouple on the inside and one on the outside of the insulation. The heat lost through the walls was calculated using the differential temperature generated by the thermocouple pair and the insulation thermal conductivity. A total of 32 thermocouple pairs were used to determine the total heat loss for each run. All temperature measurements were made with an electrically compensated datalogger. The datalogger signals were then sent to a VAX 11/750 computer for data reduction and heat transfer coefficient calculation. Because of the sometimed large temperature differences between the different surfaces, thermal radiation could not be neglected. The radiation component was calculated using a network scheme[8] with the eight tunnel surfaces (Fig. 2) modeled as infinitely long strips in the axial direction. View factors were obtained from Howell[9].

The h values were found using Eq. (1) and (2),

$$q(net) = h * (T_{wall} - T_{bulk}) \tag{1}$$

$$q(net) = q(electrical) + q(loss) + q(rad) \tag{2}$$

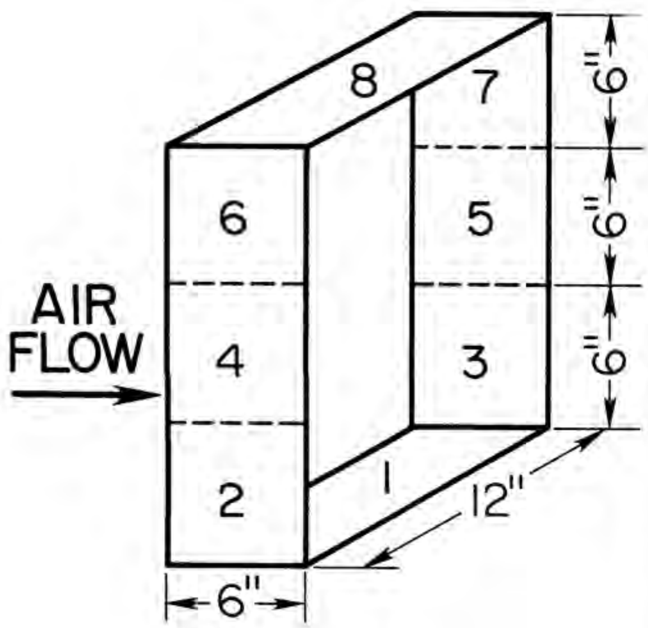

Fig. 2. Heater Pad Arrangement.

where:

q(net) = wall heat flux (in units of energy per unit area)

q(electrical) = electrical power input to heaters per unit area

q(loss) = energy lost through the tunnel's exterior walls, per unit area

q(rad) = thermal radiation (positive from the particular heater, negative toward the heater), per unit area

From these h values, the Nusselt numbers (Nu) were calculated.

Velocity measurements were made using a pitot tube array on a flow nozzle. Figure 3 shows a schematic of the complete setup. Only one heater set is shown for simplicity.

The air flows were in the fully turbulent regime[10] as described by the Reynolds number (Re). The experimental Re ranged from 6 400 to 180 000. Preliminary designs[11] have Reynolds numbers of approximately 170 000. Thus the apparatus' range covers the expected repository Reynolds number.

RESULTS

Three different heat flux profiles were tested. Figures 4 through 6 show the fully developed Nusselt number for each of the five surfaces. The line labelled "Dittus-Boelter Equation" is the classical forced convection correlation developed for turbulent flow in long tubes with constant circumferential heat flux[12], given by Eq. (3).

$$Nu = 0.023Re^{0.8}Pr^{1/3} \tag{3}$$

The runs in Fig. 4 had these boundary conditions, i.e., all heaters had the same heat flux. As would be expected, the data match the Dittus-Boelter correlation quite well at Re greater than 50 000. Below Re = 50 000, the data vary widely. For example, the Nusselt number of the floor at Re = 12 000 is approximately 110, three times the Nu given by Dittus-Boelter. From these results, it is apparent that natural convection is aiding the forced convection component of the heat transfer. Also apparent from Fig. 4 is the fact that on the ceilng the natural convection is unaiding, or opposing, the forced convection component. This again is expected, as the buoyancy-driven flows would create a stagnation layer on the ceiling, thus reducing the heat transfer as predicted by forced convection analysis.

Figures 5 and 6 show data from a heat flux profile as expected in a nuclear waste repository. The profile was calculated from estimated temperature profiles[13]. The two figures have the same flux profile but different absolute fluxes. In the forced convection region (Re greater than 50 000), the Nusselt numbers for the five surfaces differed significantly from each other. This was due to the extremely skewed input flux profile. However the slopes of the curves (the exponent of the Reynolds number) are very close to each other and to the slope of the Dittus-Boelter correlation. This similarity in slope demonstrates that although the magnitudes differ, the dependence of the Nusselt numbers on the Reynolds number is consistent in the purely forced convection region. As the Reynolds number decreases, the Nusselt numbers

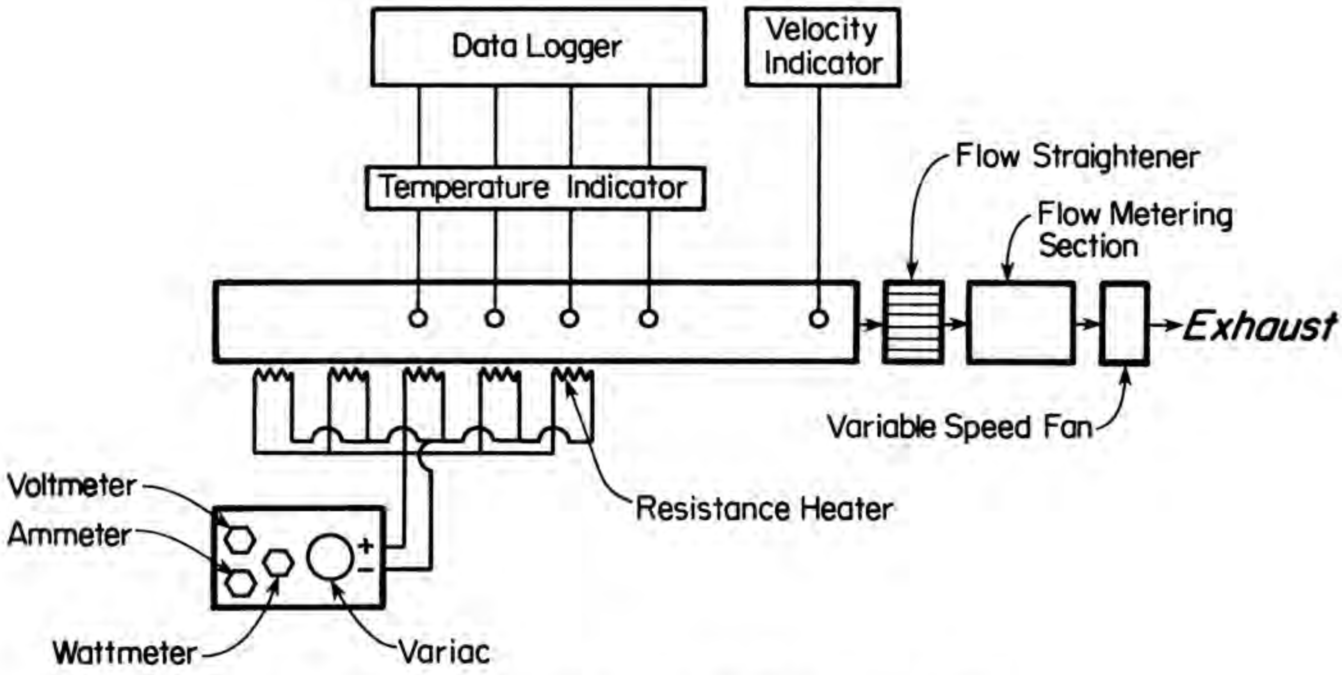

Fig. 3. Schematic of the Experimental Setup.

appear to reach a horizontal asymptote around Re = 30 000. Below this Reynolds number, the Nusselt number curves are essentially flat, indicating that as the flow decreases, the forced convection component decreases, with, however, a corresponding increase in the natural convection component. Thus in the range of Re = 6 000 to 30 000, the mixed convection Nusselt number is independent of the flow rate. Although their data were scattered, the figures by Danko and Cifka[5] show the same general trend.

This last result could be very significant if large flow rates are not required in the repository rooms, i.e. during the period that the rooms are open for emplacement activities or when the rooms are not undergoing blast cooling for retrieval. Using these results, the same heat transfer coefficient can be achieved at reduced flow rates.

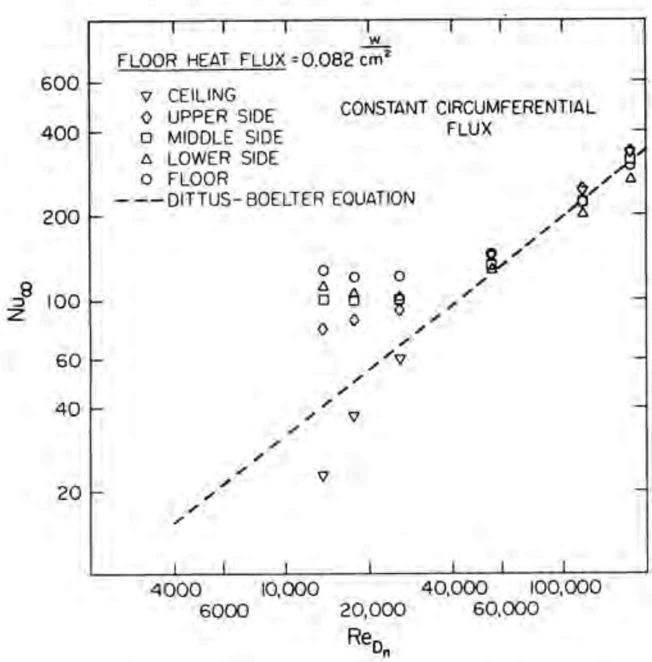

Fig. 4. Nusselt Number vs. Reynolds Number for the Constant Heat Flux Case.

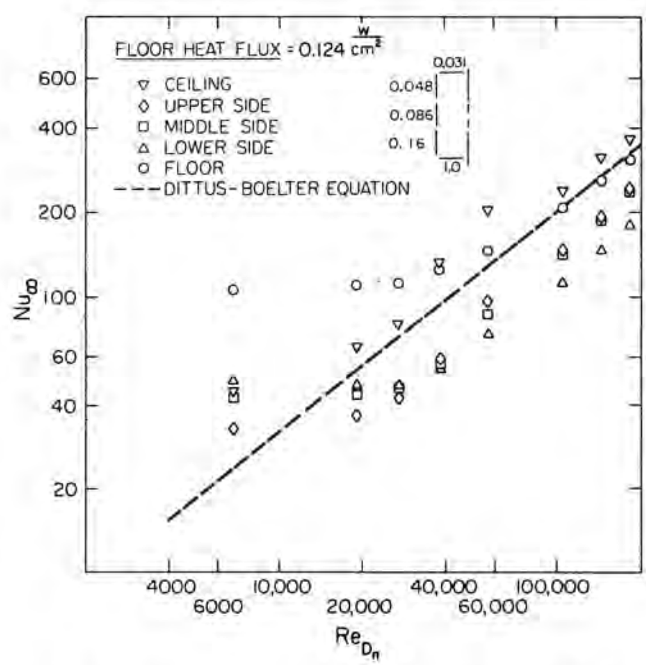

Fig. 5. Nusselt Number vs. Reynolds Number for the Indicated Heat Flux.

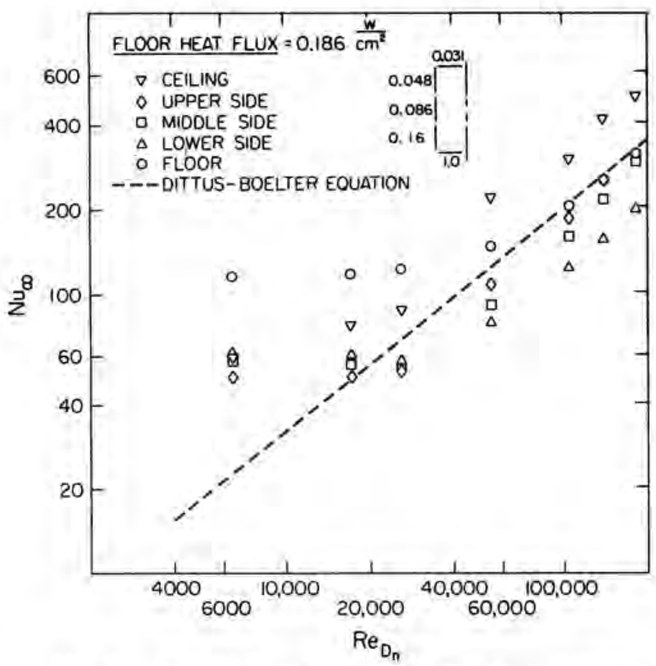

Fig. 6. Nusselt Number vs. Reynolds Number for the Indicated Heat Flux.

From these figures we determined that the actual flux profile in a waste repository depends on the flow rate, or velocity. We used an iterative approach with the HEATING5 finite difference heat conduction code. In the code, the repository tunnel was modeled as a two dimensional slice perpendicular to the tunnel axis, with the tunnel surface nodes defined similar to the experimental surfaces (see Fig. 2). The waste package was modeled as a constant temperature source, and the steady state option of the code was used. Figure 7 shows the HEATING5 model. The boundary conditions were adiabatic everywhere except for the waste package and the tunnel surfaces. The procedure was to choose an arbitrary flux profile for the experimental tunnel. Nusselt numbers were found for the five experimental surfaces. These Nusselt numbers were then input to HEATING5. The heat fluxes of the five surfaces are then calculated by Eq. (4).

$$q = h * (T,wall - T,bulk) \tag{4}$$

where

h = convection coefficient for each of the 5 surfaces input to HEATING5

T,wall = surface temperature calulated by HEATING5

T,bulk = air bulk temperature, from the experiment

From the 5 heat fluxes, a flux profile is calculated. This flux profile is set up on the experimental tunnel. The resulting Nusselt numbers are input to the code, and the procedure is carried out until the flux profile calculated by the code is within a certain percent of the one on the experimental tunnel.

The procedure was done for 2 Reynolds numbers. The results are shown in Table I. The profiles shown indicated a strong dependence of the flux ratio on the flow regime.

Figures 8 through 11 show temperature profiles in the repository for four different cases. Figures 8 and 9 are for Re = 130 000, Fig. 8 using the Dittus-Boelter relation [Eq. (3)], Fig. 9 using the

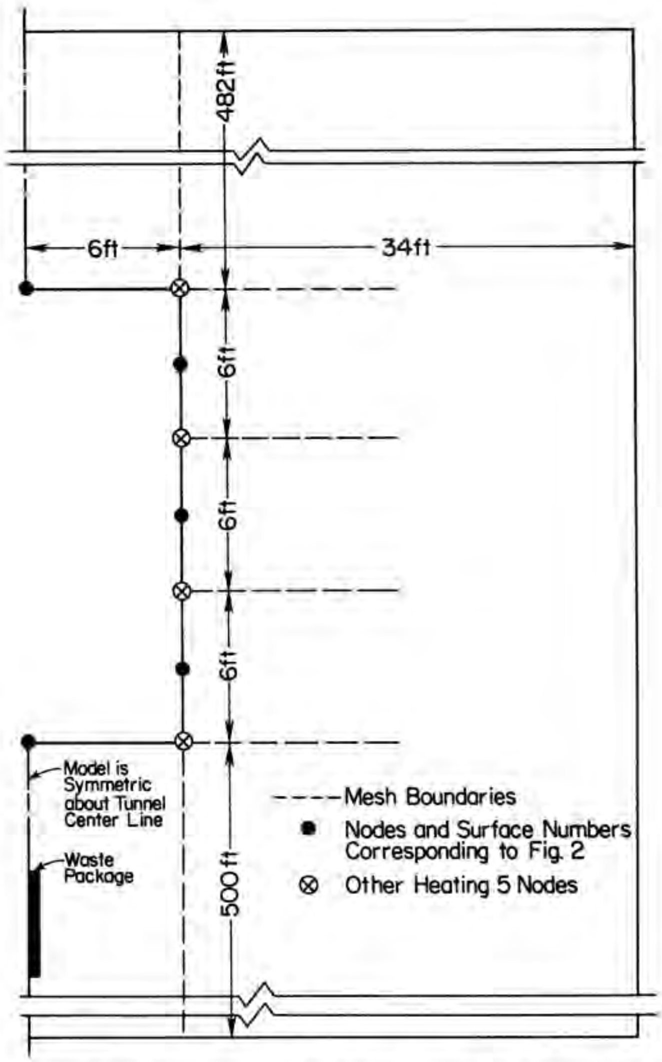

Fig. 7. HEATING5 Model of the REpository Room.

TABLE I

Iterative Heat Flux Ratios and Nusselt Numbers

HEAT FLUX RATIOS

Re	Ceiling	Upper side	Mid side	Lower side	Floor
10 000	0.16	0.41	0.51	0.79	1.0
138 000	0.61	0.50	0.48	0.64	1.0

NUSSELT NUMBERS

Re	Ceiling	Upper side	Mid side	Lower side	Floor	D-B
10 000	22	59	77	111	127	32
138 000	263	243	228	218	258	253

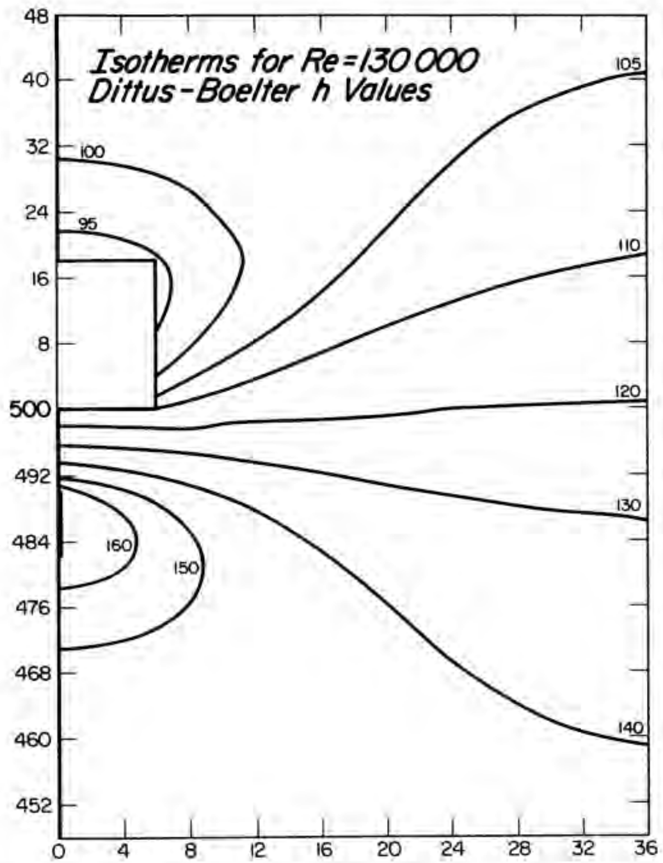

Fig. 8. Temperature Profiles for Re = 130 000 and Nu Numbers From Eq. (3). Temperatures are in Degrees F.

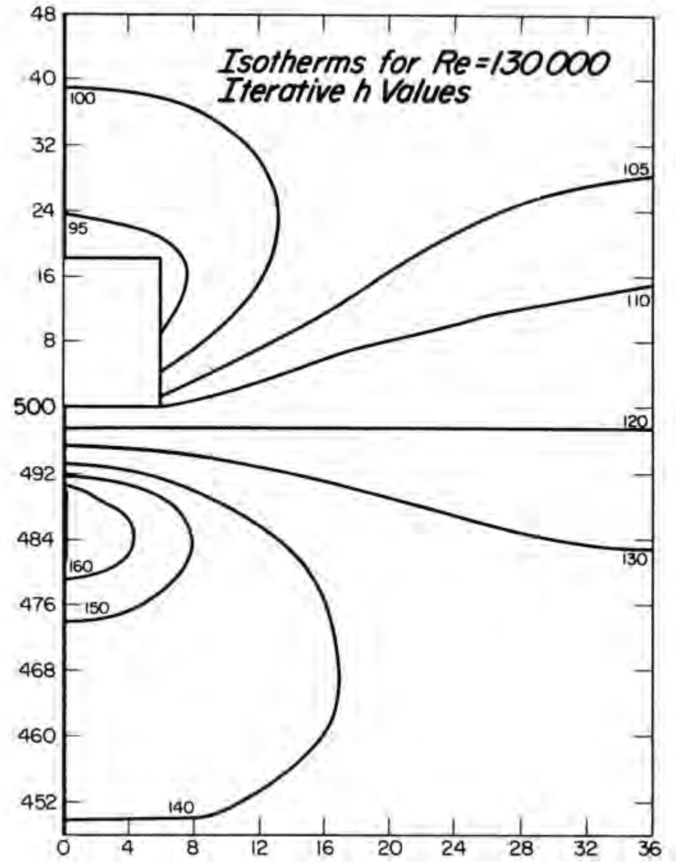

Fig. 9. Temperature Profiles for Re = 130 000 and Iterative Nu Numbers. Temperatures are in Degrees F.

iterative h values. The profiles are very similar, with Fig. 9 being slighty cooler towards the ceiling. Figures 10 and 11 are for Re = 10 000, Fig. 10 using Eq. (3), Fig. 11 using the iterative h values. As seen from the previous Fig. 4, 5, and 6, this Reynolds number is very much in the mixed convection regime. Using purely forced convection at this Reynolds number results in an overestimation of the temperature by as much as 70 C. The effect of mixed convetion is also apparent by comparing Fig. 9 and 11. Even though the Reynolds number, and therefore the flow rate, differs by a factor of 13, the temperature profiles are very close.

SUMMARY

This work presents convective heat transfer coefficients for an experimental apparatus that models a nuclear waste repository storage room. Data are shown for several circumferential heat flux ratios, including a profile expected in a room. These data indicate that for Reynolds numbers below 30 000, the convection coefficient remains essentially constant for the range down to Re = 6 000. If ventilation flow rates are in this range, significant savings can be achieved by choosing the low end of the range.

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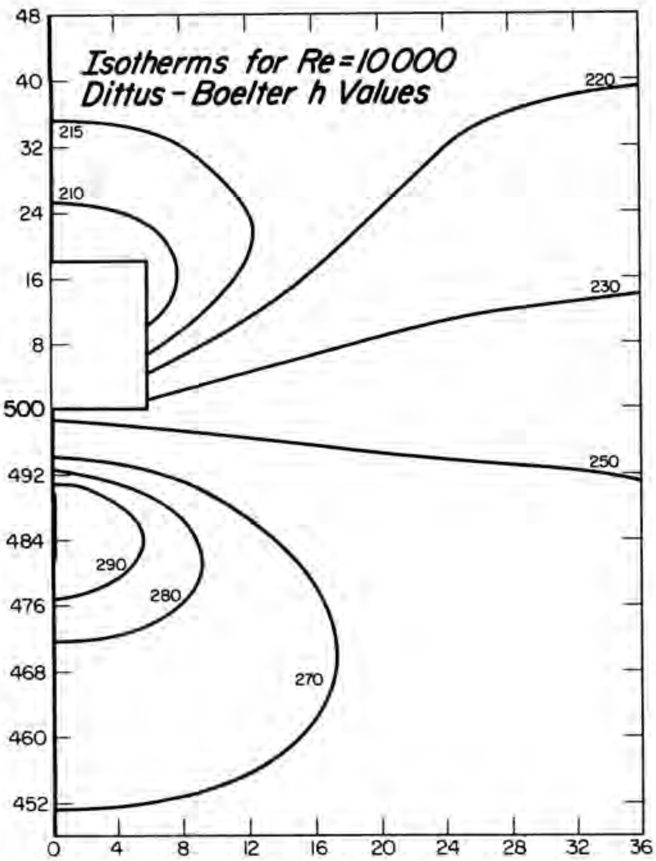

Fig. 10. Temperature Profiles for Re = 10 000 and Nu Numbers from Eq. (3). Temperatures are in Degrees F.

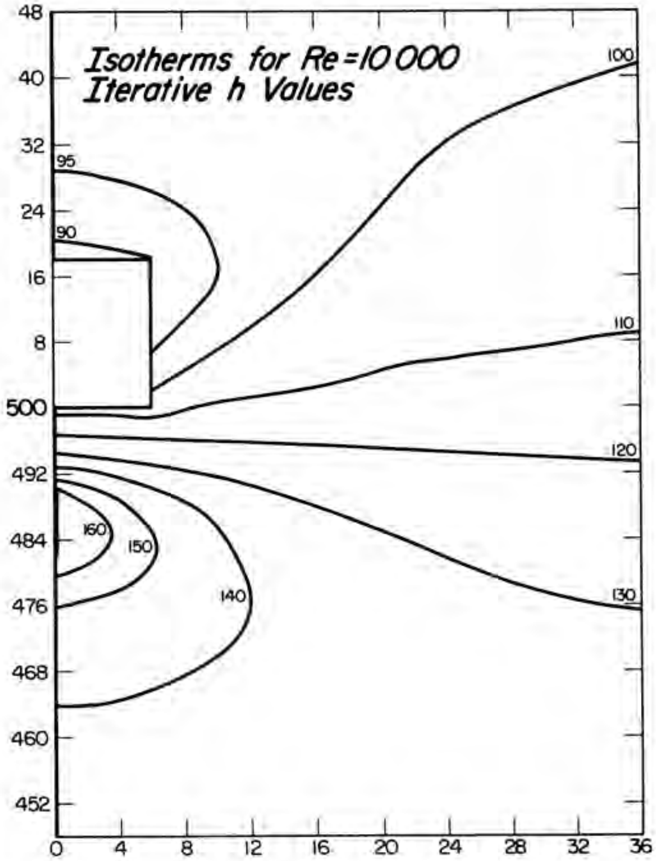

Fig. 11. Temperature Profiles for Re = 10 000 and Iterative Nu Numbers. Temperatures are in Degrees F.

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