AN EXPERIMENTAL DETERMINATION OF MIXED AND FORCED CONVECTION HEAT TRANSFER COEFFICIENTS IN A MODELED NUCLEAR WASTE REPOSITORY

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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 6000 to 180,000, covering both the forced and mixed (combined natural and forced) regimes of convection. Results indicate that the natural convection component is significant. The finite difference heat conduction code HEATING5 was used, with the Nusselt numbers obtained from the experiment, to plot temperatures around the repository room for the cases of Reynolds numbers of 10,000 and 130,000. The plots show that the use of conventional forced convection relations can result in over-estimating the room temperatures by as much as 50°C. Correlations for Nusselt number are presented for the mixed and forced convection regimes.

INTRODUCTION

During the retrievability period for high level nuclear waste and spent fuel, the underground tunnels may remain open or may be backfilled. If the tunnel is kept open during the retrievability period, ventilation will be required for both cooling and air supply for worker safety and comfort. If the drifts were backfilled, and retrieving the waste was required, ventilation would be required prior to actually removing the waste packages, again for worker safety and comfort. In addition, ventilation would be required to cool the walls of the drift down to workable temperatures (approximately 50°C).

To determine the ventilation and cooling requirements of 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 (2,3).

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. The Reynolds and Prandtl numbers of the repository were not difficult to match with the experimental duct. The Grashof number is proportional to the temperature differential between the wall and the air and the cube of a characteristic dimension. The Grashof number of the repository is large primarily from the large height of the drift (4.5 to 6 meters). As the duct was a one twelfth scale model of the drift, the duct Grashof number for the same wall-to-air differential would be $(1/12)^3$ or 1/1728 of the repository Grashof. With an increased temperature differential (as a result of using high temperature heaters), this fraction was reduced to a range of 1/500 to 1/50.

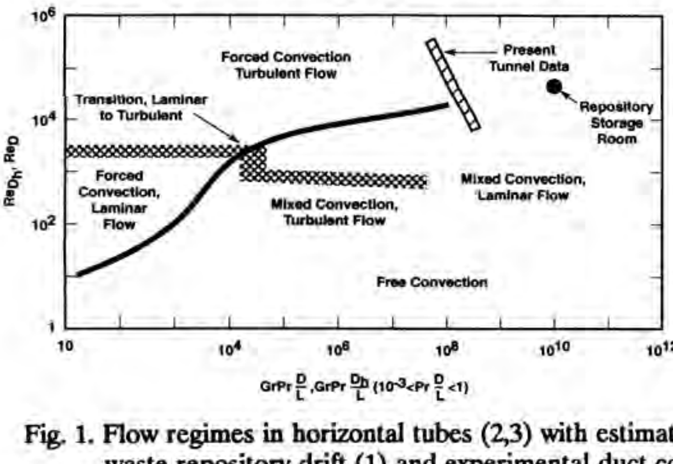

Fig. 1. Flow regimes in horizontal tubes (2,3) with estimated waste repository drift (1) and experimental duct conditions.

EXPERIMENTAL SETUP

The experimental model, or "duct", had a cross section of 0.46 meters high by 0.30 meters wide, and a total heated length of 4.4 meters. These dimensions were chosen so that the aspect ratio of the duct was consistent with preliminary designs of repository rooms. The air flows were in the fully turbulent regime, with Reynolds numbers ranging from 6000 to 180,000. The corresponding velocities were from 1.2 m/sec to 7.8 m/sec. Reynolds numbers were expected to be on the order of 170,000 in the repository (4). Thus, the experimental model's range covered the repository expected Reynolds number.

The four walls of the duct were constructed with 1.6 mm thick galvanized sheet steel, backed by insulation. The wall heat flux was provided by silicon rubber electrical resistance heaters. One circumferential bank of heaters was 0.15 meters wide. Fig. 2 shows the heater bank arrangement. A total of 29 of these circumferential banks were installed for the 4.4 meters of heated length. Each bank consisted of eight heaters arranged as follows: one pad each for the floor (surface #1 in Fig. 2) and ceiling (surface #8); three pads for each sidewall (surfaces #2 through #7). The heaters provided a continuous heat flux axially down the duct. This in effect smeared the discrete heat sources that would actually exist in the repository. Previous work (5,6) showed that this assumption is reasonable for the waste package spacing used here. The heater rows were controlled separately, with opposing sidewall heaters wired in parallel. There were five separately controllable elements per bank: 1) floor, 2) lower sidewall, 3) middle sidewall, 4) upper sidewall, and 5) ceiling. The imposed heat flux profile could be circumferentially uniform or, as in this

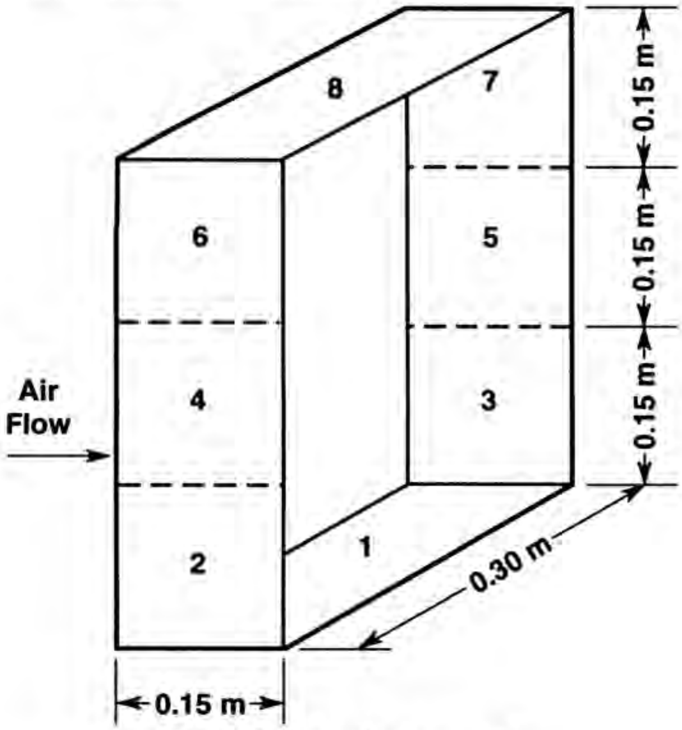

Fig. 2. Heater pad arrangement.

work, the flux profile could be set to match that found in an actual repository drift (higher flux on the floor).

Thermocouples were used to measure air and wall temperatures. Wall temperature measurements showed that the flow was fully developed by heater bank 26 (3.8 meters from the entrance). All results presented here are for this heater bank. Fig. 3 shows a schematic of the experimental system.

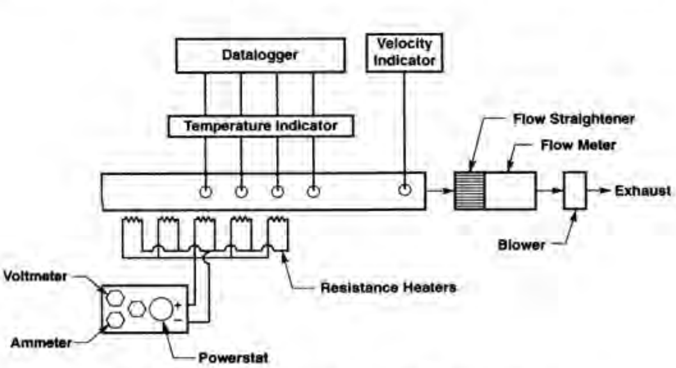

Fig. 3. Experimental system schematic.

EXPERIMENTAL PROCEDURE

The convective heat transfer coefficient was calculated for each of the five heaters. This calculation was made by using the net heat input into each surface, the wall temperature, and the bulk air temperature. The net heat flux was calculated from

$$q_{net} = q_{electric} - q_{loss} - q_{radiation} \qquad \text{(Eq. 1)}$$

where

q_{net} = net heat energy leaving the heater pad

$q_{electrical}$ = electrical power input to the heater pad, measured by power meters

q_{loss} = heat energy leaving the heater pad by conduction through the duct's walls to the room, measured by ΔT across the insulation block

q_{rad} = heat energy leaving the heater pad by thermally radiating to other duct surfaces, calculated from wall surface temperatures using a radiation network scheme (7)

The convective heat transfer coefficient (h), the nondimensional total heat transfer parameter (Nusselt number, Nu), and the natural convection heat transfer parameter (Grashof number, Gr), were calculated for each heater i as follows:

$$h_i = \frac{q_{net,i}}{A_i(T_{wall,i} - T_{bulk})} \qquad \text{(Eq. 2)}$$

where

$T_{wall,i}$ = wall temperature of heater surface (i = floor, lower sidewall, middle sidewall, upper sidewall, or ceiling)

T_{bulk} = bulk air temperature

A_i = area of the heater pad (232 cm^2 for a pair of sidewall heaters or for the floor and ceiling heaters)

$$Nu_i = \frac{h_i D_h}{k_{air}} \qquad \text{(Eq. 3)}$$

where

D_h = hydraulic diameter, defined as 4(duct flow area) ÷ (duct perimeter). With a duct height of 0.46 meters, a duct width of 0.30 meters, D_h = 0.37 meters.

k_{air} = air thermal conductivity, evaluated at the bulk temperature

$$Gr_i = \frac{g\beta(T_{wall,i} - T_{bulk})l^3}{\nu^2} \qquad \text{(Eq. 4)}$$

where

g = gravitational constant, 9.81 m/sec^2

β = volumetric expansion coefficient, evaluated at the bulk temperature

v = kinematic viscosity, evaluated at the bulk temperature

l = characteristic length

The Grashof number characteristic length, 1, of two horizontal, heated, parallel plates is the plate width for the downward facing surface, and the plate spacing for the upward facing surface. For heated, vertical walls, the characteristic length is usually the vertical distance from the bottom edge of the plate. For an enclosure, however, the vertical distance a particular surface "sees" is the distance from that surface to the top of the enclosure. For example, the upper sidewall surface, butted against the ceiling on its upper edge, had a vertical dimension of 0.15 meters, but its midpoint is 0.075 meters from the surface (the ceiling) that is impeding the natural convection flow. This surface was assigned a characteristic length of 0.075 meters. Likewise, the middle sidewall surface also had a vertical dimension of 0.15 meters, but its midpoint is 0.23 meters from the ceiling. Therefore, the middle sidewall surface was assigned a characteristic length of 0.23 meters. These alternative characteristic lengths were developed during work on the correlation presented later. The data could not be correlated well with the more conventional characteristic lengths.

HEATING5 COMPUTER PROGRAM

The computer program HEATING5 (8) was used to model the repository drift. HEATING5 is a finite-difference heat conduction program developed at the Oak Ridge National Laboratory. The program uses a point successive over-relaxation method and a modification of the Aitken δ^2 extrapolation process to solve the finite difference equations.

Previous work (9) indicates that the actual flux profile in a waste repository depends on the flow rate (Reynolds number). To try to determine the actual repository flux profile, an iterative procedure was used. The repository was modeled in HEATING5 as a two dimensional slice perpendicular to the tunnel axis, with the repository tunnel surface nodes defined similar to the experimental surfaces, with a floor, three vertical sidewall surfaces, and a ceiling. The HEATING5 node placement is shown in Fig. 4.

These surfaces were given convective boundary conditions. The tunnel was assumed to be infinite in length. The host material was salt with a temperature dependent thermal conductivity. The tunnel was 5.5 meters high and 3.7 meters wide. The experimental duct then was one-twelfth scale of the repository tunnel. Symmetry was assumed along the tunnel centerline, where an adiabatic boundary condition existed, except at the waste package location. The waste package was modeled as a constant heat flux source 2.4 meters long at the tunnel centerline, 3 meters below the floor. The gross thermal loading was considered to be 50 kW/acre (1). The source strength in the waste package plane was then 60 watts/sq.m. As the model was two dimensional, this strength was continuous in the axial direction; that is, the individual packages were not modeled. This assumption has been shown (5,6) to be reasonable for the waste package spacing used here. This was also the case for the experimental tests. Both the top and bottom boundaries were assumed to be adiabatic.

EXPERIMENT / HEATING5 HEAT FLUX PROFILE ITERATION

An iterative procedure using the HEATING5 conduction program and the experimental tunnel was used to predict the convective heat transfer coefficients and the host rock temperature profile. The procedure was to choose an arbitrary flux profile for the experimental tunnel. Convection coefficients were found for the five experimental surfaces. These experimental convection coefficients were converted to repository convection coefficients by assuming similarity, that is:

$$Nu_{\text{experiment}} = Nu_{\text{repository}} \qquad \text{(Eq. 5)}$$

As stated previously, the experiment was a one-twelfth scale of the repository, therefore

$$D_{h(\text{repository})} = 12 D_{h(\text{experiment})} \qquad \text{(Eq. 6)}$$

Using the definition of the Nusselt number and Eq. (6) and equating the thermal conductivity of the experimental air and the repository air, Eq. (5) may be solved for $h_{\text{repository}}$:

$$H_{\text{repository}} = \frac{h_{\text{experiment}}}{12} \qquad \text{(Eq. 7)}$$

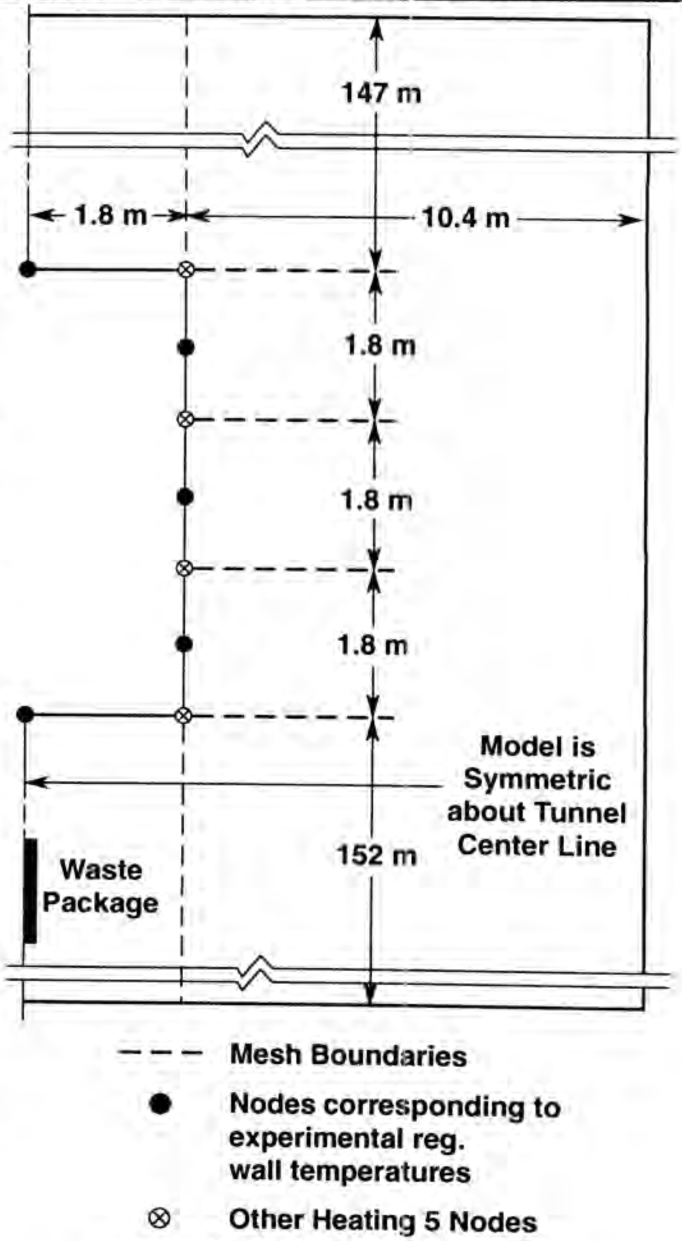

Fig. 4. HEATING5 repository model.

The five convection coefficients calculated from Eq. (7) were input to the HEATING5 model, and the host rock nodal temperatures were calculated. The five heat flux values were found from the calculated rock tunnel surface temperatures and the bulk air temperature.

$$q''_i = h_i \, (T_{\text{wall},i} - T_{\text{bulk}}) \qquad \text{(Eq. 8)}$$

From the five q''_i values, the flux profile (with respect to the floor flux) was calculated. This new flux profile was run in the experimental duct, and a new set of convection coefficients was determined. This completed one iteration. The procedure was repeated until the flux profile calculated by HEATING5 was within 3 percent of the profile that was input to the experiment. Figure 5 schematically shows the procedure.

RESULTS

The iteration was performed for two Reynolds numbers, 10,000 and 130,000. From earlier work (9), these two Reynolds

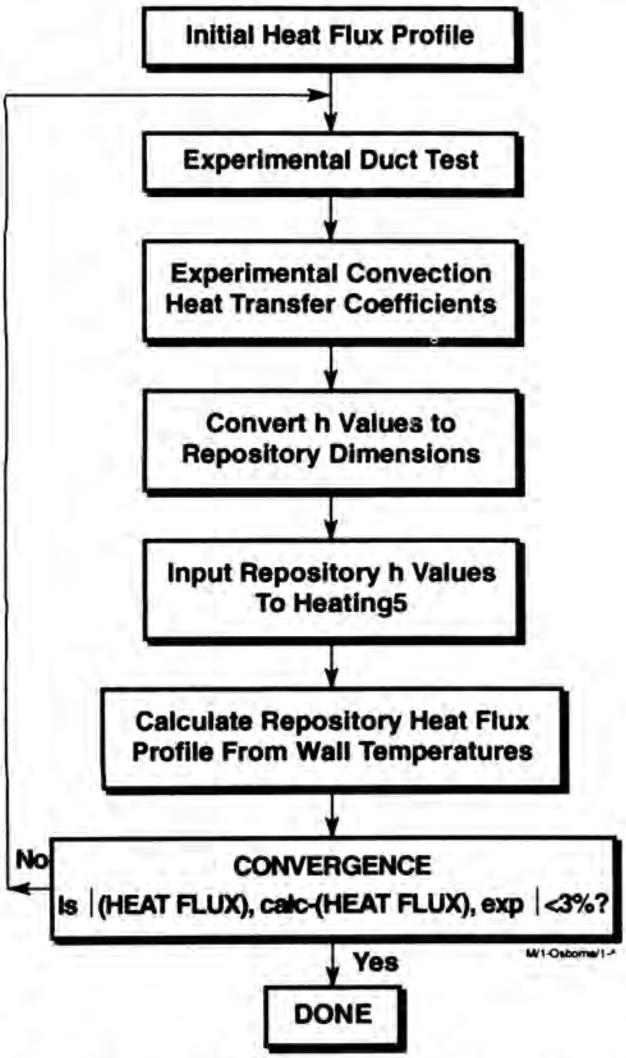

Fig. 5. HEAT5/experiment iteration for heat flux determination.

numbers are in the mixed convection and forced convection regimes, respectively. Table I shows the heat flux profiles at each step of the iterative process for both flow rates. Six steps were required for convergence of the heat flux profile for the Reynolds number of 10,000, and five steps were required for convergence for the Reynolds number of 130,000. The heat transfer results are shown in Table II. The Nusselt numbers shown indicate a strong dependence of the flux ratio on the flow regime.

Figures 6 through 9 show temperature profiles in the repository for four different cases of heat transfer coefficients. Figures 6 and 7 show the drift isotherms for Re = 130,000. For Fig. 6, the Dittus-Boelter correlation (10) was used for the heat transfer coefficients for all surfaces. Figure 7 shows drift isotherms using the iterative h values. The profiles are very similar, with Fig. 7 being slightly cooler toward the ceiling. Figs. 8 and 9 are for Re = 10,000, Fig. 8 using the Dittus-Boelter h values, Fig. 9 using the iterative h values. Re = 10,000 is very much in the mixed convection regime. Using purely forced convection relations at this Reynolds number results in an over-estimation of the floor temperature by as much as 50°C. The effect of mixed convection is also apparent by comparing Figs. 7 and 9. Even though the Reynolds number,

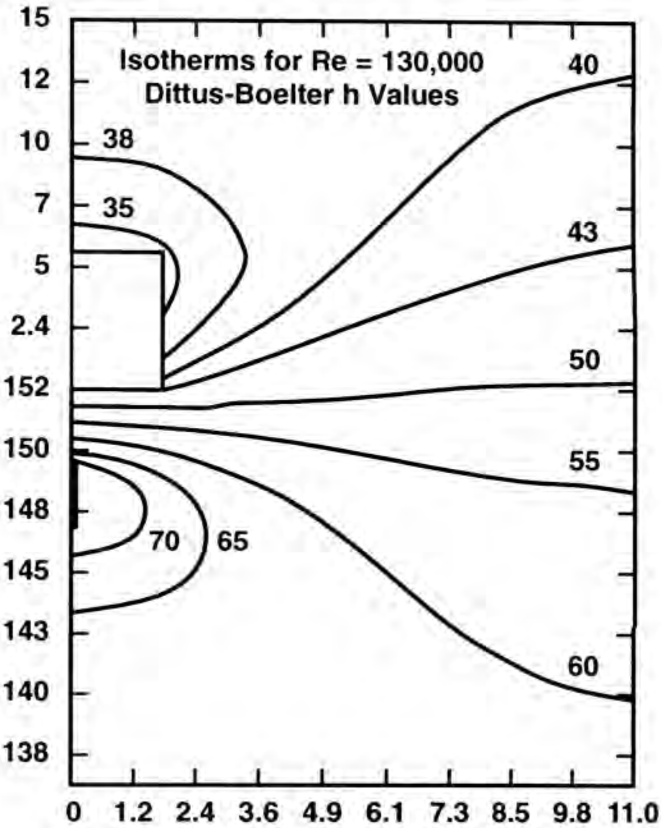

Fig. 6. HEATING5 host rock isotherms using the Dittus-Boelter (10) Nusselt number for Re = 130,000.

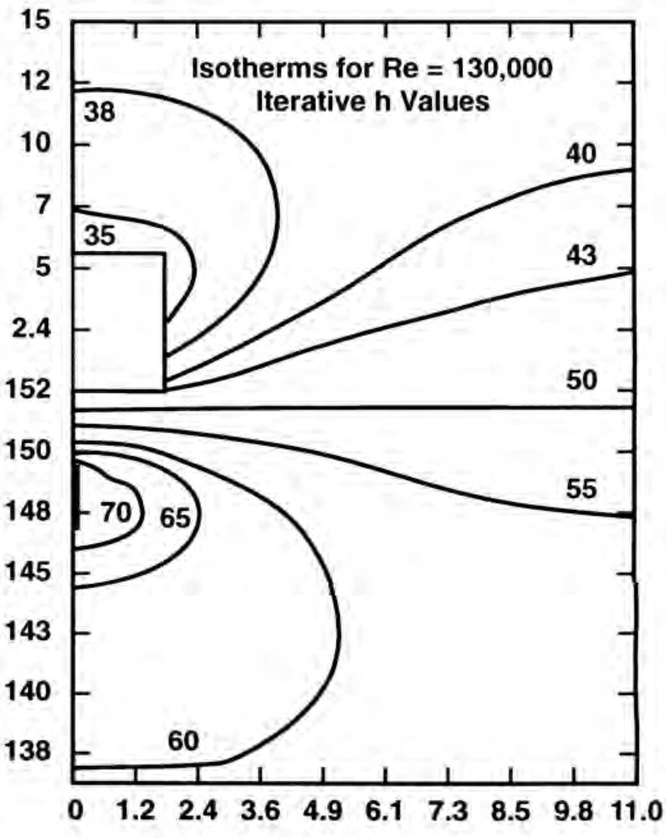

Fig. 7. HEATING5 host rock isotherms using the iterative Nusselt number for Re = 130,000.

TABLE I
HEATING5 / Experiment Iteration Heat Flux Values

Re = 10,000						
Iteration No.	1	2	3	4	5	Final Profile
Ceiling	0.28	0.17	0.17	0.16	0.13	0.13
Upper Sidewall	0.49	0.42	0.41	0.43	0.38	0.38
Middle Sidewall	0.56	0.55	0.55	0.58	0.50	0.50
Lower Sidewall	0.67	0.78	0.79	0.78	0.78	0.79
Floor	1.0	1.0	1.0	1.0	1.0	1.0
Re = 130,000						
Iteration No.	1	2	3	4	Final Profile	
Ceiling	0.56	0.86	0.76	0.62	0.61	
Upper Sidewall	0.53	0.50	0.51	0.49	0.50	
Middle Sidewall	0.47	0.48	0.50	0.49	0.48	
Lower Sidewall	0.54	0.46	0.58	0.63	0.64	
Floor	1.0	1.0	1.0	1.0	1.0	

TABLE II
Nusselt Numbers From the HEATING5/ Experimental Iteration

	Re = 10,000	Re = 130,000
Ceiling	22	263
Upper Sidewall	59	243
Middle Sidewall	77	228
Lower Sidewall	111	218
Floor	127	258
Dittus-Boelter	32	25

and therefore the flow rate, differs by a factor of 13, the temperature profiles are closer than would be expected by forced convection analyses.

Four correlations were developed which predict the Nusselt number on a tunnel surface for a given set of flow conditions. The correlations are linear regression fits of the experimental Nusselt number as a function of the Reynolds and Grashof numbers for each of the five duct surfaces. The performance of the ceiling was sufficiently different from the other four surfaces that a separate correlation is given for it. Eq. (9) for the floor and sidewalls, and Eq. (10), for the ceiling, correlate the experimental data for low Re numbers (less than 58,000) in the mixed convection regime. Eq. (10) uses the width of the duct as the characteristic dimension, as is common for a horizontal heated plate facing downward. Eq. (11) and (12) treat the cases of higher Re numbers in the forced convection regime (Re greater than 85,000). From earlier work (9), for the range of Re greater than 85,000, the Nusselt numbers of the floor and sidewalls should not have been a function of the Grashof number. Indeed, the regression analysis showed this with a exponent of zero for the Grashof number in this range.

For the floor, lower, middle, and upper sides, low Reynolds range,

$$Nu = 4.20Re^{0.172}Gr^{0.077}HFR^{0.101} \qquad \text{(Eq. 9)}$$

For the ceiling, low Reynolds range,

$$Nu = 0.00232Re^{1.02}Gr^{-0.0154}HFR^{-0.204} \qquad \text{(Eq. 10)}$$

where

Nu = Nusselt number for the duct surface, based on D_h

Gr = Grashof number for the duct surface, based on distance from ceiling for Eq. (9), based on the duct width for Eq. (10)

Re = Reynolds number, based on D_h

HFR = Heat flux ratio for the duct surface = surface heat flux / floor heat flux

Eqs. (9) and (10) are valid for $6400 < Re < 58{,}000$ and $10^5 < Gr < 10^9$. The average errors for Eqs. (9) and (10) are 8.5 and 8.3 percent, respectively.

For the floor, high Reynolds range,

$$Nu = 0.0481Re^{0.7248}HFR^{-0.050} \qquad \text{(Eq. 11)}$$

For the ceiling, high Reynolds range,

$$Nu = 0.0391Re^{0.748}HFR^{-0.0509} \qquad \text{(Eq. 12)}$$

Eqs. (11) and (12) are valid for $85{,}000 < Re < 186{,}000$ and $10^5 < Gr < 10^9$. The average errors for Eqs. (11) and (12) are 5.6 and 3.8 percent, respectively.

CONCLUSIONS

Correlations are presented for the Nusselt numbers for the five surfaces of the duct over the Reynolds number range of 6400 to 186,000. Mixed convection correlations are

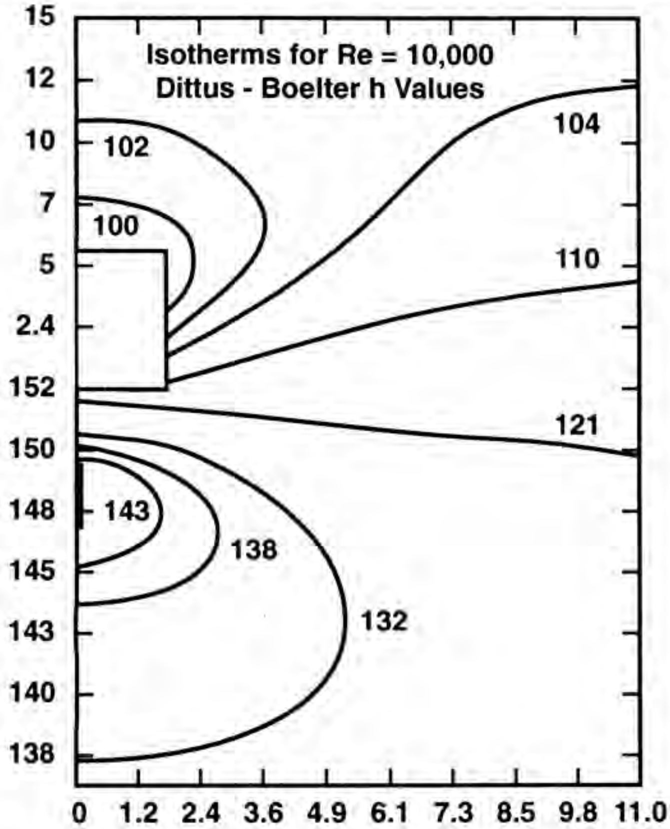

Fig. 8. HEATING5 host rock isotherms using the Dittus--Boelter (10) Nusselt number for Re = 10,000.

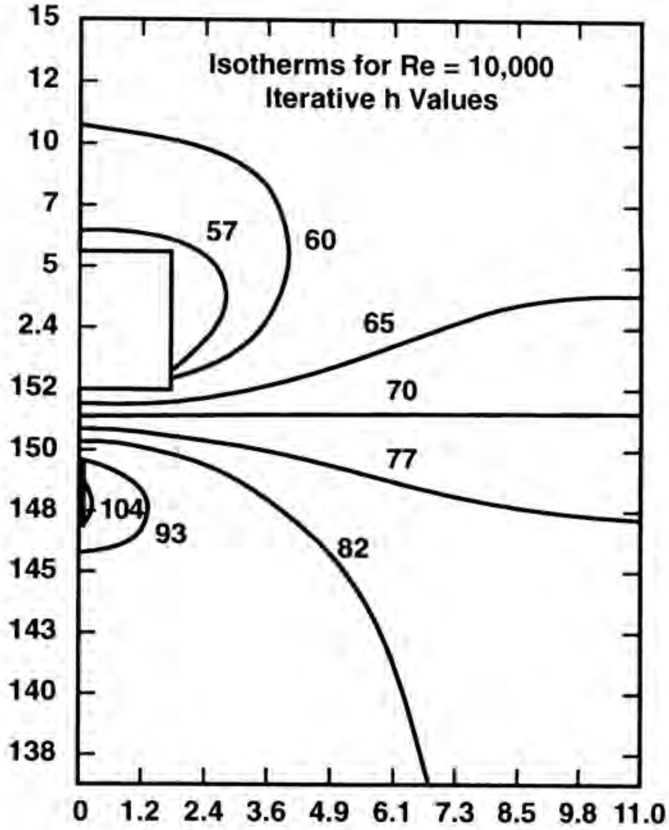

Fig. 9. HEATING5 host rock isotherms using the iterative Nusselt number for Re = 10,000.

applicable below a Reynolds number of 58,000 where the Nusselt number is a function of the Reynolds number, the Grashof number, and the surface heat flux ratio. Forced convection correlations are applicable above a Reynolds number of 85,000 where the Nusselt number is a function of the Reynolds number and the surface heat flux ratio.

These correlations are not as straight forward to use as the typical forced convection correlations, which involve the flow as the only independent variable. The correlations shown here have the heat flux ratio as an independent variable. This ratio, however, is a function of the host rock temperature profile, which in turn is dependent on the heat transfer from the tunnel surface. The mixed convection regime correlations also include the Grashof number, so the surface temperature must also be known. Again, this temperature is a function of the heat transfer from the surface. The use of these correlations is therefore an iterative process, as is any process involving mixed convection. A heat flux profile would be assumed, along with surface temperatures. The convection coefficient is calculated from the correlations using these assumptions. This convection coefficient is used to calculate the heat transfer from the tunnel surfaces, and thus the surface temperatures. The procedure is continued until the assumed parameters are within some tolerance of the calculated values.

The data presented here indicate that for Reynolds numbers below 30,000, the convection coefficient remains essentially constant for the range down to a Reynolds number of 6000. If repository ventilation flow rates are in this range, the same heat transfer can be achieved with a reduced flow rate. If the flow rate were reduced by a factor of five, pumping power could be reduced by a factor of 25. Thus, significant cost savings could be realized if these mixed convection results are used in place of the normally used forced convection analyses.

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