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CONVECTION AND THERMAL RADIATION ANALYTICAL MODELS APPLICABLE TO A NUCLEAR WASTE REPOSITORY ROOM

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MASTER

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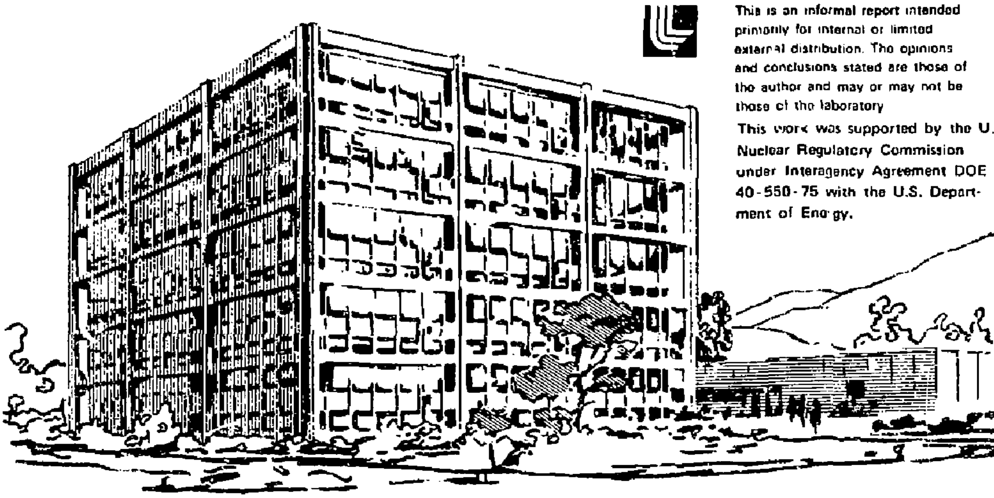

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NOMENCLATURE

A_i	-	Area of surface-i
E_{bi}	-	Black body emissive power of surface-i
F_{ij}	-	View factor surface-i to surface-j
g	-	Local acceleration due to gravity
Gr	-	Grashof number
h	-	Convection coefficient
h_s	-	Surface coefficient
J_i	-	Radiosity of surface-i
k	-	Thermal conductivity
K_{eq}	-	Equivalent conductivity
L	-	Characteristic dimension
Nu	-	Nusselt number
Pr	-	Prandtl number
q_{rad}	-	Heat transfer rate, radiation
q''_{rad}	-	Heat flux, radiation
q''_{conv}	-	Heat flux, convection
Re	-	Reynolds number
T_c	-	Ceiling temperature
T_f	-	Floor temperature
T_w	-	Vertical wall temperature
T_s	-	Surface temperature
T_∞	-	Free stream temperature
U	-	Average flow velocity
α	-	Thermal diffusivity
β	-	Volumetric thermal expansion coefficient
ν	-	Kinematic viscosity
ε	-	Surface emissivity

ABSTRACT

Time-dependent temperature distributions in a deep geologic nuclear waste repository have a direct impact on the physical integrity of the emplaced canisters and on the design of retrievability options. This report (1) identifies the thermodynamic properties and physical parameters of three convection regimes--forced, natural, and mixed; (2) defines the convection correlations applicable to calculating heat flow in a ventilated (forced-air) and in a nonventilated nuclear waste repository room; and (3) delineates a computer code that (a) computes and compares the floor-to-ceiling heat flow by convection and radiation, and (b) determines the nonlinear equivalent conductivity tables for a repository room. (The tables permit the use of the ADINAT code to model surface-to-surface radiation and the TRUMP code to employ two different emissivity properties when modeling radiation exchange between the surface of two different materials.) The analysis shows that thermal radiation dominates heat flow modes in a nuclear waste repository room.

INTRODUCTION

This report defines the heat transfer models/correlations that are applicable to calculating heat flow through the repository room space for both ventilated (forced-air) and nonventilated rooms, summarizes the work of the original investigators which led to the development of the heat transfer models, compares the relative magnitudes of convection and thermal radiation heat transfer through a repository room, and develops and uses nonlinear equivalent conductivity tables for modeling combined convection and radiation from floor-to-ceiling in a repository room. These tables allow codes such as ADINAT to model surface-to-surface radiation, and also provide a means for codes such as TRUMP to employ two different emissivity properties when modeling radiation exchange between two surfaces of differing materials.

Figure 1 shows a conceptual model for a nuclear waste repository[1] in deep geologic media. It consists of a series of long horizontal rooms, mined in a layer of bedded salt. The rooms are approximately 18 by 18 ft in cross-section. The high level waste* canisters are stowed in drilled holes equally spaced in the floor of the rooms. The bore holes are back-filled and plugged following the canister's emplacement. The tops of the emplaced canisters are approximately 10 ft below floor level.

The physical integrity of the waste canisters and the design of retrievability options are expected to be directly impacted by the time-dependent temperature distributions. The presence of a large air space above the emplaced waste canisters in a deep geologic nuclear waste repository produces significant effects on the general time-dependent temperature distribution in the geologic media.[2]

*"High level waste" as used in this report refers to both spent fuel and solidified high level waste.

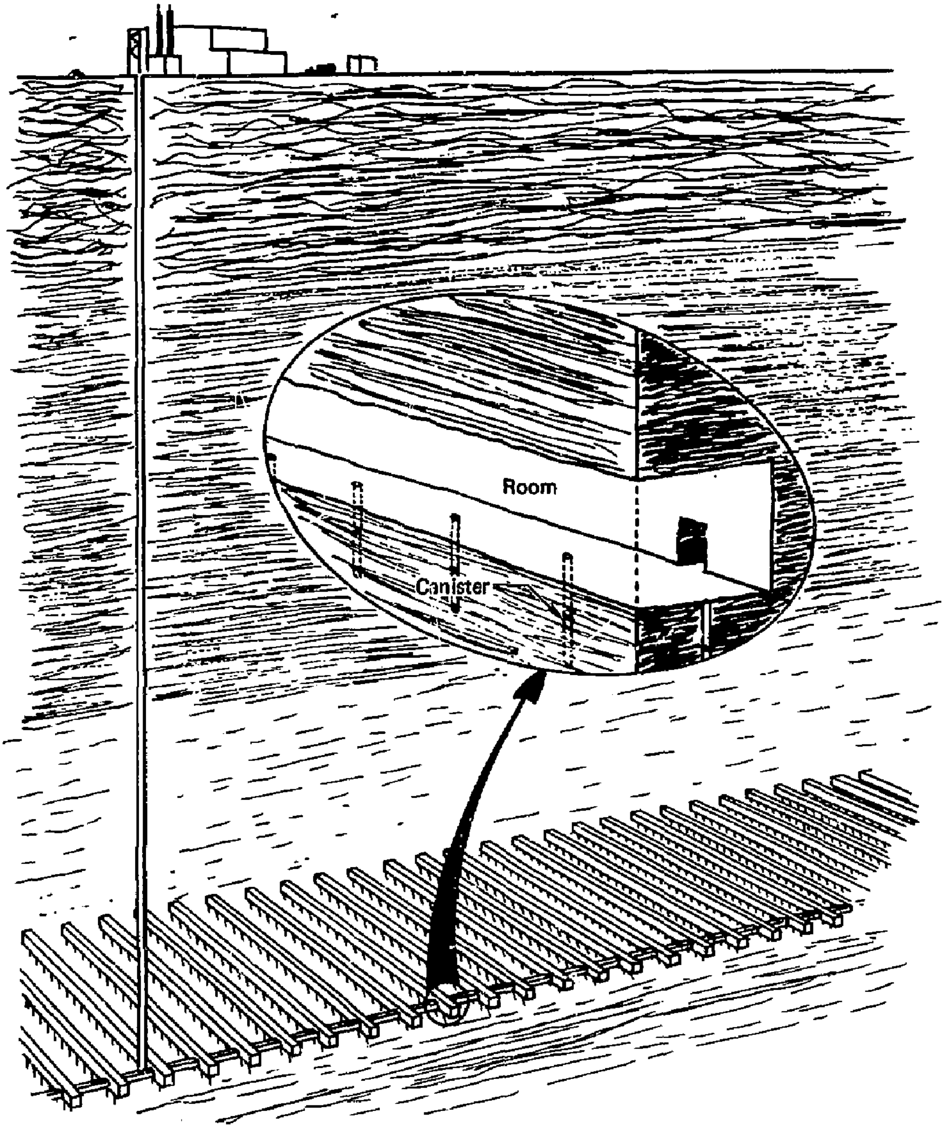

FIG. 1. Conceptual model of a deep geologic nuclear waste repository.

The extent of the effects depend on whether the rooms are ventilated,* and if so, at what flow rate and air temperature. The analysis shows that thermal radiation dominates heat flow modes in repository rooms, and that the air in the room is a more effective medium for heat transfer than salt.

*A "ventilated" room is one which is cooled by means of an externally powered air moving system.

CONVECTION MODELS

Three broad convection regimes exist: forced, natural, and mixed.[3,4] Mass flow of the convecting fluid is the mechanism by which heat is transported. Mass flow may be caused by externally induced pressure forces (forced convection) or may be induced by buoyant forces resulting from density gradients (natural convection). Sometimes, conditions exist where the forced convection effects are of the same order of magnitude as the natural convection effects; this is commonly called mixed convection.

Dimensionless groups of thermodynamic properties and physical parameters are almost always employed in development of convection heat transfer correlations. Most forced convection correlations are expressed in the form[4]:

$$Nu = f(Re, Pr) \quad . \tag{1}$$

Most natural convection correlations are expressed in the form[4]:

$$Nu = f(Gr, Pr) \quad , \tag{2}$$

where

Gr = Grashof number = $g\beta(T_s - T_\infty)L^3/\nu^2$,
Re = Reynolds number = UL/ν,
Pr = Prandtl number = ν/α, and
Nu = Nusselt number = hL/k.

CRITERION FOR MIXED CONVECTION

Mixed convection is considered "aiding" when the forced and natural convection effects reinforce each other to produce a higher surface velocity or "opposing" when the two convection effects oppose each other.[3] When aiding exists, the heat transfer coefficient is greater than either the forced component or the natural component for intuitively obvious reasons; when opposing exists, the resulting convection coefficient is less than the greater of the two components considered alone.

Gebhart[3] demonstrated by order of magnitude analysis of the Navier-Stokes equation that mixed convection exists when

$$\frac{Gr}{Re^2} \cong 1.0 \quad .$$

Holman[4] suggests the criterion that "when $(Gr/R_e{}^2)>1.0$ free convection is of primary importance." Several investigators, including Metais and Eckert[5] and Mori and Futagami,[6] have confirmed the above criterion. Metais and Eckert published a correlation for mixed convection turbulent flow which shows the convection coefficient to be very weakly dependent on the temperature difference between the surface and the free stream of the fluid. (When air is the convecting fluid, convection coefficients generally have weak temperature dependence in forced flow regimes and strong temperature dependence in natural flow regimes, as can be seen by comparing Eqs. (1) and (2).) It should be noted that the Metais and Eckert relationship correlated experimental data for the following range of (Gr/Re^2)-ratios:

$$0.49 < (Gr/Re^2) < 7.6 \quad .$$

Lloyd and Sparrow[7] showed that, for mixed flow, purely natural convection correlations accurately described the heat flow from a vertical surface when

$$Gr/Re^2 > 1 \quad .$$

Mori[8] investigated combined convection effects on a horizontal heated surface and showed that natural convection dominated the heat transfer process when

$$Gr/Re^{2.5} > 0.083 \quad .$$

Ozisik[9] summarized the results of several other investigators of mixed convection heat transfer.

CONVECTION CORRELATIONS APPLICABLE TO A VENTILATED ROOM

Previous investigators[10-12] have modeled air flow rates of 10,000 cfm per room. Average bulk air temperature was assumed to be about 80°F. For these conditions, typical floor-to-air temperature differences were about 35°F. For these conditions

$$Gr/Re^{2} = 150$$

and

$$Gr/Re^{2.5} = 0.64 \quad .$$

With reference to the work cited in the previous section (Criterion for Mixed Convection), natural convection dominates the convective heat transfer from room surfaces for ventilation conditions similar to those described above. For a repository room of cross-section 18 by 18 ft (or larger) having a bulk flow velocity of 1.0 fps (or less) and a ΔT^{*} of 10°F (or greater) the primary influence of forced flow on heat transport is that of moving fresh air in while moving out bulk air which has been heated within the room by the natural convection mechanism.

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*ΔT is the mean temperature difference between the room surface and the bulk air.

Beginning with correlations in the form of Eq. (2), McAdams[13] reduced the convection coefficient for air transfering heat from flat surfaces to the simplified forms shown below. They are applicable to heated surfaces where the Grashof number is greater than 10^9. (Repository room surfaces have associated Grashof numbers of approximately 10^{11}.) For a ventilated room the following temperature-dependent convection coefficient models appear to be most applicable:

From floor: $h = 0.22\ (T_f - T_\infty)^{1/3}$ (3)

From ceiling: $h = 0.068\ (T_c - T_\infty)^{1/4}$ (4)

From walls: $h = 0.19\ (T_w - T_\infty)^{1/3}$, (5)

where temperatures are in ^oF and h is in Btu hr-ft^2-^oF.

CONVECTION CORRELATIONS APPLICABLE TO A NONVENTILATED ROOM

For this case heat will flow from the floor to the ceiling in the manner described by Gebhart[3] and Holman[4] under the topic heading enclosed spaces. Both Holman and Gebhart summarized the experimental work of Jacob[14,15] which resulted in several empirical correlations for heat transfer through enclosed spaces. The nonventilated repository room clearly represents a horizontally enclosed space in which the floor is hotter than the ceiling. For such horizontally enclosed spaces heat will flow from floor-to-ceiling aided by buoyancy effects according to the following:

$$q'' = h_s\ (T_f - T_c) \quad . \tag{6}$$

The heat transfer coefficient, h_s, is considered to be a surface coefficient (instead of the more conventional convection coefficient) because the temperature potential is defined between the two surfaces that form the boundaries of the enclosed space.

The characteristic dimension, L, for horizontally enclosed spaces is the floor-to-ceiling distance. The Grashof number for enclosed spaces is calculated using the above characteristic dimension as shown in Eq. (7):

$$Gr = \frac{g\beta(T_f - T_c)L^3}{\nu^2} \quad . \tag{7}$$

Jacob's correlation for air between horizontally enclosed plane surfaces where the lower surface is at a higher temperature than the upper surface is

$$Nu = 0.068 \sqrt[3]{Gr} \, , \tag{8}$$

where

$$(Gr > 4 \times 10^5) \quad .$$

Using properties of 80^oF air, Eq. (8) reduces to:

$$h_s = 0.129(T_f - T_c)^{1/3} \quad . \tag{9}$$

Because the coefficient in Eq. (9) incorporates the temperature-dependent properties of the convecting air, its magnitude should be adjusted for average air temperature conditions according to the table below.

TABLE 1. Table of coefficients.

Average temperature	Coefficient
80^oF	0.129
150^oF	0.124
250^oF	0.118
350^oF	0.112
450^oF	0.106

THERMAL RADIATION

The heat transfer rate via thermal radiation between surfaces of the repository room will likely be considerably greater than the heat flow by convection. Holman[4] developed analytic models for radiation heat exchange between diffuse surfaces, using an electrical circuit analogy. Figure 2 illustrates the circuit for two surfaces exchanging heat.

FIG. 2. Electrical analogy for two surfaces exchanging thermal radiation.

The net heat transfer rate from surface 1 to surface 2 is

$$q_{rad} = \frac{\sigma(T_1^4 - T_2^4)}{(1 - \varepsilon_1)/\varepsilon_1 A_1 + 1/A_1 F_{12} + (1 - \varepsilon_2)/\varepsilon_2 A_2} \; . \tag{10}$$

For a given geometry one can determine an equivalent radiation conductivity for an air space by equating the finite difference form of Fourier's Law of Conduction with Eq. (10)

$$K_{eq} = \frac{q_{rad} L}{A\,(T_1 - T_2)} \; . \tag{11}$$

A set of nonlinear conductivity values which are a function of the surface temperatures can be developed by this process for use in large thermal codes such as ADINAT,[16] which do not have built-in surface-to-surface radiation exchange modeling capability.

COMPARISON OF CONVECTIVE HEAT FLOW WITH RADIATION HEAT FLOW

It is worthwhile to compare relative magnitudes of convection and radiation heat transfer within a repository room. Employing the relationships described in the preceding sections, a small computer code was developed for the purpose of easily comparing the floor-to-ceiling heat flow by convection and radiation. The code also computes the equivalent conductivity of the room space for nonventilated conditions. The equivalent conductivity embodies both convection and radiation contributions.

For a nonventilated room, the separate contributions of convection and radiation to the floor-to-ceiling heat transfer is illustrated in Figs. 3, 4, and 5 for temperature differences of $5^{O}F$, and $10^{O}F$. (Work by Altenbach[2] and Davis[1] indicate that the temperature difference between floor and ceiling ranges from $2^{O}F$ to $10^{O}F$ for both ventilated and nonventilated rooms.) The data plotted are based on a floor-to-ceiling view factor of 0.41 and a constant surface emissivity of 0.9. The plots clearly show that for any reasonable floor temperature, thermal radiation is the dominant heat flow mode.

A plot of equivalent conductivity of the airspace is shown in Fig. 6 for the three different ΔT's as a function of floor temperature.

An interesting, and useful, fact is observed in this plot: although the ΔT ranges from $5^{O}F$ to $20^{O}F$ all three curves fall quite closely together. This means that even though the floor-to-ceiling ΔT changes with time, the nonlinear conductivity set remains essentially unchanged. From this plot one can scale off a nonlinear conductivity set which could be used, for example, with ADINAT.

It is also significant that the equivalent conductivity of the room space is large, compared with salt. Furthermore, the equivalent room space conductivity increases dramatically as the floor temperature increases. As can be observed by the dashed curve in the lower part of the graph in Fig. 6,

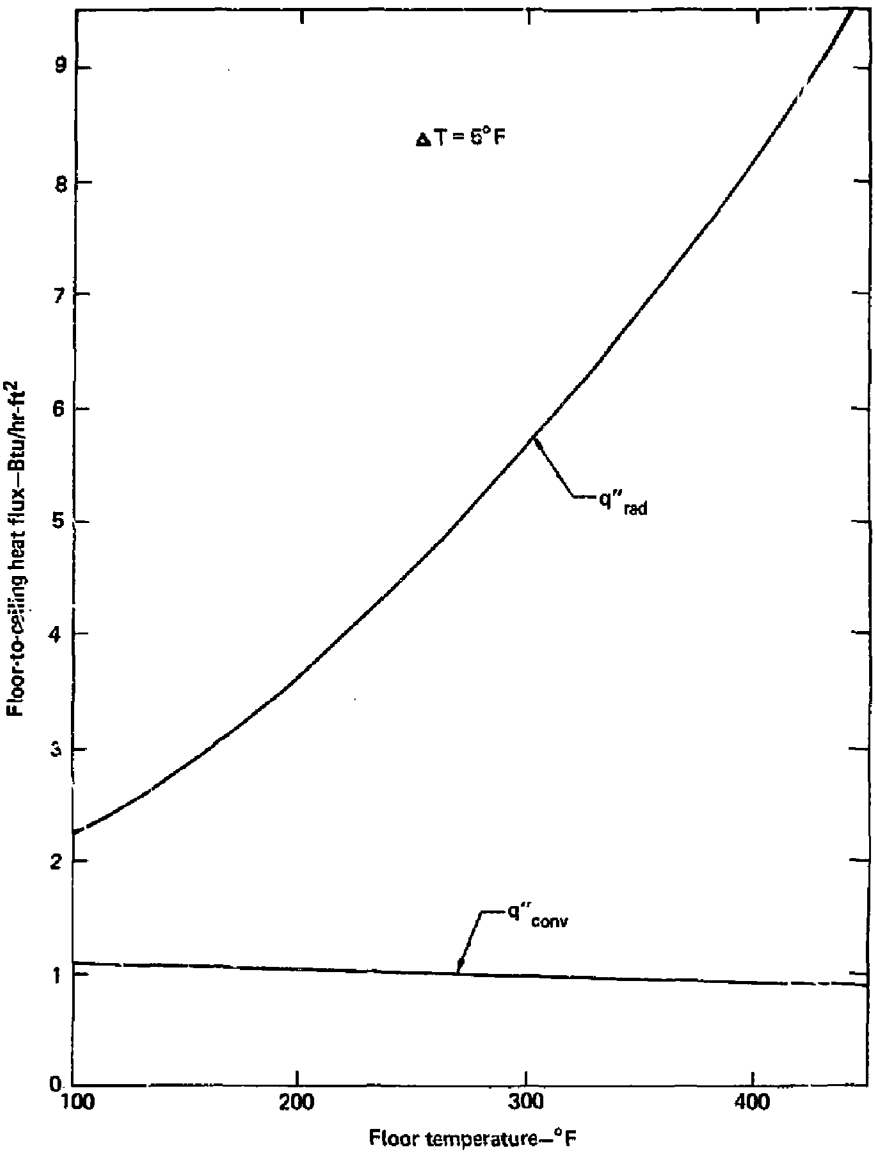

FIG. 3. Radiation heat flux from floor-to-ceiling of a repository room ($\Delta T = 5^{0}F$).

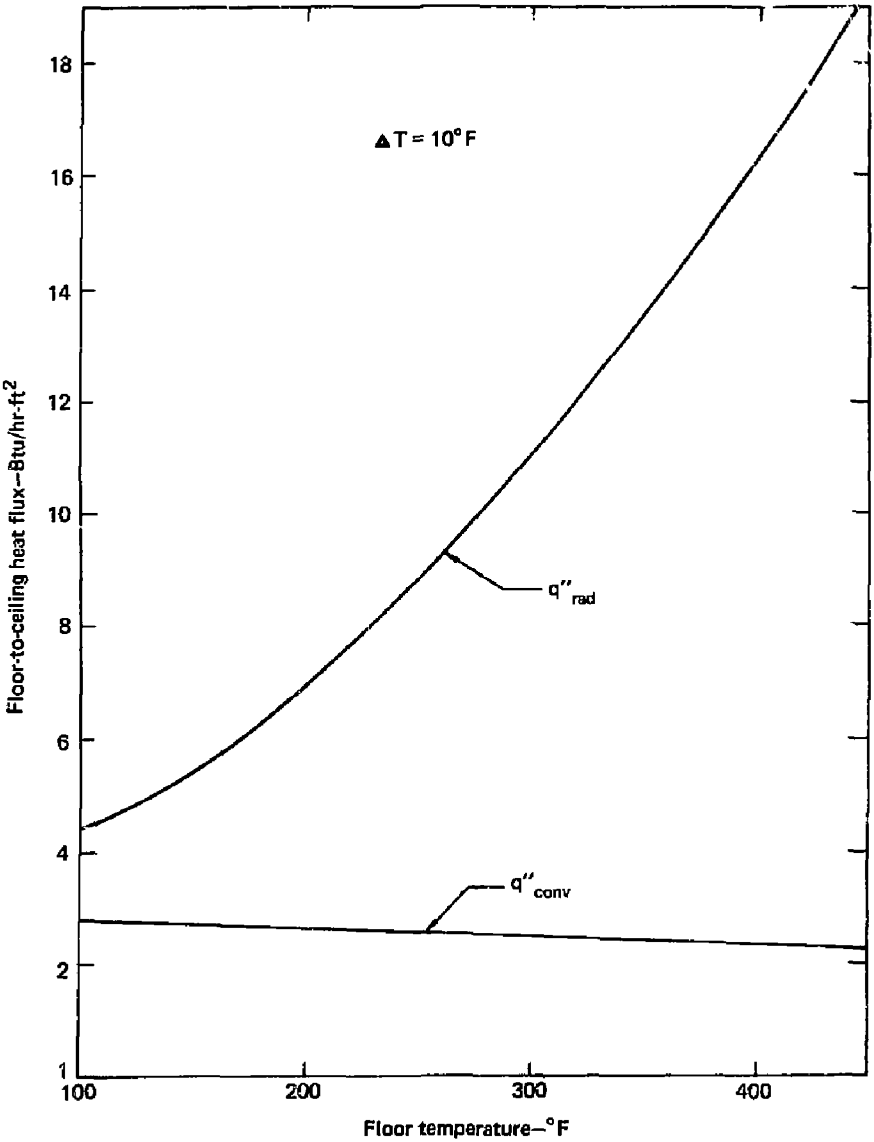

FIG. 4. Radiation heat flux from floor-to-ceiling of a repository room ($\Delta T = 10^{0}F$).

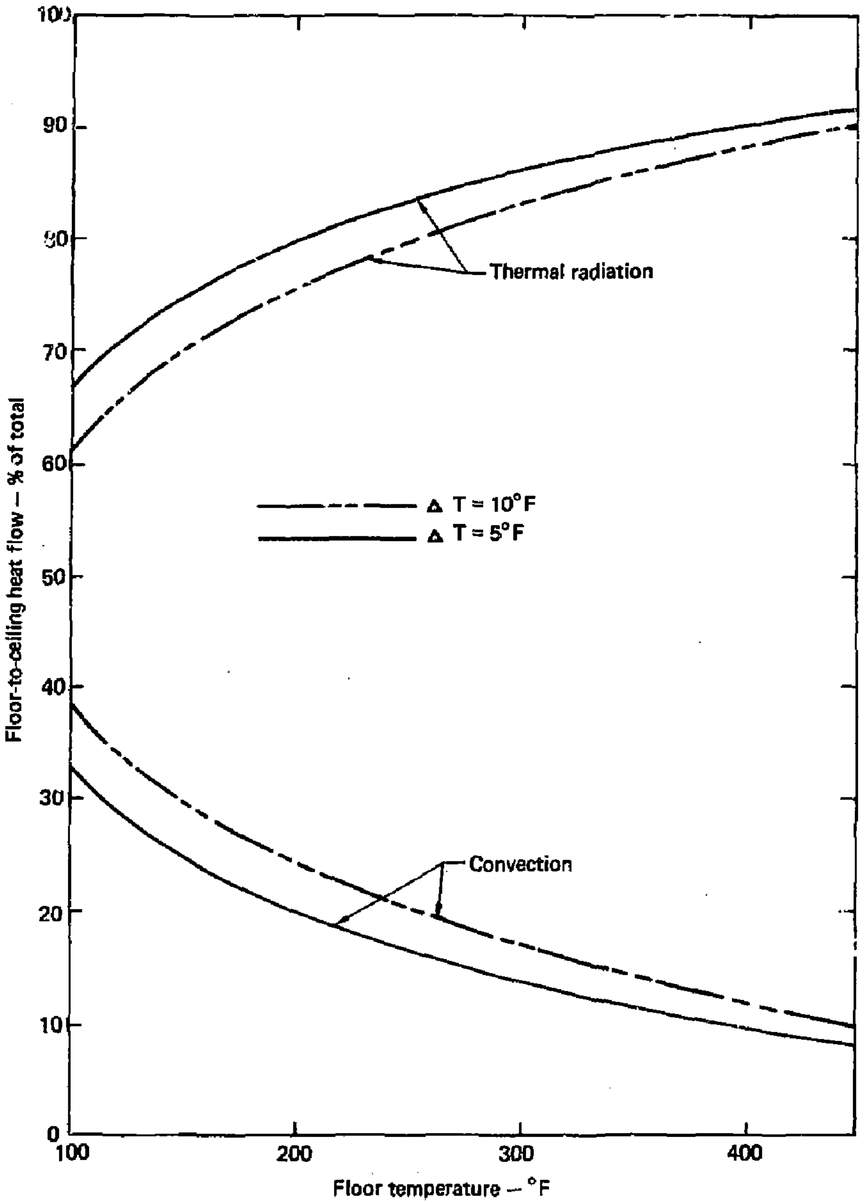

FIG. 5. Relative magnitudes of floor-to-ceiling heat transfer by convection and radiation for two different ΔT's.

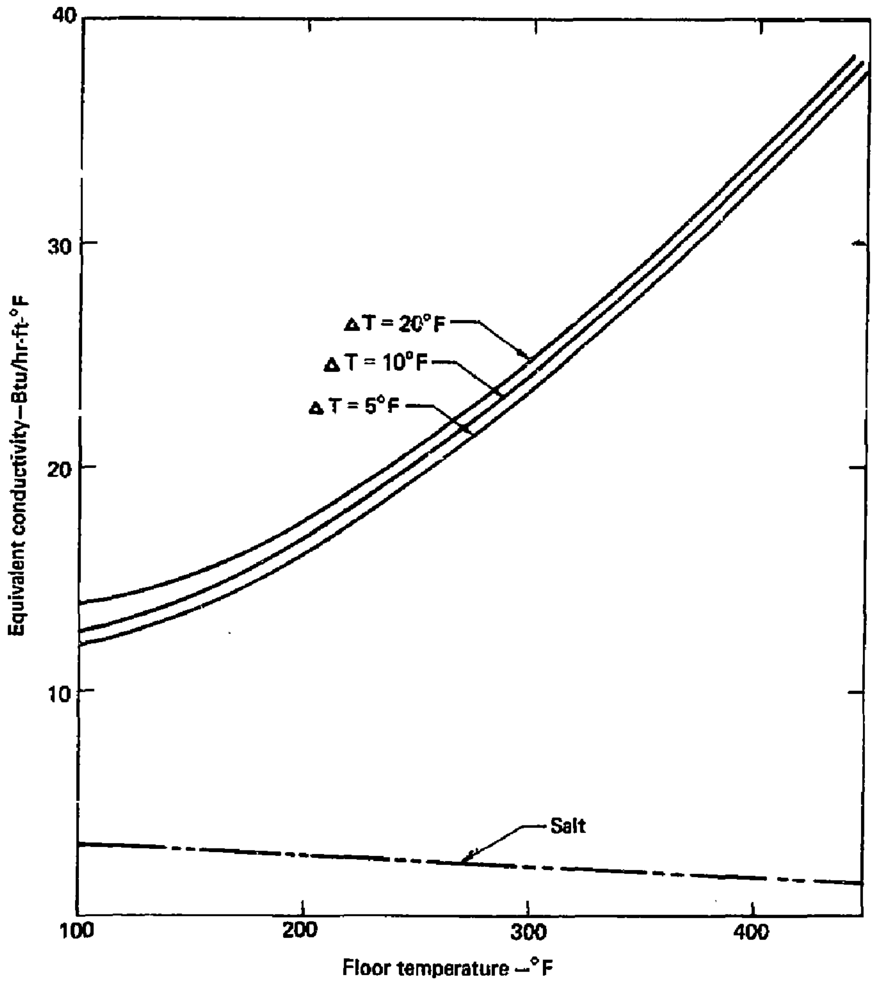

FIG. 6. Equivalent thermal conductance vs floor temperature.

the conductivity of salt decreases as the temperature increases. Hence, the presence of an air space (the room) above the emplaced canisters keeps the canisters cooler (because its effective conductance is large) even if ventilation is not used. Furthermore, backfilling of the rooms will cause the canisters to become hotter because salt is not as effective as the air space in conducting heat away from the source.

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APPENDIX

SOURCE LISTING OF A CODE WHICH DETERMINES
THE NONLINEAR EQUIVALENT CONDUCTIVITY
TABLE FOR A REPOSITORY ROOM

```
C
C THIS CODE GENERATES PLOT DATA RELATED TO COMBINED EFFECTS OF CONVEC-
C TION AND RADIATION THRU AIR SPACES IN A REPOSITORY ROOM/NON-VENTILATED.
C
C
        PROGRAM BILL(TOUT,TAPE1=TOUT,PLOT1,TAPE2=PLOT1,PLOT2,TAPE3=PLOT2)
        DIMENSION QC(4,21),QR(4,21),QT(4,21),R(4,21),RC(4,21),REQ(4,21),
     2 CONEQ(4,21),T1(4,21),T2(4,21),DT(4,21)
        CALL DEVICE(6HCREATE,4HTOUT,50000)
C
C
C      H=THE FLOOR-TO-CEILING HEIGHT
C      A=THE UNIT CELL FLOOR AREA
C      E1=EMISSIVITY OF SURFACE-1
C       E2=EMISSIVITY OF SURFACE-2
C      F12=THE FLOOR-TO-CEILING VIEW FACTOR
C      S=THE STEFAN BOLTZMANN CONSTANT
C      G=GRAVITY
C      RR=THE COMBINED SPACE AND SURFACE RADIATION RESISTANCE
C      TAV=THE BULK AIR TEMPERATURE
C      CON =THE CONDUCTIVITY OF THE AIR
C      V=THE KINEMATIC VISCOSITY OF THE AIR
C      GR=GRASHOF NUMBER
C      EK=THE CONVECTIVE EQUIVALENT CONDUCTIVITY OF AIR
C      QC=THE CONVECTIVE HEAT TRANSFER
C      QR=THE RADIATION HEAT TRANSFER
C      QT=THE TOTAL HEAT TRANSFER
C      R=THE EQUIVALENT RADIATION RESISTANCE
C      RC=THE CONVECTIVE RESISTANCE
C      REQ=THE EQUIVALENT CIRCUIT RESISTANCE
C      CONEQ=THE EQUIVALENT CONDUCTANCE
C      T1=FLOOR TEMPERATURE
C      T2=THE CEILING TEMPERATURE
C      DT=THE TEMPERATURE DIFFERENCE
C
C
       H=18.
       A=324.
       E1=0.9
       E2=0.9
       F12=0.41
       S=0.1714E-08
       G=32.174
C
       RR=(1.-E1)/(E1*A) + 1./(A*F12) + (1.-E2)/(E2*A)
C
       DO 10 I=1,4
       DO 15 J=1,20
       T1(I,J)=80.+20.*J
       T2(I,J)=T1(I,J)-5.*I
 15    CONTINUE
 10    CONTINUE
C
       WRITE(1,103)
       WRITE(1,100)
       DO 40 I=1,4
       DO 30 J=1,20
       DT(I,J)=T1(I,J)-T2(I,J)
       TAV=(T1(I,J)+T2(I,J))/2.
       IF(TAV.LT.150.)CON=0.016
       IF(TAV.LT.250.0 .AND. TAV .GE.150.)CON=0.018
       IF (TAV.LT.350.0 .AND. TAV .GE.250.)CON=0.02
       IF(TAV .GT.350.)CON=0.023
       IF(TAV.LT.150.)V=1.9E-04
       IF(TAV.LT.250.0 .AND. TAV.GE.150.)V=2.6E-04
       IF(TAV.GT.250.)V=3.0E-04
       GR=G*DT(I,J)*H**3/V**2/(TAV+460.)
       EK=0.068*CON*GR**0.333
       QC(I,J)=EK*DT(I,J)/H
       RC(I,J)=H/EK
       QR(I,J)=S*((T1(I,J)+460.)**4-(T2(I,J)+460.)**4)/(RR*A)
       QT(I,J)=QC(I,J)+QR(I,J)
       R(I,J)=DT(I,J)/QR(I,J)
       REQ(I,J)=(R(I,J)*RC(I,J))/(R(I,J)+RC(I,J))
       CONEQ(I,J)=QT(I,J)*H/DT(I,J)
 30    CONTINUE
 40    CONTINUE
C
       DO 60 I=1,4
       DO 70 J=1,20
       WRITE(1,102)T1(I,J),T2(I,J),QC(I,J),QR(I,J),QT(I,J),CONEQ(I,J)
 102   FORMAT(6E15.2)
 70    CONTINUE
       WRITE(1,100)
 60    CONTINUE
 100   FORMAT(2H  )
 103   FORMAT(10X,2HT1,13X,2HT2,13X,2HQC,13X,2HQR,13X,2HQT,10X,5HCONEQ)
       CALL EXIT
       END
```