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THERMAL ANALYSIS OF A VENTILATED
HIGH-LEVEL WASTE REPOSITORY

PREPARED FOR THE
OFFICE OF WASTE ISOLATION
OPERATED BY
UNION CARBIDE NUCLEAR DIVISION
FOR THE
U. S. ENERGY RESEARCH AND DEVELOPMENT ADMINISTRATION
UNDER SUBCONTRACT NO. 86Y-16527V
APRIL 1977

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EXECUTIVE SUMMARY

In earlier work, studies have been made of the long and short-term temperature distributions for a high-level waste storage facility in a bedded salt formation. These studies showed that ventilation of the entire repository can have an important effect on the thermal environments.

In this study the cooling response of a single ventilated storage room in an otherwise unventilated array of rooms is examined. Sensitivities to important parameters and effects of variations in storage room design are investigated. The approach has been to model a baseline repository as a finite array of rooms containing high-level waste and to calculate the temperature distributions in the vicinity of a single ventilated storage room within the array using standard auxiliary programs for transient heat transfer.

The results of the calculations show that ventilation provides a thermal sink in the heated system inducing temperature gradients in the formation significantly different from the unventilated case. An asymptotic cool-down limit exists for the storage room temperature. The value of this minimum temperature depends upon the inlet air temperature, the ventilation flow rate and the convective heat transfer coefficient. For inlet air at 75°F and 50,000 cfm and a heat transfer coefficient of 0.8 BTU/hr-°F-ft², the limit is about 100°F. The limit is essentially independent of the ventilation scheduling history of the room.

The time required to cool a room to reasonable operating temperatures depends upon the length of time the room is sealed prior to ventilation and upon inlet air temperature and the heat transfer coefficient. A storage room which has been sealed for 5 years will achieve temperatures of about 180°F. Approximately 4 months would be required in order to cool the storage room floor to a temperature of 120°F with a flow rate of 50,000 cfm at an inlet air temperature of 75°F, assuming a convective heat transfer coefficient of 0.8 BTU/hr-°F-ft². Approximately 2 months would be needed to cool the exhaust air to 120°F. For large air flow rates the time required to cool the storage room is essentially independent of the flow rate. Additionally, preliminary cooling of the room does not significantly reduce the cooling time.

Variations in the repository design have been shown to have an effect on the cooling characteristics. Increasing the storage room surface area by 25% over the baseline model depresses the cool-down temperatures by only about 4 F⁰ and decreases cooling times by 20% (on the order of 4 weeks less). Modifications in the canister design or width have virtually no effect on the cooling. However placing the waste deeper beneath the storage rooms and/or using longer canisters can lower the operating temperatures and cooling times significantly. Burying the waste 6-ft deeper would lower room temperatures as much as 20 F⁰. Another design parameter which affects the cooling significantly is the areal heat load in the repository. Reducing the canisters from 3.5 kW power density for 10 year old waste (108.5 kW/acre) to 2.0 kW/canister (62 kW/acre) reduces cooling temperatures by more than 20 F⁰ and reduces cooling times to a few weeks or less. The cooling times are nearly independent of the conductivity of the geologic formation in spite of the fact that temperatures during storage depend strongly on this property.

Convective heat transfer coefficients necessary to predict temperatures in the repository range from 0.2 to 1.5 BTU/hr-ft²-F and it will be necessary to perform a full-scale experiment in order to determine accurate temperatures during and after retrieval operations.

It has been determined that the temperature increase in the air brought from the surface down the supply shaft to the storage room level is about 5-7 F⁰ per 1000 feet, the largest portion due to adiabatic heating of the air which cannot be mitigated without refrigeration techniques.

It has also been demonstrated that temperature increases in regions immediately adjacent to repository rooms containing stored high-level waste are modest (less than 20 F⁰ 20-ft from the panel in the first 5 years of storage). Hence excavation or other operations should not be seriously restricted 30 or more feet from the panel in this time frame.

Finally it has been determined that for 3.5 kW canisters distributed in the repository to provide an areal heat load of 108.5 kW per acre, both the 1% and 25% unit-cell thermal criteria are easily met. Thermal criteria for the temperatures within the canister, itself, also do not appear to be restrictive since these temperatures can be markedly reduced by appropriate canister design considerations.

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1. INTRODUCTION

Early design efforts for a geologic waste repository were predicated on the concept of non-retrievability. High-level radioactive waste would be solidified and emplaced beneath storage rooms mined in salt formations and the rooms would then be back-filled with crushed salt which would reconsolidate, thus making retrievability of the waste canisters very difficult and expensive. The present design philosophy, however, requires easy retrievability. A first phase of the repository operation would require retrievability as an option at least until the entire repository is converted to a permanent waste isolation facility. Many of the concepts and design criteria previously developed must be re-examined in the light of this new requirement.

Previous studies have shown that high temperatures can be expected in the neighborhood of closed rooms where waste canisters are located.^{1,2,3,4} In addition, it has been shown that acceptable operating temperatures can be maintained in the storage rooms during the retrieval cycle by continuous ventilation of the entire repository.^{3,4} Since it is unlikely that the entire repository will be ventilated continuously, the effectiveness of intermittent ventilation of a single room in a panel of sealed storage rooms is examined; and, in particular, ventilation just prior to a retrieval operation is studied.

The model used for the ventilation calculations is discussed in Section 2. Based on the conclusions of previous work,⁴ the model is a two-dimensional representation of a panel which provides realistic temperatures in the vicinity of the room of interest for at least 20 years of operation. The technique used to extrapolate the results of the two-dimensional model to provide temperatures down the three-dimensional storage room is discussed in Appendix A.

This model is applied to a number of room ventilation scenarios in Section 3. Cooling and heating rates and their dependence upon ventilation characteristics such as air flow rate and inlet air temperature are examined. The effect of heat removal prior to actual ventilation for retrieval operations is also examined.

Effects of repository design variations upon the cooling characteristics of the storage rooms are presented in Section 4. Cooling rates and temperatures are studied as functions of parametric changes in storage room dimensions, canister design, location and spacing, the heat generation rate and thermal conductivity of the formation.

The uncertainties in the choice of the convection heat transfer coefficient are discussed in Section 5. Some of the considerations needed to make a more reliable estimate of this important parameter are presented and appropriate experiments are also discussed.

The heating of ventilation air as it travels down the supply shaft is discussed in Section 6. This effect is important for the ventilation system design. The main sources of heating and estimated temperature rises are given.

The storage room temperatures that may be expected during retrieval operations are compared with federal recommendations for work standards in hot environments in Section 7.

Since excavation work may take place near to storage rooms containing high-level waste, temperatures in the neighborhood of a sealed panel of such rooms are discussed in Section 8.

In Section 9 detailed near-field temperatures in the vicinity of the 3.5 kW canister are reported. Impacts of canister design variations on the temperatures in this vicinity are also discussed. Comparison of the canister temperatures with closed-form solutions for a simplified one-dimensional model is made in Appendix B.

Finally, in Section 10 a summary of the conclusions of this report is presented.

2. SPECIFICATION OF THE PROBLEM

2.1 PHYSICAL DESCRIPTION

This study considers the thermal environments generated within a high level waste (HLW) storage complex located in bedded salt at a depth 2000 feet below the surface of the earth. A schematic of a section of a typical HLW repository is shown in Figure 2-1 (taken from Reference 8). The HLW complex includes several storage panels connected via main corridors. Each storage panel consists of an 18 x 30 x 2500-ft branch corridor and 64 storage rooms. Each storage room is 18 x 18 x 558-ft and the pillar between any two rooms, is 60-ft wide.

A typical loading of 32 canisters of HLW is assumed to be emplaced beneath the floor down the centerline of each room on a pitch of 17.5-ft. The receptacle for each waste canister is a 21-in diameter hole, 18-ft deep, that may be lined with a 2-in thick carbon steel sleeve. The stainless steel canister is 10 ft long although the waste only fills the bottom 8-ft of the vessel. The canister will be encased in a carbon steel overpack. A 7-ft concrete thermal plug is inserted into the hole after the canister is in place thus completing the retrievable storage of the waste. Each canister is assumed to contain 6.28 cubic feet of HLW and to generate approximately 3.5 kilowatts of heat at the time of emplacement. The decay factors for the heat generation rate are given in Table 2.1.

The geologic formation immediately around the rooms and below is salt and the material above is a mixture of shale and salt. The thermal properties of a typical bedded salt formation and of the canister materials are given in Tables 2.2 and 2.3. The thermal conductivity of the salt and shale region is taken to be that for a homogenous mixture of 80 percent salt and 20 percent shale (by volume). The anistropy in the upper region due to the presence of shale is neglected here. A revision would be required if substantial amounts of shale or other geologic materials were to be found immediately above the actual mined room.

Figure 2-1 Schematic of NPTS Repository for High Level Waste

TABLE 2.1^a

DECAY FACTORS FOR TYPICAL WASTES
RESULTING FROM THE REPROCESSING^b OF
LIGHT WATER REACTOR FUEL

<u>Time^c, t (years)</u>	<u>P(t)/P₀</u>
0	1.0
2	0.917
5	0.845
7	0.800
10	0.737
12	0.697
15	0.638
20	0.576
30	0.450
40	0.351

a) See Reference 3.

b) Reprocessing, 90 days after reactor shutdown.

c) t is considered to be zero ten years after fuel has
been discharged from reactor.

TABLE 2.2

THERMAL PROPERTIES OF THE GEOLOGIC
FORMATION AND OF THE CANISTER MATERIALS

	DENSITY (lb/ft ³)	HEAT CAPACITY (BTU/lb-°F)	THERMAL CONDUCTIVITY (BTU/hr-ft ² -°F)
Air ^a	0.07	0.24	d
Stainless Steel 304L ^a	488	0.11	9.5
Carbon Steel ^a	490	0.11	26
Glass Waste ^b	200	0.22	0.58
Salt ^c	135	0.22	d
80% Salt - 20% Shale Mix	138	0.216	d
Concrete	144	0.20	0.54

a) Ref. 5.

b) Ref. 6.

c) Ref. 3.

d) See Table 2.3.

TABLE 2.3

THERMAL CONDUCTIVITIES (BTU/hr -ft-°F)

<u>T(°F)</u>	<u>Salt^a</u>	<u>80% Salt - 20% Shale Mixture^a</u>	<u>Air^b</u>
32	3.53	2.29	0.014
122	2.90	2.06	0.016
212	2.43	1.85	0.018
302	2.08	1.68	0.019
392	1.80	1.53	0.021
482	1.60	1.41	0.023
572	1.44	1.31	0.024
662	1.33	1.23	0.026
752	1.20	1.14	0.028

a) See Ref. 3.

b) See Ref. 5.

The assumed ambient temperatures of the geologic formation as a function of depth below the surface of the earth are provided in Table 2.4. With this assumed geothermal gradient, the ambient temperature in the storage rooms is roughly 93°F. However, the results of the ventilation calculations presented here are not strongly dependent on the ambient temperatures in the room or in the salt.

2.2 COMPUTATIONAL MODEL FOR VENTILATION STUDIES

A two-dimensional modeling technique was successfully used during previous studies to calculate the thermal environments of a HLW storage room containing 5.1 kW canisters.⁴ This same technique has been employed in this study to determine the impact of ventilating a single room in a finite array of HLW storage rooms.

A schematic of the two-dimensional repository model is presented in Figure 2-2. The boundaries at the top and bottom of the array are located sufficiently far from the heat generation sources that they have little impact upon the storage room temperatures during the first 30 years of burial. The boundary which bisects room 1 is a plane of symmetry and locates room 1 in the center of the finite array of rooms. A seven-room array was selected on the basis that the thermal environments calculated for the central storage rooms duplicated those calculated for an infinite array over the first 30 years of waste storage. When all rooms are sealed, the seven-room array successfully provides a thermal environment for the central room equivalent to the environment of a single room in the repository. Furthermore, the impact of ventilating the central room in the seven-room array is essentially the same as would occur when a single room of a typical HLW repository is cooled in a similar fashion.

The volumetric heat generation rate of the sources below the rooms is equivalent to that of 3.5 kW canisters buried at 17.5-ft pitch. This gives an areal heat load of 108.8 kW per acre. The steel sleeve, overpack and canister wall have all been neglected in the ventilation calculations since they have negligible impact on room temperatures. Radiative heat transfer between rooms surfaces has been explicitly included with an emissivity of $\epsilon = 0.9$.

TABLE 2.4
GEOHERMAL PROFILE^a

<u>Distance below surface of Earth (ft).</u>	<u>Ambient Temperature (°F)</u>
0	59.0
400	63.0
710	72.5
1100	74.0
2000	92.8
3500	124.0

a) See Ref.1.

2-8

Figure 2-2 SCHEMATIC MODEL-HIGH LEVEL
WASTE REPOSITORY

The two-dimensional calculations have been made with the heat conduction code HEATING5⁷. An auxillary program to extend the two-dimensional seven-room array calculations to include three-dimensional aspects of the storage room is described in Appendix A. Mesh spacing and time steps were selected such that further refinements of these parameters along with adjustments of the iteration scheme and convergence criteria would not change calculated temperatures by more than 4 percent. However, in some cases where transitions are quite sudden as in the onset of ventilation, the calculated temperatures oscillate about the true temperatures due to the finite mesh spacing. In these cases the mesh spacing was further reduced to ensure that averaging over these oscillations provided results appropriate for continuous media.

3. COOLING RATES

A likely scenario for a panel of storage rooms is that the rooms would be open and ventilated during the placement of the waste canisters and for a relatively short period of time thereafter. The entire panel of rooms would then be sealed until possible retrieval operations at some future time. While the panel is sealed, energy storage would occur throughout the local salt formation. Temperatures within the local salt formation and the room can be expected to increase to high levels due to heat generation in the waste, and it would be necessary to cool a particular room as a part of the retrieval operations.

Cooling by ventilation provides a heat sink that may induce severe temperature gradients in the formation while removing large amount of energy from the salt. The time required to cool a particular room would depend upon the distribution of stored energy in the salt, the thermal properties of the medium, and the heat removal capability of the ventilating air. Calculated, cooling behavior of a single ventilated room in a previously sealed panel is presented for a variety of retrieval conditions in the remainder of this section.

3.1 VENTILATING A SEALED ROOM

Calculations were performed for the thermal response of the seven-room array due to the ventilation of the central room using the previously described calculationed model. Figure 3-1 shows the calculated maximum floor temperatures in a room which is ventilated after being sealed for 1 year, 5 years, 10 years and 15 years, respectively. The calculations were made for an inlet air temperature of 75°F, a flow of 50,000 cfm, and for convective heat transfer coefficients of 0.2 and 0.8 BTU/hr-ft²-°F. Although the precise value of this coefficient is not well-known, it is believed that these values provide a useful range (See Section 5). In each case the floor temperature follows the sealed room profile until cooling is initiated. The temperature falls quickly and then, asymptotically reaches a lower cool-down limit. The lower limit in each case closely approximates the temperature that would have been attained had the room been ventilated continuously. Earlier work has shown that temperatures of

Figure 3-1 IMPACT OF SINGLE-ROOM VENTILATION INITIATED AFTER VARIOUS STORAGE TIMES

continuously ventilated storage rooms are determined by the heat removal capability of the ventilation air and are essentially independent of ambient temperatures for the formation.² In view of this, it is not surprising that the final cool-down temperatures are independent of the length of time the rooms are sealed. On the other hand, Figure 3-1 reveals that the time needed to reach this limit does depend somewhat upon the length of the sealed period and increases as the duration of the sealing period is increased.

A more detailed look at the cooling times for the ventilated room is given in Figures 3-2 and 3-3 where room temperatures are plotted versus time after venting. The crossing of the 10-year and 15-year temperature profiles in Figure 3-3 reflects the fact that the storage room begins to cool off after 10 years due to the decaying heat source. It is evident that the cooling rates immediately after venting are faster for the longer storage periods. This observation can be explained by the higher temperatures achieved when the room has been sealed for a longer period of time. However, the room that is sealed longer requires more cooling time to obtain a given temperature. As the room cools, the rate of cooling also decreases. This behavior is due to decreasing temperature gradients in the formation causing slower conduction of heat toward the room. The severe gradients and the changes in these gradients with time are likely to induce additional stresses in the salt. These particular stresses are not considered in the usual mechanical studies since ventilation is not ordinarily taken into account.

A useful measure for facility and equipment design is the cooling time needed to reach a particular operating temperature. Data extracted from Figures 3-2 and 3-3 are given in Table 3.1 for ventilating air at an inlet temperature of 75°F and a flow rate of 50,000 cfm. It is, in general, more difficult to cool a room the longer it is sealed. For example, cooling times to reach floor temperatures on the order of 130°F vary from several months to 5 years for different vent initiation times for the values of h_c examined here.

The temperature of the air in the ventilated room decreases faster than the floor temperature does. The cooling times to achieve reasonable

Figure 3-2 FLOOR TEMPERATURES FOR VARIOUS COOLING TIMES, $h_c = 0.2$

Figure 3-3 FLOOR TEMPERATURES FOR
VARIOUS COOLING TIMES, $h_c = 0.8$

TABLE 3.1

COOLING TIMES TO REACH TEMPERATURES $(T_{\text{inlet}} = 75^{\circ}\text{F}, f = 50,000 \text{ cfm})$

	Vent at 1 yr.	Vent at 2 yrs.	Vent at 5 yrs.	Vent at 10 yrs.	Vent at 15 yrs.
Maximum Floor Temperature ($^{\circ}\text{F}$) ($h_c = 0.2$)					
120	2 mo	6 mo	∞	∞	∞
130	<1 wk	7 wk	18 mo	3 yr	5 yr
140	0	3 wk	5 mo	1 yr	2 yr
150	0	<1 wk	11 wk	6 mo	9 mo
Maximum Floor Temperature ($^{\circ}\text{F}$) ($h_c = 0.8$)					
120	3 wk	4 wk	16 wk	6 mo	7 mo
130	<1 wk	2 wk	9 wk	18 wk	20 wk
140	0	<1 wk	7 wk	11 wk	13 wk
150	0	<1 wk	4 wk	8 wk	10 wk
Air Exhaust Temperature ($^{\circ}\text{F}$) ($h_c = 0.2$)					
90	4 wk	6 wk	7 wk	8 wk	11 wk
100	<2 wk	4 wk	6 wk	7 wk	7 wk
120	<1 wk	2 wk	4 wk	5 wk	5 wk
Air Exhaust Temperature ($^{\circ}\text{F}$) ($h_c = 0.8$)					
90	5 wk	8 wk	12 wk	17 wk	24 wk
100	3 wk	4 wk	9 wk	11 wk	12 wk
120	<1 wk	2 wk	7 wk	9 wk	9 wk

operating temperatures (on the order of 100°F) are approximately 6 to 8 weeks when the room has been sealed for at least 2 years. These maximum air temperatures as a function of cooling time for the case of 75°F inlet air, a flow rate of 50,000 cfm and h_c of 0.2 and 0.8 are also given in Table 3.1, and, for the example of ventilation begun after 10 years, illustrated in Figure 3-4.

3.2 COOLING RATE SENSITIVITY TO VENTILATION CHARACTERISTICS

Many contributing factors for the thermal environment of a HLW storage room are fixed by the initial design. These include considerations such as room size, canister design, depth of canister emplacement and heat generation rate. However, there are several controllable aspects to the system with the mode of ventilation being the most practical and the easiest to employ at the retrieval stage. In this section the dependence of the cooling times upon the ventilation flow rate and inlet air temperature is investigated.

The calculations of storage room temperatures were made with ventilation being initiated at 5, 10 and 15 years with a flow rate of 50,000 cfm. The results are qualitatively the same for ventilation at each time and results for cooling at 10 years are shown in Figures 3-5 and 3-6. In this case the effects of varying the ventilation inlet air temperature are shown for heat transfer coefficients of 0.2 and $0.8 \text{ BTU/hr-ft}^2\text{-}^{\circ}\text{F}$ respectively. A striking feature of the results is the regular, nearly linear dependence of the cool-off limit of the floor temperatures upon the inlet air temperature. Furthermore, the time needed to reach this limit is very nearly the same for all values of the inlet temperature. Therefore, cooling times to achieve a given room temperature can be reduced by cooling the ventilating air. For example, for $h_c = 0.8$, the time needed to reduce the maximum floor temperature to 130°F is decreased from 11 months to 2 months by using 60°F inlet air rather than 100°F air. This illustrates the importance of knowing the degree to which the ventilating air is warmed prior to entering the open storage room from its passage down the main shaft (Section 6) and through the branch corridors (Section 3.3).

The effect of varying the air flow rate for the 10-year case

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Figure 3-4 MAXIMUM AIR TEMPERATURES IN VENTILATED ROOM
(Room vented at 10 yrs)

Figure 3-5 MAXIMUM FLOOR TEMPERATURES IN
 VENTILATED ROOM FOR VARIOUS
 INLET AIR TEMPERATURES, $h_c = 0.2$

**Figure 3-6 MAXIMUM ROOM FLOOR
TEMPERATURES IN VENTILATED
ROOM FOR VARIOUS INLET AIR
TEMPERATURES, $h_c = 0.8$**

PA0858-11-1

with an inlet air temperature of 75°F is shown in Figure 3-7. The heat transfer coefficient is fixed at the lower value of $0.2 \text{ BTU/hr-ft}^2\text{-}^{\circ}\text{F}$ although this coefficient can be expected to increase as the air velocity is increased. Hence the actual dependence of the flow rate is likely to be even more pronounced than is demonstrated in Figure 3-7. Even with a constant h_c , it is evident the cooling rates depend upon the ventilation flow rate. However, there is a limit to the effectiveness of the heat removal in this way. For example, there is very little difference in the final temperatures reached when 50,000 cfm (2.6 fps) or 150,000 cfm (7.7 fps) are employed. For these high rates the energy is essentially carried out as quickly as it is transmitted into the storage room. In view of this saturating dependence upon the air flow rate it would appear that the heat removal could be improved by distributing the flow over several rooms, thus gaining the advantage of the additional heat sinks due to the cooling of adjacent rooms. However, some of this advantage may be lost since the heat transfer coefficient in general depends upon the velocity. The results of a calculation for this effect is shown in Figure 3-7 where 3 rooms are ventilated with 150,000 cfm air. The air flow in the central room is reduced by a factor of 3 but the results are virtually unchanged from that obtained directing the entire air flow through a single room. As a consequence, there appears to be no apparent advantage for this multiple room approach unless, of course, the adjacent rooms would also need to be cooled within a short period of time.

3.3 PRELIMINARY COOLING

It may be possible to improve the cooling properties of the system, by removing a portion of the stored energy in the salt prior to the actual ventilation at the retrieval stage. It is clear that any reduction of the energy content of the system should affect the time needed to lower temperatures to some predetermined level at later times. In this section a number of possibilities for the mode of early energy removal are explored.

Figure 3-8 shows the effect of precooling by ventilating at various early stages. The solid curve is calculated for full ventilation in the

Figure 3-7 MAXIMUM FLOOR TEMPERATURES IN VENTILATED ROOM vs. FLOW RATE (Room vented at 10 yrs.)

Figure 3-8 MAXIMUM FLOOR TEMPERATURE
PRECOOLING SEQUENCE

first year, sealing the room until 15 years and ventilating thereafter with 75°F inlet air, a flow rate of 50,000 cfm, and a heat transfer coefficient of 0.2 BTU/hr-ft²-°F. After the first 2 years, this curve becomes indistinguishable from that where the preliminary heat removal operation is not performed. It is clear this early pre-cooling has little effect. The amount of heat removed in the first year after placement of the HLW canister is insignificant, probably because temperatures in the region of the room have not been built up to allow very efficient heat removal. A one-year ventilation period at 5 and 10 years has a somewhat more noticeable effect as shown by the dotted and dashed curves respectively. The effect is slightly more pronounced when the precooling is conducted at 10 years since the temperatures at this time are generally higher and energy removal efficiency is increased. Nevertheless, in neither case is energy removal from the system sufficient to reduce the cooling times at 15 years significantly. These calculations clearly illustrate the superficial nature of short-term cooling effects.

The ability of continuous bleed-through ventilation to remove energy prior to full ventilation is illustrated in Figure 3-9. In this case air at 5000 cfm (0.26 fps) is allowed to flow through a storage room for 10 years prior to retrieval operations at which time the room is ventilated with 50,000 cfm. It is clear that the bleed-through of this magnitude has little effect in future cooling and removes little energy from the system during the precooling stage.

Finally, Figure 3-10 shows the cooling effect of natural convection from an unvented storage room into the branch corridor. In this calculation the branch corridor is maintained at 75°F by a ventilation flow for 10 years. The "chimney" or Venturi effect is neglected here and can be expected to enhance the cooling of the room at the near end and heat the room somewhat further from the branch corridor. In this case the convection process is somewhat different than for heat transfer from the walls into the storage room and a heat transfer coefficient of 0.8 BTU/hr-ft²-°F is used for heat transfer from the storage room into the corridor. The temperature reduction, although of some importance for the first 100 feet, is not important for the larger part of the storage

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Figure 3-9 EFFECT OF BLEED-THROUGH VENTILATION ON STORAGE ROOM FLOOR TEMPERATURES

Figure 3-10 AIR TEMPERATURES IN
STORAGE ROOM BRANCH
CORRIDOR VENTILATED AT
74°F

room. It is not expected that ventilation of the branch corridor alone will substantially aid in the cooling of the storage rooms. Heat conduction however will warm the branch corridor air and will, along with heat transfer from the walls of the corridor, affect temperatures of air fed from the branch corridor into a vented storage room. It may be necessary to seal off the storage rooms with insulating bulkheads to reduce this effect. More complete calculations should be performed to determine the full impact of this heating.

4. PARAMETRIC STUDIES

Ventilation requirements expressed in terms of air flow rates and input air temperatures were discussed for the baseline high-level waste repository design in the previous section. In most cases ventilation times on the order of months are required to cool the storage rooms to temperatures appropriate for retrieval operations. However, the ventilation can couple with the storage facility design in important ways which may influence the cooling characteristics. The question arises whether it is possible to alter the effectiveness of the cooling with reasonable changes in the design. In this section, it will be shown that the system is reasonably stable and that small design changes, in general, have little impact on the cooling characteristics. However, there are some important exceptions such as depth of canister storage and intensity of the heat generation sources.

A constant heat transfer coefficient ($h_c = 0.2 \text{ BTU/hr-ft}^2\text{-}^\circ\text{F}$) and ventilation characteristics of 75°F inlet air with a velocity 2.6 fps (50,000 cfm for the baseline model) were assumed for the following illustrative examples.

4.1 STORAGE ROOM SIZE

The cooling characteristics of a storage room can be altered somewhat by varying the surface area. In the baseline model the storage room is 18 ft wide by 18 ft high. The calculations have been performed extending the room surfaces by increasing the perimeter by 25%. This increase was accomplished through a 50% increase in storage room height and, separately, through a 50% increase in storage room width. Both techniques for increasing the room surface area produced similar impacts on room cooling. The comparison of the baseline model results with those for the increased room surface area case are shown in Figure 4-1. It can be seen that the reasonably attainable storage room temperatures are about 4°F cooler for the larger room. Further, the time required to attain these temperatures

Figure 4-1 EFFECT OF STORAGE ROOM SIZE
ON STORAGE ROOM FLOOR
TEMPERATURES

is slightly reduced basically because the temperatures at the onset of ventilation are somewhat cooler in the latter case. The cooler initial temperatures are readily understood in terms of extended room surfaces for which heat can be directly radiated away from the hot floors to the cooler surfaces. In the calculations shown in Figure 4-1 the air flow rate is held constant. If the air velocity had been held fixed by increasing the air flow rate, the difference between the two curves would have been somewhat larger, about 5°F. Clearly the cooling characteristics would be only slightly affected by room size.

4.2 CANISTER DESIGN

Early work^{2,3} suggested that temperatures in the vicinity of the storage room do not depend sensitively upon the canister design. Calculations performed for the ventilation characteristics show a similar result. Although temperatures within the canister are strongly affected by the size and conductance properties of the canister, the thermal response in the vicinity of the storage room is essentially independent of such detail in the canister design. Calculations for a canister with twice the radius of that given by the baseline design or with a conductivity only 10% of the design value give results which are indistinguishable from the heavy curve shown in Figure 4-1.

4.3 DEPTH OF CANISTER

The emplacement depth of the buried waste with respect to the storage room can have an important effect upon the room cooling behavior. In the baseline design the top of the waste is 10 feet below the storage room floor. The effects of waste burial 10, 13 and 16 feet below the storage room are shown in Figure 4-2. Both the room temperatures and cooling times are significantly reduced largely due to the fact that room temperatures at the onset of ventilation are much lower. For burial 6 feet below the baseline design depth, the temperatures are lowered by more than 20 F° and the cooling times are reduced by a factor of two. Longer canisters which are buried at a 10-ft depth will have similar impact on the storage room cooling properties. The results for a 16-ft 3.5 kW canister are also shown in Figure 4-2.

4.4 HEAT GENERATION RATE

Changing the heat generation rate of the stored canisters has a large impact on the cooling characteristics of the storage room. Figure 4-3 shows the results of calculations for canisters initially at 5 kW, 3.5 kW and 2.0 kW. In each case high-level waste which is 10 years old at the time of burial was considered. The temperature after 1 year of ventilation for 2 kW canisters is 20 F° lower than for baseline 3.5 kW canisters. The heat generation rate would appear to be a very sensitive design parameter with respect to the cooling response of a storage room. The heat load can also be obtained by altering the spacing between canisters with the areal heat load being the overall controlling factor. Table 4.1 gives the asymptotic cool-down temperature as a function of the overall heat loading per acre of the repository.

4.5 CONDUCTIVITY OF THE GEOLOGIC MEDIUM

Although the conductivity of the geologic media in the vicinity of the repository is not in the strictest sense a design parameter, some control can be imposed by the choice for the location of the repository. However, the conductivity of rock salt varies widely throughout the salt bed and variations as much as 100% at a single location are not unexpected. Core samples from a single location near Carlsbad, New Mexico vary by as much as a factor of 2 throughout depths from 1000 to 2700 feet.⁹ The influence of variations in the conductivity upon the cooling response of a storage room is shown in Figure 4-4. The curves shown result by multiplying the temperature-dependent conductivity of salt of Table 2.3 by factors of 0.67, 1.0 and 1.50, respectively. The reflected changes in repository room cooling behavior are quite modest, temperature changes being less than 5 F°. This should not imply that the thermal response of the system is not sensitive to the conductivity but only that the room cooling rate is not strongly influenced. Even more drastic changes in the conductivity such as those found by going to other media appear to have a severe effect on the predicted cooling characteristics. A calculation for a medium having a conductivity of 0.25 BTU/hr-ft-°F corresponding

Figure 4-3 EFFECT OF HEAT GENERATION RATE ON COOLING OF STORAGE ROOM FLOORS

Figure 4-4 EFFECT OF FORMATION
CONDUCTIVITY, K ON COOLING
OF STORAGE ROOM FLOORS

Table 4.1

COOL-DOWN TEMPERATURES AS FUNCTION OF AREAL HEAT LOAD
(F = 50,000 cfm, Inlet = 75°F)

<u>Heat Load (kW/acre)</u>	<u>Cool-Down Temperature(°F)</u>
158	146
108.5	131
62	114

Figure 4-5 EFFECT OF FORMATION
CONDUCTIVITY ON COOLING OF
STORAGE ROOM FLOORS

to some types of limestone is contrasted in Figure 4-5 with the results from the salt formation. In this case, although the 5-year storage room floor temperature is much higher for the low conductivity limestone case, the room cools quickly to temperatures lower than for the salt and the cooling times to reach comparable temperatures are nearly the same for the two cases. This implies that cooling systems designed for a repository in salt may not be required to be very different for other media.

5. DETERMINATION OF THE CONVECTIVE HEAT TRANSFER COEFFICIENT

A large source of uncertainty in the thermal analysis of a vented storage room in a high-level waste repository is directly related to the choice of coefficient for the convection of heat from the salt into ventilated regions as shown in Figure 3-1. The predicted temperatures for storage rooms ventilated with 75°F air differ by as much as 20°F depending upon whether a heat transfer coefficient of 0.2 or 0.8 BTU/hr-ft²-°F is used. Appropriate theoretical values are difficult to establish due to the irregular features of the storage room surfaces. Further, experimental data do not exist for the repository configuration. However, predictions based on idealized models can be made and several of these will be discussed in this section. At the same time, experimental data for systems related to the ventilated storage room can be correlated in terms of the usual dimensionless variables. These parameterizations can be extrapolated to the storage room configuration and several accepted correlations will also be discussed here. It will be seen that both the predictive models and current phenomenological correlations permit a wide range of values for the convective heat transfer coefficient so that additional information is needed to provide an adequate thermal description of the repository.

5.1 DUCT CORRELATIONS

For fully-developed turbulent air flow the 558-ft storage room can be represented as a duct of large cross-sectional area. It is well-established that a Dittus-Boelter-type correlation is applicable to such a situation for ducts of circular cross section¹⁰ and that for fully-developed flow the correlation for rectangular ducts approaches the parameterizations for circular ducts.^{11,12,13} The Dittus-Boelter convective heat transfer coefficient for the circular duct is

$$h_c = 0.23 \text{ Re}^{0.8} \text{ Pr}^{0.4} k/d$$

where Re is the Reynolds number, Pr is the Prandtl modulus, k is the thermal conductivity of air and d is the hydraulic diameter of the duct. The 0.4 power of the Prandtl number corresponds to heat removal by the convective fluid and all quantities are evaluated at the bulk temperature. The maximum temperatures for a storage room ventilated after 5 years using the Dittus-Boelter and several other correlations are shown in Figure 5-1. The corresponding values of average heat transfer coefficient are given in Table 5.1.

The Dittus-Boelter relation assumes a fully-developed turbulent boundary layer along the entire length of the storage room¹⁰ and for this problem predicts an average heat transfer coefficient of about 0.36 BTU/hr-ft²-°F. The leading-edge laminar boundary layer¹⁴ is assumed to be negligible, since this effect reduces the heat transfer coefficient by less than 0.5%. However, entrance effects associated with the developing turbulent boundary layer can be important, particularly for distances less than 30 diameter lengths from the entrance to the duct.^{15, 16} A correlation suggested by Eckert and Drake¹⁷ which approximately takes into account these entrance effects has the form

$$\frac{hc}{h_{\infty}} = 1 + 1/3 \left(\frac{d}{x}\right)^{2/3}$$

where h_{∞} is the coefficient for fully-developed flow, x is the distance from the entrance, and d is the hydraulic diameter. This correction leads to an average coefficient of about 0.39 BTU/hr-ft²-°F. This does not appear to be a large change from the uncorrected coefficient. However, the above correction is may not be reliable since it is based on data for which entrance effects are relatively minor and the actual correction may be significantly larger when the entrance region forms a substantial portion of the duct.¹⁵ The length of the entrance region for the storage room can be estimated from the Blasius relation for the turbulent boundary layer thickness.¹⁸

$$\delta = \frac{0.375 x}{Re_x^{0.2}}$$

**Figure 5-1 MAXIMUM ROOM TEMPERATURES
FOR SELECT HEAT TRANSFER
COEFFICIENTS**

TABLE 5.1

HEAT TRANSFER CORRELATIONS AND MEAN
HEAT TRANSFER COEFFICIENT

Correlation	$\bar{h}_c$ (BTU/hr-°F-ft ²)
Dittus-Boelter, $h_c = \frac{0.023 Re_d^{0.8} Pr^{0.4} k}{d}$	0.36
Dittus-Boelter with entrance correction, $h_c = h_\infty [1 + 1/3 (\frac{d}{x})^{.67}]$	0.39
Prandtl (flat plate), $h_c = \frac{0.029 Re_x^{.8} Pr k}{x [1 + 2.12 Re_x^{-.1} (Pr-1)]}$	0.20
Colburn (flat plate), $h_c = 0.029 \frac{Re_x^{.8} Pr^{.33} k}{x}$	0.30
Temperature dependence, $h_c = h_\infty (\frac{T_B}{T_W})^{0.575}$	0.80-0.86
Roughness correlation, $h_c = h_\infty 1.5$	0.55
$h_c = h_\infty 3.5$	1.28
Bouyancy correlation, $h_c = h_\infty + 0.6$	0.96

Here the Reynold's number, Re_x , is calculated with the distance x . This relation is used to estimate the entrance length, L_e , needed for the turbulent boundary thickness to reach one-half the diameter of the duct and thus define the onset of fully-developed flow. The entrance length is then

$$L_e = 2.67 \left(\frac{d}{2}\right) Re_{L_e}^{0.2}$$

Table 5.2 contains values of the starting length for which $d/2 = 9$ ft. Since the storage room is 558-ft long, the flow throughout most of the storage room is not fully-developed. In this case, the applicability of duct correlations such as the Dittus-Boelter relation becomes questionable. An appropriate correlation would have to account for the fact that the boundary layer thickness may be much smaller than the diameter of the storage room.^{12,19,20,21}

5.2 FLAT PLATE CORRELATIONS

In the region where the boundary layer thickness is much smaller than the width of room surfaces it becomes reasonable to regard the room surfaces as flat plates parallel to the air flow. Data exist for a wide range of the Reynolds number. A theoretical relation for the local heat transfer coefficient which has proven successful in describing much of the data for turbulent flow was given early by Prandtl²² as

$$h_c(x) = \frac{0.029 (Re_x)^{.8} (Pr) k/x}{[1 + 2.12 Re_x^{-.1} (Pr-1)]}$$

where the Reynolds number, Re_x , is calculated with the distance x .

TABLE 5.2

APPROXIMATE ENTRY LENGTHS FOR WHICH
FLOW IS PARTIALLY-DEVELOPED.

<u>Flow Rate (cfm)</u>	<u>Velocity (fps)</u>	<u>Entrance Length (ft)</u>
10,000	0.51	389
25,000	1.29	489
50,000	2.57	582
75,000	3.86	643
100,000	5.14	692

A calculation using this relation is shown in Figure 5-1 and the corresponding mean coefficient is $0.20 \text{ BTU/hr-ft}^2\text{-}^\circ\text{F}$. Colburn²³ found that a useful parameterization for flat plates is

$$h_c(x) = \frac{0.029 [Re_x]^{0.8} Pr_k^{1/3}}{x}$$

which results in an average coefficient somewhat larger than that of Prandtl. Efforts to extend the Prandtl expression to the case of fully-developed turbulent flow in tubes and ducts result in heat transfer coefficients which are about 10% lower than those calculated from the Dittus-Boelter relation.^{24,25}

The phenomenological correlations and theoretical relations described in this section address only a small portion of the uncertainty attached to the heat transfer coefficient since they normally apply to idealized systems. For more realistic cases, additional uncertainties exist and some of these are discussed below.

5.3 CORRELATION UNCERTAINTIES

The integral nature of most heat transfer experiments and the complexity of the theoretical description of the convection process in turbulent flow imply that large uncertainties may be attached to the simple parameterizations used to correlate the data. The standard tube and duct correlations usually fit the data with a total deviation of no better than 20-40% even for restricted ranges of the moduli.^{26,27} Further, extrapolations to new ranges of parameter variations cannot be expected to be even this reliable. For example, correlations which depend upon the Reynolds number as $Re^{.8(10)}$ or $Re^{.75(11)}$, which are used to fit data for essentially the same regime, lead to large differences when extrapolated away from this regime.

Another important consideration with such parameterizations is the formal definition of the heat transfer coefficient. The assumption is usually made that the heat transfer is uniform^{11,12} or that the surface and/or bulk temperatures are fixed.^{28,29} These assumptions are not appropriate in the present application to the storage room. In this case, these quantities vary with position, depending upon the heat transfer far upstream as well as at the local position.

5.4 SIMPLIFYING APPROXIMATIONS

In order to simplify the parameterizations a number of approximations are explicitly and implicitly made. The neglect of an important factor may be difficult to account for with the simple parameterizations if they are extrapolated to a new situation. A recognized problem with the parameterizations has been the temperature dependence of the physical parameters involved. All temperature-dependent properties are evaluated at the bulk temperature of the coolant¹⁰ for the Dittus-Boelter relation while the properties are evaluated at an average film temperature for other correlations. When there is a large difference in temperatures, neither assumption may be valid.

Sieder and Tate³¹ suggested a correction term for the case when the material properties depend strongly on temperature. A term of this type used by Colburn²³ and others³² to account for large temperature differences for air and water cooling. The correction suggested by Hausen³³ is

$$h_c = h_B \left(\frac{T_B}{T_W} \right)^{0.575}$$

where h_B is the coefficient with all properties evaluated at the bulk temperatures and T_B and T_W are the bulk and wall temperatures, respectively. Both h_B and T_B are local values for this calculation. Although this expression may not be any more appropriate than that evaluated at an average film temperature, it may be useful in evaluating the sensitivity to this dependence. The effect of such a correction on the temperature curves is shown in Figure 5-1. The resulting average heat transfer coefficient of 0.86 is compared with others in Table 5.1.

Another simplification is to neglect the irregularity of the storage room walls. The relative roughness factors for the storage rooms are expected to range from 2×10^{-3} to 5×10^{-2} and will, in fact, vary with time as the surfaces in the salt formation creep. The Moody friction enhancement factors³⁴ range from 1.25 to 3.5 in this case. However it is not at all clear that the standard Moody friction corrections apply to the partially-developed storage room flow in the same way that they do for flow through tubes or pipes. In this case the roughness factors alone could cause major uncertainty in the parameterized heat transfer coefficients. The resulting temperature profiles and enhanced transfer coefficients are given in Figure 5-1 and Table 5.1 respectively.

A third simplification in these correlations has been to neglect natural convection processes. Since the interaction of forced and natural convection is not well understood, it is fortunate that the mixing of these processes is usually unimportant. The relative importance of these two processes is usually given for partially-developed

forced flow by the ratio of the Grashof number, Gr , to the square of the Reynolds number.⁵ Approximate values for this ratio are shown in Table 5.3 for the storage room configuration with a temperature difference of $20\text{ }^{\circ}\text{F}$ between the walls and air for various air velocities. These values indicate that for reasonable values of the flow rate, the mixing of these processes may be important and can lead to an important ambiguity in the calculated heat transfer. In some cases the combined effects are estimated simply by adding the coefficients calculated separately for the two processes³⁵ although there is little experimental evidence to support this approach. The natural convection coefficient for the confined storage room in this case is about $0.6\text{ BTU/hr-ft}^2\text{-}^{\circ}\text{F}$ and the result of adding the Dittus-Boelter coefficient and this value is shown in Figure 5-1 and Table 5.1 respectively.

5.5 DETERMINATION OF THE HEAT TRANSFER INTO STORAGE ROOMS

It is obvious that the existing correlations are inadequate to predict the coefficients for convective heat transfer required for the thermal analysis of the storage rooms. The expected envelope determined from the standard parameterization (for storage room temperatures) is shown in Figure 5-1. In order to specify these temperatures more precisely than this and to evaluate the heat load on the ventilating system for the repository accurately, it may be necessary to perform measurements of this heat transfer at a mine of the approximate size and configuration planned for the repository. However, an experiment to determine the local heat transfer coefficient is not likely to be meaningful since a complete mapping of this coefficient as a function of wall temperature, air temperature and velocity as well as the position down the storage room would be subject to local fluctuations in time, position and temperature. Moreover the results of such a careful experiment may not be general enough to warrant the considerable expense entailed. On the other hand, a measurement of the total heat transferred into the storage room should not only be more amenable to standard experimental techniques, but should also permit the extraction of generalized, mean heat transfer

TABLE 5.3

RATIO OF GRASHOF NUMBER TO SQUARE OF
REYNOLDS NUMBER FOR VARIOUS FLOW RATES.

V(fps)	F(cfm)	Gr/Re ²
1.0	20,000	36.00
2.5	50,000	5.80
5.0	100,000	1.40
10.0	200,000	0.36

coefficients by comparison with numerical calculations. Such an integral experiment would likely provide access to temperature dependence simply by lowering the temperature of the ventilating air with the usual refrigeration techniques available. Heated ventilating air might also provide interesting data for comparison.

6. HEATING OF VENTILATION AIR IN THE SUPPLY SHAFT

A number of investigations have been performed on the sources of energy in underground mines.³⁶ These studies conclude that the more important sources heating the ventilation air as it is pumped down the ventilation supply shaft are

1. Compression of the air in the lower regions of a vertical shaft due to the weight of the air column,
2. Heat transfer from the rocks around the shaft into the air as it flows down the shaft,
3. Friction that causes kinetic energy to be transformed into heat as the air travels down the shaft,
4. Conversion of kinetic energy into heat upon redirection of the air flow from the vertical shaft into the horizontal repository corridors.

An estimate of the air temperature increase for each of these phenomena is important in order to determine if the repository ventilation requirements are adequate or whether engineering measures should be employed to mitigate their impact. In order to estimate the air temperature increases the repository design described in Reference 8 will be employed with the exception that the mine level is assumed to be 2000 feet below the surface of the earth. This repository design includes a ventilation supply shaft with a cross sectional area of about 190 ft² and a perimeter of about 59 feet. The shaft is assumed to transport dry air at flow rates up to 500,000 cfm (an air velocity of about 44 fps).

The temperature rise due to the weight of the air in the vertical shaft can be computed from the following equation for adiabatic compression:

$$\frac{T}{T_0} = \frac{P}{P_0}^{\frac{\gamma-1}{\gamma}}$$

where,

T = Air temperature at point of interest down the shaft ($^{\circ}\text{R}$)

T_0 = Air temperature at top of the shaft ($^{\circ}\text{R}$)

P = Air pressure at the point of interest down the shaft in absolute units

P_0 = Air pressure at the top of the shaft

γ = Ratio of heat capacities of air, C_p/C_v .

If the air enters the supply shaft at a pressure of about 1 atmosphere, the increased pressure at the bottom of a 2000 foot column of air would be on the order of 0.07 atmosphere. Then since γ for dry air of 1.41, the increase in the temperature of inlet 60°F (520°R) air is about 10.3°F . For humid air the specific ratio is smaller and the temperature rise will be somewhat less. For example for humidity of about 50%, corresponding to $\gamma = 1.24$, the temperature rise will be 6.7°F .

In a similar simple fashion, an estimate of the heat transferred to the ventilation air from the rocks surrounding the shaft can be determined from an iterative solution of the equation:

$$\Delta T_a = \frac{h_c S (T_w - T_a)}{C_p \rho v A}$$

where, ΔT_a is the incremental increase in air temperature in the shaft; h_c is the convective heat transfer coefficient; S is the incremental surface area of the shaft; $T_w - T_a$ is the difference between the shaft wall temperature and the air temperature; C_p , ρ , and v are the heat capacity, density and velocity of the air; and A is the cross sectional area of the shaft. For the configuration being analyzed, the rock temperature variation with surface depth, x , is taken as

$$T(x) = 60.0 + 0.165 \times (^\circ\text{F}).$$

This corresponds to an average natural geothermal flux and could vary widely from site to site.³⁷ Increased temperatures in the formation due to the stored waste may also be a factor. Applying roughness factors for the concrete shaft walls, the convective heat transfer coefficient will vary between 0.4 and 1.6 BTU/hr-ft²-°F. For a value of 1.0 BTU/hr-ft²-°F, the calculation yields a temperature increase of 3.6 F° during the traversal of the 2000 foot shaft.

The increase in air temperature due to friction and turbulence can be estimated assuming a roughness factor of 0.01-ft for the concrete-lined shaft, corresponding to a relative roughness of 8×10^{-4} . Since the anticipated flow rate and assumed shaft design yield a Reynolds number of about 2.5×10^6 , a friction factor of about 0.018 can be read from a Moody diagram.³⁴ This friction factor coupled with the assumed ventilation design with dry air yields an air temperature increase on the order of 0.7 F° for the 2000 foot shaft.

The redirection of the ventilation air to the various mining levels will also contribute to a velocity head loss and thus a temperature rise in the air. In the worse case a sharp right angle turn of the shaft itself may be considered. In this case the velocity head loss is half that for the frictional losses down the shaft, so that the maximum temperature rise would be less than 0.35 F°.³⁸

These simple calculations show that the largest contribution to ventilation air heating in the shaft is due to the adiabatic compression of the air. The total heating of the air due to these various sources will be on the order of 10 F° to 15 F° depending on the specific design characteristics and the properties of the geologic location of the repository. Since the largest increase in temperature is from the adiabatic compression, it is not possible to reduce this effect without refrigeration. Other engineering measures, such as insulating and smoothing the surfaces of the shaft, may reduce the heat transfer from the surrounding media and from frictional dissipation, but these contributions do not appear to be of sufficient magnitude to warrant any extraordinary procedures.

7. REPOSITORY THERMAL ENVIRONMENTS COMPARED WITH FEDERAL RECOMMENDATIONS

OSHA³⁹ and MESA⁴⁰ have produced recommendations for work standards in hot environments. These recommendations are intended to protect workers exposed to high temperatures by minimizing the risk of heat disorder and illness. Both recommendations draw upon the earlier considerations of heat stress by NIOSH⁴¹ and, consequently are very similar. In general the recommendations attempt to relate heat stress or external effects and heat strain or body response phenomena to simple, readily measurable quantities. The complexities of these phenomena render this a difficult, if not impossible, task. Consequently, within the health and scientific communities there is not a consensus regarding the recommendations of their applicability as a standard.

The complexity of heat stress and heat strain can be readily seen from the large number of interaction factors that are involved. There are more than a dozen factors instrumental in heat transfer between the body and the environment--air temperature, barometric pressure, temperature and emissivity of solids in the work area, air velocity, ambient water vapor pressure, skin temperature, skin vapor pressure, effective surface of the skin, skin color, pulmonary ventilation, clothing, etc. The heat transfer takes place through several avenues: evaporation, radiation, convection, and sometimes conduction. The physiologic strain of dissipating metabolic heat from the body core is the result of a half-dozen functions: metabolic heat production, external work, skin temperature, circulation of blood to vital organs and skin, and sweat production.

Studies have shown that there are definite physiological changes when workers are exposed to a high-temperature working environment.

The pulse rate and rectal temperature of the worker are good indicators of the heat strain experienced by the worker. However, this type of monitoring is not practical when hundreds of employees performing various tasks are involved, and each task imposes a different heat load to the respective worker. Moreover, each worker has a different tolerance to the heat stress because of varying acclimation and differing physical and physiological characteristics. Several techniques have been developed to relate environmental measurements to the physiological responses of the workers. The most recently developed technique, the Wet-Bulb Globe Temperature (WBGT) index is the relationship employed in both the OSHA and MESA recommendations.

The WBGT index is based on measurements of temperature alone. For indoor evaluation of heat stress, that is when solar radiation is absent, the index is defined as:

$$WBGT = 0.7 WB + 0.3 GT$$

where WB is the temperature measured by a wet-bulb thermometer. The GT is the black-globe temperature, which is determined by a thermometer placed in the center of 6-in diameter thin copper spherical shell which is painted flat black. For outdoor use, the dry-bulb or normal temperature reading, DB, becomes a part of the WBGT formula:

$$WBGT = 0.7 WB + 0.2 GT + 0.1 DB$$

Both the OSHA³⁹ and MESA⁴⁰ recommendation utilize the NIOSH⁴¹ definition of a hot working environment:

"Hot environmental condition means any combination of air temperature, humidity, radiation and wind speed that exceeds a WBGT of 79°F."

For environments which exceed this WBGT limit, the OSHA recommendations suggest acceptable threshold WBGT values for various types of work loads (see Table 7.1). The most appropriate work load level for retrieval operations within the repository appears to be "moderate" during which the air flow will be high. Thus the appropriate threshold WBGT value would be 87°F. The American Conference of Governmental Industrial Hygienists have adapted threshold work/rest ratios for moderate work loads in hot environments as a function of WBGT value.⁴² These work/rest ratios are

Continuous work.....	80.0°F WBGT
75% work, 25% rest each hour.....	82.4°F WBGT
50% work, 50% rest each hour.....	84.9°F WBGT
25% work, 75% rest each hour.....	88.0°F WBGT

The MESA recommendations suggest that these WBGT values for a given work/rest ratio can be exceeded if work practices and medical surveillance are effected to ensure that the worker's body temperature does not exceed 100.4°F.

Figure 7-1 gives a comparison of WBGT values that are predicted for various humidity conditions within the present high-level waste repository design during retrieval operations. The WBGT values presented in this figure were computed for the burial of 3.5 kW canisters at a 17.5 foot pitch in each 558 ft. room. It was assumed that the room had been sealed for 5 years prior to the retrieval operations at which time 75⁰ air was pumped into the rooms. The WBGT values presented in Figure 7.1 represent maximum values expected within a repository room under the assumed conditions, that is, near the ventilation exhaust at the end of room.

In order that WBGT values for other repository conditions may be estimated readily Table 7.2 is included. Knowing the wall temperature and the dry air temperature at a given location within a repository, the WBGT can be estimated for an assumed percent humidity.

Table 7.1

OSHA RECOMMENDED THRESHOLD WBGT VALUES

WORK LOAD	Threshold WBGT Values (^o F)	
	Low air velocity (Up to 300 fpm)	High air velocity (300 fpm or above)
Light (Level 2) (200 kcal/hr or below)	86	90
Moderate (Level 3) (201 to 300 kcal/hr)	82	87
Heavy (Level 4) (Above 300 kcal/hr)	79	84

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Figure 7-1 CALCULATED WBGT VALUES IN A VENTILATED NPTS STORAGE ROOM

TABLE 7.2

WET BULB GLOBE TEMPERATURES FOR VARIOUS
NPTS REPOSITORY CONDITIONS

Dry Air Temp. (°F)	Percent Humidity	Wet-Bulb Temp. (°F)	WBGT (°F) For Wall Temperatures (°F)			
			100	120	140	160
70	10	46	62	68	74	80
	30	53	67	73	79	85
	50	58	71	77	83	89
	70	63	74	80	87	93
	100	70	79	85	92	98
90	10	58	71	77	83	89
	30	67	77	83	89	95
	50	74	82	88	94	100
	70	81	87	93	99	105
	100	90	93	99	105	111
120	10	76		89	95	101
	30	88		98	104	110
	50	98		105	111	117
	70	108		112	118	124
	100	120		120	126	132

8. DEVELOPMENT OF NEARBY FACILITIES

It is expected that some repository operations would be performed near regions which contain stored high-level waste materials. For example in the design of Reference 8, new panels would be excavated 30 ft from a panel which may contain stored waste. It is possible that the thermal environment associated with these previously stored waste materials could affect nearby operations. Appropriate heat transfer calculations were performed which provide temperature profiles in the affected regions. Temperature increases due to previously stored waste materials in a region where excavation for a new panel of storage rooms would take place are shown in Figure 8-1. The calculation assumed a sealed panel of storage rooms without ventilation and that all of the waste had been loaded simultaneously and instantaneously. Even for this extreme case the temperature increases at 30 feet from the panel are not extreme. Less than a 20 F° increase was experienced 5 years after the placement of the waste. The anticipated excavation operation would probably take place before this time. It was also determined that spatial variation of temperatures due to the nonuniform distribution of waste canisters in the sealed panel are less than 2 F° at 30 feet from the panel. Hence the smoothing effect of the large thermal conductivity of salt eliminates the concern about particular hot spots at this distance.

In the more realistic situation, temperature profiles would be slightly lower than those shown in Figure 8-1 since the waste materials would necessarily be loaded more gradually and in the presence of ventilation. For example, the arrival rate of 3.5 kW canisters is expected to be about 10 per day.⁴³ This loading schedule would reduce each of the profiles shown in Figure 8-1 by less than 2 F°. The effect of ventilation during the loading operation would also be small since normally a panel would be filled in less than one year and the temperatures near a storage room are virtually unaffected after a 1 year initial period of ventilation (this effect can be seen in Figure 3-8). Hence the curves in Figure 8-1 are not overly conservative and provide a reasonable indication of the temperatures to be expected in the formation near the sealed storage panel. It does not appear that these temperatures represent a serious restriction upon operations in the affected regions.

Figure 8-1 TEMPERATURES NEAR STORAGE ROOMS CONTAINING HIGH-LEVEL WASTE

9. THERMAL ENVIRONMENTS NEAR 3.5 kW CANISTER

As part of the design process, several thermal criteria were established.^{1,44} One of these criteria states that no more than 1% of the salt volume within a unit cell and between horizontal plane at the top and bottom of the waste shall exceed 482°F (250°C).

For the present configuration of one row of HLW canisters per room with a pitch of 17.5 feet between canisters and 78 feet between room centerlines, 1% of the salt volume would be contained in a 14-inch annulus around the canister or sleeve. A second criterion is that no more than 25% of the salt in this unit cell shall exceed 392°F (200°C). This volume corresponds to an annulus nearly 10 feet thick around the canister.

In order to determine whether these criteria might affect the design of a repository for 3.5 kW HLW canisters, the thermal environments in the immediate vicinity of such a canister were calculated. This calculation required a more detailed model of the canister/overpack package and its immediate surrounding than was used in the ventilation calculations.

The model for this analysis employed an infinite array of canister unit cells. Each unit cell was represented as a right circular cylinder of 21-foot radius having an initial areal thermal power density of 108.5 kW/acre. Details of the canister, overpack, concrete plug and the crushed salt were explicitly taken into account and are shown in Figure 9-1.

Temperatures in the immediate vicinity of the waste canister peak about 5 years after emplacement. The isotherms for this time are presented in Figure 9-2. From this figure, it is seen that the bedded salt temperatures several inches beyond the crushed salt region reach no more than about 300°F (well below the 482°F criteria). A more detailed presentation of temperature versus distance from the centerline of the canister is shown in Figure 9-3. This profile is evaluated at the midwaste level; consequently, the temperatures shown in Figure 9-3 are maximum temperatures for the unit cell at the specified time. From Figure 9-3, it is seen that none of the rock salt exceeds 392°F (the 25% salt volume criterion) crushed salt will reach 482°F. Hence the predicted temperatures for the 3.5 kW canisters fall well within the thermal criteria for the unit cell.

Figure 9-1 DETAILS OF WASTE RECEPTACLE AND CANISTER

PA0818-11-1

Figure 9-2 ISOTHERMS IN IMMEDIATE
VICINITY OF CANISTER
TIME = 5 YRS

PA0818-10-3

**Figure 9-3 TEMPERATURE vs RADIAL
DISTANCE FROM A 3.5 kw
CANISTER, 5 YEARS AFTER BURIAL**

In these calculations the thermal conductivity of the crushed salt was taken to be 10% of that for the rock salt. The temperatures in the canister depend strongly upon the thermal properties of the crushed salt and materials in the canister. However the temperatures in the rock salt were influenced very little by this parameter. Analytic solutions treating the canister as an infinitely long cylinder (which provides a reasonable estimate for temperatures near to the canister) imply that temperatures in the rock salt are essentially unaffected by the description of the canister while the temperatures in the canister/overpack/crushed salt environment are inversely proportional to the conductivity, depend upon the radii of interfaces between materials and are reduced by radiation across air gaps. These analytic solutions are described in Appendix B.

In order to compare these effects for the canister, additional results of calculations for the detailed unit cell after 5 years of storage are shown in Figure 9-4. The result for the standard model shown in Figure 9-3 with a waste conductivity of $0.58 \text{ BTU/hr-ft-}^{\circ}\text{F}$ are repeated in Curve 1 of Figure 9-4, while the situation for Curve 2 is the same except that the waste conductivity is increased to 5.8. For both of these calculations the emissivity for the steel surfaces was assumed to be 0.7. Curve 3 was calculated with an emissivity of 0., that is, for no radiation across the air gap. The dependence of the temperature increases upon radiation and thermal conductivity suggested by the closed-form solutions of Appendix B appears to be qualitatively correct. The full calculations of internal canister temperatures shown in Figure 9-4 agree to within 15% with the predictions of the analytic model. However, the closed-form solutions are less reliable outside the canister; and for a thermal conductivity which is temperature-dependent, the results of Appendix B cannot be used at all. Nevertheless the main conclusions remain true: temperatures within the canister can be significantly reduced without a corresponding increase in the exterior environment by increasing the thermal conductivity, by increasing radiative transport or by increasing the diameter of the canister.

PA0818-14-1

**Figure 9-4 TEMPERATURE vs. RADIAL
DISTANCE FROM A 3.5 kw
CANISTER, 5 YEARS AFTER BURIAL**

10. CONCLUSIONS

This study was primarily concerned with a parametric analysis of the cooling characteristics of a typical ventilated storage room in a HLW repository. The thermal environments associated with the typical storage room were calculated for various assumed modes of repository operation. Existing heat transfer codes were found to be inadequate for the required analysis and an essentially new calculational procedure was developed to meet the special needs of this investigation.

It was clearly shown that a unique minimum cool-down wall temperature exists for each assumed mode of repository operation. The value of this minimum temperature was shown to be dependent upon the inlet air temperature, the ventilation flow rate, and the assumed value of the convective heat transfer coefficient. Further, it was found that this cool-down limit was essentially independent of the elapsed time between panel closure and the initiation of ventilation as well as on the ventilation history of the room. This minimum temperature was found to depend in a nearly linear fashion on the inlet air temperature for the range of parameters considered. Also, the ventilation time needed to reach this limit is nearly independent of the inlet air temperature.

The time required to cool the ventilated room to a particular operating temperature was also shown to depend on the elapsed time before ventilation as well as on the inlet air temperature and the assumed value of the convective heat transfer coefficient. This cooling time was found to depend on the ventilation flow rate for flows less than 50,000 cfm and became essentially independent of the flow rate for larger flows. Further, preliminary cooling of a storage room did not significantly decrease the cooling times and a continuous low flow rate of air through the room during storage was found to have a negligible effect on the required cooling times. An interesting and important generalization is that for most of the operational modes considered, the cooling times required to achieve satisfactory retrieval conditions were on the order of months.

P The effect of variations in the basic repository design on the cooling characteristics of a storage room were also studied. Increasing the room size which effectively increases the surface area for heat transfer made only a modest impact on the cool-down temperatures. Modifications in canister design had virtually no effect on the cooling. However, a deeper emplacement and/or using a longer canister did lower the operating temperatures and cooling times. Further, decreasing the areal heat load in the repository did significantly affect the cooling characteristics. Finally, cooling times were found to be nearly independent of the thermal conductivity of the host geologic formation

It was determined that excavation and other repository operations would not be restricted in regions adjacent to rooms already containing stored wastes. The temperature increases beyond about 30 feet from a sealed panel were found to be less than 20 F°. Also, both the 1% and 25% unit-cell thermal criteria for the repository were easily satisfied by the thermal conditions associated with the repository design under consideration.

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APPENDIX A.

ITERATIVE SOLUTION OF REPOSITORY ROOM ENVIRONMENTS

In specifying the thermal environments within a repository room, both the maximum air temperature and the maximum room surface temperature occur at the end of the room at which the ventilation exhaust is located. It would be possible to determine these temperatures utilizing a full, 3-dimensional computer simulation of the repository room taking into account the mass transfer down the length of the 558-foot storage room. Such an approach can be accomplished using a heat transfer code like TRUMP which explicitly considers mass transfer and its impact on heat transfer. However, such a computation is very complex and expensive because of the extreme mesh spacing and time step requirements which must be met to generate accurate results. The complexity leads to excessive model set-up time, long computer execution times, core-size limitation problems and a lack of flexibility within the model to perform parametric analyses.

Fortunately, such a comprehensive calculation may not be necessary in many cases. Three-dimensional unit-cell calculations¹ have demonstrated that the heat transfer rate, and consequently the air and room surface temperatures, vary slowly with distance down the repository room. This variation is sufficiently slow that an iterative solution of the heat and mass transfer equations for a succession of incremental slices of the repository will yield the temperatures as a function of distance down the repository room. The incremental equation to be solved is

$$F\rho C_p \Delta T_a = hS(T_w - T_a).$$

¹Thermal Operating Conditions in a National Waste Terminal Storage Facility, Y/OWI/Sub-76/47950, Science Applications, Inc., September 1976.

In this equation,

F = air flow rate

ρ = air density

C_p = specific heat capacity of the air

ΔT_a = local temperature change of the air

h_c = local convective heat transfer coefficient

S = surface area of the incremental slice of the repository

T_w = surface temperature of the incremental slice of the repository room

T_a = air temperature in the incremental slice of the repository room

For each iteration the increments are chosen sufficiently small that variations in temperatures across each slice do not prevent convergence of the iterative procedure. In order to guarantee this convergence the transfer time across each slice must be much smaller than the characteristic time for transfer of heat to the air from the walls. The latter rate of heat transfer is of the order of an hour or more so that for normal air velocities the procedure will be accurate even for storage room slices as large as 18 ft.

The wall temperature and heat transfer history for each incremental slice of the repository room is obtained from standard heat transfer codes using either an explicit mass transfer representation of the air flow or representing the heat removal through transfer of heat to a sink of fixed bulk temperature. These histories are then parameterized in terms of the bulk air temperatures and heat transfer coefficients down the storage room and these parameterizations are used to solve the heat transfer equations. The iterative solution to these heat transfer equations for the temperatures for the entire storage room agree with the full three-dimensional mass-transfer calculations to better than 4% which is within the uncertainty expected for such large-scale heat transfer calculations.

The program written to perform the complete storage room ventilation calculations is compact and efficient. It uses as input floor,

P ceiling and wall temperatures calculated with the HEATING5 and TRUMP programs for various choices of the independent variables of air temperature and convective heat transfer coefficient. The output of the program is storage room floor and air temperatures as a function of the inlet air temperature and air flow rate for any parameterization of the heat transfer coefficient.

APPENDIX B.

ANALYTIC SOLUTIONS FOR TEMPERATURES IN CYLINDERS

For the simple case of infinitely long cylinders where the heat flow is radial only, the appropriate equation is

$$\rho C_p \frac{\partial T}{\partial t} = \frac{1}{r} \frac{\partial}{\partial r} \left[k r \frac{\partial T}{\partial r} \right] + q$$

where ρ is the density, c_p is the specific heat capacity, k is the thermal conductivity and q is the volumetric heat generation rate. When the time rate-of-change of the temperature is small, as when temperatures peak or when heat capacities are small, the equation reduces approximately to

$$\frac{1}{r} \frac{\partial}{\partial r} \left[r \frac{\partial T}{\partial r} \right] + \frac{q}{k} = 0.$$

The solution to this equation is

$$kr \frac{\partial T}{\partial r} = C - \frac{qr^2}{2}$$

for the flux and

$$T = \frac{C}{k} \ln r - \frac{qr^2}{4k} + D$$

for the temperature. Here C and D are constants to be evaluated from the boundary conditions. It is assumed that flux across any interface is conserved. This is equivalent to the earlier assumption that the time

variation of temperature is slow. Then, since the temperature at the origin is finite, the solutions are

$$T = T_0 - \frac{qr^2}{4k_0}$$

in the inner heat-generating region and

$$T = T_i - \frac{qR_i^2}{2k_i} \ln \frac{r}{R_i}$$

in the cylindrical shell with inner radius R_i and conductivity k_i . T_0 and T_i are the temperatures at the origin and on the inner surface of the shell, respectively. In this case temperature increases are inversely proportional to the conductivity of the region and depend logarithmically upon the radius.

A list of temperature increases across each material in the configuration of Section 9 is given in Table B.1 along with values calculated from the equations for the one-dimensional model. The temperature increases are essentially independent of increases in other regions and can be greatly influenced by an appropriate choice of conductance properties and geometry. The metal overpack and canister walls both help to reduce temperatures within the canister. The values in Table B.1 also reflect the importance of radiation in decreasing the temperature drop across the air gap, reducing it from about 500 F° without radiation to about 50 F°. The temperatures in the salt bed are essentially independent of these effects. The overall temperature rise is proportional to the heat generation rate per unit length. However, this parameter also determines the flux of heat into the salt bed and thus influences temperatures in that region as well.

TABLE B-1

TEMPERATURE INCREASES ACROSS MATERIALS NEAR CANISTER AFTER 5 YEARS STORAGE.

Material	Conductivity BTU/hr-°F-ft	R_a (ft)	R_b (ft)	T (°F)	ΔT_{model} (F°) ¹
Waste	0.58	0.000	0.500	172.	173.
Stainless Steel Wall	9.50	0.500	0.531	1.25	1.27
Air Gap	0.02	0.531	0.563	54.2	51.3 ²
Carbon Steel Overpack	26.00	0.563	0.583	0.26	0.27
Crushed Salt	0.16	0.583	0.750	300.	315.

1) ΔT_{model} calculated from closed-form expressions for infinite cylinders.

2) Calculated temperature rise includes approximate correction for radiation across gap. Without this correction, one-dimensional model gives temperature rise of 514 F°.