MASTER

**TECHNICAL SUPPORT FOR GEIS:
RADIOACTIVE WASTE ISOLATION
IN GEOLOGIC FORMATIONS**

Volume 19

Thermal Analyses

April 1978

Prepared By

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**UNION
CARBIDE**

**OFFICE OF WASTE ISOLATION
OAK RIDGE, TENNESSEE**

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TECHNICAL SUPPORT FOR GEIS:
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Volume No.	Volume Title	Prepared by*
TM-36/1	Executive Summary	SAI
TM-36/2	Commercial Waste Forms, Packaging and Projections for Preconceptual Repository Design Studies	SAI
TM-36/3	Stratigraphies of Salt, Granite, Shale, and Basalt	D&M
TM-36/4	Baseline Rock Properties-Salt	D&M
TM-36/5	Baseline Rock Properties-Granite	D&M
TM-36/6	Baseline Rock Properties-Shale	D&M
TM-36/7	Baseline Rock Properties-Basalt	D&M
TM-36/8	Repository Preconceptual Design Studies: Salt	PBQD
TM-36/9	Drawings for Repository Preconceptual Design Studies: Salt	PBQD
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TM-36/13	Drawings for Repository Preconceptual Design Studies: Shale	PBQD
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TM-36/15	Drawings for Repository Preconceptual Design Studies: Basalt	PBQD
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TM-36/19	Thermal Analyses	SAI
TM-36/20	Thermomechanical Stress Analysis and Development of Thermal Loading Guidelines	D&M
TM-36/21	Groundwater Movement and Nuclide Transport	D&M
TM-36/22	Nuclear Considerations for Repository Design	SAI
TM-36/23	Environmental Effluent Analyses	SAI

*These documents were prepared by Science Applications, Inc. (SAI), 800 Oak Ridge Turnpike, Oak Ridge, Tennessee 37830; Dames & Moore (D&M), 20 Haarlem Avenue, White Plains, New York 10603; and Parsons Brinckerhoff Quade & Douglas, Inc. (PBQD), 1 Penn Plaza, New York, New York 10001.

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ABSTRACT

This volume, Y/OWI/TM-36/19, "Thermal Analyses," is one of a 23-volume series, "Technical Support for GEIS: Radioactive Waste Isolation in Geologic Formations," Y/OWI/TM-36, which supplements the "Contribution to Draft Generic Environmental Impact Statement on Commercial Waste Management: Radioactive Waste Isolation in Geologic Formations," Y/OWI/TM-44. The series provides a more complete technical basis for the preconceptual designs, resource requirements, and environmental source terms associated with isolating commercial LWR wastes in underground repositories in salt, granite, shale and basalt. Wastes are considered from three fuel cycles: uranium and plutonium recycling, no recycling of spent fuel and uranium-only recycling.

This volume discusses the thermal impacts of the isolated high level and spent-fuel wastes in geologic formations. A detailed account of the methodologies employed is given as well as selected results of the analyses.

PREFACE

Project Background

One of the major problems related to the production of electricity by light-water nuclear reactors is the management of radioactive wastes generated by the use of nuclear fuel. However, the subject is considered amenable to a rational solution, and the technology involved is considered to be well within the capabilities of our present-day technological base.

An important step toward the realization of an effective waste management program is the preparation of the generic environmental impact statement for commercial waste management. A pivotal issue in waste management is the means of providing long-term, permanent storage of these wastes in a manner to best assure their isolation from the biosphere. Analyses spanning two decades have generated the widely supported concept for providing final isolation of these nuclear wastes in deep geologic formations. Therefore, the Office of Waste Isolation* was assigned the responsibility for preparation of those sections of this generic statement dealing with deep geologic waste isolation.

The original concept for deep geologic disposal was first advanced in 1957 when a National Research Council Advisory Committee of the National Academy of Sciences suggested the burial of solid radioactive wastes in salt deposits. To date, the majority of the research, development, and demonstration (RD&D) activities have been in salt. The current United States Department of Energy (DOE) National Waste Terminal Storage (NWTS) program calls for the selection of two sites overlying suitable salt formations by 1979, followed by the construction and start-up in 1985 of one NRC-licensed repository designed for the permanent disposal/isolation of commercial nuclear wastes in a salt formation at one of these two sites. In addition to this activity in salt, vigorous RD&D programs have been initiated to determine the appropriateness of various hard rocks as host media for a repository.

*Operated by Union Carbide Corporation-Nuclear Division for the Department of Energy.

The deep-geologic-isolation portion of the generic statement considers repositories located in salt, granite, shale, or basalt. The repositories are designed to handle wastes from the nuclear fuel cycle in which the spent fuel is considered a waste (no reprocessing), or from either of two fuel cycles that include reprocessing--the cycle with uranium and plutonium recycle and that with only uranium recycle. To prepare this contribution, the Office of Waste Isolation contracted with Dames & Moore, Parsons Brinckerhoff Quade & Douglas, Inc., and Science Applications, Inc. In order to prepare this description, generic sites were defined, preconceptual repository designs completed, and resource requirements and effluents from the repositories identified. The preconceptual repository designs for the salt host formation were based on more than two decades of analysis and in situ experimentation. The data base upon which the repository design for the non-salt host formations were based is more more sparse since repository-oriented analyses of these formations have been proceeding only for the last couple of years. For each of the host rocks additional analyses were performed during the conduct of these studies. Details of this additional technical work are described within the twenty-three volumes of this report.

For an overview of these preconceptual repository design studies, the study objectives and scope, and the major study assumptions, the reader is referred to Volume 1 of this series, the "Executive Summary" (Y/OWI/TM-36/1).

Volume Summary (Y/OWI/TM-36/19)

Of major concern for the repository is the heat generated by the waste. Heat generation, particularly for the high-level wastes and spent fuel assemblies is expected to be intense for long periods of time and may have important impacts on the waste matrix itself, on the protective canister, and on the geologic formation. Nearly all of the thermal analyses for this report have been performed by Science Applications, Inc. with supplementary work performed by Dames & Moore, and Parsons Brinckerhoff Quade and Douglas. Temperature fields generated in the repository have been utilized by Dames & Moore for the thermomechanical stress studies of

the repository (see TM-36/20, Thermomechanical Stress Analysis). Thermal analysis at distances far removed from the repository have been used by Dames & Moore to study the geo-hydrology of the formation (see TM-36/21, Ground Water Movement), by Dames & Moore and Science Applications, Inc. to study nuclide transport behavior in the formation (see TM-36/21 Nuclide Transport) and by Dames & Moore to study surficial uplift (see TM-36/20, Thermal Loading Guidelines). Detailed waste package response to the thermal impacts has been studied by Science Applications, Inc. (see this volume) and the role of convective heat removal has been analyzed by Parsons, Brinckerhoff, Quade and Douglas (see this volume). The thermal loadings based on these analyses have been used by Parsons, Brinckerhoff, Quade and Douglas to determine mine layout and mining schedules. The major role of the analyses in this volume has been to provide the predicted temperature fields for the detailed studies mentioned. These studies have resulted in limits to the thermal loadings such that no adverse consequences occur in these study areas. These limits are a function of the geology as well as the waste form in the repository. The controlling factor determining these limits appears to be the requirement for storage-room stability and waste-package integrity throughout a suitable retrievability period. Other factors such as long-term consequences and impacts upon nuclide migration are important but appear to be much less restrictive than those imposed by the retrievability option.

1.0 INTRODUCTION

1.1 OVERVIEW

This volume examines the thermal impacts from the isolation of radioactive wastes in preconceptual repository designs in various geologic formations. The analyses reported here discuss the temperature fields and histories resulting from three different waste types from alternative LWR fuel cycles:

- Reprocessed waste resulting from uranium and plutonium recycle (Total Recycle or Cycle I)
- Spent unprocessed reactor fuel (SURF or Cycle II)
- Reprocessed waste resulting from the recycle of uranium only (U-only Recycle or Cycle III).

Specific characteristics and composition of these various wastes are detailed in Volume 2 of this document (Y/OWI/TM-36/2 Commercial Waste Forms).

A single-level repository for these wastes is assumed to lie approximately 2000 ft below the surface of the earth in a geologic formation of one of several generic types:

- Domed or bedded salt
- Granite (crystalline)
- Shale (argillaceous)
- Basalt (volcanic)

The repository studied here consists of underground facilities including mined storage rooms with holes in the floors of the storage rooms for the emplacement of canisters containing the waste material. The temperature fields resulting from the emplacement of these heatgenerating wastes will have consequences in the three regions: (1) the near-field region encompassing the storage rooms and pillars between the storage

rooms, (2) the very-near-field, which includes the immediate vicinity and the interior of the waste canisters, and (3) the far-field, that is, at distances much removed from the sources. The emplacement loads are limited by these consequences. The major intent of these studies is to determine the thermal loading limits based upon criteria imposed in the near-field, very-near-field, and far-field zones.

Associated with the studies reported here, near-field temperatures have been provided to thermo-mechanical stress analysts to determine impacts on storage-room closure (TM-36/20, Thermo-mechanical Stress Analysis). In addition, thermal studies for the very-near-field have been used to determine loadings for individual canisters which would not result in thermal degradation of the canister prior to the end of a retrievability period. Separate studies of the long-term, far-field temperatures have been made and the results have been used in the geologic-stress analysis (TM-36/20, Thermo-mechanical Stress Analysis) and in the hydrological studies in order to determine water flow behavior in the vicinity of the repository (TM-36/21 Ground Water Movement).

Earlier work has already provided considerable thermal analysis for repositories in salt,¹⁻⁴ and these studies have served as benchmarks for the present analysis in granite, shale, and basalt. At the same time new results for salt are also reported here. The remainder of this section outlines the methodologies used in these studies and presents input data relevant to the thermal analysis.

Section 2.0 gives the detailed quantitative approaches and some selected results for the near-field studies. Section 3.0 provides the details and results of the very-near-field analyses. The technical approach and selected results from the far-field studies are presented in Section 4.0. The analysis for convective heat removal from the canister and storage room is given in the Appendices. This work has been performed by Science Applications, Incorporated. Additional studies for the convective heat removal performed by Parsons, Brinckerhoff, Quade and Douglas, Inc. are described in the Supplement to this volume.

1.2 METHODOLOGY

1.2.1 Physical Models

1.2.1.1 Room-and-Pillar Models

In these analyses the zone in the vicinity of the storage rooms is referred to as the near-field. Of greatest concern in this case is an understanding of temperatures in this zone only in the early time periods during which operations would take place within the storage rooms. Accurate knowledge of the immediate vicinity of the canisters (the very-near-field), or at distances far removed from the storage rooms and yield pillars, are not emphasized in this case. As shown later in this report, the near-field temperatures are not strongly influenced by precise details of the very-near-field, and the time periods of interest are sufficiently short that far-field effects are also of little consequence. The room-and-pillar thermal response is primarily influenced by the average waste emplacement density, the time-dependent heat generation rate, the average depth of the heat sources relative to the storage rooms, the thermal properties of the rock in the near-field, and the size of the storage rooms. Also of importance are the amount of ventilation through the room and thermal radiation behavior in the storage rooms.

In general, the increased temperatures in the vicinity of the storage room will induce thermo-mechanical stresses in this region. An understanding of the temperatures in this zone, without recourse to detailed knowledge of the very-near-field or far-field, should provide the opportunity for a reasonable understanding of the temperature-induced stresses. At the same time, large-scale integrity of the host rock and its influence on the room-and-pillar zone will not be well understood in such a model. Similarly, the very-near-field stresses induced around canisters cannot be analyzed from these data since the very-near-field temperatures are not accurately represented.

1.2.1.2 Very-Near-Field Models

By very-near-field is meant the zone in the immediate vicinity of waste canisters and interior to the waste package. Peak temperatures are usually achieved in this case very early, in the first five years or less.

For very close canister spacings, peak temperatures may occur at slightly later times. Factors which most strongly impact the very-near-field are details of the canister design, very-near-field thermal properties, individual canister thermal loadings, and average canister spacing. In this case, accurate treatment of the storage rooms or other aspects of room-and-pillar zone (the near-field) do not have a major influence on the thermal response of this zone other than in the average temperature field provided by the surrounding canisters.

Temperatures calculated for the interior of the waste package include the effects of details in the package design in such a model. Calculated temperatures somewhat removed from the canister (more than 5-10 feet away for the generic formation materials considered) are not expected to be accurate. These may, however, provide representation of averages over the temperature distribution at these distances.

1.2.1.3 Far-Field Models

By far-field is meant a region sufficiently far removed from the source that details of the near-field do not significantly have a bearing. In this case, the distances of concern are on the order of hundreds of feet away from the repository, and the time periods of interest are hundreds of years or longer. The most important factors for the far-field response is the average waste emplacement density, the average formation thermal properties, stratigraphy, and geometry, and the long-term time-dependence of the waste-heat generation rate. In addition, heat-removal effects due to ventilation of the repository during the operational phase or due to water flow in aquifers and heat removal at the surface of the earth can be of importance. Far-field models provide temperature fields of use in understanding the long-term thermal and mechanical response of the repository host formation and its surrounding strata. These models, however, cannot be expected to provide accurate representation of the temperatures within the repository itself.

1.2.2 Thermal Criteria

The waste emplacement in the repository is necessarily limited by the consequences of the thermal loadings. Some of the limiting conditions are⁵

- Near-Field
Room and pillar stability through suitable periods of time to allow the option of retrieval of the canisters.
- Very-Near-Field
Canister recoverability throughout the retrievability period and waste-matrix integrity for even longer periods.
- Far-Field
 - Limited temperature rises in overlying aquifers and near the surface of the earth.
 - Integrity of host rock and surrounding rock units.
 - Limited surficial uplift and subsidence.

The impact of emplacement upon these conditions is studied through a step-by-step approach utilizing each of the three models. Near-field temperature profiles and histories are analyzed to determine an envelope of local thermal loading densities (kW/acre) and average canister spacings that suggest the storage rooms will remain open throughout the retrieval period. Based on these average loadings, very-near-field studies are then conducted to determine maximum canister heat-generation rates which predict canister recoverability during this period. These studies may require a modification in the local areal loading limits. Finally, far-field studies are conducted to determine if the total heat loadings for the repository based on the average local loadings satisfactorily meet the far-field conditions. In the event it becomes necessary, the average local loading can then be reduced. In this fashion the criteria result in design parameters based upon these thermal loading limits. Once the design has been established, actual thermal conditions in the repository appear to be controllable in terms of the choice of actual thermal loadings.

1.2.3 Quantitative Models

1.2.3.1 Approximations

The thermal analysis reported here is subject to certain assumptions and approximations. Basically, the models must be simple in order to allow reasonable and tractable analysis. At the same time, the simplifications made must be consistent with the requirement that quantitative results be conservative.

The following approximations help in achieving these goals:

- Only high-level waste and spent-fuel canisters are considered.
- The entire repository is assumed to be loaded simultaneously and instantaneously.
- Thermal properties based on reasonable estimates are assumed.
- The effects of stress upon thermal properties are not included.
- The presence of water is neglected in the thermal analysis.
- Heat removal due to convection in storage rooms or around canisters or due to water flow is generally neglected.*
- Only simplified horizontal stratigraphies are assumed.
- No compaction or subsidence of the formation is considered.

Additional simplifications for a specific model are detailed later.

1.2.3.2 Numerical Models

Data for the impact of emplacement of waste in deep geologic repositories are generally lacking. Project Salt Vault¹ provided valuable information; but, in general, understanding of thermal impacts must rely on predictive models which can be benchmarked to experimental data as they become available. Relatively sophisticated models for this analysis have

*The specific effects of convective cooling of the canisters and storage rooms are discussed in the appendices. Moreover heat removal due to storage-room ventilation has been discussed in Reference 6.

been developed,^{2-4, 6-14} and many of these have been incorporated and updated here for these studies. These quantitative models are sketched here:

- Room-and-Pillar Models

The repository is modeled as an infinite array of storage rooms of unlimited length, with idealized waste canisters buried below the floor of the storage rooms. In this case a unit cell containing a single canister is studied. The remaining canisters and storage rooms are taken into account with adiabatic boundary conditions at planes of symmetry. Upper and lower boundaries are sufficiently far removed that they have no impact on the storage room temperatures during the time periods of interest.

- Very-Near-Field Models

In this case a unit cell is also utilized. The cylindrical cell containing one canister has a radius corresponding to half the average canister separation. Insulating boundary conditions at the cylindrical, vertical wall of the unit cell effectively take into account the surrounding canisters in the infinite repository and provide conservative values for the temperatures near the canister. Again, upper and lower boundaries are sufficiently far removed from the region of interest. The storage room is not modelled but details of the canister are explicitly taken into account.

- Far-Field Models

The repository is modeled as a disc source, the heat-generation material assumed to be homogeneously spread throughout the disc. Vertical boundaries and the lower horizontal plane boundary are far enough removed from the region of interest to have no consequence during the time

span of interest. Layers of material corresponding to the generic stratigraphy are assumed to be laterally unbounded. Boundary conditions at the surface of the earth take into account an ambient geothermal flux, radiation from and to the sky, and convection.

In spite of simplifications imposed on these models, they are still too complex to be treated by analytic methods; therefore, numerical approaches are used. Analytic solutions, however, have been used as checks for the numerical approaches, particularly for the very-near-field and far-field cases. In these cases, idealized examples can be calculated by both techniques for comparison purposes. The numerical analysis has been made utilizing two different heat conduction codes, TRUMP¹⁵ and HEATING5,¹⁶ as well as supplementary routines and codes. TRUMP is a versatile finite-differences code for solving the diffusion equations and has been tested against a wide variety of problems for accuracy and efficiency of calculation. The HEATING5 code is the latest version in a series of HEATING programs which have been used successfully in general heat transfer problems for more than 10 years, and which have been useful in many types of heat-transfer problems associated with the nuclear fuel cycle. Both codes have been successfully applied to the heat-conduction problem associated with geologic waste isolation in the past several years.^{2-4,6,7} Details of the application of these codes to the present studies are described later.

1.2.3.3 Input Data

1.2.3.3.1 Waste Forms and Packaging

Complete descriptions of the various waste forms and packages are given in Y/OWI/TM-36/2. Appropriate data relevant to the thermal analyses are summarized here.

The canisters containing HLW or spent-fuel assemblies are assumed to be emplaced in the repository 10 years after discharge from the reactor. The calculations reported here have utilized the approximate relative thermal-decay rates for wastes from each of the three cycles shown in Table 1-1. The values given are for PWR wastes and are subject to modifications

TABLE 1-1
RELATIVE HEAT-GENERATION DECAY PROPERTIES OF
WASTES FOR EACH CYCLE^a

Year After Emplacement ^b	Total Recycle ^c	SURF Cycle	U-Recycle
0	1.0	1.0	1.0
2	0.91	0.96	0.96
3	0.87	0.89	0.90
4	0.84	0.86	0.87
5	0.81	0.84	0.85
6	0.78	0.82	0.83
10	0.69	0.75	0.76
15	0.60	0.69	0.70
20	0.52	0.62	0.64
25	0.46	0.57	0.59
50	0.25	0.39	0.41
500	0.015	0.080	0.076
5,000	0.0025	0.016	0.015
50,000	0.00029	0.0025	0.0024
500,000	0.000071	0.00043	0.000091

^aSee Y/OWI/TM-34, "Nuclear Waste Projections and Source Term Data for FY 1977"

^bAssumes waste is 10 years old at emplacement.

^cAssumes a 3:1 mixture of waste from U and MOX fuels.

on the order of 1 percent if BWR wastes are assumed. The values given were extracted from the source strengths for PWR wastes in Y/OWI/TM-36/2 derived from calculations with the computer code ORIGEN.¹⁷ Furthermore, the values for Cycle I can change by up to 5 percent depending upon the amount of mixed oxide ($\text{UO}_2 + \text{PuO}_2$) fuel blended with virgin UO_2 fuel. The results for the very-near-field calculations are virtually insensitive to these differences in the relative decay rates, since the peak temperatures of interest usually occur within the first three years after emplacement. The choice of a 1:3 blend of mixed oxide and UO_2 fuel rather than pure mixed oxide fuel results in less than a 3 percent decrease in the predicted temperature rises in the first 25 years after emplacement for the room-and-pillar calculations.

The high-level waste forms under consideration are calcined and borosilicate glass. Detailed descriptions for the waste packaging and the active volume in each case are given in Table 1-2. The power density for the waste from each cycle is also given in Table 1-2, although it should be kept in mind that the actual waste content in a HLW canister can be altered by the relative amount of inert material present. In the detailed calculations, the canisters are assumed to be overpacked, and the description assumed for the overpack is given in Table 1-3. Details of the spent-fuel assemblies assumed here are given in Table 1-4.

1.2.3.3.2 Material Thermal Properties

Thermal properties assumed for the materials are given in Tables 1-5 to 1-9. The thermal properties of HLW have not yet been established, but those given in Table 1-5 have been used in recent thermal analyses.¹⁸ For the very-near-field calculations, properties of the materials in the immediate vicinity of the canisters can be important. It is generally assumed that drilled holes into which canisters are placed are backfilled with crushed material extracted from the formation when the storage rooms and canister holes were created. The thermal properties assumed for this backfill material are given in Table 1-5. For all but the basalt formation the baseline thermal conductivities for the crushed material are arbitrarily taken to be 10 percent of the values for the intact formation material. The conductivity of crushed basalt is arbitrarily taken to be 0.15 Btu/hr-ft-°F.

TABLE 1-2
CANISTER DESCRIPTION

Cycle	Waste Type	Outside Diameter (in)	Nominal Wall Thickness (in)	Overall Length (ft)	Active Volume of Waste (ft ³)	Thermal Power of Waste ^e (Watts/MTHM)
I-Total Recycle	HLW (Glass)	12.75 ^a	0.375	10	6.28	1341 ^f
	HLW (Calcined)	8.625 ^b	0.322	10	2.79	1341 ^f
II-SURF Cycle	PWR-Spent Fuel	14 ^c	0.375	16		1193
	BWR-Spent Fuel	10.75 ^d	0.375	16		996
III-U only Recycle	HLW (Glass)	12.75 ^a	0.375	10	6.28	1250
	HLW (Calcined)	8.625 ^b	0.322	10	2.79	1250

^aIndicated canister is 12 in. schedule 40S pipe.

^b8 in. schedule 30S.

^c14 in. schedule 30S.

^d10 in. schedule 40S.

^eTen years after discharge from reactor.

^fAssumes 1:3 blend of mixed oxide (UO₂ + PuO₂) fuel and virgin UO₂ fuel. The initial thermal power for mixed oxide fuel wastes is 2268 watts/MTHM and for UO₂ fuel wastes is 1032 watts/MTHM.

TABLE 1-3
OVERPACK DESCRIPTION

Cycle	Waste Type	Outside Diameter (in)	Overall Length (ft)
I-Total Recycle	HLW (Glass)	14	11
	HLW (Calcined)	10.75	11
II-SURF Cycle	PWR-Spent Fuel	16	16.5
	BWR-Spent Fuel	12.75	16.5
III-U Only Recycle	HLW (Glass)	14	11
	HLW (Calcined)	10.75	11

Note: Wall thickness of all overpacks is 0.375 in.

TABLE 1-4
SPENT FUEL ASSEMBLY DESCRIPTION*

Item	Dimension
Length (ft)	12
Number of pins	289 (17 x 17)
Pitch-to-diameter ratio	1.36
Fuel rod OD (in)	0.38
Cladding thickness (in)	0.021
Fuel pin OD (in)	0.32

*See Reference Engineering Standardized Safety Analysis Report, Westinghouse Electric Company. 1974.

TABLE 1-5
THERMAL PROPERTIES OF MATERIALS.

Material	Density (lb/ft ³)	Heat Capacity (BTU/lb-°F)	Conductivity (BTU/hr-ft-°F)
Vitrified HLW ^a	187	0.20	0.7
Calcined HLW ^a	81	0.20	0.2
UO ₂ ^b	684	c	c
Zircalloy Cladding ^b	412	d	d
Helium ^e	0.004	1.24	0.13
Stainless Steel 304L ^c	488	0.11	9.5
Carbon Steel ^e	490	0.11	26
Air ^e	0.07	0.24	f
Concrete	144	0.2	0.54
Salt ^g	135	0.22	f
Crushed Salt	135	0.22	h
Granite ^g	167	i	f
Crushed Granite	167	0.2	h
Shale ^g	140	i	f
Crushed Shale	140	0.2	h
Basalt ^g	180	i	f
Crushed Basalt	180	i	0.15
Sand ^e	98	0.2	0.2

^aSee Larson, D.E., HLW Canister Thermal Characterization for the Sealed Cask Concept, AR14-LD-104, 1974.

^bSee Steam - Its Generation and Use, Babcock and Wilcox, 1972.

^cSee Table 1-6.

^dSee Table 1-7.

^eSee Kreith, F., Principles of Heat Transfer, International Press, Inc., 3rd Edition, 1973.

^fSee Table 1-8.

^gSee TM-36/4-7. (Baseline Rock Properties)

^hConductivity for crushed materials taken to be 10% of uncrushed materials. See Table 1-8.

ⁱSee Table 1-9.

TABLE 1-6
THERMAL PROPERTIES OF UO_2^a

Temperature (°F)	Heat Capacity (BTU/lb-°F)	Conductivity (BTU/hr-ft-°F)
300	4.05	0.056
1000	2.37	0.067
2000	1.56	0.072
3000	1.33	0.075

^aSee Steam - Its Generation and Use, Babcock and Wilcox, 1972.

TABLE 1-7
THERMAL PROPERTIES OF ZIRCONIUM ALLOY CLADDING ^a

Temperature (°F)	Heat Capacity (BTU/lb-°F)	Conductivity (BTU/hr-ft-°F)
70	0.070	8.17
200	0.073	8.42
400	0.077	8.92
600	0.081	9.42
800	0.085	10.0
1000	0.086	10.9
1200	0.086	11.8

^aSee Steam - Its Generation and Use, Babcock and Wilcox, 1972.

TABLE 1-8

ASSUMED THERMAL CONDUCTIVITIES FOR FORMATION MATERIALS AND AIR (BTU/hr-ft-°F)

T(°F)	Salt ^a	Granite ^a	Shale ^a	Basalt ^a	Air ^b
32	3.53	1.65	0.97	0.67	0.014
122	2.90	1.56	0.93	0.69	0.016
212	2.43	1.48	0.89	0.73	0.018
302	2.08	1.41	0.88	0.76	0.019
392	1.80	1.35	0.87	0.79	0.021
482	1.60	1.30	0.87		0.023
572	1.44	1.24	0.86	0.86	0.024
662	1.33	1.20	0.86		0.026
752	1.20	1.15	0.85	0.90	0.028

^aSee TM-36/4-7 (Baseline Rock Properties)^bSee Kreith, F., Principles of Heat Transfer, International Press, Inc., 3rd Edition, 1973.

TABLE 1-9
ASSUMED HEAT CAPACITIES FOR FORMATION MATERIALS (BTU/lb-°F)

T(°F)	Salt*	Granite*	Shale*	Basalt*
32	0.22	0.21	0.19	0.17
212	0.22	0.22	0.20	0.19
392	0.22	0.23	0.21	0.22

*See TM-36/4-7 (Baseline Rock Properties)

Properties for additional materials of detailed stratigraphies for the far-field calculations are provided in Section 4.

The ambient formation temperatures in each case are assumed to be described by the values in Table 1-10. These values were assumed in previous analyses for repositories in salt¹ and are consistent with the wide range of possibilities for the other formations as well (TM-36/4-7, Baseline Rock Properties). For most considerations, temperature rise over ambient is the item of interest; and the precise choice for the initial temperatures is therefore not important. Since the repository is generally taken to be at a depth of about 1500 to 2500 feet, the ambient formation temperature is assumed to have the form

$$T = T_0 + gz$$

where $T_0 = 51^\circ\text{F}$, z is the depth below the surface of the earth in feet and g is the assumed geothermal gradient, 0.021°F/ft .

TABLE 1-10
ASSUMED AMBIENT FORMATION TEMPERATURES*

Depth Below Earth's Surface (ft)	Temperature (°F)
0	59
400	63
750	72.5
1100	74
3500	124
4000	135

*See Bradshaw, R.L., and McClain, W.C., Project Salt Vault: A Demonstration of the Disposal of High-Activity Solidified Wastes in Underground Salt Mines, ORNL-4555, Oak Ridge National Laboratory, 1971.

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2.0 NEAR-FIELD CALCULATIONS

2.1 INTRODUCTION

It is anticipated that the storage rooms in the repository will need to remain open for some pre-determined period of time to allow the retrievability of the waste canisters. The primary objective of the near-field thermal analysis is to provide input to thermomechanical studies of room-closure conditions. The basic concern is that the induced heat load within a repository room be controlled sufficiently to predict acceptable creep in the case of salt repositories and to minimize the possibility of rock-fracturing or failure in the case of the non-salt repositories throughout the retrievability period. Accurate temperature profiles and histories in the pillars and in the rock surrounding the storage rooms are of greatest importance. The time periods of interest are on the order of 50 years or less.

The factors of most importance with respect to the near-field temperature distributions over these time frames include:

- Average local thermal loadings
- Storage room geometry and rock characteristics
- Average depth of waste below storage-room floor

The key factor in determining near-field temperatures appears to be the average local thermal loading. The local loading refers to heat-generation capacity associated with a single storage room including the width extending from pillar centerline to pillar centerline and is usually expressed in units of kW per acre. This loading is then determined by the individual canister loadings and the density of canisters in the storage room as well as the separation between storage rooms.

Temperatures in the vicinity of the storage room are affected by the volume of rock mined for the storage rooms and depend upon the heat capacity and density of the formation material. Also, radiative heat-transfer properties of storage-room surfaces can be of some influence.

The average depth of the emplaced wastes below the storage room affects the time at which peak temperatures are obtained in the pillars.

The most important approximations in the near-field models are:

- The unit-cell concept is applied
- Details of the waste package and the formation in the immediate vicinity of the canisters are not specified
- Effects of mechanical stress on thermal properties are neglected
- Ventilation and convection in the storage room are neglected.

The utilization of the unit cell enables the treatment of a small section of the repository so that fine resolution may be obtained. The remaining parts of the repository are accounted for with appropriate adiabatic boundary conditions. The repository is then viewed as an infinite, uniformly distributed array of unit cells, each containing a small number of canisters (in the case of this study, one). Furthermore, the unit-cell approximation implies that all the waste is loaded simultaneously. These approximations allow a degree of simplification and are conservative in that, if the gross area of the repository including haulage ways and other non-storage space and sequence of emplacement were included, perhaps lower temperatures would result.

For the generic formations considered, the actual details of the waste package are not of importance for distances greater than about 5 feet from the sources, as will be shown later.

The effect of stresses on thermal properties is neglected. Temperature dependence of these properties, however, is taken into account.

2.2 METHODOLOGY

The near-field unit-cell configurations that have been analyzed are shown in Figures 2-1 to 2-3. The unit cell in this case is represented

SALT FORMATION

SPENT FUEL

FIGURE 2-1. STORAGE ROOM AND CANISTER CONFIGURATION FOR SALT

FIGURE 2-2. STORAGE ROOM AND CANISTER CONFIGURATION FOR SHALE

FIGURE 2-3. STORAGE ROOM AND CANISTER CONFIGURATION FOR GRANITE AND BASALT

in Cartesian coordinates. The storage rooms are idealized as infinitely-long rectangular openings in the rock. Since the calculations are made utilizing an x-y-z coordinate system, special features such as rounded ceilings for stress relief are not explicitly included. These details were not felt to be important for the thermal analysis, although the height of the storage rooms was modified so that the storage-room volume corresponds to that of the design. In the storage room, any natural or forced convective heat transfer into the storage room is neglected but radiative heat transfer between walls and between floor and ceiling is included. Details of the canister package and of backfill around the canister are neglected. Only the active volume of the waste is modeled. Further, because the Cartesian coordinate system is used, the waste volume is modeled as a parallelepiped rather than as a cylinder. Earlier work has shown this leads to a less than 3 percent error in the calculated temperatures more than a few feet from the waste.²

Adiabatic boundary conditions are applied at the planes of symmetry at mid-pillar and down the room centerline to account for the surrounding heat sources. The boundary conditions at the top and bottom of the unit cell are taken to be isothermal, consistent with the assumed geothermal gradient. Nevertheless, the top and bottom planes are sufficiently far removed, 1000 feet above and below the storage room, that they have no impact in the first 25 years after waste emplacement.

Analytic models to check the numerical calculations are not practical. In place of an analytic verification, a check of the accuracy of the calculations was obtained by varying mesh spacings and time steps until convergence* to within 5 percent or better resulted. Typical mesh spacings chosen for the calculations are given in Tables 2-1 and 2-2 for the finite-differences approach.

* Convergence in this case means the agreement between results before and after modifications in the mesh structure and not in the sense of iteration. For example once a nodal structure has been found that is roughly self-consistent with temperature gradients in the problem, all mesh spacings are reduced. This process is repeated until the results before and after the reduction differ at most by 5 percent.

TABLE 2-1

MESH SPACING FOR NEAR-FIELD STUDIES IN GRANITE AND BASALT

z(depth below surface, ft)				x(distance from waste centerline normal to storage room, ft)		y(distance from waste centerline down storage room, ft)*
HLW		Spent Fuel		HLW	Spent Fuel	HLW
1000	2010	1000	2010	0.0	0.0	0.0
1400	2011	1400	2011	0.5	0.5	0.5
1700	2014	1700	2017.5	1.0	1.0	1.0
1800	2017	1800	2024	2.0	2.0	2.0
1850	2018	1800	2025	3.5	3.5	3.1
1880	2019	1880	2026	5.0	5.0	4.41
1905	2020.5	1905	2027	7.0	7.0	
1925	2023	1925	2028.5	9.0	9.0	
1940	2026	1940	2031	10.0	10.0	
1950	2031	1950	2036	11.5	11.5	
1960	2040	1960	2043	13.0	13.0	
1970	2050	1970	2050	15.0	15.0	
1977	2060	1977	2060	17.5	17.5	
1980	2070	1980	2070	20.5	21.5	
1990	2082	1990	2082	24.0		
2000	2100	2000	2100	29.0		
2150	2003	2150	2003	34.0		
2005	2300	2005	2300	39.0		
2007.5	2600	2007.5	2600			
2009	3000	2009	3000			

*Only needed for 3D models for HLW.

TABLE 2-2
MESH SPACING FOR NEAR-FIELD STUDIES IN SHALE

z(depth below surface, ft)				x(distance from waste centerline normal to storage room, ft)	
HLW		Spent Fuel		HLW	Spent Fuel
1000	2000	1000	2000	0.0	0.0
1300	2004	1300	2004	0.5	0.5
1600	2008	1600	2009	1.0	1.0
1750	2009	1750	2010	2.0	2.0
1800	2010	1800	2011	3.5	3.5
1975	2011	1975	2017.5	5.0	5.0
1925	2017	1925	2024	7.0	7.0
1935	2018	1935	2025	9.0	9.0
1940	2019	1940	2026	10.0	10.0
1945	2021	1945	2028	11.0	11.0
1950	2028	1950	2050	12.5	12.5
1955	2040	1955	2065	13.5	13.5
1960	2060	1960	2100	15.5	15.5
1965	2100	1965	2150	18.0	18.0
1970	2150	1970	2250	21.0	21.0
1975	2250	1975	2500	25.0	25.0
1980	2500	1980	2500	30.0	30.0
1982	3000	1982	3000	35.0	35.0
1991		1991		40.0	

These conditions are applied to two different models in the analysis, the three-dimensional (3D) model described above and the two-dimensional (2D) model illustrated in Figure 2-4. In the 2D model the heat source is modelled in the limit of simultaneously diluting the waste in the canisters and reducing the separation between canisters such that the total loading in the unit cell is preserved. In this limit the heat source is uniformly distributed down the length below the storage rooms and heat transfer parallel to this direction can be neglected. Although the 3D and 2D models differ in important respects and calculated temperatures in the vicinity of the waste are expected to be different due to the differing local thermal intensities, temperatures far from the source should be very similar in the 2D and 3D models. In fact, 3 percent agreement between the models was obtained to within 3 feet of the waste for similar calculations in salt² which has the highest thermal conductivity of the formation materials considered.* A comparison for basalt, the formation with the smallest conductivity, is shown in Figure 2-5. The mesh spacing in the dimensions common to both models are the same, while in the 3D model the mesh spacing in the remaining dimension (down the storage room) is essentially the same as that horizontally out to the mid-pillar plane. The physical properties of the waste are assumed to be the same as for the formation material except that the waste region generates heat.

Temperature profiles at the waste level from the waste to the boundary of the unit cell are shown in Figure 2-5 for various times. It is clear that in spite of large differences in temperature gradients in the vicinity of the waste, the mesh spacing used is adequate to provide good agreement (better than 10 percent agreement in calculated temperature rises) more than 5 feet from the waste canisters. This example demonstrates the appropriateness of the 2D model for the room-and-pillar calculations, and all

*Strictly speaking, identification of the thermal response of the media with thermal conductivity alone is not adequate. For example, for the formations considered here, the rock conductivities are temperature-dependent. The thermal response then depends upon temperature gradients as well as physical properties of the materials. Nevertheless approximations which are valid for the thermal response of both salt and basalt would appear to be applicable to granite and shale as well.

FIGURE 2-4. 2D AND 3D UNIT CELLS FOR HLW

FIGURE 2-5. COMPARISON OF 2D AND 3D NEAR-FIELD CALCULATIONS FOR BASALT

results provided in this volume for the near-field stress studies have been generated utilizing this model.

2.3 SELECTED RESULTS OF NEAR-FIELD CALCULATIONS

The approach employed to interface the thermal analyses with the thermo-mechanical stress calculations was to initially generate snapshots in time of near-field temperature profiles for selected thermal loadings. The thermo-mechanical stress calculations determined the appropriate areal thermal loading to predict room-and-pillar stability. In some cases, supplementary calculations were performed for the stress analysis. Finally, refined temperature profiles were generated for the selected areal thermal loading.

Thermal and thermo-mechanical stress calculations for salt have been performed earlier, in some cases using more sophisticated stress models than those employed in these studies for the non-salt rocks.¹⁻⁴ However, during these studies some thermal calculations for salt were repeated to benchmark the thermal models. For each of the host rock cases snapshots of the temperature profiles were generated at intervals of 1 year for up to 30 years. The range of areal heat loadings spanned by the calculations were from 60 kW/acre to 400 kW/acre for the non-salt rocks and from 30 kW/acre to 200 kW/acre for salt.

In this subsection, selected temperature profiles are presented from those calculations most closely representing the limiting areal thermal loading for each host rock as determined by the thermo-mechanical stress analyses. Figures 2-6 through 2-9 show isothermal profiles in the near-field region for Total Recycle HLW buried in salt (at 150 kW/acre), granite (at 190 kW/acre), shale (at 120 kW/acre), and basalt (at 190 kW/acre), respectively. Calculations were also performed for Uranium-only Recycle HLW. The results of these latter calculations differ from the Total Recycle HLW results by at most 5 percent. Similar results were also obtained for spent-fuel isolation in the various rocks. These results are shown in Figures 2-10 through 2-13 for the same loadings as presented for HLW.

FIGURE 2-6
NEAR-FIELD ISOTHERMAL PROFILES FOR SALT-HLW

FIGURE 2-7
NEAR-FIELD ISOTHERMAL PROFILES FOR GRANITE-HLW

FIGURE 2-8

NEAR-FIELD ISOTHERMAL PROFILES FOR SHALE-HLW

FIGURE 2-9
NEAR-FIELD ISOTHERMAL PROFILES FOR BASALT-HLW

FIGURE 2-10

FIGURE 2-11

FIGURE 2-12

NEAR-FIELD ISOTHERMAL PROFILES FOR SHALE-SPENT FUEL-120 KW/ACRE

FIGURE 2-13
NEAR-FIELD ISOTHERMAL PROFILES FOR BASALT-SPENT FUEL-190 KW/ACRE

Selected locations for display of calculated temperature histories are shown in Figure 2-14. These locations, correspond to near the upper corner of the storage room, mid-pillar at the storage room floor level and mid-pillar at mid-waste level. The calculated temperature histories are displayed in Figures 2-15 through 2-18 and are for both HLW (Cycle I) and spent fuel.

2.4 THERMAL CRITERIA

Calculations for temperature profiles in the vicinity of the storage room have been used as input to mechanical stress analyses. The massive amount of data needed for the stress analysis (temperature histories throughout the region of the room and pillar) are conveniently labeled by a single number, the local areal heat load associated with the temperatures. Fortunately this number is a key design parameter which defines the average waste distribution in the storage room at emplacement. The stress analysis determines an areal thermal loading limit based on room-and-pillar stability throughout the retrievability period. These limits, determined from the thermo-mechanical stress analysis for 5-year retrievability, are given in Table 2-3 for repositories in salt, granite, shale, and basalt respectively. The limits based on 25-year retrievability of spent-fuel are also given in Table 2-3.

The limits for granite, shale, and basalt have been determined from linear thermo-mechanical analyses based on the temperature profiles discussed in volume 20 of this report. The limits for salt were determined in earlier work for similar repository descriptions and formation stratigraphies.³ Supplementary analyses for the present study indicate that the limits for salt are probably conservative. For example, for a stratigraphy and storage-room design shown in Figure 2-19 Cheverton and Turner⁴ predict an areal loading limit of about 158 kW/acre for 10-year old HLW. This limit is controlled by the criterion that less than 25 percent of the volume of salt in the unit cell bounded by horizontal planes at the top and the bottom of the waste never exceeds 200°C(392°F).⁵ The loading limit as a function of canister separation in the design of Reference 4

FIGURE 2-14. MAP LOCATIONS FOR NEAR-FIELD TEMPERATURE HISTORIES

SALT-150 KW/ACRE

--- SPENT FUEL

— HLW

200

C

B

A

C

B

A

TEMPERATURE RISE (°F)

100

ELAPSED TIME (YRS.)

0

5

10

15

20

25

FIGURE 2-15. NEAR-FIELD TEMPERATURE HISTORIES FOR SALT

FIGURE 2-16. NEAR-FIELD TEMPERATURE HISTORIES FOR GRANITE

FIGURE 2-17. NEAR-FIELD TEMPERATURE HISTORIES FOR SHALE

FIGURE 2-18. NEAR-FIELD TEMPERATURE HISTORIES FOR BASALT

TABLE 2-3

MAXIMUM AREAL LOADINGS (kW/acre) BASED ON NEAR-FIELD
THERMO-MECHANICAL STRESS STUDIES

Formation	HLW and Spent Fuel (5 Year Retrievability)	Spent Fuel (25 Year Retrievability)
Salt ^a	150	36
Granite ^b	190	80
Shale ^b	120	55
Basalt ^b	190	80

^aBased on Russell, J.E., "Areal Thermal Loading Recommendations for Nuclear Wastes Repository in Salt", OWI/TM-37, draft, Oak Ridge, TN, August, 1977.

^bBased on Y/OWI/TM-36/20 "Thermomechanical Stress Analysis and Development of Thermal Loading Guidelines."

Figure 2-19. Three-Dimensional Model of a Unit Cell of the High-Level Mine. See Reference 4.

are shown in Figure 2-20. The limit based on the same 25 percent criterion calculated for the present repository description (Figure 2-1) is also shown in Figure 2-19 and is seen to vary between about 140 kW/acre and about 180 kW/acre depending upon the canister separation. If the 25 percent criterion were based on temperatures achieved only in the first 5 years after emplacement rather than maximum temperature values, the limit would be even higher, 240 kW/acre, and this criterion would probably no longer control the areal loading limits.

The stratigraphy for salt assumed in the present calculation is shown in Figure 2-21 and differs from that in Reference 4. Other differences between the system in Reference 4 and that calculated here include differences in the assumed ambient temperatures, in the waste heat generation rates, in the room-and-pillar configuration, and in the waste distribution and placement. Of the differences between the descriptions here and in Reference 4, the largest effect is due to the difference in stratigraphy. In Reference 4, layers of low-conductivity, interbedded shale were located very near to the waste canisters, effectively insulating the region around the waste canister and increasing temperatures in this region. This effect alone amounts to a 30 percent difference in the local areal load limit. The different depths assumed for the repository in the formation, generates about a 10 percent difference in the loading limit, since the 25 percent criterion refers to the actual temperatures in the unit cell rather than temperature changes from ambient. The difference in thermal decay rate assumed for the waste is small for the first 5 years but can lead to important differences in temperatures achieved after 25 or 30 years. This fact was noticed by Cheverton and Turner when they showed the strong dependence of their results upon the age of the waste at emplacement.⁴ This difference between the two heat-generation rates accounts for about a 5 percent difference in the local loading limit. A difference in the predicted limit of about 20 percent is due to the difference in room and pillar configuration alone. The Cheverton and Turner calculations shown here were for a unit cell about 40 feet wide⁴ while the unit cell for the present case is 78 feet wide.

FIGURE 2-20. AREAL LOADING LIMIT BASED ON 25% SALT CRITERION^a

FIG. 2-21 GENERIC STRATIGRAPHIC SECTION—SALT

It is clear that in any considerations for the local loading limits, the particular description for the repository and stratigraphy will have a major impact and specific design considerations such as local loading limits may differ considerably from the generic values suggested here.

2. REFERENCES

1. Cheverton, R. D. and Turner, W. D., Thermal Analysis of National Radioactive Waste Repository: Progress through June 1971, ORNL-4726, 1971.
2. Office of Waste Isolation/Union Carbide Corporation, Nuclear Division, Thermal Operating Conditions in a National Waste Terminal Storage Facility, Y/OWI/SUB-76/47950, Oak Ridge, TN, 1976.
3. Russell, J. E., "Areal Thermal Loading Recommendations for Nuclear Wastes Repository in Salt," OWI/TM-37, draft. Oak Ridge, TN, August, 1977.
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5. Office of Waste Isolation/Union Carbide Corporation, Nuclear Division, "The Selection and Evaluation of Thermal Criteria for a Geologic Waste Isolation Facility in Salt," Y/OWI/SUB-76/07220, Oak Ridge, TN, 1976.

3.0 VERY-NEAR-FIELD CALCULATIONS

3.1 INTRODUCTION

In the thermal analyses associated with design studies for geologic repositories for reprocessed HLW and unprocessed spent fuel, special care must be taken with the treatment of the temperatures and heat transfer in the immediate vicinity of the waste canisters. Although temperature profiles at locations somewhat removed from the canisters do not depend upon the detailed description of the canister and its immediate surroundings, temperatures within the waste and canister materials are strongly affected by the emplacement design. Therefore, extreme care is needed in modeling the very-near-field environment in order to ensure accurate results.

The purposes of this analysis are to predict maximum canister temperatures both within the waste and in the protective packaging and to compare these values with limitations which may exist. Severe close-in temperatures may impact canister oxidation and cracking rates and diffusion rates of the nuclides within the waste matrix, increase the decomposition rates of the waste form, and have physical consequences in the rock immediately around the canister.

The key factors for the very-near-field temperatures include:

- Canister and canister-well design
- Canister power load
- Local thermal properties of the formation rock
- Short-term thermal decay rates of the waste
- Average local canister spacing.

In these very-near-field calculations, it is assumed that the stainless steel canister is in a carbon steel overpack and the overpacked canister is placed into a well, which for some cases is lined with a carbon steel sleeve. The annulus between the receptacle wall and the overpack or sleeve may then be backfilled with, for example, crushed rock. Details

of this entire canister and canister-well design influence the heat transfer from the waste and, consequently, the temperatures of the waste and the canister. Unfortunately, the detailed thermal response of the backfill zone is difficult to quantify. Furthermore, enhanced temperatures in the rock may introduce thermo-mechanical stresses sufficient to cause cracking of the formation material around the canister. The extent of this fractured zone and its thermal response are also not well defined. For this analysis, the term "backfill region" includes the fractured zone as well as the region backfilled with crushed rock. Thermal properties of the backfill region are assumed arbitrarily; sensitivity studies have been performed on these properties and are reported later. Since details of the waste package design are somewhat better established, they are explicitly taken into account. Details of the design which would affect the heat transfer in the vertical direction, above or below the canister, are of little significance, although the models also take these into account.

Waste-dilution factors which determine the individual canister loadings are crucial in determining the maximum temperatures achieved in the waste and canister and in determining when these will occur. For the smaller initial canister thermal loadings, temperatures peak early; therefore, in these cases slight variations in the waste heat-generation rates is not important in determining the magnitude of peak temperatures. Also, the average local repository loading has minor impact for these peak temperatures.

Other major approximations made in modeling the very-near-field situation are

- A cylindrical unit cell is utilized
- Convection in air spaces between canister and overpack, or between overpack and sleeve or receptacle wall is neglected
- Only Total Recycle for HLW wastes is considered
- The models for the waste form are simple
- Phase changes of materials are not included
- The presence of any water in the very-near-field is neglected.

The unit cell approximation assumes an infinite, uniform distribution of canisters. Moreover, because axial symmetry is assumed for the unit cell, uniform distribution in all directions is implied. In fact, the pitch of the canisters down the storage room is much smaller than the separation between storage rooms. Furthermore, the unit-cell approximation assumes that all the waste is emplaced simultaneously. The time for loading a local area would probably be less than one year, a somewhat shorter time than that needed for temperatures to peak.

Convection in gaps between the various components of the canister design, and between the overpack and the sleeve or the rock is neglected. This approximation should be fairly accurate for the assumed mode of operation in which the storage room is sealed as soon as it is filled with waste.

For HLW the relative decay rate for Cycle I waste is used for both Cycle I and Cycle III, since the results for Cycle III are higher than for Cycle I waste by at most 2 percent in temperature rise. The active volumes of HLW are modeled as solid homogeneous blocks of waste material. The model for temperatures outside spent-fuel canisters also utilize the solid homogeneous block concept, although the model for the internal temperatures takes into explicit account the detailed structure of the spent-fuel assemblies.

Neglecting the presence of water in the very-near-field of the waste is very appropriate for salt calculations. Although some groundwater inflow into the non-salt repositories is predicted for the generic stratigraphies, the impact of heat on flow near the canister is not known. It is possible that the very-near-field temperature gradient would drive the groundwater away from the canister well or thermally induced fracturing could increase water inflow into the canister well. However, if the average groundwater inflow into the generic granite or basalt repositories (that is about one-half ounce per square foot per day) is assumed to traverse every surface of a canister well, approximately five percent of the heat might be removed from the immediate canister vicinity by total vaporization during the retrievable period. Therefore, the simplifying assumption of

neglecting the presence of water appears warranted, especially since it leads to conservative heat loads.

In these analyses, phase transition temperatures for the waste or any other materials involved and their respective heats of transformation are not taken into account.

3.2 VERY-NEAR-FIELD TEMPERATURE LIMITATIONS

Temperature limitations for HLW and spent fuel in the verynearfield environment are given in Table 3-1. These limitations are taken from Reference 1. The justification for the choices for these limits given in Reference 1 are summarized here.

Austenitic stainless steel undergoes changes in structure during long-term exposures in air at temperatures in the range 400 to 900°C. The observed effect is an increased susceptibility to stress cracking upon subsequent exposure to water at atmospheric temperatures.² Pacific Northwest Laboratory workers have stated that precautions should be taken to maintain the canister wall temperatures below 400°C during handling and storage after filling, in order to avoid this sensitization. Argonne National Laboratory workers have quoted a maximum temperature of 343°C (650°F) for 100 year retrievable storage in air.³ ARCHO workers⁴ have recently recommended maximum temperatures of 277°C (530°F), and 427°C (800°F), respectively, for normal and abnormal conditions in long-term (100 year) retrievable, sealed-cask storage. The abnormal conditions could prevail for an indefinite period. It is concluded that the limit of 375°C for fiveyear retrievable geologic storage is reasonable, especially since the canister will probably be at the maximum temperatures for more than several years.

Carbon steel is probably less susceptible to significant changes in structure and corrosion susceptibility during long-term heating at temperatures $\geq 400^\circ\text{C}$, but there is no information on this. Accordingly, the same temperature limit as that for stainless steel has been specified.

The transition temperature for typical borosilicate waste glasses is about 500°C.⁵ The softening temperature is a few degrees higher.

TABLE 3-1

MAXIMUM ACCEPTABLE TEMPERATURE OF WASTE AND CONTAINER DURING
RETRIEVABLE GEOLOGIC STORAGE^a

Item	Maximum Acceptable Temperature, °C (°F)	
HLW		
Canister	375	(707)
Solid Waste		
glass	500	(932)
calcined	700	(1292)
sintered glass ceramic	800	(1472)
Spent Fuel Element	200	(392)

^aSee Jenks, G.H., "Maximum Acceptable Temperature of Waste and Containers During Retrievable Geologic Storage", Y/OWI/TM-42, Oak Ridge, TN, August, 1977.

Significant increases in cracking and in leach rates have been observed in test glasses which were heated for periods of a few months in the temperature range 500 to 800°C.^{2,6} Migration of heavy, separate phases in the glass might take place above the softening temperature. This could be of special concern with the HLW from U-only-recycle, which contains most of the Pu generated during reactor operation. The solubility of Pu in a waste glass may be only about 2 percent by weight,⁷ in which case about 50 percent of this Pu would occur as a separate phase in the glass. Hoskins and Berreth⁸ have suggested 550°C for the maximum temperature of waste glass during storage. Considerations of the above information have led to the proposal of 500°C for the maximum temperature to which borosilicate waste glass should be exposed during retrievable geologic storage.

The limitation of 700°C for a calcine waste matrix is that proposed by Hoskins and Berreth⁸ on the basis that significant cesium volatility occurs at temperatures greater than 750°C. The 800°C limits for structured glass ceramic waste, also from Hoskins and Berreth,⁸ is based on the preliminary evidence that this material is stable to at least 800°C.

The limit value of 200°C for the spent fuel elements is based on the following considerations. The surfaces of the zircaloy hulls will become very reactive during storage within a sealed canister as a result of the degradation of the protective oxide film which will occur during heating in the absence of an oxidant.⁹⁻¹¹ No complete analyses have been made of hazards which might result from the above effects and how these would depend upon the temperatures in the contained wastes.

3.3 METHODOLOGY

3.3.1 HLW Model

Details of the physical model assumed for a typical reference HLW canister emplaced in the repository are shown in Figure 3-1. In this case, the radius of the canister corresponds to that assumed for vitrified waste. The dimensions associated with both vitrified and calcined HLW are given in Tables 1-2 and 1-3. It is assumed that the canister is overpacked and, after emplacement, that the space between the overpack and formation is backfilled with crushed formation material.

Figure 3-1. DETAILS OF WASTE RECEPTACLE AND CANISTER

Since the HLW canister arrangement in the very-near-field is roughly cylindrically symmetric about the axis of the canister, this symmetry is assumed for the entire problem. The cylindrical unit cell extends out to a radius R where an insulating boundary condition is applied. In this way the results correspond to those of a large array of uniformly spaced canisters with a pitch of $2R$. For most cases, the separation between canisters in different storage rooms is likely to be much larger than the average separation between canisters down the storage room. Moreover, in some cases it is anticipated that storage-room designs will allow for two or more rows of canisters down the storage room. Therefore, the actual canister spacing may be much less uniform than assumed in this model. Although the cylindrical symmetry allows the accurate representation of very-near-field processes such as the radial conduction and thermal radiation between canister and overpack, more detailed treatments taking into account nonuniformity in the canister distribution may be necessary in order to gain a true understanding of the very-near-field thermal response.

Mesh spacings used to make calculations for the reference case are given in Table 3-2. These mesh spacings have been found to be satisfactory for the reference cases in all four generic formations. As design variables such as canister radius or backfill thickness are changed, modification to these mesh spacings become necessary. The local areal thermal loading is given by the individual canister thermal load divided by πR^2 . Therefore, the value of the unit-cell radius, R , must also change as cases for various areal loadings are considered.

The temperature of the waste at emplacement is assumed to be 350°F . The calculations are insensitive to this value, since there is so little energy associated with this temperature. The ambient formation temperature is given in Table 1-10. The ambient formation temperature can be important since the temperature limitations are expressed in terms of temperatures rather than temperature rises. The boundaries at the top and bottom of the unit cell are sufficiently far removed, 250 feet above and below the unit cell, that they affect the temperatures by less than 3 percent in the first 5 years after emplacement. Since the peak temperatures are reached within this time frame, these boundaries are considered to be

TABLE 3-2

TYPICAL MESH LINES FOR VERY-NEAR-FIELD MODELS FOR HLW

Z, Vertical (ft) ^a	R, Radial (ft) ^a
1750	0
1900	0.333
1960	0.500
1980	0.531
2000	0.563
2004	0.583
2006	0.750
2006.75	1.083
2007	1.583
2007.21	2.000
2008	3.000
2008.03	4.500
2010	6.230
2011	8.780
2014 ^b	10.800
2017	12.900
2018	15.000
2018.03	17.000
2018.25	19.050 ^c
2019	
2021	
2025	
2030	
2050	
2100	
2250	

^aFor this case HLW region is bounded by the 2010 and 2018 Z mesh lines and by the 0.500 radial mesh line.

^bMidwaste level.

^cThis farthest radial mesh line provides an areal thermal loading of 120 kW/acre for 3.14 kW canisters.

sufficiently far removed from the canisters. Imposed at the top (1750 ft. depth) of the unit cell is an isothermal boundary temperature of 87.5°F and at the bottom (2250 ft. depth), of 98°F. These are chosen to maintain the ambient formation temperatures in the absence of the heat source. The storage room is not modelled.

Thermal radiation across gaps in the canister and between the canister wall and the overpack is taken into explicit account. The calculations are not sensitive to the emissivity of the steel surfaces. Results calculated for emissivity values from 0.5 to 1.0 differ by less than 3 percent. A very small emissivity (≤ 0.3) could have much more severe impacts, however.

As an example of calculations with this model, a canister containing 10-year old waste placed in salt is considered with a canister load of 3.5 kW. A unit cell radius of 21 feet is chosen which corresponds to an areal loading of about 110 kW/acre. Temperatures in the canister reach maximum values within the first 5 years after emplacement. The isotherms at 5 years after emplacement are shown in Figure 3-2. From this figure, it is seen that the bedded salt temperatures several inches beyond the canister reach no more than about 300°F. A more detailed presentation of temperature versus distance from the centerline of the canister is shown in Figure 3-3. This profile is evaluated at the midwaste level; consequently, the temperatures shown in Figure 3-3 are maximum temperatures for the unit cell at the specified time.

In this example, the thermal conductivity of the crushed salt was arbitrarily taken to be 10 percent of that for the rock salt, and the conductivity for the waste is taken to be 0.58 Btu/hr-ft-°F. Conduction in the air gaps is included. 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 salt away from the canister were influenced very little by these properties. Analytic solutions treating the canister as an infinitely long cylinder provide a reasonable estimate for maximum temperatures very near the canister. This implies that temperatures in the salt are essentially unaffected by the internal details of the canister. On the other hand, the temperatures in the canister/overpack/crushed-salt environment are inversely proportional to the conductivities, and depend upon the radii of interfaces between materials.

**Figure 3-2 ISOTHERMS IN IMMEDIATE VICINITY OF
CANISTER, TIME = 5 YRS.**

PA818-10-4

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

The results of parametric studies are shown in Figure 3-4. The effect of arbitrarily increasing the waste conductivity by a factor of 10 is shown by Curves 1 and 2 in Figure 3-4. It can be seen that the temperature rise in the waste is roughly inversely proportional to the conductivity of the waste. The dependence of the 5-year temperature profile on thermal radiation is shown in Curves 1 and 3 of Figure 3-4. Although the difference between calculations for an emissivity of 0.7 or 0.5 is small, the effect for no radiation, $\epsilon=0$, is important and can increase internal temperature rises by a factor of 3 or more in the present example.

3.3.2 Comparison of HLW Model with Analytic Solution

Analytic expressions for the temperatures in the example case provide similar conclusions. For the steady-state solution to obtain temperatures in an infinitely long cylinder of waste, the temperature rise in the waste may be computed using¹²

$$\Delta T_{\text{waste}} = \frac{q R_{\text{waste}}^2}{4 k_{\text{waste}}}$$

Where q is the volumetric heat generation rate, R_{waste} is the radius of the waste and k_{waste} is conductivity of the waste. The solution for the temperature rise in a cylindrical shell of conductivity k is given by

$$\Delta T = \frac{q R_i^2}{2k} \ln \left(\frac{R_o}{R_i} \right)$$

Where R_i is the inner radius and R_o is the outer radius of the shell. The temperature rises in these materials from these expressions are compared with the numerical results in Table 3-3. It can be seen that the agreement is reasonable. Although these expressions hold only for the steady state, the temperature changes are evidently slow enough at 5 years that gross errors are not introduced. The temperatures in the salt far from the canister have the poorest agreement as might be expected from the infinite-cylinder approximation that was necessary.

PA818-14-2

Figure 3-4. TEMPERATURE vs. RADIAL DISTANCE FROM A 3.5 kw CANISTER, 5 YEARS AFTER BURIAL

TABLE 3-3
TEMPERATURE INCREASES ACROSS VERY-NEAR-FIELD MATERIALS
AFTER FIVE YEARS' STORAGE

Material	Conductivity (BTU/hr-°F-ft)	Inner Radius (ft)	Outer Radius (ft)	$\Delta T(^{\circ}\text{F})$	$\Delta T_{\text{analytic}}(^{\circ}\text{F})^a$
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 ^b
Carbon Steel Overpack	26.00	0.563	0.583	0.26	0.27
Crushed Salt	0.16	0.583	0.750	300.	315.

^a $\Delta T_{\text{analytic}}$ is calculated from closed-form expressions for infinite cylinders.

^bCalculated temperature rise includes approximate correction for radiation across gap. Without this correction, the one-dimensional model gives a temperature rise of 514.°F.

The conclusions from the numerical and analytical expressions appear to be:

- Temperatures outside the canister are roughly insensitive to internal canister properties and radiation.
- Temperature rises are roughly inversely proportional to the thermal conductivity.
- Temperature rises in a material depend strongly on the thickness or radius of the material.

3.3.3 Spent-Fuel Model

Since the temperatures outside the canister do not depend strongly on internal details of the spent-fuel assembly, the model for the temperatures outside a spent-fuel canister assumes the waste is a solid block of material. The gross waste and canister description are given in Tables 1-2 and 1-3. The mesh spacing associated with this model is given in Table 3-4. This model includes a possible backfill zone.

The crude model for spent-fuel canisters contains only the canister and the waste. Radiation between the waste and the canister is explicitly taken into account with an emissivity of 0.6. The model canister has the same radius as the steel sleeve liner illustrated in Figure 3-1. Temperature rise between the sleeve and overpack and between the overpack and canister is estimated from the calculated rise between the model canister and waste with appropriate geometry corrections taken into account. This approximation is based on the assumption that radiation is the predominant heat transfer process in this case. This can be checked by comparing the computed heat transfer with the expression infinite concentric cylinders

$$q = \sigma \epsilon A (T_1^4 - T_2^4)$$

where σ is the Stefan-Boltzmann constant, ϵ is the emissivity, A is the appropriate average surface area and T_1 and T_2 are the temperatures of the two surfaces bounding the gap. In general, the agreement is better

TABLE 3-4

TYPICAL MESH LINES FOR VERY-NEAR-FIELD MODELS FOR
PWR SPENT FUEL ASSEMBLIES^a

Z, Vertical (ft)	R, Radial (ft)
1750	0
1900	0.200
1960	0.244
1980	0.500
2000	0.729
2004	0.760
2006	0.927
2007	1.260
2008	1.760
2009	2.260
2010	3.260
2010.03	4.760
2011	7.972 ^b
2014	
2017	
2023.03	
2023.50	
2024	
2026	
2030	
2035	
2050	
2100	
2250	

^aDetails of the spent-fuel assembly are not taken into account. These details are taken into account by the method of Cox¹³, using shroud temperatures calculated in this model.

^bThis farthest radial mesh line provides an areal thermal loading of 120 kW/acre for 0.55 kW PWR spent-fuel assemblies.

than 1 percent so that the predicted canister temperatures are expected to be quite accurate.

The internal waste temperatures require more sophisticated treatment of the spent-fuel assembly. Detailed treatments for spent-fuel assembly temperatures can be found in references 13 and 14 when the canister temperature is specified. Using canister temperatures from the exterior temperature model, maximum internal temperature is given implicitly by

$$\frac{\sigma(T_1^4 - T_n^4)}{Q/A_1} - \frac{(1-\epsilon)(1+mA_1)}{\epsilon A_n} = Z$$

In this expression σ is the Stefan-Boltzmann constant, ϵ is the average emissivity of the fuel rods, m is the number of pins in the assembly, Q is the heat-generation rate in the pin, A_1 is the surface area of the pin, and A_n and T_n are the inner area and temperature of the canister wall respectively. T_1 is the temperature of the central fuel pin or maximum temperature of the assembly. The configuration factor, Z , is given in reference 13 for the geometry of interest. For PWR spent-fuel assemblies, specified in Table 1-4, Z is 47.5. A value of 0.5 is assumed for ϵ . The temperature rise within the canister is then given by:

$$\Delta T = T_1 - T_n$$

Results using these expressions are presented in Section 3.4.2.

3.4 RESULTS OF VERY-NEAR-FIELD CALCULATIONS

3.4.1 HLW

Ten-year-old vitrified waste diluted to a specific volume of 3 ft³ per metric ton of heavy metal (MTHM) provides a heat generation rate of 2.8 kW for the canister specified in Table 1-2. A loading of 2.5 kW can be obtained for calcined waste by diluting the waste to 1.5 ft³/MTHM. Calculations for the maximum canister and waste temperatures for these waste packages are shown in Figures 3-5 and 3-6. The temperatures are shown as a function of the average areal thermal loading in the repository or, equivalently, the average canister spacing. The maximum temperatures are noticeably higher for the calcined waste case than for the vitrified waste case. This difference is due largely to the smaller radius of the canister and the higher volumetric heat generation rate of the specified calcined waste as well as the smaller conductivity of the calcine matrix. It can be seen from Figure 3-5 that for all media other than salt, the maximum waste criteria of 932°F for glass is exceeded for the canister load of 2.8 kW in the present model. Similarly from Figure 3-6 the calcine temperatures exceed the allowable 1292°F for all media other than salt for 2.5 kW/canisters.

The temperatures are noticeably higher for the lower thermal conductivity formation materials; salt provides the lowest canister and waste temperatures for a given thermal loading. In many cases, the thermal resistance is so high that canister and waste temperature criteria are exceeded even when canisters are separated very far from each other. The maximum temperatures cannot be reduced arbitrarily for the non-salt formations by reducing the areal thermal load, and individual canister loads must be reduced or changes in the canister design must be considered if temperature criteria are to be met. Figures 3-7 to 3-10 show the results of variation of the canister loading. In these calculations the areal thermal load for each formation has the value determined from the near-field thermo-mechanical studies, which predict storage room and pillar stability. The reference design for the vitrified HLW studies consists of a 12-inch ID canister and a 6-inch zone of backfill or fractured formation material. The ID for the reference calcined waste canisters is only

FIGURE 3-6. CANISTER AND WASTE TEMPERATURES VS. IDEAL HEAT LOAD

FIGURE 3-7. MAXIMUM CANISTER AND WASTE TEMPERATURE VS. CANISTER HEAT LOAD

8 inches. The canister thermal loadings which allow the canister and waste criteria of Table 3-1 to be met are listed in Table 3-5. Uncertainties in the loading limits based upon 10 percent uncertainty in the predicted temperature rises are also shown in Table 3-5. It can be seen that the canister temperatures are limiting for each case shown in Table 3-5. An optimal situation would arise if the canister were designed so that canister loading limits based on both temperature limits were equal.

The canister thermal loading limits in Table 3-5 are based on a specific design. Calculations similar to those illustrated in Figures 3-7 to 3-10 have also been made for variations in the reference design. The canister loading limits resulting from variations in the average areal loading (or canister separation), in the canister radius, in the thickness of the backfill zone, in the effective conductivity of the backfill zone, and in the conductivity of the waste matrix are given in Tables 3-6 through 3-10, respectively. As before, the uncertainty in these limits based on a theoretical uncertainty of 10 percent in the calculated temperatures is also given in the Tables.

It appears that in many cases it may be possible to increase canister loadings with appropriate modification to canister or repository design. For example, it appears that canister loads in salt could be increased from 3.2 kW/can to 4 kW/can simply by increasing the radius of the canister from 6 to about 10 inches or by decreasing the thickness of the backfill zone from 6 to about 3 inches.

These studies do not preclude other design changes which may be implemented to reduce canister and waste temperatures. The convective heat-removal approach discussed in the Appendices may be useful in this respect.

If the canister is not backfilled after emplacement, thermal radiation from the overpack surface to the canister receptacle wall could enhance the heat transfer away from the canister. Estimates for the temperature drop across the gap between the overpack and the receptacle wall can be made from the drops between the canister and the overpack. Based on an emissivity of 0.6 and on the limiting canister temperature criterion in Table 3-1, these estimates lead to canister loading limits of approximately

TABLE 3-5
MAXIMUM CANISTER LOADINGS^a

Waste Form	kW/can			
	Salt	Granite	Shale	Basalt
Vitrified Waste ^b	3.2 ± .4	1.7 ± .3	1.2 ± .2	1.3 ± .3
Calcined Waste ^b	2.6 ± .4	1.6 ± .2	1.1 ± .2	1.1 ± .3
Vitrified Waste ^c	3.6 ± .4	2.3 ± .3	1.6 ± .2	1.9 ± .3
Calcined Waste ^c	2.9 ± .4	2.0 ± .2	1.5 ± .2	1.7 ± .3

^aBased on areal thermal loadings for 5-year retrievability. Reference vitrified waste canister has 12 in. ID, 6 inches crushed rock backfill. Calcined waste canister has 8 in. ID, 6 inches crushed rock backfill. Waste is 10 years old at emplacement.

^bBased on limiting canister temperatures. Note that these limits are controlling.

^cBased on limiting waste temperatures.

TABLE 3-6
MAXIMUM CANISTER LOADS FOR DIFFERENT AREAL LOADINGS^a

Salt		Granite		Shale		Basalt	
kW/acre	kW/can	kW/acre	kW/can	kW/acre	kW/can	kW/acre	kW/can
100	3.3 $\pm$.4	130	1.7 $\pm$.3	80	1.2 $\pm$.4	130	1.3 $\pm$.3
150	3.2 $\pm$.4	190	1.7 $\pm$.3	120	1.2 $\pm$.4	190	1.3 $\pm$.3
200	3.1 $\pm$.4	250	1.6 $\pm$.3	160	1.2 $\pm$.4	250	1.3 $\pm$.3

^aBased on limiting canister temperatures. Reference canister for 10-year-old HLW has 12 in. ID and 6 in. annulus of crushed rock backfill.

TABLE 3-7

MAXIMUM CANISTER LOADS FOR DIFFERENT CANISTER RADII^a

Canister Radius (in)	kW/can			
	Salt	Granite	Shale	Basalt
2	2.3 ± .3	1.1 ± .2	1.0 ± .1	0.8 ± .2
4	2.8 ± .3	1.4 ± .2	1.1 ± .1	1.0 ± .2
6	3.2 ± .4	1.7 ± .3	1.2 ± .2	1.3 ± .3
8	3.5 ± .5	2.0 ± .3	1.3 ± .2	1.5 ± .3
12	3.7 ± .5	2.3 ± .3	1.4 ± .2	1.6 ± .3

^aBased on limiting canister temperatures and areal thermal loadings for 5-year retrievability. Reference canister for 10-year-old HLW has 6 in. annulus of crushed rock backfill.

TABLE 3-8
MAXIMUM CANISTER LOADS FOR VARYING BACKFILL GEOMETRY^a

Backfill Annulus Thickness (in)	kW/can			
	Salt	Granite	Shale	Basalt
0	5.3 ± .7	3.7 ± .5	1.8 ± .2	2.6 ± .4
4	3.8 ± .5	2.1 ± .3	1.4 ± .2	1.5 ± .3
6	3.2 ± .4	1.7 ± .3	1.2 ± .2	1.3 ± .3
8	2.8 ± .4	1.4 ± .3	0.9 ± .1	1.1 ± .2
12	2.5 ± .3	1.2 ± .2	0.2 ± .1	0.9 ± .2
18	2.5 ± .3	1.2 ± .2	---	0.9 ± .2

^aBased on limiting canister temperatures and areal thermal loadings for 5-year retrievability. Reference canister for 10-year-old HLW has 12 in. ID, 6 in. crushed rock backfill.

TABLE 3-9

MAXIMUM CANISTER LOADS FOR VARYING BACKFILL CONDUCTIVITY^a

Backfill Material Conductivity (BTU/hr-ft-°F)	kW/can			
	Salt	Granite	Shale	Basalt
0.05	0.6 ± .1	0.3 ± .1	0.6 ± .1	0.6 ± .2
0.10	1.4 ± .2	1.1 ± .2	1.2 ± .2	0.9 ± .2
0.15	1.9 ± .3	1.7 ± .3	1.6 ± .2	1.3 ± .3
0.20	2.4 ± .3	2.1 ± .3	1.8 ± .2	1.7 ± .3
1.0	4.3 ± .5	3.5 ± .5	3.2 ± .4	5.1 ± .7
1.5	4.6 ± .5	3.7 ± .5	3.4 ± .4	6.3 ± .9
2.0	4.7 ± .6	3.8 ± .5	3.5 ± .4	7.2 ± 1.0
2.5	4.8 ± .6	3.8 ± .5	3.5 ± .4	7.7 ± 1.0

^aBased on limiting canister temperatures and areal thermal loadings for 5-year retrievability. Reference canister for 10-year-old HLW has 12 in. ID, 6 in. crushed rock backfill.

TABLE 3-10

MAXIMUM CANISTER LOADS FOR VARYING WASTE CONDUCTIVITY^a

Waste Conductivity (BTU/hr-ft-°F)	kW/can							
	Salt		Granite		Shale		Basalt	
	(b)	(c)	(b)	(c)	(b)	(c)	(b)	(c)
0.2	3.2 ± .5	2.6 ± .4	1.7 ± .3	1.6 ± .2	1.2 ± .2	1.2 ± .2	1.7 ± .3	1.4 ± .3
0.45	3.2 ± .4	3.3 ± .5	1.8 ± .3	2.1 ± .4	1.2 ± .2	1.5 ± .2	1.7 ± .3	1.8 ± .3
0.7	3.2 ± .4	3.6 ± .6	1.8 ± .3	2.3 ± .4	1.2 ± .2	1.6 ± .2	1.7 ± .3	1.9 ± .3
1.0	3.2 ± .4	3.7 ± .6	1.8 ± .3	2.4 ± .5	1.2 ± .2	1.7 ± .3	1.7 ± .3	2.0 ± .3
2.0	3.3 ± .4	3.8 ± .6	1.8 ± .3	2.6 ± .4	1.3 ± .2	1.8 ± .3	1.8 ± .3	2.1 ± .3

^aBased on areal thermal loadings for 5-year retrievability. Reference canister for 10-year-old HLW has 12 in. ID, 6 in. crushed rock backfill.

^bBased on limiting canister temperatures.

^cBased on limiting vitrified waste temperatures.

4.5, 4.0, 3.5 and 3.4 kW per canister for the salt, granite, shale, and basalt formations respectively.

From Table 3-9 it is evident that the heat transfer away from the canister is quite sensitive to the conductivity of the backfill material. It may be possible, therefore, to increase the canister loading limit by backfilling with material of higher thermal conductivity. It is likely that the crushed salt backfill material would reconsolidate increasing the conductivity in this zone to values approaching that of the uncrushed formation material.

For a specific design, the canister thermal loading and waste density are directly related; and, given a canister loading limit, this value may be used to derive a maximum waste density for a specific canister design. Equivalently, minimum waste-dilution factors can be derived. Values of minimum specific volumes for the waste associated with the baseline design are given in Table 3-11. It should be noted that the optimum value for one formation differs from that for another formation. Hence, an optimized canister for one repository will not be interchangeable with another. It is also clear that the dilution factors cannot be set arbitrarily. For example, in no case is a specific volume of $2 \text{ ft}^3/\text{MTHM}$ acceptable without changes from the baseline design. Nevertheless, the necessary alterations to the canister or canister well can be determined in a straight forward way from the preceding Tables 3-6 through 3-10. For example, a waste dilution of $2 \text{ ft}^3/\text{MTHM}$ could be acceptable for the salt repository if the backfill layer were reduced to a thickness of about 2.5 inches or less.* The uncertainties expressed in Tables 3-5 to 3-10 suggest that conservative values for parameters may be necessary to ensure that canister temperature criteria are met.

3.4.2 Spent Fuel

Results for spent-fuel canisters based on 5-year retrievability are shown in Figure 3-11. The waste is assumed to be 10-year-old PWR spent-fuel assemblies, so that the canister load at emplacement is 0.55 kW/can.

*The specific volume scales inversely with the canister heat load. For example, for HLW in salt, by reducing the specific volume from $2.63 \text{ ft}^3/\text{MTHM}$ to $2.0 \text{ ft}^3/\text{MTHM}$, the canister load increases from 3.2 kW to 4.2 kW.

TABLE 3-11

MINIMUM WASTE DILUTION FOR REFERENCE CANISTERS^a

Waste Type	Active Volume (ft ³)	Watt/ MTHM	Specific Volume (ft ³ /MTHM)			
			<u>Salt</u>	<u>Granite</u>	<u>Shale</u>	<u>Basalt</u>
<u>Total Recycle</u> <u>(UO₂ and MOX Blend)</u>						
Vitrified (3:1 Blend)	6.28	1341	2.63	4.68	7.02	4.95
Calcined (5:1 Blend)	2.79	1238	1.33	2.30	3.14	2.88
<u>U-Only Recycle</u>						
Vitrified	6.28	1250	2.45	4.35	6.54	4.62
Calcined	2.79	1250	1.34	2.37	3.17	2.96

^aBased on areal thermal loadings for 5-year retrievability. Reference canister has 12 in. ID, 6 in. backfill of crushed material.

Table 3-12 summarizes the maximum temperatures expected in the canisters and assemblies for each of the four generic repositories for a variety of conditions. In many of the non-salt cases, the temperatures exceed the limits in Table 3-1. As pointed out for HLW, the temperatures are extremely sensitive to design changes. For example, a reduction in the backfill thickness can significantly reduce these values. The temperatures in the formation are much lower for spent fuel than for HLW, so that cracking of the formation may be reduced, and the backfill zone may be reduced in this way. However, even for the extreme case of solid formation around the canister, the temperature limits for the granite and shale areal thermal-loading design are nearly exceeded, and the limit for the basalt areal thermal-loading design is exceeded.

The effect of placing the spent-fuel canisters in the canister well without backfilling is also shown in Table 3-12. In this case radiation across the 6-inch gap occupied by the backfill material in the earlier calculation reduces the temperatures below the temperature criterion in every case except for basalt.

The maximum temperatures for 25-year retrievability are also seen in Table 3-12. Although much less severe, in several cases these values, too, exceed the limits for spent fuel assemblies in Table 3-1. However, when there is no backfill material around the canisters, in every case the 25-year temperatures fall below the limits.

For the situations shown in Table 3-12, areal loading must be reduced in some cases in order to keep spent fuel assembly temperatures below the 200°C (392°F) limit. Maximum areal loadings for these cases, consistent with the near-field considerations, are shown in Table 3-13. In the cases studied for the basalt repository, it appears that areal loadings may be limited by very-near-field factors rather than room closure considerations.

TABLE 3-12 MAXIMUM EXPECTED TEMPERATURES IN PWR SPENT-FUEL CANISTERS (⁰F)

	<u>Formation</u>	<u>Sleeve</u>	<u>Overpack</u>	<u>Canister^a</u>	<u>Spent Fuel Assembly^b</u>
5 Year Retrievalability (6 inches of crushed backfill material around canister)	Salt	303	326	349	402
	Granite	358	377	396	442
	Shale	358	377	396	442
	Basalt	413	429	444	483
25 Year Retrievalability (6 inches crushed backfill material around canisters)	Salt	193	214	235	290
	Granite	296	308	320	361
	Shale	306	322	338	376
	Basalt	362	374	386	418
5 Year Retrievalability (Solid Material around canisters.)	Salt	241	264	287	352
	Granite	300	319	338	393
	Shale	299	318	337	392
	Basalt	389	404	419	461
5 Year Retrievalability (6 inch gap around canisters)	Salt	240	263	286	351
	Granite	296	315	334	390
	Shale	295	314	333	389
	Basalt	372	387	402	447
25 Year Retrievalability (6 inch gap around canisters)	Salt	151	172	193	257
	Granite	274	286	298	342
	Shale	264	280	296	340
	Basalt	320	332	344	381

^aMaximum Acceptable Temperature is 375⁰C (707⁰F).

^bMaximum Acceptable Temperature is 200⁰C (392⁰F).

TABLE 3-13 MAXIMUM LOCAL LOADINGS FOR PWR SPENT-FUEL
CANISTERS (KW/ACRE)

<u>FORMATION</u>	<u>5-YEAR RETRIEVABILITY</u>			<u>25-YEAR RETRIEVABILITY</u>
	<u>(a)</u>	<u>(b)</u>	<u>(c)</u>	<u>(a)</u>
Salt	141	150	150	36
Granite	142	189	190	80
Shale	88	120	120	55
Basalt	115	136	145	68

(a) Six inches crushed backfill material around canisters.

(b) Solid material around canisters.

(c) Six-inch gap around canisters.

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4.0 FAR-FIELD ANALYSES

4.1 INTRODUCTION

In this section, the calculations performed to provide thermal profiles at distances far removed from the HLW or spent fuel emplaced in the repository over very long time frames are described. These temperature distributions are of importance in themselves. However, they are also needed for the complete assessment of the long-term thermo-mechanical stress in the formation and thermally-induced changes in the groundwater flow. Far-field thermal studies for repositories in salt have been conducted earlier and have led to significant understanding of the long-term impacts of waste isolation.¹ The bulk of the present work has been performed for the generic granite, shale, and basalt formations, although some special cases for salt have been studied for comparison and are included in the present section. The most important factors influencing these temperatures include:

- the over-all thermal loading for the entire repository
- the long-term time-dependence of the heat generation rate
- mean far-field thermal properties
- boundary conditions at the surface of the earth.

For distances sufficiently far removed from the sources, details of the repository design should not be of significance. This assumption lifts the requirement to resolve these details in the model and simplifies the problem considerably. The trade-off is that temperatures calculated for the immediate vicinity of the repository cannot be expected to be accurate. However, it is expected that reasonable estimates of these temperatures may be obtained from the near-field analyses. Additional simplifications made in the calculational model include:

- Stratigraphy in the formation is assumed flat and unbounded in the horizontal direction
- Thermal properties of the various layers are assumed to be uniform and unaffected by the stresses induced
- The effect of groundwater flow on thermal properties or heat removal is neglected.

The last approximation may be very important, particularly for heat removal. The convection currents in aquifers induced by the heat sources may have capacity to remove substantial portions of the generated heat. Any heat removal by this process would have a stabilizing effect and would reduce the convective currents.

4.2 METHODOLOGY

4.2.1 Far-Field Model

The stratigraphies assumed for the generic granite, shale, and basalt formations along with associated thermal conductivities of materials are given in Figures 4-1 through 4-3. The repository is modeled as a disc with a radius of about 1 mile at a depth which depends on the geologic formation in which the repository is located. The model used in the analysis is shown in Figure 4-4. Symmetry about the axis of the repository is assumed in the model. The thermal properties of the repository, except for heat generation, are taken to be the same as those of the surrounding material. The vertical boundaries are taken to be at least 3 repository radii from the centerline; insulating boundary conditions are imposed there. Any error that would occur due to this boundary would lead to conservative results. However, the boundaries are sufficiently far removed that very little effect is felt in the formation directly above the repository even during the long time scales of interest. The boundary condition at the bottom of the model which is also far removed from the repository is taken to be isothermal and consistent with the ambient temperature distribution assumed for the formation. At the surface of the earth, convective cooling by surface breezes, radiation from the sun, and reradiation back to the sky, as well as the geothermal flux, are explicitly included.

DEPTH(FT.)

FIGURE 4-1. STRATIGRAPHY AND PROPERTIES ASSUMED FOR GRANITE FAR-FIELD CALCULATIONS

FIGURE 4-2. STRATIGRAPHY AND PROPERTIES ASSUMED FOR SHALE FAR-FIELD CALCULATIONS. SPECIFIC HEAT FOR ALL MATERIALS IS 0.22 BTU/LB-°F

Because of the number of different materials involved and the complexity of the boundary conditions at the surface of the earth, a numerical solution is chosen as the most feasible approach. However, because of the large spatial dimensions and time span involved, great care must be given to assure accuracy. Convergence in the usual sense may not provide the necessary accuracy, and analytic solutions for simple cases have also been developed for comparison purposes. In addition, although simple analytic models are not adequate to provide detailed treatment of the formation temperatures, general trends and dependencies on various parameters can be approximately ascertained from the closed-form expressions so that the analytic model serves a useful secondary purpose. The next subsections are devoted to the numerical model, the analytic solutions, and the results of parametric studies.

4.2.2 Numerical Model

The mesh lines assumed for the numerical calculations are shown in Figures 4-5 through 4-7 for the repositories in the generic formations. The mesh spacing is chosen to reduce errors which might accrue due to the variation in the thermal gradient.

In the far-field model, specification of the initial temperature distribution is straightforward in terms of a constant temperature gradient. When several materials are present, the calculated temperatures will then include the errors associated with the approach to the correct ambient (steady-state) distribution. The proper initialization of the temperature distribution requires consistent heat balances in the formation including at the earth's surface. The situation is viewed in Figure 4-8. The heat balance at the earth's surface is given by

$$q''_{\text{solar}} = q''_{\text{conv}} + q''_{\text{rerad}} + q''_{\text{geo}}$$

where the terms correspond to the solar radiation, convection, reradiation and geothermal heat fluxes. This can be rewritten

$$q''_{\text{solar}} = h_s (T_s - T_{\text{air}}) + \epsilon_s \sigma (T_s^4 - T_{\text{sky}}^4) + k_{\text{eq}} \frac{\Delta T_0}{d}$$

RADIUS (FT.)

DEPTH (FT.)

Repository

FIGURE 4-6. MESH FOR THE SHALE CONFIGURATION

RADIUS(FT.)

DEPTH (FT.)

FIGURE 4-7. MESH FOR THE BASALT CONFIGURATION

FIGURE 4-8. INITIALIZATION PROCESS USED FOR THE NUMERICAL MODEL

where q''_{solar} is the incident solar flux, h_s is an overall heat transfer coefficient at the surface, T_s is the temperature at the surface of the earth, T_{air} is the ambient air temperature, ϵ_s is the emissivity of the surface of the earth, σ is the Stefan-Boltzmann constant, T_{sky} is the effective sky temperature for reradiation, k_{eq} is an effective conductivity for the formation and $\Delta T_o/d$ is the mean geothermal gradient. To an extent these phenomenological parameters need to be consistent with each other; that is, they cannot be selected independently of one another. For a heat-transfer coefficient of the form

$$h_s = 0.1 + 0.3V \frac{\text{Btu}}{\text{hr-}^\circ\text{F-ft}}$$

where V is the wind velocity in miles per hour, the parameters fall roughly in the following ranges in the U. S.:^{4,5}

- q''_{solar} from 300 to 600 Langley's
- Wind velocity from 8 to 11 miles per hour
- Ambient air temperature from 50 to 70°F
- Surface emissivity from 0.1 to 0.4
- Sky temperature 10 to 40°F below the ambient air temperature depending upon average cloudiness.

For the present analysis the values in Table 4-1 have been assumed. The equivalent conductivity can be readily calculated, since the initial temperature distribution is assumed to be in steady state. Consequently, the geothermal flux is everywhere constant. Therefore

$$q''_{\text{geo}} = k_1 \frac{\Delta T_1}{\Delta Z_1} = k_2 \frac{\Delta T_2}{\Delta Z_2} = k_3 \frac{\Delta T_3}{\Delta Z_3} = \dots \quad (4-1)$$

TABLE 4-1
PARAMETERS ASSUMED FOR FAR-FIELD THERMAL ANALYSIS

Parameter	Value
Mean solar flux, q'' solar	73.4 BTU/hr-ft ²
Mean wind velocity, V	10 miles per hour
Ambient air temperature, T_{air}	66.°F
Surface emissivity	0.3
Effective sky temperature, T_{sky}	45.°F
Mean geothermal gradient, $\Delta T_t/d$	0.021° F/ft

Then, since

$$\Delta T_o = \Delta T_1 + \Delta T_2 + \Delta T_3 + \dots$$

ΔT_o can be evaluated.

$$\Delta T_o = \Delta T_1 + \frac{k_1}{k_2} \frac{\Delta Z_2}{\Delta Z_1} \Delta T_1 + \frac{k_1}{k_3} \frac{\Delta Z_3}{\Delta Z_1} \Delta T_1 + \dots \quad (4-2)$$

and

$$q''_{geo} = k_1 \frac{\Delta T_1}{\Delta Z_1} = k_{eq} \frac{\Delta T_o}{d} \quad (4-3)$$

With k_{eq} known, equation (4-1) can be solved for the surface temperature. The values calculated for granite, shale and basalt are 77.8°F in each case. These values are somewhat high, reflecting the high value of T_{sky} and the geothermal gradient chosen for these formations. Nevertheless, these values are used as a basis since the quantities of interest are the temperature rises above ambient. Once T_s has been evaluated, the ambient formation temperature can be readily evaluated from (4-1). These values are then used as a basis for the numerical calculations.

4.2.3 Analytic Model

The analytic model represents the repository as a disc source in a semi-infinite medium. For purposes of comparison with the numerical model surface temperatures are studied. In this case, the model is simplified to include only the effects of convection and the geothermal flux. The effects of solar radiation and reradiation are accounted for approximately with an effective heat-transfer coefficient. The basic equation is therefore:

$$\frac{1}{\alpha} \frac{\partial T}{\partial \theta} = \frac{1}{r} \frac{\partial}{\partial r} \left(r \frac{\partial T}{\partial r} \right) + \frac{\partial^2 T}{\partial Z^2} \quad (4-4)$$

Subject to the conditions:

- At $\theta = \text{time} = 0$, $T = \begin{cases} \frac{\alpha Q}{k} & \text{for } r < a, Z = 0 \\ 0 & \text{for } r > a \text{ or } Z \neq 0 \end{cases} \quad (4-5)$

- $k \frac{\partial T}{\partial r} = 0 \text{ at } r = \infty$

- $k \frac{\partial T}{\partial Z} = \begin{cases} g \text{ at } z = -\infty \\ h_c (T - T_{\text{air}}) \text{ at } z = d \end{cases} \quad (4-7)$

where Q is the strength of an instantaneous source, $z = 0$, g is the ambient geothermal flux, h_c is the effective surface heat transfer coefficient, T_{air} is the effective air temperature, k is the formation thermal conductivity and α is the diffusivity ($\alpha = k/\rho c$). The surface is a distance d above the repository.

Equation (4-4) is linear, so superposition can be applied. This enables the extension of the solution for an instantaneous source to the more practical case of time-dependent source simply by convolution. It also allows the separation of the external fluxes from the problem. The solution for $Q = 0$ is⁴

$$T = T_{\text{air}} + \frac{g(Z-d)}{k} + T_0 \operatorname{erf}\left(\frac{Z-d}{2\sqrt{\alpha\theta}}\right) + T_0 \exp\left(-\frac{h_c(Z-d)}{k}\right) + \frac{h_c^2}{k^2} \alpha\theta \operatorname{erfc}\left(\frac{Z-d}{2\sqrt{\alpha\theta}} + \frac{h_c}{k}\sqrt{\alpha\theta}\right) \quad (4-8)$$

Here T_0 is the difference between the ambient air temperature, T_{air} , and surface temperatures. The actual temperature is obtained by adding (4-8) to the homogeneous solution found by solving the problem for T_0 , T_{air} and g equal to zero.

Carslaw and Jaeger⁴ give a homogeneous solution for a disk source in an infinite conducting medium. In general, the expression is an integral that cannot be expressed except in terms of tabulated functions; but on the axis of the disk (which is hottest anyway), it simplifies to

$$T(Z) = \frac{Q\sqrt{\alpha}}{2\pi a^2 k \sqrt{\pi\theta}} \left(1 - \exp\left(-\frac{a^2}{4\alpha\theta}\right)\right) \exp(-Z^2/4\alpha\theta) \quad (4-9)$$

where a is the radius of the disk. The surface boundary condition is approximately satisfied by subtracting the mirror-image solution to obtain

$$T(Z) = \frac{Q\sqrt{\alpha}}{2\pi a^2 k \sqrt{\pi\theta}} \left(1 - \exp\left(-\frac{a^2}{4\alpha\theta}\right)\right) \left(\exp\left(-\frac{Z^2}{4\alpha\theta}\right) - \exp\left(-\frac{(2d-Z)^2}{4\alpha\theta}\right)\right) \quad (4-10)$$

and substituting into the boundary condition

$$\frac{\partial T}{\partial Z} = -\frac{h_c T}{k} \quad \text{at } Z = d$$

giving

$$T(Z=d) = \frac{Qd}{2\pi a^2 h_c} \sqrt{\pi\alpha\theta}^{3/2} \left(1 - \exp\left(-\frac{a^2}{4\alpha\theta}\right)\right) \exp(-d^2/4\alpha\theta) \quad (4-11)$$

Actually, this solution is only a Green's function; the actual heat source has time-dependence. Equation (4-11) for the instantaneous inhomogeneous solution can be written

$$T = QT_0(r, \theta) \quad (4-12)$$

To get the correct time dependence,

$$T(r, \theta) = \int_0^\theta q(\theta') T_0(r, \theta - \theta') d\theta' \quad (4-13)$$

where $q = \frac{dQ}{d\theta}$, the heat generation rate of the source. Finally, (4-13) and (4-8) are added to get the total solution for the time-dependent heat source.

Two limiting cases are of interest. If a , the radius of the disc, is very small, the expression for a point source is obtained:

$$T_{\text{point}}(Z=H) = \frac{qd}{8h_c(\pi\alpha)^{3/2}\theta^{5/2}} \exp(-d^2/4\alpha\theta)$$

whereas if a is very large, the expression for a plane source is obtained:

$$T_{\text{plane}}(Z=H) = \frac{q''d}{2h_c(\pi\alpha)^{1/2}\theta^{3/2}} \exp(-d^2/4\alpha\theta)$$

Here q'' is the power per unit area.

4.2.4 Parametric Studies with Analytic Model

Detailed analysis of the formation temperature distribution is not possible with the analytic model developed above. Nevertheless parametric studies with model are useful to study general trends. The equation to be evaluated for surface temperature rise is given by

$$\Delta T_s = \frac{d}{h_c 2\pi a^2 (\pi\alpha)^{1/2}} \int_0^\theta \frac{q(\theta')}{(\theta - \theta')^{3/2}} \left(1 - \exp(-a^2/4\alpha(\theta - \theta')) \right) \exp(-d^2/4\alpha(\theta - \theta')) d\theta'$$

When the source term, $q(\theta')$ is constant, the temperature rise can be expressed with error functions. However, when $q(\theta')$ is a complicated function of time, then the integral must be evaluated numerically. Figures 4-9 to 4-11 were generated with spent fuel as the heat source. This function is given in tabular form in Table 1-1. The numerical integration was performed with the routines developed by Boris⁶ which are based on Gaussian integration formulae with controlled accuracy.

The results shown in Figures 4-9 to 4-10 are for an initial load of 100 kW/acre and a surface heat-transfer coefficient, h_c , of 3 watts/m²°C (0.52 Btu/hr-ft°F). The entire formation is taken to be salt with a constant conductivity of 2.4 Btu/hr-°F-ft. Note that in the expression for temperature rise (4-11), the results are proportional to the heat load, provided the same time-dependence is used, and inversely proportional to the surface heat-transfer coefficient. Figure 4-9 shows the effect of thermal diffusivity, α . Typical values for thermal diffusivity of the basic materials encountered in repository analyses are given in Table 4-2. Figure 4-10 shows the effect of repository radius on the surface temperature. As may be noted, as the radius becomes large, a maximum temperature rise will be reached. Figure 4-10 shows the effect of depth of the repository under the ground.

The results for spent fuel shown in the previous figures can be used to assess the temperature rise that may be encountered at the surface of

FIGURE 4-9. ANALYTICAL RESULTS; EFFECT OF THERMAL DIFFUSIVITY

FIGURE 4-10. ANALYTICAL RESULTS; EFFECT OF SOURCE RADIUS

FIGURE 4-11. ANALYTICAL RESULTS; EFFECT OF DEPTH OF BURIAL

TABLE 4-2
TYPICAL THERMAL DIFFUSIVITIES

Formation	$\alpha(\text{m}^2/\text{sec})$	$\alpha(\text{ft}^2/\text{hr})$
Salt	2.2×10^{-6}	0.085
Granite	1.2×10^{-6}	0.045
Shale	6.7×10^{-7}	0.026
Basalt	5.7×10^{-7}	0.022

the earth under a variety of conditions. The variables that can be considered are areal heat load, the surface heat-removal rate (as specified by the heat-transfer coefficient, h , the rock or earth thermal properties by means of the thermal diffusivity, the radius of the repository, and the depth of burial.

4.2.5 Comparison of Numerical Calculations and Analytic Solutions

The results of two comparisons between numerical and analytical results are shown in Figures 4-12 and 4-13. The cases consider a formation with $\alpha = 2.2 \times 10^{-6} \text{ m}^2/\text{sec}$ and $\alpha = 5 \times 10^{-7} \text{ m}^2/\text{sec}$. In each case the solid curve gives the analytical results, while the individual circled points are calculated by the numerical model. It should be noted that the analytical solution is an approximation for long time. This explains the improved agreement for very long time frames. The analyses presented in this section show that reliable far-field temperature histories can be computed. The results are only applicable far removed from the source and after long elapsed times. The accuracy of the far-field calculations is estimated to be about 15 percent. The accuracy can be increased by decreasing the mesh spacing and by using more detailed thermal decay profiles. However, in view of the uncertainties in the generic thermal properties, the neglect of aquifers and the other approximations made, such improvement does not seem warranted.

4.3 FAR-FIELD RESULTS

Detailed results have been generated for a wide range of cases. These cases are for three stratigraphies, granite, shale, and basalt with decay-heat profiles of HLW and spent-fuel waste. Each stratigraphy and waste form was analyzed for areal heat loadings ranging from 90 to 400 kW/acre.

The node configurations for each formation are shown in Figures 4-5 to 4-7. The corresponding stratigraphies are shown in Figure 4-1 to 4-3. Because far-field temperature changes are not expected to be very large and temperatures near the heat source, although large at times, are not representative of the near field, the thermal properties in all cases were

FIGURE 4-12. COMPARISON OF NUMERICAL VS. ANALYTICAL

FIGURE 4-13. COMPARISON OF NUMERICAL VS. ANALYTICAL

assumed to be constant and are given in Figures 4-1 and 4-3. For all materials, a specific heat of $0.22 \text{ Btu/lbm-}^{\circ}\text{F}$ has been used. The heat source was assumed to be uniformly distributed in a layer of rock 50 feet thick. Thus, with a heat load of 100 kW/acre , the equivalent heat generation rate can be calculated to be $0.155 \text{ Btu/hr-ft}^3$. Far-field thermal results have been transmitted to analysts for the thermo-mechanical stress studies (TM-36/20) and for the hydrology studies (TM-36/21). Selected isotherms throughout the formations are displayed in TM-36/21. Supplementary temperature histories at specific locations are presented here. The results are for two locations of interest, the temperature history at the surface of the earth, and those below the surface where large-scale water movement may be present. The effect of such aquifers has not been accounted for, although they may significantly affect the temperature distributions. Figures 4-14 through 4-17 show temperature histories in the granite configurations while Figures 4-18 through 4-21 show those for shale. Histories for basalt are shown in Figures 4-22 to 4-25. As may be noted the temperature rise is nearly proportional to the heat load. This effect is shown in Figure 4-26, where the peak temperatures on the axis of the repository are plotted as a function of heat load. This plot also shows that, in spite of very small temperature increases at the surface, at relatively shallow depths below the earth's surface, the temperature rise can be significant.

FIGURE 4-14. TEMPERATURE AT 750 FT. BELOW SURFACE IN THE GRANITE CONFIGURATION WITH U AND PU RECYCLE WASTE

FIGURE 4-15 TEMPERATURES AT 750 FT. BELOW SURFACE IN THE GRANITE CONFIGURATION WITH SPENT FUEL

FIGURE 4-16. TEMPERATURE RISES AT THE SURFACE IN THE GRANITE CONFIGURATION WITH U AND PU RECYCLE WASTE

FIGURE 4-17. TEMPERATURE RISES AT THE SURFACE IN THE GRANITE CONFIGURATION WITH SPENT FUEL

FIGURE 4-18. TEMPERATURE AT 175 FT. BELOW THE SURFACE IN THE SHALE CONFIGURATION WITH U AND PU RECYCLE WASTE

FIGURE 4-19. TEMPERATURES AT 175 FT. BELOW THE SURFACE IN THE SHALE CONFIGURATION WITH SPENT FUEL

FIGURE 4-20. TEMPERATURE RISES AT THE SURFACE IN THE SHALE CONFIGURATION WITH U AND PU RECYCLE WASTE

FIGURE 4-21. TEMPERATURE RISES AT THE SURFACE IN THE SHALE CONFIGURATION WITH SPENT FUEL

FIGURE 4-22. TEMPERATURES AT 300 FEET BELOW THE SURFACE IN BASALT U AND PU RECYCLE

FIGURE 4-23. TEMPERATURES AT 300 FEET BELOW THE SURFACE IN BASALT SPENT FUEL

FIGURE 4-24 TEMPERATURES AT THE SURFACE FOR BASALT-U + PU RECYCLE

FIGURE 4-25 TEMPERATURES AT THE SURFACE FOR BASALT-SPENT FUEL

FIGURE 4-26 EFFECT OF HEAT LOAD ON MAXIMUM TEMPERATURE AT A SPECIFIED DEPTH

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APPENDIX A

CONVECTIVE HEAT REMOVAL FROM A SPENT FUEL REPOSITORY IN SALT

A.1 INTRODUCTION

Preliminary calculations have been performed on the convective heat removal from spent-fuel canisters emplaced in the salt formation below a repository storage room. Recent work has determined that substantial amounts of energy generated by the waste canisters can be removed from the repository by ventilation air,¹ and the present analysis confirms this result for a wide range of repository and canister conditions (air flow rates, inlet air temperatures, and convective heat transfer coefficients). This work has been performed by Science Applications, Inc. and concludes that even for flow rates as low as 8000 cfm in the storage room, more than 80 percent of the heat generated in the first 25 years of emplacement of the spent-fuel canisters in the repository can be removed by ventilation with 70°F inlet air. Therefore, the repository could conceivably be loaded for above the limits predicted to induce storage room closure before the end of a retrievability period.

The approach has been to model the storage room as a collection of unit cells (see Section 2) with the assumption that temperature variations from unit cell to unit cell are sufficiently slow that each cell can be treated individually, accounting for the other unit cells by appropriate boundary conditions. This approach has been found to be valid for surprisingly large flow rates in the storage room.¹ In this way heat transfer from the canister directly to the storage room and indirectly through the formation can be modeled explicitly using numerical approaches which have already been developed. The total heat transfer into the storage room is then parameterized on the storage-room temperatures and heat-transfer coefficients. Finally, a separate analysis for the entire storage room is conducted based upon the parameterized results from the unit cell calculations, taking into account heat transfer to and from the formation down the entire length of the storage room to determine the net heat removal from the repository due to the ventilation.

This uncoupled procedure is approximate but necessary in order to treat the detailed behavior of the very-near-field zone involving characteristic distances of less than 1 foot as well as the large scale behavior of the near-field zone involving distances on the order of 1000 feet. A complementary analysis has been performed by Parsons, Brinckerhoff, Quade and Douglas using a different approach. This work, described in the Supplement to this volume, provides results in substantial agreement with those presented in this Appendix.

A.2 Methodology

The requirement for room and pillar stability throughout the retrievability period imposes restrictions on the maximum spent-fuel canister emplacement density in the storage rooms. When the retrievability period is ended, these requirements can be lifted and other, less stringent, limits become controlling. The objective of this analysis was to determine if, by removing heat throughout the retrievability period from the formation by ventilation of the storage rooms and/or the spent-fuel canisters, the repository could be loaded above the temporary limits imposed by the room and pillar stability requirements for the retrievability period.

The physical model for the canister and canister sleeve arrangement is seen in Figure A-1. It is assumed that heat is transferred to the air space around the canister by convection. In the calculational model the convective heat transfer to the air is treated as a boundary condition at a fixed local sink temperature, T_0 , so that all of the fluid dynamics associated with the ventilation air is implicitly modeled in terms of a single phenomenological parameter, the convective heat-transfer coefficient, h . The heat-transfer coefficient could be assigned values characteristic of forced convection or, through a model, describe the energy transfer achieved by free convection. In the present analysis, the heat transfer from the storage-room walls was presumed to be turbulent forced convection, and that from the canister and its liner as free convection described by the model given in Appendix B.

The very-near-field model for the canister unit cell is shown in Figure A-2. Based on the analysis of Appendix B, the coefficient for convective heat transfer to the air space around the canister is taken to be 0.27 Btu/hr-ft²°F. Heat is also transferred to the formation by radiative heat transfer across the air gaps. The emissivity in this case is chosen to be 0.6. Therefore in addition to the direct heat transfer to the storage room by way of the air circulating around the canister, energy may also be transferred by way of the salt formation and convection off the storage-room walls. These paths are both explicitly included in the model and are calculated for a range of values of the local storage-room temperature and the storage-room heat-transfer coefficient.

Once the results for the unit cell are known and a model for the storage-room heat-transfer coefficient is chosen, the heat removal characteristics of the entire storage room can be calculated. In this case the ventilated storage room is modeled in an array of unventilated storage rooms using the treatment of reference 1. Therefore the results are conservative. Calculations have been made for ventilation of additional rooms in the panel, holding the total air flow rate for the combined system fixed to the single-room value, and the results do not differ substantially from those for the single-room case.¹

A.3 Results

Table A-1 shows the predicted heat transfers to the ventilation air resulting from different choices of local air temperatures (T_0) and storage-room wall heat-transfer coefficients (h) for an areal thermal loading of 60 kW/acre of spent fuel. The waste is 5 years old at emplacement. Shown are the instantaneous and cumulative heat transfers. These unit-cell calculations illustrate that an average air temperature in a repository room of 60° would remove more than that which is generated, thus removing heat from the formation. Furthermore, an average air temperature in a repository room of 120° would remove approximately 50 percent of the heat generated during a 20-year retrievability phase of a repository with continuous ventilation, but

FIGURE A-2: UNIT CELL CONFIGURATION FOR CONVECTIVE HEAT REMOVAL

TABLE A-1 HEAT TRANSFER FROM WASTE CANISTER AND FORMATION TO LOCAL
VENTILATING AIR^{a)}

TIME AFTER EMPLACEMENT	INSTANTANEOUS HEAT GENERATION RATE	CUMULATIVE WASTE HEAT GENERATED	LOCAL AIR TEMPERATURES	STORAGE ROOM HEAT TRANSFER COEFFICIENT	STORAGE ROOM HEAT TRANSFER RATE	CUMULATIVE STORAGE ROOM HEAT TRANSFER
τ (YRS)	q_w (BTU/hr)	Q_w (BTU)	T_o ($^{\circ}F$)	h (BTU/HR- F° ft ²)	q_s (BTU/HR)	Q_s (BTU)
5	1870	14100	60	.2	3160	19400
			90	.2	1980	11800
			90	.6	2010	11900
			120	.2	830	4400
10	1570	22500	60	.2	2440	32200
			90	.2	1650	20500
			90	.6	1690	20800
			120	.2	910	9400
20	1270	36500	60	.2	1840	52600
			90	.2	1360	35300
			90	.6	1360	35500
			120	.2	880	18800

^aSpent-fuel waste is 5 years old at emplacement. Average unit-cell loading is 60 kW/acre. Canister-region heat-transfer coefficient = 0.27 BTU/hr - $^{\circ}F$ - ft².

only on the order of 30 percent of the heat generated through the first 5 years of heat generation.

These unit-cell calculations provide the basis to estimate heat removal for the entire storage room. The results of this analysis are shown in Table A-2. As indicated above, the model for the storage-room heat-transfer coefficient in the vicinity of the canister is given in Appendix B. The value used is 0.27 Btu/hr-ft²°F.

It is assumed that the flow in the storage room is turbulent and fully-developed. The standard duct relationship is applicable in this case and it is assumed the heat-transfer coefficient satisfied the Dittus-Boelter² relationship:

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

Where Re is the Reynolds number, Pr is the Prandtl number, k is the conductivity of the air and d is the hydraulic diameter of the storage room.

The exhaust air temperature and total convective heat removal by the ventilation air in a 3500 ft storage room are shown for several inlet air temperatures (60, 80, and 100°F), and the flow rates per room of 8, 16, and 32 kcfm. The calculations were made for an initial areal heat load of 60 kW/acre (580 kW/room) and results are presented at 5, 10, and 20 years after emplacement.

It can be seen that a ventilation-air flow rate of 16 kcfm will remove essentially 90 percent of the heat generated by the waste for inlet air temperatures of 60°F and 80°F. Even the 100°F case removed energy equivalent to 45, 58, and 70 percent of the energy generated by the waste for at 5, 10, and 20 years after emplacement, respectively. The exhaust air temperatures from the storage room remain below 140° for all conditions in Table A-2 for the 16 kcfm cases. A flow rate of 16 kcfm corresponds to an average air velocity of 0.56 miles per hour in the 18' x 18' storage room.

Also shown in Table A-2 are results at 5 years after emplacement for no direct convective heat removal from the canister. Even in this case

TABLE A-2 CONVECTIVE HEAT REMOVAL IN STORAGE ROOM^{a)}

TIME AFTER EMPLACEMENT	FLOW RATE	INLET AIR TEMPERATURE	EXHAUST AIR TEMPERATURE	HEAT REMOVAL RATE	HEAT GENERATION RATE	PER CENT HEAT REMOVED
τ (Years)	G KCFM	T_o °F	T_{ex} °F	q_{out} KW	q_{gen} KW	$q_{out}/q_{gen} \times 100$ %
5 ^b	8	60	137	185	342	54
	16		123	301		88
	32		103	407		119
	8	80	138	140		41
	16		127	227		66
	32		112	307		90
	8	100	139	95		28
	16		132	153		45
	32		122	207		61
10 ^b	8	60	142	197	287	69
	16		119	283		99
	32		97	351		122
	8	80	145	157		55
	16		127	224		78
	32		109	277		97
	8	100	149	117		41
	16		135	166		58
	32		121	205		71
20 ^b	8	60	141	195	233	84
	16		113	252		108
	32		90	290		124
	8	80	147	161		69
	16		123	208		89
	32		105	240		103
	8	100	153	127		55
	16		134	164		70
	32		120	189		81
5 ^c	8	60	134	176	342	51
	16		114	200		76
	32		96	346		101
	8	80	137	136		40
	16		121	194		57
	32		106	252		74
	8	100	141	98		29
	16		128	135		39
	32		118	172		50

^aSpent-fuel waste is 5 years old at emplacement. Repository is loaded 60 kW/acre.

^bCanister-region heat transfer coefficient = 0.27 BTU/hr - ft² - °F.

^cNo convective heat removal from canister.

the heat removal rate in the ventilated storage room is very high. For example for 80°F ventilating air the situation with the direct pathway to the storage room is less than 20% more effective at heat removal than if the pathway is closed. This illustrates that when the direct heat-transfer path is closed, the salt, itself, provides adequate heat transfer to allow substantial cooling of the formation.

The current criterion to ensure that rooms in salt will stay open for at least a 25-year retrievability period is that the net average loading (after heat removal) be less than 36 kW/acre during this time. It appears from Table A-2 that with an average loading (before heat removal) of 60 kW/acre, the ventilated repository rooms will remain open with flow rates as low as 8000 cfm for inlet air temperatures as low as 70°F. From the analysis this appears to be true whether or not direct convective heat removal from the canisters occurs.

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APPENDIX B

THERMOSYPHON-COOLED SPENT-FUEL CANISTER

The thermosyphon is a system which produces the transfer of energy, mass, and momentum through the harnessing of buoyancy forces created by temperature-induced density differences. The configuration shown in Figure B-1 is commonly referred to as an open thermosyphon, and is representative of more complicated real systems including the problem of cooling a spent-fuel canister.

This calculational model clearly identifies the system parameters (such as characteristic temperatures, fluid properties, and thermosyphon dimensions) which most strongly influence the energy transfer achieved by the thermosyphon. The desired effects of interest such as heat transfer rates are parametrically expressed in terms of these system variables. The existence of the thermosyphon model eliminates or at least reduces the user's need to directly consider the detailed phenomenological aspects of the system such as convective heat transfer mechanisms, internal flow patterns, temperature fields, variations in fluid properties, etc.

This approach places a great reliance on currently available empirical and analytical results, such as those of Japikse and Winter¹ as presented in Figure B-2. These correlations have been tested in the classical sense and may provide a reasonable basis for the present analysis.

It is assumed that the heat transfers from each of these surfaces contributes to the overall thermosyphon effect with the performance of each being described by the appropriate characteristic dimension of the thermosyphon model: $a = a_1$ for the canister surface, and $a = a_2$ for the surface of the concrete liner. It is presumed that the flow patterns required for successful thermosyphon operation can be achieved through the appropriate proportioning of flow areas of the hot and cold legs.

The uncertainty in the resulting heat-transfer coefficients can be estimated using the thermosyphon correlations for the range of

FIGURE B-1. OPEN THERMOSYPHON

$\log_{10} (Nu_o \cdot f_{or}) = M_1 \log_{10} f_{or} + b_1$ $\log_{10} (\dot{m}) = M_2 \log_{10} f_{or} + b_2$ $\log (f_{mc} - f_o) = M_3 \log_{10} f_{or} + b_3$			
$M_1 = 1.26, \quad M_2 = 0.272 \text{ and } M_3 = 0.984$			
Pr	b_1	b_2	b_3
∞	-0.243	0.172	-0.414
100	-0.246	0.161	-0.406
10	-0.265	0.0904	-0.354
3	-0.295	-0.00823	-0.285
1	-0.343	-0.137	-0.204

Nondimensional groups

Gr , Grashof number = $\beta g \Delta T a^3 / \nu^2$;

Pr , Prandtl number = $\mu C_p / k$;

Ra , Rayleigh number = $Gr Pr$;

Nu_o , Nusselt number (used for open system)

$$= ha/k = \dot{Q} / 2\pi(T_1 - T_0)k;$$

Nu_d , Nusselt number (used for closed system)

$$= hd/k = \dot{Q} / \pi(T_{1,1} - T_{1,2})k;$$

t , $\beta g a^4 T / \nu^2 L$;

t_{oi} , $\beta g a^4 (T_1 - T_0) / \nu^2 L$;

t_{ci} , $\beta g a^4 (T_{1,1} - T_{1,2}) / \nu^2 L$;

FIGURE B-2. THERMOSYPHON RELATIONSHIPS. (FROM JAPIKSE AND WINTER, REF. 1)

characteristic dimensions appropriate to the present problem. The thermosyphon expressions¹ for the conditions of this problem provide the following minimum and maximum heat-transfer coefficients:

$$h_1 = 0.112 (T_1 - T_o)^{0.26}$$

$$h_2 = 0.128 (T_2 - T_o)^{0.26}$$

The effects of differences between the problem analyzed and the thermosyphon, can be estimated from the relation:

$$\frac{\delta h}{h} = 0.26 \frac{\delta(\Delta T)}{\Delta T}$$

So that a 10 percent uncertainty in the temperature difference results in about a 3 percent uncertainty in the heat transfer coefficient.

Finally, the calculation of the convective transfer of energy by a thermosyphon requires that the characteristic temperatures be known. However, for the fixed-source problem, these temperatures can be calculated by standard heat-transfer codes which require as input the heat transfer coefficients for the convecting surfaces. Therefore, an iterative procedure can yield these simultaneous requirements. The relationship between the calculated ΔT 's and the ΔT 's required to produce the heat-transfer coefficients predicted by the thermosyphon model, is shown in Figure B-3. The intersection of corresponding curves defines the value of the heat-transfer coefficient such that the computed ΔT value is consistent with the $h(\Delta T)$ correlation predicted by the thermosyphon model. It is noted that the value of the heat-transfer coefficient does not have a strong dependence on ΔT and would assume a nearly constant value for many problems. The value of the heat-transfer coefficient for the thermosyphon model is roughly 0.27 Btu/hrft²°F.

An approach different from a thermosyphon model would have been to consider a flow annulus around the canister thin enough to ensure

FIGURE B-3. EVALUATION OF THERMOSYPHON HEAT TRANSFER COEFFICIENT

established laminar flow patterns throughout the heated length, and then to estimate the heat transfer according to the following equation due to Ehret and Hahnemann²

$$Nu_b = \frac{hb}{k} = 3.75$$

Where Nu_b is the Nusselt number based upon the channel diameter, b , and k is the conductivity of the air. Adaptation of this coefficient to the present analysis would be more difficult for the following reasons: (1) conduction across the thin gap would have to be included, (2) excessive heating of the air and resulting decrease in convective heat removal would occur due to the low mass flows, and (3) the sink temperature, with respect to which the heat-transfer coefficient is defined, is different from that required by the standard codes. This mode of convective heat removal would have to be more seriously considered if the canister significantly enhances the overall heat-removal characteristics. It is interesting to note that, assuming reasonable values for the parameters, the value of this coefficient is 0.18 Btu/hr-F°-ft². The relative closeness of the values would be expected, since laminar convection is the heat-removal mechanism in both correlations.

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SUPPLEMENT
REFINED CONVECTIVE HEAT REMOVAL REPORT*

*This work has been performed by Parsons Brinckerhoff Quade and Douglas, Inc.

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1.0 INTRODUCTION

As part of the analysis of retrievability in salt, heat transfer studies were conducted to evaluate mine ventilation requirements of the 25-year retrievable option. Under this option, canisters containing spent fuel elements would be emplaced in a repository, and if desired, could be retrieved intact within 25 years of emplacement. The use of ventilation to remove sizable quantities of heat from the repository in order to control room closure rates during the retrievable period was one of two design approaches studied. (The other approach was simply to reduce the areal thermal loading of the canisters.)

The ventilation approach provides for a constant flow of ventilating air to be supplied to the repository storage rooms over the 25-year retrievable period. Under this design approach, concrete sleeves which serve to isolate the waste canisters from the surrounding salt would be utilized. These sleeves, which would be vertically located below the repository storage room floors, enable a portion of the heat generated by the spent fuel wastes to be convected via buoyancy directly to the storage room ventilating air stream without first interacting with the salt in the immediate vicinity of the canisters. This approach would reduce the thermal loading on the surrounding salt and would therefore permit higher thermal loadings at the time of emplacement.

OWI has specified that repository rooms will remain open over the 25-year retrievable period if 60% or less of the 60 KW/acre heat generated by 10-year old spent fuel remains in the formation. The capacity of the ventilation system required to remove the other 40% of the heat was determined from heat transfer studies. These studies were performed in order to characterize the thermodynamic interrelationship among the spent fuel canisters, the concrete sleeves, the salt formation, and the ventilating air flow through the storage rooms.

As part of this effort, a near-field analysis on the concrete sleeves was performed. The purpose of the analysis was threefold: 1) to determine the feasibility of the free ventilating aspect of the design; 2) to estimate what percentage of heat generated by the canister is removed by convection to the room above; and 3) to examine the temperature distribution in the vicinity of the canister. In conjunction with this near-field analysis, a second analysis was undertaken, in cooperation with SAI, to investigate the heat transfer relationship among the air flow in the repository storage rooms, the surrounding salt formation, and the performance estimates for the concrete sleeves.

For the purpose of analysis, two concrete sleeve configurations were considered. The first configuration consisted of two concentric concrete cylinders enclosing the protective steel liner which houses the canister. Air spaces are provided between the two concrete cylinders and between the inner cylinder and the steel liner with connecting air spaces at the bottom. This configuration would be located in a vertical hole in the floor of the storage room with the outer concrete surface in contact with the salt formation.

The second configuration studied had the outer concrete cylinder removed while maintaining the resulting void between the inner concrete cylinder and the salt. This configuration was also used to evaluate the performance of concrete sleeves emplaced in trenches. The configuration dimensions used in the analysis were based on structural and radiation shielding considerations and therefore do not necessarily represent optimum design dimensions with regard to free convection heat removal.

In concept, air would flow down the outer channel and up the inner channel, past the steel liner. The driving force for this flow is the heat generated by the canister. Due to the transfer of this heat away from the canister, temperature differences between the inner and outer columns of air are created. These temperature differences cause differences in air density which in turn result in pressure differences which cause the air to flow.

In the thermodynamic analysis of the sleeve configurations, steady state heat transfer models were first developed to determine what portion of the heat generated by a canister reaches the storage room above and what portion is absorbed by the surrounding salt medium. In these models, the storage room air temperature above the canister and the temperature of the salt adjacent to the concrete sleeves were considered invariant with time. The heat flux to the salt formation was then used as input to a subsequent analysis which investigated the transient response of the temperature of the surrounding salt. The integrated results of both efforts provided an estimate of the transient heat removal behavior of the concrete sleeve configurations.

The storage room heat transfer analysis was carried out with the aid of previously verified heat transfer computer programs.^{1,2} In this analysis, a matrix of simulations was performed in which the air flow rate and entering air temperature to the storage rooms were varied. The resulting air temperature distributions along the length of the storage rooms over the retrievability period served to evaluate the heat removing capability of the ventilation concept over the range of air flows studied.

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2. Subway Environmental Design Handbook, Volume II, Subway Environment Simulation Computer Program (SES), October 1975, National Technical Information Service No. PB-254790.

2.0 METHODOLOGY

The placement of spent fuel canisters in free convecting concrete sleeves provides, in concept, two heat transfer paths away from the canisters: convection to the storage room air above and conduction to the surrounding salt. The percentage of heat initially removed via each of these paths does not, however, determine the overall percentage of heat that remains in the salt formation. Some of the heat convected from the canisters to the storage room may reenter the formation at the storage room wall surfaces; some of the heat initially conducted to the surrounding salt may, in turn, reach the storage room and be convected away by the ventilating air.

In order to study these thermodynamic phenomena over the entire length of the storage room, an analytical model was developed to characterize the performance of the concrete sleeve configurations as a function of the air temperature above the canister. In conjunction with this effort and with heating studies performed by SAI, an analysis was carried out utilizing the Heat Sink subprogram of the Subway Environment Simulation (SES) program. This analysis evaluated the air temperature distribution over the entire length of a storage room. The combination of these efforts provided sufficient information to identify the required ventilation capacity for removing heat from the repository to insure 25-year retrievability.

2.1 CONCRETE SLEEVE ANALYSIS

The concrete sleeve analysis was undertaken to investigate the feasibility of free convection as a means of removing heat from the canister and to determine the percentage of generated heat that is removed through this mode. The concrete sleeve analytical model is a combination of a steady-state model of the sleeve configuration and a transient model of the temperature response of the salt surface adjacent to the sleeve.

The steady-state model focuses on the performance of the sleeve configuration as influenced by those parameters that are used in describing the heat transfer process. The local storage room air temperature and the salt surface temperature adjacent to the sleeve serve as boundary conditions in the mathematical model.

The transient model focuses on the temperature response of the salt surface adjacent to the sleeve as influenced only by the transfer of heat at this surface predicted by the steady-state model. The thermal effects of adjacent canisters and of the storage room air above were not addressed in estimating the transient temperature response at this surface.

2.1.1 Concrete Sleeve Steady-State Model

The concrete sleeve configurations for which steady-state mathematical models were developed are shown in Figures 2-1 and 2-2. Simplified schematics which correspond to the models are shown in Figure 2-3.

The assumptions included in the mathematical model for the concentric concrete sleeve configuration are:

- o The radial heat flow from the canister is axial symmetric, i.e., the thermal effects of adjacent canisters are considered negligible.
- o Axial heat conduction in relation to radial heat conduction in the concrete sleeves is negligible.
- o The canister heat generation rate is constant. Although the canister heat generation rate decays with time, the time frame associated with any significant reduction is such that the decay rate does not impact upon the free convective performance of the sleeve.
- o The temperature of the salt adjacent to the concrete sleeve is constant. Although the temperature of the salt can and usually will vary with time as a result of heat flux to the salt, this variation is relatively slow in comparison to the temperature response of the sleeve configuration. Consequently, a quasi-steady-state situation in the sleeve configuration exists.

- o Convective heat transfer in the sleeve configuration can be adequately represented using average channel air temperatures and average channel surface temperatures.
- o At the top of the sleeve configuration, it is assumed that there is no interaction, either thermally or aerodynamically, between the entering and leaving air flows and that the entering air temperature is constant and corresponds to the average local air temperature in the storage room above.
- o The heat transfer coefficient at the surfaces of the air channels can be estimated using an empirical expression for natural convection of heat from a vertical cylinder in the turbulent range. At the outset of the analysis, a brief literature search was undertaken to determine the appropriate magnitude of the heat transfer coefficient at the channel surfaces. The result of this effort, however, did not produce the desired information thought to be applicable to the particular geometric configuration under investigation and for the particular range of Grashof numbers anticipated. Due to time constraints, an alternative approach was taken whereby the magnitude of the heat transfer coefficient was estimated using a simplified expression for free convection from a vertical cylinder, followed by a sensitivity study of the resulting solution to the magnitude of this coefficient.
- o The flow of air in each channel is uni-directional.
- o The air density of each column and, in turn, the net buoyant force on the air can be adequately estimated using the average air temperature in each column of air.

The following notation has been utilized to describe this system mathematically:

- 1 = subscript of parameters at the outer surface of the steel liner housing the canister
- 2 = subscript of parameters at the inner surface of the inner sleeve
- 3 = subscript of parameters at the outer surface of the inner sleeve
- 4 = subscript of parameters at the inner surface of the outer concrete sleeve
- s = subscript of parameters at the salt surface
- q = heat generation rate of the canister (BTU/hr.)
- A = surface area (ft.²)
- h = convective heat transfer coefficient (BTU/hr.-ft.²-°F)
- h_r = radiative heat transfer coefficient (BTU/hr.-ft.²-°F)
- R = radial distance from center of canister (ft.)
- T_{a1} = average air temperature in the inner channel (°F)
- T_{a2} = average air temperature in the outer channel (°F)
- T_{a2i} = temperature of air entering outer channel (°F)
- T_{a2o} = temperature of air leaving outer channel (°F)
- T_{a1i} = temperature of air entering inner channel (°F)
- T_{a1o} = temperature of air leaving inner channel (°F)
- T_{salt} = temperature of salt in contact with outer sleeve (°F)
- CFM = volume air flow rate through the configuration (ft.³/min.)
- C_p = specific heat of air (BTU/lb_m-°F)
- ρ = density of air (assumed constant and evaluated at 100°F (lb_m/ft.³))
- L = length of the canister (ft.)
- K = thermal conductivity of concrete sleeves (BTU/hr.-ft.-°F)

With the above assumptions, the radial flow of heat in the sleeve configuration via conduction and radiation, and the vertical flow of heat via convection can be described by the following energy equations:

$$q = A_1 h_1 (T_1 - T_{a1}) + A_1 h_{r1-2} (T_1 - T_2) \quad (2.1.1a)$$

where

$$h_{r1-2} = (0.9)(0.172) \left[\frac{\left(\frac{T_1 + 460}{100} \right)^4 - \left(\frac{T_2 + 460}{100} \right)^4}{T_1 - T_2} \right]$$

$$h_1 = 0.21 (T_1 - T_{a1})^{1/3} \quad (\text{See Appendix B})$$

$$A_1 h_{r1-2} (T_1 - T_2) = A_2 h_2 (T_2 - T_{a1}) + \frac{2\pi KL}{\ln \left(\frac{R_3}{R_2} \right)} (T_2 - T_3) \quad (2.1.1b)$$

$$\frac{2\pi KL}{\ln \left(\frac{R_3}{R_2} \right)} (T_2 - T_3) = A_3 h_{r3-4} (T_3 - T_4) + A_3 h_3 (T_3 - T_{a2}) \quad (2.1.1c)$$

$$A_3 h_{r3-4} (T_3 - T_4) = A_4 h_4 (T_4 - T_{a2}) + \frac{2\pi KL}{\ln \left(\frac{R_5}{R_4} \right)} (T_4 - T_{salt}) \quad (2.1.1d)$$

$$A_1 h_1 (T_1 - T_{a1}) + A_2 h_2 (T_2 - T_{a1}) = (CFM)(\rho)(C_p)(T_{1o} - T_{1i})(60) \quad (2.1.1e)$$

$$A_3 h_3 (T_3 - T_{a2}) + A_4 h_4 (T_4 - T_{a2}) = (CFM)(\rho)(C_p)(T_{2o} - T_{2i})(60) \quad (2.1.1f)$$

Since,

$$T_{20} = T_{1i}$$

$$T_{20} - T_{2i} = 2(T_{a2} - T_{2i})$$

$$T_{10} - T_{1i} = 2(T_{a1} - T_{1i}) = 2(T_{a1} - T_{20}) = 2(T_{a1} - 2T_{a2} + T_{2i})$$

Equations 2.1.1e and 2.1.1f can be rewritten in terms of the average channel air temperatures, T_{a1} and T_{a2} , and the local average storage room temperature, T_{2i} :

$$A_1 h_1 (T_1 - T_{a1}) + A_2 h_2 (T_2 - T_{a1}) = (CFM)(\rho)(C_p)(2)(T_{a1} - 2T_{a2} + T_{2i})(60) \quad (2.1.1g)$$

$$A_3 h_3 (T_3 - T_{a2}) + A_4 h_4 (T_4 - T_{a2}) = (CFM)(\rho)(C_p)(2)(T_{a2} - T_{2i})(60) \quad (2.1.1h)$$

To complete the description of the steady-state problem mathematically, an additional equation is necessary to relate the volume air flow rate, the geometry of the channel, and the buoyant force on the air as a function of the average air temperature of each column. The resulting equation is derived in Appendix A:

$$(CFM)^2 = 7.58 \times 10^8 \left[\frac{1}{T_{a2} + 460} - \frac{1}{T_{a1} + 460} \right] \quad (2.1.1i)$$

For a given air inlet temperature of the outer channel corresponding to the local storage room temperature, T_{a2i} , the heat generation rate of the canister, q , and the temperature of the salt in contact with the outer sleeve, T_{salt} , the solution to the problem reduces to solving for seven unknowns, T_1 , T_2 , T_3 , T_4 , T_{a1} , T_{a2} , and CFM - with six linear equations having temperature dependent coefficients, 2.1.1a-d, g, h, and one non-linear equation, 2.1.1i.

The problem was programmed on a computer and solved utilizing an iterative procedure. The basic sequence of operations in the procedure was as follows:

1. Assume values for temperature dependent heat transfer coefficients.
2. Assume value for CFM.
3. Solve the set of linear equations.
4. Substitute resulting temperatures in non-linear equation and solve for CFM.
5. Re-compute temperature dependent heat transfer coefficients.
6. Estimate new value for CFM based on previous re-computed values.
7. Return to step 3 until all equations are simultaneously satisfied.

Although the non-linear equation makes possible more than one solution to the mathematical problem, only the solution that satisfies the physics of the actual problem need be considered.

The assumptions upon which the mathematical model for the second concrete sleeve configuration is based coincide with those used for the first configuration studied. The removal of the outer concrete sleeve, however, requires that the governing equations previously set forth be modified to reflect the change in the configuration. These modifications are as follows:

- o Equation 2.1.1c replaced by

$$\frac{2\pi KL}{\ln\left(\frac{R_3}{R_2}\right)}(T_2 - T_3) = A_3 h_{r3-s}(T_3 - T_{salt}) + A_3 h_3(T_3 - T_{a2}) \quad (2.1.1j)$$

- o Equation 2.1.1d eliminated.

- o Equation 2.1.1f replaced by

$$A_3 h_3(T_3 - T_{a2}) + A_s h_s(T_s - T_{a2}) = (CFM)(\rho)(C_p)(2)(T_{a2} - T_{2i})(60) \quad (2.1.1k)$$

- o Equation 2.1.1i replaced by

$$(CFM)^2 = 1.26 \times 10^9 \left[\frac{1}{T_{a2} + 460} - \frac{1}{T_{a1} + 460} \right] \quad (\text{See Appendix A})(2.1.11)$$

- o All other equations remain the same.

The method of solution previously described was also employed in solving this second set of equations.

2.1.2 Surrounding Salt Transient Model

The rate of heat flux between the sleeve configuration and the surrounding salt formation is dependent upon the temperature of the adjacent salt surface. The salt surface temperature is dependent upon the heat conduction history of the surrounding formation and the heat flux rate history at the salt surface. An interdependence thus exists between the salt surface temperature and the heat flux to the formation. Since the steady-state model predicts the heat flux to the formation as a function of the salt surface temperature, only a heat conduction model is required which, with the heat flux rate history at the surface as input, can predict the corresponding salt surface temperature response.

For the purpose of estimating the transient response of the salt surface temperature, the solution to the heat conduction problem of a region bounded internally by the cylinder having radius $r=a$, with zero initial temperature and constant heat flux, Q , per unit time and per unit area at $r=a$, was used. The solution given by Carslaw and Jaeger¹ for the temperature response at $r=a$ for small values of the parameter kt/a^2 is:

$$T = \frac{2Q}{K} (kt)^{1/2} \left[\text{ierfc}(0) - \frac{(kt)^{1/2}}{a} i^2 \text{erfc}(0) \right] \quad (2.1.2a)$$

where

T = the temperature rise at $r=a$ ($^{\circ}\text{F}$)

Q = heat flux at $r=a$ (BTU/hr.-ft.^2)

k = thermal diffusivity of region material ($\text{ft.}^2/\text{hr}$)

K = thermal conductivity of region material ($\text{BTU/hr.-ft.}^{\circ}\text{F}$)

a = radius at surface (ft.)

t = time (hr.)

$\text{ierfc}(0)$ = first integral of complementary error function evaluated at zero

$i^2 \text{erfc}(0)$ = second integral of complementary error function evaluated at zero

For large values of time t , the solution is approximated by:

$$T = \frac{Qa}{2K} \left[\ln \left[\frac{4kt}{Ca^2} \right] + \frac{a^2}{2kt} \ln \left[\frac{4kt}{Ca^2} \right] + \frac{a^2}{2kt} + \dots \right] \quad (2.1.2b)$$

where

$$C = e^\gamma$$

$$\gamma = 0.57722 \dots \text{ (Euler's constant)}$$

The above solution is for the problem of constant heat flux at the internal surface at radius $r=a$. Since the problem for which this model was chosen to represent corresponds to a heat flux at the salt surface which is not constant (it is a function of the salt surface temperature), a procedure was adopted in developing the input for the solution so that the variation in heat flux and its surface temperature dependency could be reflected somewhat in the resulting temperature response at the salt surface.

The procedure involved first estimating the average heat flux to the salt for a given change in salt surface temperature from its original value (which was assumed equal to the temperature of the undistributed formation). This was done by integrating the heat flux to the salt provided by the steady state sleeve models over the entire change in salt surface temperature. For the given change in surface temperature and the corresponding average heat flux occurring over this change, the appropriate equation was solved to determine the time required for this change to occur.

Through repeated application of this procedure to total changes in the salt surface temperature, an estimate of the transient behavior of the salt surface temperature was developed. This, in turn, provided an estimate of the transient flow of heat between the canister and the surrounding salt.

2.2 STORAGE ROOM VENTILATION ANALYSIS

In addition to the direct exchange of heat between the canisters and the storage room air via the free convecting sleeves, storage room air temperatures are influenced directly by the heat conduction history of the surrounding salt. The rate of heat flux between the air and the storage room walls is dependent on the difference in their temperatures and on the convective heat transfer coefficient. Heat transfer studies were performed to ascertain the heat flux between the storage room air and the surrounding salt along the length of the rooms, and the resulting room air temperatures so that the heat removal capability of ventilating air through the repository storage rooms could be evaluated.

In the initial studies performed by SAI, the heat generation rates correspond to those of spent fuel (which is ten years old at emplacement) and an areal loading of 60 KW/acre. The room geometry corresponded to that shown in Figure 2-4. A matrix of cases was evaluated by varying the pertinent parameters: air flow rate through each storage room (16, 32, 64×10^3 CFM), temperature of the air entering the room (60°F, 80°F, 100°F), and the performance of the free convecting concrete sleeves (100%, 50%, 25% efficient). The performance efficiency is the percentage of heat generated that is directly convected to the storage room air above. The remaining heat was considered a heat source located appropriately below the storage room floor. In these parametric studies, the performance of the concrete sleeves was assumed explicitly and the initial temperature of the surrounding salt was assumed to correspond to the undisturbed formation temperature (92.8°F).

Based on the concrete sleeve performance analysis, it was found that for the range of storage room air temperatures that might be encountered, the efficiency of the sleeves in general tended to approach 100% within two years after emplacement. That is, almost all of the heat generated at this time would be convected directly to the storage room air above, and

the remaining small percentage of the generated heat would be conducted through the salt adjacent to the sleeves. Consequently, the heat removal capability of the ventilating air through the storage rooms could be estimated by evaluating only the heat flux between the ventilating air and the storage room walls while assuming that all of the generated canister heat is deposited directly into the storage room air above.

To implement this approach so that the impact of design changes (with respect to room layouts, thermal loading, etc.) on ventilation requirements could be evaluated promptly, the Heat Sink subprogram of the Subway Environmental Simulation (SES) program was employed. This subprogram computes the effects of long-term heat addition to underground tunnels from the surrounding earth, as a function of the thermal characteristics of the earth and the velocity dependent transfer of heat between the air in the tunnel and the tunnel walls. A complete description of the mathematical models that constitute this subprogram and their interrelationship along with a description of the entire SES program is presented in the Subway Environmental Design Handbook?

In the storage room ventilation analysis, the Heat Sink Subprogram was first used to evaluate those cases evaluated initially by SAI for free ventilating sleeves having a thermal performance efficiency of 100%. These evaluations were performed to support the findings of each of these independent analyses as well as to determine the impact on the results of the Heat Sink subprogram's limitations with regard to its applicability to multi-storage room layouts (more than two storage rooms in parallel).

Based on the results of these evaluations which indicated good agreement with the results of SAI, subsequent evaluations were performed for the room geometry of the final design proposed by PBQ&D for the entire 25-year retrievable option in salt; shown in Figure 2-5. For this analysis, the heat generation rates correspond to that of spent fuel which is 10 years old at emplacement and an areal loading of 60 KW/acre.

A room air flow rate of 9000 CFM was investigated with room entering air temperatures of 60°F, 80°F, and 100°F. In evaluating the heat removal capability of the ventilating air for these three cases five years after emplacement, two evaluations were performed for each case with the difference being the heat generation rate. Since the Heat Sink subprogram assumes the heat generation rate to be constant at any given location in the storage room over the elapsed time period, heat generation rates corresponding to 10 and 15 year old spent fuel were considered so that the solution to the actual problem of a decaying heat rate could be bracketed.

2. REFERENCES

1. Carslaw, H.S. and Jaeger, J.C., Conduction of Heat in Solids, Clarendon Press, Oxford, 1959.
2. Subway Environmental Design Handbook, Volume II, Subway Environment Simulation Computer Program (SES), National Technical Information Service No. PB=254790.

FIG. 2-1 CONCENTRIC SLEEVE CONFIGURATION

FIG. 2-2 SINGLE SLEEVE CONFIGURATION

FIG. 2-3 SIMPLIFIED CONFIGURATION SCHEMATICS

FIG. 2-4 PRELIMINARY STORAGE ROOM GEOMETRY

ROOM LENGTH = 3500'

FIG. 2-5 FINAL STORAGE ROOM GEOMETRY

3.0 DISCUSSION OF RESULTS

3.1 CONCRETE SLEEVE STEADY-STATE ANALYSIS

The steady state performance analysis of the concrete sleeve configurations indicates that the percentage of generated canister heat that is freely convected to the storage room air is essentially proportional to the temperature difference between the salt adjacent to the sleeve and the storage room air above. In the six cases studied, 100% of the generated canister heat was found to be convected to the storage room air where the aforementioned temperature differences ranged from as high as 23°F to as low as 7°F. When these temperature differences were exceeded, extraction of ambient heat from the formation occurred. Conversely, when the temperature difference was less than these values, a portion of the generated heat was absorbed by the adjacent salt formation.

For the steady state analysis, six cases were studied. In each case the percentage of generated canister heat removed through the free convection mode was evaluated over a range of storage room air temperatures for a constant adjacent salt temperature. Three salt temperatures were considered: 93°F, 113°F, and 133°F. A brief discussion of each case is presented below and results are summarized in Table 3-1.

3.1.1 Case 1

This case examines Configuration 1 (shown in Figure 2-3) which corresponds to the canister encased in two concentric concrete sleeves buried in a vertical hole below the storage room floor, with the outer sleeve assumed to be in perfect contact with the surrounding salt. The results are presented in Tables 3-2, 3-3, and 3-4. The percentage of generated heat convected to the storage room and the air flow rate through the configuration are shown plotted in Figures 3-1 and 3-2, respectively.

The results indicate that when the difference in temperature between the salt in contact with the outer sleeve and the room air is approximately 13°F, 100% of the generated heat is removed via convection to the storage room. Larger differences in these temperatures are shown to result in the extraction of ambient heat from the formation. This is accompanied by higher air flow rates through the sleeve configuration which is consistent with an increase in the free convecting driving force: heat transfer to the air flow. As the air temperature above the canister increases, larger percentages of the generated heat are transferred to the salt. This causes higher temperature gradients in the radial direction to occur and results in higher canister surface temperatures.

3.1.2 Case 2

This case was performed to evaluate the sensitivity of the steady state model to the magnitude of the convective heat transfer coefficient at the channel wall surfaces. Case 1 was repeated with the expression for the heat transfer coefficient modified by multiplying it by a factor of 0.5. The results are shown in Tables 3-5, 3-6 and 3-7 and in Figures 3-3 and 3-4.

A reduction in the convective heat transfer coefficient results in a decrease in the free convective performance of the configuration in comparison to Case 1. This is expected since the only mechanism of heat transfer to the air that was considered has been reduced. Only a proportional increase in the temperature differences between the air and the channel surfaces could counteract the reduced heat transfer coefficient. This is shown to occur to some degree but not sufficiently to equal the convective performance of Case 1.

Although the reduction in convective heat transfer coefficient ranged between 40% and 50%, the reduction in performance was only 18% for the 100% performance condition and progressively decreased as the performance decreased. The canister surface temperature did show, however, an increase in temperature, but only on the order of 10°F to 15°F.

3.1.3 Case 3

This case examines Configuration 2 (shown in Figure 2-2) which has the outer concrete sleeve removed. The removal of the outer sleeve provides for a larger outer air channel which results in reduced resistance to flow compared to that of Configuration 1. In addition, the canister diameter has increased from 12 in. to 16 in., thus providing more surface area for heat transfer. As expected, this configuration is more efficient in terms of heat removal by convection. The results are presented in Tables 3-8, 3-9 and 3-10 and Figures 3-5 and 3-6.

3.1.4 Case 4

This case is intended to study the performance of the sleeve configuration studied in Case 3 when it is placed in a trench as shown in Figure 3-7. The width of the trench corresponded to the diameter of the hole used in Case 3 and the center to center spacing of the canisters was taken as 5.5 ft. In the mathematical model, the salt surface area in contact with the air surrounding the sleeve, A_4 , was reduced to comply with the actual surface area available per unit canister. The geometric shape factor used in computing the radiant heat transfer between the sleeve and the surrounding salt was unchanged. The results, which are presented in Tables 3-11, 3-12 and 3-13 and Figures 3-7 and 3-8 indicate that the placing of the canister assembly in a trench rather than in a hole has little impact on its free convective performance.

3.1.5 Case 5

Case 5 investigates the free convective performance of the assembly as a function of canister heat generation rate. This was accomplished by repeating Case 4 with the heat generation rate per canister reduced from 1.0 to 0.5KW. The results are presented in Tables 3-14, 3-15 and 3-16 and Figures 3-9 and 3-10.

The results indicate that for air temperatures resulting in more than 100% of the generated canister heat convected to the room above, a reduction in the generated heat has little impact on the quantity of heat extracted from the formation. A comparison of Tables 3-11 and 3-14 for 60°F air temperature indicates the quantities of heat extracted from the formation to be approximately 0.94 and 1.03KW, respectively. This fact accounts for the rather minor reduction in air flow through the configuration even though the quantity of generated heat is reduced in half. The reduction in generated heat is clearly evident in the accompanying reduction of the canister surface temperature.

3.1.6 Case 6

This case considers the effect of reducing the geometric shape factor for radiant heat transfer between the sleeve and the walls of the trench to account for interference due to the adjoining canister assemblies in the trench. Case 5 was repeated with the shape factor between A_3 , the sleeve outer surface, and A_4 , the surface of the trench, reduced from a value of 1.0 to 0.7. The results are presented in Tables 3-17, 3-18 and 3-19 and Figures 3-11 and 3-12. A comparison of cases 5 and 6 indicates that the change in the shape factor has essentially no effect on the free convective performance of the sleeve.

Although the steady-state sleeve analysis does not consider conduction through the adjoining salt as a heat flow path between the canister and the storage room air in parallel with free convection, the results of the analysis can be used to estimate the significance of this heat flow path in relation to convection. An examination of the steady-state results for Case 1 indicates that when the salt surface temperature adjacent to the sleeve is 13°F greater than the local storage room air temperature above, 100% of the generated canister heat is convected to the storage room air. However, since a 13°F temperature difference exists between the salt surface and the storage room air, a parallel flow of heat must occur through

the salt to the storage room air. As a first order approximation, the magnitude of this heat flow was computed using the storage room floor surface area per unit canister and the distance between the center of the canister and the storage room floor as the effective cross section and length of the heat flow path, respectively. Not including the additional thermal resistance at the storage room floor surface, a quantity of heat equal to approximately 6% of the generated heat would be conducted through the salt to the storage room air. For Cases 2 through 6, this percentage ranges between 4 and 10%. When there is no potential for conduction heat flow to the storage room air, i.e., the local storage room air temperature and the salt surface temperature adjacent to the sleeve are equal, between 60 and 70% of the generated heat is still shown to be convected to the storage room air for Cases 1 through 6. The implications are 1) that the convective mode is the predominant heat flow mechanism by which generated canister heat is transferred to the storage room air, and 2) that the results of the steady-state model are essentially valid even though the conduction flow path to the storage room air was not addressed in the model.

3.2 CONCRETE SLEEVE TRANSIENT ANALYSIS

The results of the transient analysis indicate that the performance efficiency of the sleeves tends to approach 100% within a period of two years after emplacement. The clear implication is that the major resistance to heat flow from the canister is the surrounding salt and that the temperatures in the sleeve assembly adjust accordingly to accommodate the free convection mode of heat transfer.

The transient analysis investigated the temperature response of the salt surface adjacent to the sleeve and the corresponding performance of the sleeve configuration. The results presented herein correspond to the transient analysis of Configuration 2. The results of Case 3 (for the steady state analysis) were used in this analysis to provide the heat flux to the salt as a function of the salt surface temperature and the storage room air temperature above the canister. Storage room air temperature of 60°F, 90°F and 120°F were considered. The initial temperature of the surrounding salt was considered uniform and equal to the undisturbed formation temperature.

The heat flux to the salt given as a function of salt surface temperature is presented in Table 3-20 for the three storage room air temperatures considered. From these values, average heat flux rates occurring over changes in salt temperature were computed. The time required to affect these changes in salt surface temperature were evaluated using these average heat flux rates as input to the transient model. The results of this effort are presented in Tables 3-21, 3-22 and 3-23 and plotted in Figures 3-14, 3-15 and 3-16. A more comprehensive presentation of these results is given in Table 3-24 and is graphically depicted in Figures 3-17 and 3-18.

Tables 3-21, 3-22 and 3-23 give the time required in days for the salt surface temperature adjacent to the sleeve to change from its original value of 93°F to the values shown. This temperature variation is shown plotted versus time in Figures 3-13, 3-14 and 3-15. Based on these Figures, Table 3-24 was generated which gives the salt surface temperatures and the corresponding heat flux to the salt at various points in time after emplacement. This information is shown plotted in Figures 3-16 and 3-17 as a function of the storage room air temperature above the canister.

Referring to Figures 3-16 and 3-17, it can be seen that at the time of emplacement the performance of the sleeves varies significantly as a function of the local air temperature above the canister. For 60°F air,

more heat is extracted from the salt than is actually generated by the canister. At the other extreme (120°F air), almost all of the heat generated by the canister is transferred to the adjacent salt. As time elapses however, the performance of the sleeves approaches a condition whereby the transfer of heat between the sleeves and the surrounding salt reduces significantly and the majority of heat that is transferred convectively to the storage room air is the generated canister heat. The figures indicate that an equivalent of more than 90% of the generated canister heat is convected to the storage room air two years after emplacement.

3.3 STORAGE ROOM ANALYSIS

The results of the storage room analysis, which utilized the SES program to evaluate the heat removal capability of ventilating air through the repository storage rooms, indicate that air flow rates as low as 9000 CFM per room will remove sufficient quantities of heat from the repository to satisfy the requirements for 25-year retrievability.

Table 3-25 presents the results of the analysis performed with the SES program. In this analysis all of the canister generated heat was dumped directly into the storage room air and distributed uniformly over the length of the room. Air flow rates of 16, 32 and 64 x 10³ CFM per room were evaluated for a room inlet temperature of 60°F. The heat generation rate was held constant over the evaluation period at the value at burial (10 year old spent fuel, 60 KW/acre). The results indicate that for an air flow rate as low as 16,000 CFM per room, an amount of heat equivalent to 81% of the canister generated heat is removed by the ventilating air five years after emplacement. Figures 3-18, 3-19 and 3-20 show the distribution of average air and wall surface temperature over the length of the room for the three air flow rates. For 16,000 CFM the formation rejects heat to the ventilating air over the first 900 feet of the room. For the remaining 2600 feet of the room, the formation

absorbs a quantity of heat equal to the rejected formation heat plus an amount of heat equivalent to 19% of the canister generated heat. Although the results indicate the net heat gain by the formation to be essentially zero for 32,000 CFM through the room, Figure 3-19 indicates a net exchange of heat between the formation associated with the front and rear halves of the room.

The convective heat transfer coefficient at the wall surface was computed internally by the program in accordance with the calculation procedure described in Appendix C. When free convection did not predominate, the velocity dependent coefficient corresponded to 0.12, 0.22 and 0.41 BTU/hr.-ft.²-°F for the three air flows, respectively.

A comparison of these results with those of SAI indicates relatively good agreement. At five years after emplacement, differences in the ratio of heat removed by the ventilating air to heat generated by the canisters were found to be on the order of 7%, 4% and 1% of those ratios shown in Table 3-25 for 16,000, 32,000 and 64,000 CFM, respectively. In this comparison, the SAI results were for the end storage rooms of the panel.

In addition to analyzing the problem with a room inlet temperature of 60°F, the SAI analysis considered room inlet temperatures of 80°F and 100°F. The decay of the canister heat generation rate with time was also considered in these evaluations. The results indicate that the percentage of generated heat extracted by the ventilating air increases with time over the retrievable period. This was found to hold for all three room inlet temperatures studied. Consequently, if the percentage of generated heat removed by a particular storage room ventilation rate is sufficient during the initial years of the 25-year retrievable period, then the percentage of generated heat removed by the ventilating air over the entire period will therefore be more than adequate to ensure retrievability.

The results of analyzing final room design with an air flow rate of 9000 CFM per room are shown in Figure 3-21. The heat generation rate corresponds to spent fuel which is 10 years old at emplacement and an areal

loading of 60 KW/acre. Room air inlet temperatures of 60°F, 80°F and 100°F were studied. The analysis was performed twice, with constant heat generation rates corresponding to that at burial and five years after emplacement. The percentage of generated heat that is removed by the formation five years after emplacement as a function of the room inlet temperature, is bracketed by the two solutions shown in Figure 3-21. For the range of inlet temperatures considered, sufficient quantities of the generated heat are shown to be removed by a ventilation rate of 9000 CFM per room to satisfy the requirements for 25-year retrievability.

The results of an analysis performed by SAI for the same room geometry and thermal loading are also shown in Figure 3-21. The air flow rate through the storage room in the SAI analysis, however, is 7500 CFM. In addition to the difference in air flow rates, the convective transfer of heat from the concrete sleeves to the storage room air was implicitly modeled in the SAI analysis, in contrast to the approach followed in this analysis where all of the generated heat is dumped directly into the storage room air. In order to compare results of the PBQ&D and SAI analyses, the results of the SAI analysis were modified to reflect a room air flow rate of 9000 CFM. The modification was accomplished by assuming the room exhaust temperature remained unchanged due to this minor increase in flow. The result of this modification was an increase in the quantity of heat removed by the ventilating air proportional to the air flow rates. These modified results, which are also shown in Figure 3-21, compare remarkably well with the results of the PBQ&D analysis.

FIG. 3-1 PERCENTAGE OF HEAT TO AIR (BASED ON 1 KW HEAT GENERATION RATE OF THE CANISTER) VS AIR INLET TEMPERATURE FOR DIFFERENT SALT SURFACE TEMPERATURES (CASE 1)

FIG. 3-2 FLOW RATE OF AIR VS AIR INLET TEMPERATURE FOR VARIOUS SALT SURFACE TEMPERATURES
(CASE 1)

FIG. 3-3 PERCENTAGE OF HEAT TO AIR (BASED ON 1 KW HEAT GENERATION RATE OF THE CANISTER) VS AIR INLET TEMPERATURE FOR VARIOUS SALT SURFACE TEMPERATURES
(CASE 2)

FIG. 3-4 FLOW RATE OF AIR VS AIR INLET TEMPERATURE
(CASE 2)

FIG. 3-5 PERCENTAGE OF HEAT TO AIR (BASED ON 1 KW HEAT GENERATION RATE OF THE CANISTER) VS AIR INLET TEMPERATURE FOR VARIOUS SALT SURFACE TEMPERATURES
(CASE 3)

FIG. 3-6 FLOW RATE OF AIR (CFM) VS AIR INLET TEMPERATURE
(CASE 3)

FIG. 3-7 PERCENTAGE OF HEAT TO AIR (BASED ON 1 KW HEAT GENERATION RATE OF THE CANISTER) VS AIR INLET TEMPERATURE (CASE 4)

FIG. 3-8 FLOW RATE OF AIR (CFM) VS AIR INLET TEMPERATURE
(CASE 4)

FIG. 3-9 PERCENTAGE OF HEAT TO AIR (BASED ON 0.5 KW HEAT GENERATION RATE OF THE CANISTER) VS AIR INLET TEMPERATURE (CASE 5)

FIG. 3-10 FLOW RATE OF AIR (CFM) VS AIR INLET TEMPERATURE
(CASE 5)

FIG. 3-11 PERCENTAGE OF HEAT TO AIR (BASED ON 0.5 KW HEAT GENERATION RATE OF THE CANISTER) VS AIR INLET TEMPERATURE (CASE 6)

FIG. 3-12 FLOW RATE OF AIR (CFM) VS AIR INLET TEMPERATURE
(CASE 6)

FIG. 3-13 VARIATION OF T_{SALT} AS A FUNCTION OF TIME FOR $T_{AIR} = 60^{\circ}F$
INITIALLY $T_{SALT} = 93^{\circ}F$

FIG. 3-14 VARIATION OF T_{SALT} AS A FUNCTION OF TIME, FOR $T_{\text{AIR}} = 90^\circ\text{F}$
INITIALLY $T_{\text{SALT}} = 93^\circ\text{F}$

FIG. 3-15 VARIATION OF T_{SALT} AS A FUNCTION OF TIME, FOR $T_{\text{AIR}} = 120^{\circ}\text{F}$
INITIALLY $T_{\text{SALT}} = 93^{\circ}\text{F}$

FIG. 3-16 VARIATION OF HEAT TO SALT, Q_w , AS A FUNCTION OF T_{AIR} FOR VARIOUS TIME t

FIG. 3-17 VARIATION OF PERCENTAGE OF HEAT TO THE AIR AS A FUNCTION OF T_{AIR} FOR VARIOUS TIME t

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FIG. 3-18 AVERAGE AIR AND WALL SURFACE TEMPERATURE DISTRIBUTION THROUGHOUT STORAGE ROOMS - 16000 CFM

FIG. 3-19 AVERAGE AIR AND WALL SURFACE TEMPERATURE DISTRIBUTION
THROUGHOUT STORAGE ROOMS - 32,000 CFM

FIG. 3-20 AVERAGE AIR AND WALL SURFACE TEMPERATURE DISTRIBUTION
THROUGHOUT STORAGE ROOMS - 64,000 CFM

FIG. 3-21 COMPARISON OF SAI AND PBQ&D RESULTS - 9000 CFM

Table 3-1 Summary of Results for Cases 1-6

	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6
Configuration	1	1	2	2	2	2
Canister diameter(in)	12	12	16	16	16	16
Expression of h	$0.21 \times \Delta T^{1/3}$	<u>$1/2 \times 0.21 \times \Delta T^{1/3}$</u>	$0.21 \times \Delta T^{1/3}$	$0.21 \times \Delta T^{1/3}$	$0.21 \times \Delta T^{1/3}$	$0.21 \times \Delta T^{1/3}$
Inner surface area of formation	$2\pi \times (\frac{28}{12})' \times 15'$	$2\pi \times (\frac{28}{12})' \times 15'$	$2\pi \times (\frac{31}{12})' \times 15'$	<u>$5-1/2' \times 2 \times 15'$</u>	<u>$5-1/2 \times 2 \times 15'$</u>	<u>$5-1/2' \times 2 \times 15'$</u>
Heat generation rate per canister, q	1 Kw	1 Kw	1 Kw	1 Kw	<u>0.5 Kw</u>	<u>0.5 Kw</u>
Shape factor from A ₃ to A ₄	1	1	1	1	1	<u>0.7</u>
$\Delta T = (T_{salt} - T_{air})$ for 100% heat removal by air (°F)	~13	~23	~10	10~12	7~8	7~8

Table 3-1 Cont'd.

	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6
CFM corresponding to 100% heat removal by air	160~165	160~165	190~195	190~195	~145	~145

Table 3-2 Results for Case 1, Configuration 1 Repository - $T_{salt}=93^{\circ}\text{F}$

T_{air} ($^{\circ}\text{F}$)	ΔT ($^{\circ}\text{F}$)	h_1	h_2	h_3	h_4	hr_1	hr_2	T_1	T_2	T_3	T_4	T_{a1}	T_{a2}	CFM	% by air*
60	33	.754	.506	.542	.550	1.163	1.008	128.7	96.4	85.6	86.4	82.4	68.4	192.3	153.22
70	23	.750	.501	.514	.495	1.200	1.033	134.6	102.6	91.2	89.7	89.1	76.6	179.1	126.39
80	13	.745	.493	.478	.422	1.236	1.057	140.3	108.6	96.5	92.9	95.6	84.8	165.1	101.17
90	3	.740	.484	.433	.301	1.273	1.079	145.8	114.2	101.5	95.7	102.0	92.8	150.2	78.43
100	-7	.733	.468	.364	.298	1.309	1.100	151.5	119.9	106.3	98.2	108.8	101.0	136.0	58.70
110	-17	.732	.459	.261	.419	1.345	1.122	156.5	125.1	110.6	100.8	115.0	108.8	119.6	37.50
120	-27	.723	.442	.245	.488	1.373	1.139	160.8	129.5	114.2	103.3	120.1	115.8	98.9	17.46
130	-37	.720	.438	.333	.527	1.394	1.154	163.8	132.5	117.1	105.3	123.5	121.1	72.8	1.56

$T_{salt} = 93^{\circ}\text{F}$

Remarks: Case 1. Configuration 1 repository

$$\Delta T = T_{salt} - T_{air} (^{\circ}\text{F})$$

h_1, h_2, h_3, h_4 are free convection heat transfer coefficients at surfaces A_1, A_2, A_3, A_4 respectively, $\text{Btu/hr.ft}^2 \text{ } ^{\circ}\text{F}$

hr_1, hr_2 are radiant-heat-transfer coefficients between surfaces A_1, A_2 & A_3, A_4 respectively, $\text{Btu/hr.ft}^2 \text{ } ^{\circ}\text{F}$

T_1, T_2, T_3, T_4 are temperatures at surfaces A_1, A_2, A_3, A_4 respectively, $^{\circ}\text{F}$

T_{a1}, T_{a2} are average air temperatures in the inner and outer air channels respectively, $^{\circ}\text{F}$

* % heat picked up by air is based on per canister heat generation rate $q = 1 \text{ kw} = 3415 \text{ Btu/hr}$

Table 3-3 Results for Case 1, Configuration 1 Repository - $T_{\text{salt}}=113^{\circ}\text{F}$

T_{air} (°F)	ΔT (°F)	h_1	h_2	h_3	h_4	h_{r1}	h_{r2}	T_1	T_2	T_3	T_4	T_{a1}	T_{a2}	CFM	% by air
60	53	.754	.521	.592	.632	1.210	1.070	135.7	104.7	94.5	99.4	89.3	72.0	211.8	209.08
70	43	.750	.516	.570	.594	1.250	1.097	141.9	111.2	100.4	103.0	96.3	80.4	199.9	179.76
80	33	.745	.510	.544	.548	1.290	1.125	147.8	117.5	106.1	106.6	103.1	88.7	187.6	151.51
90	23	.736	.530	.525	.509	1.314	1.151	153.8	123.3	111.4	109.8	109.8	97.0	175.1	125.42
100	13	.736	.497	.478	.422	1.367	1.177	159.2	129.3	116.8	113.1	116.1	105.0	160.9	99.31
110	3	.731	.487	.431	.303	1.404	1.200	164.5	134.8	121.6	115.9	122.3	112.9	146.2	76.53
120	-7	.724	.472	.359	.294	1.442	1.222	170.0	140.4	126.2	118.4	129.0	121.1	132.1	56.71
130	-17	.718	.457	.239	.418	1.479	1.245	175.1	145.5	130.4	121.1	135.2	129.0	116.1	35.22
140	-27	.713	.444	.266	.486	1.509	1.264	179.3	149.7	133.9	123.6	140.3	136.0	95.9	15.35
150	-37	.710	.441	.335	.525	1.530	1.279	182.2	152.8	136.7	125.5	143.5	141.1	70.3	- 0.45

$T_{\text{salt}} = 113^{\circ}\text{F}$

* Same remarks as Table 3-2

Table 3-4 Results for Case 1, Configuration 1 Repository - $T_{\text{salt}}=133^{\circ}\text{F}$

T_{air} ($^{\circ}\text{F}$)	ΔT ($^{\circ}\text{F}$)	h_1	h_2	h_3	h_4	hr_1	hr_2	T_1	T_2	T_3	T_4	T_{a1}	T_{a2}	CFM	% by air
60	73	.754	.534	.634	.695	1.257	1.132	142.6	112.7	103.2	111.9	96.1	75.6	228.8	268.95
70	63	.750	.529	.615	.664	1.299	1.162	148.9	119.4	109.3	115.8	103.3	84.1	217.9	237.83
80	53	.745	.524	.595	.630	1.341	1.192	155.1	126.0	115.3	119.6	110.3	92.5	206.7	207.50
90	43	.741	.519	.572	.592	1.383	1.222	161.1	132.3	121.1	123.3	117.1	100.8	195.1	178.09
100	33	.737	.513	.545	.547	1.425	1.250	167.0	138.5	126.6	126.8	123.8	109.1	183.1	149.75
110	23	.733	.508	.515	.493	1.466	1.279	172.7	144.4	131.9	130.1	130.3	117.2	170.4	122.78
120	13	.728	.500	.477	.421	1.506	1.305	178.1	150.0	137.0	133.3	136.6	125.3	156.9	97.45
130	3	.722	.490	.429	.305	1.544	1.330	183.3	155.4	141.6	136.2	142.7	133.1	142.3	74.66
140	-7	.715	.475	.354	.289	1.584	1.353	188.7	106.8	146.0	138.6	149.2	141.2	128.4	54.79
150	-17	.708	.459	.216	.416	1.622	1.376	193.8	165.9	150.2	140.3	155.4	149.1	112.8	33.19
160	-27	.703	.445	.282	.484	1.654	1.397	197.9	170.0	153.6	143.8	160.4	156.1	93.0	13.30
170	-37	.700	.443	.355	.522	1.676	1.413	200.7	172.9	156.3	145.8	163.6	161.2	68.1	- 2.43

$T_{\text{salt}} = 133^{\circ}\text{F}$

* Same remarks as Table 3-2

Table 3-5 Results for Case 1, Configuration 1 Repository - T_{salt}=93°F

T _{air} (°F)	ΔT (°F)	h ₁	h ₂	h ₃	h ₄	hr ₁	hr ₂	T ₁	T ₂	T ₃	T ₄	T _{a1}	T _{a2}	CFM	by air ⁸
60	33	.420	.315	.310	.300	1.236	1.037	142.7	105.7	92.4	89.9	78.8	66.8	179.2	124.49
70	23	.415	.308	.293	.271	1.272	1.058	148.1	111.4	97.2	92.6	86.2	75.4	167.0	103.36
80	13	.411	.301	.274	.234	1.307	1.078	153.5	116.9	101.7	95.1	93.4	84.0	154.2	83.45
90	3	.407	.292	.250	.178	1.341	1.097	158.6	122.1	105.0	97.3	100.6	92.5	140.7	65.23
100	-7	.401	.281	.216	.129	1.375	1.114	163.6	127.2	108.9	99.3	108.0	101.1	127.3	49.82
110	-17	.395	.270	.167	.211	1.408	1.132	168.5	132.3	113.6	101.3	115.1	109.6	112.6	33.46
120	-27	.390	.258	.094	.254	1.437	1.147	172.6	136.1	116.7	103.3	121.4	117.4	94	17.81
130	-37	.386	.249	.174	.280	1.460	1.160	175.8	139.4	119.3	105.0	126.0	123.8	69	4.22

T_{salt} = 93°F

Remarks: Case 2. Configuration 1 repository

using $h = \frac{1}{2} \times 0.2 \times \Delta T^{1/3}$

* Other remarks same as Table 3-2

Table 3-6 Results for Case 2, Configuration 1 Repository - $T_{\text{salt}}=113^{\circ}\text{F}$

T_{air} ($^{\circ}\text{F}$)	ΔT ($^{\circ}\text{F}$)	h_1	h_2	h_3	h_4	hr_1	hr_2	T_1	T_2	T_3	T_1	T_{a1}	T_{a2}	CFM	$\frac{\text{lb}}{\text{hr}}$ by air
60	53	.423	.328	.338	.343	1.287	1.109	149.6	114.7	102.9	104.5	89.4	69.6	197	168.12
70	43	.419	.322	.325	.323	1.325	1.133	155.3	120.7	107.9	107.4	91.9	78.3	185.9	145.07
80	33	.415	.316	.310	.299	1.363	1.155	160.9	126.4	112.7	110.2	99.3	87.0	174.5	122.84
90	23	.410	.309	.293	.271	1.401	1.178	166.3	132.0	117.3	112.8	106.6	95.6	162.5	101.59
100	3	.406	.301	.273	.234	1.437	1.199	171.5	137.3	121.7	115.3	113.8	104.2	149.9	81.59
110	-7	.401	.293	.248	.179	1.473	1.219	176.5	142.4	125.7	117.6	120.8	112.6	136.5	63.30
120	-17	.395	.282	.213	.124	1.508	1.237	181.4	147.3	129.5	119.5	128.0	121.2	123.1	47.85
130	-27	.389	.269	.139	.211	1.543	1.255	186.1	152.0	133.1	121.6	135.1	129.6	108.7	31.48
140	-37	.384	.257	.118	.253	1.572	1.270	190.2	156.0	136.1	123.5	141.3	137.5	90.3	15.94
150	-47	.380	.249	.181	.278	1.596	1.284	193.3	159.1	138.6	125.2	145.8	143.7	65.4	2.54

$T_{\text{salt}} = 113^{\circ}\text{F}$

* Same remarks as Table 3-5

Table 3-7 Results for Case 2, Configuration 1 Repository - $T_{\text{salt}}=133^{\circ}\text{F}$

T_{air} ($^{\circ}\text{F}$)	ΔT ($^{\circ}\text{F}$)	h_1	h_2	h_3	h_4	hr_1	hr_2	T_1	T_2	T_3	T_4	T_{a1}	T_{a2}	CFM	by air
60	73	.426	.340	.362	.377	1.339	1.183	156.5	123.6	113.2	118.6	89.8	72.3	212.8	215.
70	63	.422	.334	.351	.361	1.380	1.208	162.4	129.7	118.4	121.7	97.5	81.2	202.5	190.61
80	53	.418	.329	.338	.343	1.420	1.233	168.2	135.7	123.4	124.7	105.0	89.9	192	166.72
90	43	.414	.323	.325	.322	1.460	1.258	173.8	141.5	128.3	127.6	112.5	98.6	181.2	143.50
100	33	.410	.316	.310	.299	1.500	1.283	179.2	147.1	132.9	130.3	119.8	107.3	170	121.20
110	23	.405	.309	.292	.271	1.538	1.306	184.5	152.5	137.4	133.0	127.0	115.8	158.2	100.13
120	13	.401	.301	.272	.235	1.577	1.328	189.6	157.7	141.6	135.5	134.1	124.3	145.3	79.79
130	3	.396	.293	.246	.181	1.613	1.349	194.5	162.7	145.5	137.8	141.0	132.7	132.5	61.45
140	-7	.390	.282	.209	.119	1.650	1.367	199.2	167.4	149.1	139.7	148.1	141.2	119.1	46.00
150	-17	.384	.269	.150	.209	1.636	1.386	203.9	172.0	152.6	141.8	155.2	149.7	105.1	29.63
160	-27	.378	.257	.133	.252	1.717	1.403	207.9	175.9	155.5	143.7	161.3	157.5	86.9	14.16
170	-37	.374	.249	.188	.277	1.740	1.417	210.8	178.9	157.9	145.4	165.6	163.6	62.0	.97

$T_{\text{salt}} = 133^{\circ}\text{F}$

* Same remarks as Table 3-5

Table 3-8 Results for Case 3, Configuration 2 Repository - $T_{salt}=93^{\circ}\text{F}$

T_{air} ($^{\circ}\text{F}$)	ΔT ($^{\circ}\text{F}$)	h_1	h_2	h_3	h_4	hr_1	hr_2	T_1	T_2	T_3	T_{a1}	T_{a2}	CFM	% by air*
60	33	.704	.500	.564	.598	1.147	1.037	122.1	97.8	89.3	84.3	69.9	251.1	216.2
70	23	.700	.491	.523	.527	1.172	1.046	126.1	101.9	92.6	89.1	77.2	226.2	161.57
80	13	.694	.478	.472	.432	1.195	1.055	130.1	105.7	95.7	93.8	84.3	199.6	113.55
90	3	.690	.465	.405	.238	1.221	1.063	134.2	109.6	98.7	98.8	91.5	172.3	74.57
100	-7	.683	.446	.294	.374	1.245	1.071	138.1	113.3	101.4	103.8	98.7	142.9	43.52
110	-17	.679	.434	.217	.465	1.258	1.075	104.2	115.3	102.8	106.5	103.9	101.2	15.65

$T_{salt} = 93^{\circ}\text{F}$

Remarks: Case 3. Configuration 2 repository.

$$\Delta T = T_{salt} - T_{air} \quad (^{\circ}\text{F})$$

h_1, h_2, h_3, h_4 are free convection heat transfer coefficients at surfaces A_1, A_2, A_3, A_4 , respectively, $\text{Btu/hr.ft}^2, ^{\circ}\text{F}$.

hr_1, hr_2 , are radiant-heat-transfer coefficients between surfaces A_1, A_2 & A_3 & A_4 respectively, $\text{Btu/hr.ft}^2, ^{\circ}\text{F}$.

T_1, T_2, T_3 , are temperatures at surfaces A_1, A_2, A_3 , respectively, ($^{\circ}\text{F}$)

T_{a1}, T_{a2} are average air temperatures in the inner and outer air channels respectively, $^{\circ}\text{F}$

* % heat picked up by air is based on per canister heat generation rate $q = 1 \text{ KW}$

Table 3-9 Results for Case 3, Configuration 2 Repository - $T_{\text{salt}}=113^{\circ}\text{F}$

T_{air}	ΔT ($^{\circ}\text{F}$)	h_1	h_2	h_3	h_4	hr_1	hr_2	T_1	T_2	T_3	T_{a1}	T_{a2}	CFM	% by air
60	53	.705	.522	.634	.703	1.220	1.135	133.2	110.6	103.1	95.2	75.5	288.9	339.49
70	43	.701	.513	.602	.653	1.247	1.145	137.4	114.7	106.5	100.1	82.9	267.2	274.48
80	33	.697	.504	.566	.596	1.273	1.155	141.4	118.7	109.8	104.8	90.2	244.2	214.27
90	23	.692	.494	.524	.526	1.298	1.166	145.3	122.5	112.9	109.5	97.3	219.8	159.56
100	13	.687	.482	.472	.430	1.322	1.173	149.1	126.2	115.8	114.0	104.4	193.7	111.47
110	3	.682	.468	.402	.238	1.348	1.182	153.0	129.9	118.5	113.8	111.5	166.6	72.43
120	-7	.575	.449	.282	.372	1.372	1.190	156.7	133.4	121.0	123.6	118.6	137.4	41.47
130	-17	.670	.437	.237	.462	1.385	1.193	158.7	135.2	122.2	126.1	123.6	96.2	14.42

$T_{\text{salt}} = 113^{\circ}\text{F}$

* - Same remarks as Table 3-8

Table 3-10 Results for Case 3, Configuration 2 Repository - $T_{\text{salt}}=133^{\circ}\text{F}$

T_{air} ($^{\circ}\text{F}$)	ΔT ($^{\circ}\text{F}$)	h_1	h_2	h_3	h_4	hr_1	hr_2	T_1	T_2	T_3	Ta_1	Ta_2	CFM	% by air
60	73	.707	.542	.693	.783	1.298	1.240	144.5	123.5	117.1	106.1	81.1	321.2	480.45
70	63	.703	.534	.667	.744	1.326	1.251	148.8	127.7	122.6	111.1	88.5	301.7	407.13
80	53	.699	.526	.638	.700	1.354	1.262	152.9	131.8	123.9	116.0	95.9	281.4	337.70
90	43	.694	.517	.605	.651	1.381	1.272	156.9	135.7	127.1	120.7	103.2	260.1	272.58
100	33	.690	.508	.568	.594	1.407	1.282	160.8	139.5	130.2	125.3	110.4	237.7	212.27
110	23	.685	.498	.525	.524	1.452	1.288	164.5	143.1	133.1	129.8	117.5	213.8	157.52
120	13	.680	.486	.471	.429	1.458	1.300	168.1	146.6	135.8	134.2	124.5	188.1	109.44
130	3	.673	.471	.400	.238	1.483	1.309	171.8	150.1	138.3	138.8	131.5	161.3	70.35
140	-7	.666	.452	.269	.371	1.508	1.316	175.4	153.4	140.6	143.5	138.5	132.2	39.54
150	-17	.662	.440	.253	.459	1.520	1.320	177.1	155.1	141.7	145.8	143.4	91.6	13.30

$T_{\text{salt}} = 133^{\circ}\text{F}$

* Same remarks as Table 3-8

Table 3-11 Results for Case 4, Configuration 2 Repository - $T_{salt}=93^{\circ}\text{F}$

T_{air} ($^{\circ}\text{F}$)	ΔT ($^{\circ}\text{F}$)	h_1	h_2	h_3	h_4	hr_1	hr_2	T_1	T_2	T_3	T_{a1}	T_{a2}	CFM	% by air*
60	33	.708	.510	.568	.610	1.137	1.034	120.2	96.3	88.4	81.9	68.5	242.8	194.00
70	23	.703	.498	.527	.538	1.165	1.044	124.3	100.8	92.0	87.5	76.2	220.2	148.44
80	13	.695	.483	.475	.440	1.192	1.054	129.5	105.0	95.4	93.1	83.8	196.3	107.79
90	3	.690	.465	.406	.242	1.220	1.063	134.1	109.6	98.7	98.7	91.5	171.9	74.04
100	-7	.682	.442	.286	.333	1.248	1.072	138.6	113.7	101.6	104.4	99.1	145.8	46.29
110	-17	.675	.422	.259	.434	1.266	1.077	141.6	116.4	103.4	108.3	105.3	109.3	19.85

$T_{salt} = 93^{\circ}\text{F}$.

Remarks: Case 4. Configuration 2 repository

Smaller A_1 for convection heat transfer

$\Delta T = T_{salt} - T_{air}$ ($^{\circ}\text{F}$.)

h_1, h_2, h_3, h_4 are free convection heat transfer coefficients at surfaces A_1, A_2, A_3, A_4 respectively, Btu/hr.ft². $^{\circ}\text{F}$.

hr_1, hr_2 are radiant-heat-transfer coefficients between surfaces A_1, A_2, A_3, A_4 respectively, Btu/hr.ft². $^{\circ}\text{F}$.

T_1, T_2, T_3 are temperatures at surfaces A_1, A_2, A_3 , respectively, $^{\circ}\text{F}$.

T_{a1}, T_{a2} are average air temperatures in the inner and outer air channels respectively, $^{\circ}\text{F}$.

* % heat picked up by air is based on per canister heat generation rate $q = 1 \text{ KW}$.

Table 3-12 Results for Case 4, Configuration 2 Repository - $T_{salt}=113^{\circ}\text{F}$

T_{air} ($^{\circ}\text{F}$)	ΔT ($^{\circ}\text{F}$)	h_1	h_2	h_3	h_4	hr_1	hr_2	T_1	T_2	T_3	T_{a1}	T_{a2}	CFM	Z by air
60	53	.711	.537	.640	.716	1.203	1.131	130.2	108.0	101.6	91.2	73.3	277.2	296.43
70	43	.706	.526	.608	.666	1.233	1.142	135.0	112.7	105.4	96.9	81.1	257.2	242.82
80	33	.701	.514	.571	.608	1.262	1.153	139.6	117.2	108.9	102.4	88.8	236.3	192.76
90	23	.695	.502	.528	.536	1.290	1.163	144.1	121.5	112.3	107.9	96.4	214.1	146.84
100	13	.688	.486	.475	.438	1.318	1.172	148.5	125.7	115.5	113.2	103.9	190.5	105.91
110	3	.682	.469	.402	.241	1.347	1.182	152.9	129.8	118.5	118.7	111.5	166.2	71.91
120	-7	.673	.445	.273	.382	1.375	1.190	157.1	133.8	121.2	124.3	119.0	140.2	44.09
130	-27	.667	.425	.275	.481	1.393	1.195	160.0	136.3	122.8	128.0	125.0	103.8	19.21

$T_{salt} = 113^{\circ}\text{F}$.

* Same remarks as Table 3-11

Table 3-13 Results for Case 4, Configuration 2 Repository - $T_{\text{salt}}=133^{\circ}\text{F}$

T_{air} ($^{\circ}\text{F}$)	ΔT ($^{\circ}\text{F}$)	h_1	h_2	h_3	h_4	h_{r1}	h_{r2}	T_1	T_2	T_3	T_{a1}	T_{a2}	CFM	% by air
60	73	.715	.562	.700	.798	1.272	1.234	140.3	119.9	115.1	100.6	78.1	307.0	413.29
70	63	.710	.552	.673	.758	1.304	1.246	145.2	124.6	118.9	106.4	85.9	288.9	353.16
80	53	.705	.541	.604	.714	1.335	1.258	149.9	129.2	122.6	112.0	93.7	270.2	295.91
90	43	.699	.530	.611	.654	1.365	1.269	154.5	133.7	126.1	117.5	101.4	250.7	241.87
100	33	.694	.518	.573	.606	1.395	1.280	159.0	138.0	129.4	122.9	109.0	230.1	191.44
110	23	.688	.505	.529	.534	1.425	1.290	163.3	142.1	132.5	128.2	116.5	208.3	145.20
120	13	.681	.490	.474	.437	1.453	1.299	167.5	146.1	135.5	133.4	124.0	185.1	104.04
130	3	.674	.472	.398	.241	1.483	1.309	171.7	150.1	138.3	138.8	131.5	160.9	69.84
140	-7	.655	.448	.258	.380	1.511	1.317	175.8	153.8	140.8	144.1	138.9	134.9	42.03
150	-17	.658	.429	.288	.478	1.529	1.322	178.4	156.1	142.2	147.6	144.8	98.7	16.71

$T_{\text{salt}} = 133^{\circ}\text{F}$.

* Same remarks as Table 3-11

Table 3-14 Results for Case 5, Configuration 2 Repository - $T_{\text{salt}}=93^{\circ}\text{F}$

T_{air} (F)	T (°F)	h_1	h_2	h_3	h_4	h_{r1}	h_{r2}	T_1	T_2	T_3	T_{a1}	T_{a2}	CFM	% by air
60	33	.594	.449	.547	.609	1.066	1.029	102.7	89.8	86.4	80.1	68.7	224.5	305.95
70	23	.586	.431	.501	.537	1.091	1.038	107.3	94.1	89.8	85.4	76.3	198.6	216.90
80	13	.580	.412	.441	.440	1.116	1.045	111.6	98.1	93.1	90.6	83.8	169.7	138.57
90	3	.571	.386	.355	.255	1.140	1.056	115.8	102.0	96.0	95.8	91.2	137.3	75.08
100	-7	.559	.348	.028	.368	1.162	1.063	119.6	105.3	98.4	100.8	98.4	98.5	28.36

$T_{\text{salt}} = 93^{\circ}\text{F}$

Remarks: Case 5. Configuration 2 repository

* % heat picked up by air is based on per canister heat generation-rate $q = 0.5\text{kW}$

Other remarks same as Table 3-11 (Case 4)

Table 3-15 Results for Case 5, Configuration 2 Repository - $T_{\text{salt}}=113^{\circ}\text{F}$

T_{air} ($^{\circ}\text{F}$)	ΔT ($^{\circ}\text{F}$)	h_1	h_2	h_3	h_4	hr_1	hr_2	T_1	T_2	T_3	T_{a1}	T_{a2}	CFM	% by air
60	53	.502	.482	.625	.715	1.132	1.125	113.3	101.8	99.8	89.6	73.5	263.8	509.45
70	43	.595	.468	.590	.665	1.160	1.136	118.0	106.3	103.5	95.2	81.2	242.1	403.61
80	33	.588	.452	.551	.607	1.187	1.147	122.6	110.6	106.9	100.6	88.9	218.8	305.12
90	23	.580	.434	.503	.535	1.213	1.156	126.9	114.6	110.2	105.8	96.5	193.3	215.48
100	13	.573	.414	.442	.439	1.238	1.166	131.1	118.5	113.2	110.8	103.9	164.9	136.60
110	3	.563	.388	.352	.254	1.263	1.174	135.1	122.1	115.9	115.8	111.2	132.7	72.79
120	-7	.551	.350	.132	.356	1.285	1.180	138.7	125.3	118.1	120.7	118.3	94	26.38

$T_{\text{salt}} = 113^{\circ}\text{F}$

* Same remarks as Table 3-14

Table 3-16 Results for Case 5, Configuration 2 Repository - $T_{\text{salt}}=133^{\circ}\text{F}$

T_{air} ($^{\circ}\text{F}$)	T ($^{\circ}\text{F}$)	h_1	h_2	h_3	h_4	hr_1	hr_2	T_1	T_2	T_3	T_{a1}	T_{a2}	CFM	$\%$ by air
60	73	.612	.514	.689	.798	1.201	1.229	123.9	113.9	113.5	99.2	78.2	296.5	742.73
70	63	.604	.500	.661	.757	1.231	1.241	128.8	118.5	117.2	104.9	86.1	277.4	623.32
80	53	.597	.486	.629	.713	1.260	1.252	133.5	122.9	120.8	110.4	93.8	257.4	510.03
90	43	.590	.471	.594	.663	1.289	1.263	138.1	127.2	124.2	115.8	101.5	236.2	403.35
100	33	.583	.455	.554	.605	1.317	1.273	142.4	131.3	127.5	121.1	109.1	213.4	304.21
110	23	.574	.437	.505	.534	1.303	1.283	146.6	135.1	130.5	126.1	116.6	188.4	213.93
120	13	.566	.416	.442	.437	1.369	1.292	150.5	138.8	133.3	131.0	124.0	160.3	134.63
130	3	.556	.390	.349	.254	1.394	1.300	154.3	142.4	135.8	135.8	131.2	128.4	70.58
140	-7	.543	.352	.164	.364	1.416	1.307	157.8	145.2	137.7	140.5	138.2	89.7	24.53

$T_{\text{salt}} = 133^{\circ}\text{F}$

* Same remarks as Table 3-14

Table 3-17 Results for Case 6, Configuration 2 Repository - $T_{\text{salt}}=93^{\circ}\text{F}$

T_{air} ($^{\circ}\text{F}$)	ΔT ($^{\circ}\text{F}$)	h_1	h_2	h_3	h_4	hr_1	hr_2	T_1	T_2	T_3	T_{a1}	T_{a2}	CFM	% by air
60	33	.591	.440	.535	.610	1.061	.717	102.0	88.9	85.1	76.7	68.5	222.3	296.83
70	23	.585	.427	.494	.538	1.089	.725	106.9	93.6	89.2	85.2	76.2	197.4	212.74
80	13	.580	.412	.442	.440	1.116	.732	111.6	98.1	93.1	90.6	83.8	169.7	138.65
90	3	.572	.392	.369	.249	1.143	.740	116.2	102.5	96.7	96.1	91.3	139.4	78.60
100	-7	.562	.364	.228	.373	1.168	.747	120.5	106.4	99.9	101.2	98.6	103.3	32.76

$T_{\text{salt}} = 93^{\circ}\text{F}$

Remarks: Case 6. Configuration 2 repository

Shape factor = 0.7

* Other remarks same as Table 3-14 (Case 5)

Table 3-18 Results for Case 6, Configuration 2 Repository - $T_{\text{salt}}=113^{\circ}\text{F}$

T_{air} ($^{\circ}\text{F}$)	ΔT ($^{\circ}\text{F}$)	h_1	h_2	h_3	h_4	hr_1	hr_2	T_1	T_2	T_3	T_{a1}	T_{a2}	CFM	$\frac{1}{8}$ by air
60	53	.597	.467	.606	.717	1.223	.782	112.0	100.0	97.2	82.9	73.2	260.4	489.24
70	43	.591	.456	.575	.667	1.153	.791	117.0	140.9	101.6	94.6	81.0	239.3	389.34
80	33	.585	.444	.539	.608	1.182	.800	121.9	109.7	105.7	100.2	88.7	216.7	296.36
90	23	.579	.430	.497	.536	1.210	.808	126.6	114.2	109.6	105.6	96.4	192.2	211.56
100	13	.573	.415	.443	.439	1.239	.815	131.1	118.5	113.2	110.8	103.9	165.	136.85
110	3	.565	.394	.366	.248	1.266	.823	135.5	122.7	116.6	116.1	111.4	134.8	76.35
120	-7	.555	.366	.212	.371	1.291	.830	139.5	126.4	119.5	121.1	118.5	98.6	30.53

$T_{\text{salt}} = 113^{\circ}\text{F}$

* Same remarks as Table 3-17

Table 3-19 Results for Case 6, Configuration 2 Repository - $T_{\text{salt}}=133^{\circ}\text{F}$

T_{air} ($^{\circ}\text{F}$)	ΔT ($^{\circ}\text{F}$)	h_1	h_2	h_3	h_4	hr_1	hr_2	T_1	T_2	T_3	T_{a1}	T_{a2}	CFM	% by air
60	73	.604	.494	.666	.800	1.187	.852	121.9	111.1	109.6	98.1	77.8	292.3	709.83
70	63	.599	.484	.640	.759	1.219	.861	127.1	116.2	114.0	104.0	85.7	273.7	597.35
80	53	.592	.472	.612	.715	1.250	.871	132.2	121.1	118.3	109.7	93.6	254.2	490.26
90	43	.586	.460	.580	.664	1.281	.880	137.1	125.8	122.4	115.3	101.3	233.5	389.49
100	33	.580	.447	.543	.606	1.311	.889	141.8	130.4	126.3	120.7	109.0	211.4	295.75
110	23	.573	.433	.499	.534	1.341	.897	146.3	134.7	129.9	125.9	115.6	187.3	210.29
120	13	.566	.417	.443	.437	1.369	.904	150.6	138.8	133.4	131.0	124.0	160.5	135.04
130	3	.558	.396	.364	.248	1.397	.912	154.8	142.8	136.6	136.1	131.4	130.5	74.12
140	-7	.547	.368	.193	.369	1.423	.918	158.6	146.3	139.2	140.9	138.4	94.2	28.47

 $T_{\text{salt}} = 133^{\circ}\text{F}$

* Same remarks as Table 3-17

Table 3-20 Heat Flux to Salt, Q_w , as a Function of T_{air} and T_{salt}

$T_{air} = 60^{\circ}F$

$T_{salt} (^{\circ}F)$	93	91	89	87	85	83	81
$Q_w (Btu/hr)$	-3968.25	-3585.94	-3211.46	-2845.05	-2487.05	-2138.64	-1798.13
$T_{salt} (^{\circ}F)$	79	77	75	73	71	70	
$Q_w (Btu/hr)$	-1467.01	-1145.84	- 835.06	- 535.94	- 248.39	- 109.34	

$T_{air} = 90^{\circ}F$

$T_{salt} (^{\circ}F)$	93	94	95	96	97	98	99	100
$Q_w (Btu/hr)$	867.8	758.71	640.76	522.80	395.93	269.06	134.7	.33

$T_{air} = 120^{\circ}F$

T _{salt} (°F)	93	95	97	99	101	103	105	107	
Q _w (Btu/hr)	3415.00	3330.67	3245.46	3038.51	3006.85	2870.51	2731.73	2561.30	
T _{salt} (°F)	109	111	113	115	117	119	121	123	125
Q _w (Btu/hr)	2390.88	2186.36	2015.93	1811.42	1570.38	1365.87	1127.27	888.67	650.07

Table 3-21 Transient Response Time of Adjacent Salt - $T_{air}=60^{\circ}F$

$T_{salt} (^{\circ}F)$	$\Delta T_{salt} (^{\circ}F)$	$\bar{Q}_w$ (Btu/hr)	$\bar{Q}_w/A$ (Btu/hr.ft ²)	$\frac{K \Delta T_{salt}}{a \bar{Q}_w/A}$	$\frac{kt}{a^2}$	t(d)
-	-	-	-	-	-	-
-	-	-	-	-	-	-
-	-	-	-	-	-	-
93 - 85	- 8	- 3219.55	- 13.22	.6793	.585	1.67
93 - 83	- 10	- 3039.40	- 12.48	.8995	1.18	3.36
93 - 81	- 12	- 2862.07	- 11.76	1.1455	2.6	7.41
93 - 79	- 14	- 2687.69	- 11.04	1.4236	5.6	15.95
93 - 77	- 16	- 2516.37	- 10.34	1.7371	12	34.19
93 - 75	- 18	- 2348.24	- 9.64	2.0951	27	76.92
93 - 73	- 20	- 2183.49	- 8.97	2.5030	64	182.34
93 - 72	- 21	- 2102.86	- 8.64	2.7235	101	287.75
93 - 71	- 22	- 2022.23	- 8.31	2.9720	168	478.64
93 - 70	- 23	- 1945.71	- 7.99	3.2315	265	755

$T_{air} = 60^{\circ}F$

Table 3-22 Transient Response Time of Adjacent Salt - $T_{air}=90^{\circ}F$

$T_{salt} (^{\circ}F)$	$\Delta T_{salt} (^{\circ}F)$	$\bar{Q}_w (Btu/hr)$	$\bar{Q}_w/A (Btu/hr.ft.^2)$	$\frac{K \Delta T_{salt}}{a \bar{Q}_w/A}$	$\frac{kt}{a^2}$	$t(d)$
93-95	2	754.55	3.099	.7245	.67	1.91
93-96	3	696.6	2.8611	1.1771	2.85	8.12
93-97	4	636.5	2.6141	1.7177	11.5	32.76
93-98	5	575.24	2.3626	2.3757	49.	139.6
93-99	6	512.31	2.1042	3.2011	265.	755.

$T_{air} = 90^{\circ}F$

Table 3-23 Transient Response Time of Adjacent Salt - $T_{\text{air}}=120^{\circ}\text{F}$

T_{salt} ($^{\circ}\text{F}$)	ΔT_{salt} ($^{\circ}\text{F}$)	$\bar{Q}_w/A$ (Btu/hr.ft 2)	$\frac{K \Delta T_{\text{salt}}}{a \bar{Q}_w/A}$	$\frac{kt}{a^2}$	$t(\text{d})$
93-101	8	13.25	.6778	.58	1.652
93-103	10	13.00	.8636	1.025	2.92
93-105	12	12.75	1.0566	1.95	5.55
93-107	14	12.47	1.2603	3.62	10.31
93-109	16	12.18	1.4747	6.4	18.23
93-111	18	11.86	1.7038	11.2	31.91
93-113	20	11.53	1.9473	19.3	54.99
93-115	22	11.19	2.2071	34	96.87
93-117	24	10.83	2.4878	61.5	175.22
93-119	26	10.46	2.7904	115.	327.64
93-121	28	10.07	3.1214	225.	641.03
93-123	30	9.67	3.4827	470.	1339.05
93-125	32	9.26	3.8194	920.	2621.13

$T_{\text{air}}=120^{\circ}\text{F}$

Table 3-24 Variation in Adjacent Salt Temperature as a Function Elapsed Time After Emplacement

T_{air} (°F)	$t(d)$	$T_{salt}(°F)$	$Q_w/A(Btu/hr.ft^2)$	$Q_w(Btu/hr)$	Percent heat to Air
60	0	93	-15.51	-3968	216.2
90	0	93	3.56	867.8	74.59
120	0	93	14.03	3415.	0
60	10	80	-6.07	-1632.57	148
90	10	96.2	2.04	497.4	85.4
120	10	107	10.52	2561.3	25
60	30	77.5	-5.04	-1226.13	135.9
90	30	96.9	1.63	395.93	88.4
120	30	111	8.98	2186.36	36
60	100	74.5	-2.51	-610.72	117.9
90	100	97.75	1.235	300.78	91.2
120	100	115.5	7.19	1750.5	48.7
60	365 (= 1 yr)	71.4	-1.26	-305.9	108.96
90	365	98.65	0.746	181.7	94.7
120	365	119.5	5.365	1306.22	61.75
60	730 (= 2 yr)	70	-.696	-109.34	103.20
90	730	99	.55	134.7	96.06
120	730	121	4.63	1127.27	66.99
120	1825 (= 5 yrs)	124	3.16	769.37	77.47
120	2190 (= 6 yrs)	124.5	2.915	709.72	79.22

Table 3-25 Convective Heat Removal From Storage Room*

Time After Emplacement (Years)	Flow Rate (10 ³ CFM)	Heat Generation Rate (KW/room)	Room Inlet Temperature (°F)	Room Exhaust Temperature (°F)	Heat Removed by Ventilation (KW/Room) (%)	
5	16	600	60	150	486	81
5	32	600	60	117	606	101
5	64	600	60	93	684	114

* Room Layout - PEQD Study No. 11
10 Year Old Spent Fuel, 60 KW/acre

4.0 CONCLUSIONS

The concept of removing sizable portions of heat from the repository by means of ventilation was investigated in the present analysis in relation to the 25-year retrievable option. Included in the analysis was a study of the heat removal behavior of specialized concrete designs intended for preventing thermal loading by the canisters from interacting directly with the salt. The analysis indicates that free convective heat transfer will be sufficient to remove virtually all the heat generated by the canisters to the storage room air stream within a period of two years after emplacement. The ventilation required to prevent room closure over the 25-year retrievable period was also investigated. An air flow rate of 9000 CFM per room was found to satisfy the heat removal requirements and was used as a basis for determining the overall capacity of the repository ventilation system for removing heat. Because of economic constraints in determining the capacity of the repository ventilation system for heat removal, the selection of the storage room air flow rate was based on providing the minimum ventilation rate that would satisfy the requirements set forth by OWI for 25-year retrievability, i.e., 60% or less of the heat generated by 10-year old spent fuel buried at 60 KW/acre remains in the formation. The minimum ventilation rate was based on SAI computer studies of air stream and formation temperatures for this particular option, and the PBQ&D studies described in this report.

The feasibility of the ventilation concept depends heavily on the ability of the ventilating air stream to absorb sizable portions of the canister generated heat without the requirement of inordinate quantities of ventilation. The results of the analysis have shown that sufficient quantities of heat can be removed from the repository storage rooms with air flows having velocities that are hardly perceivable. Although the

utilization of specially designed concrete sleeves to facilitate the transfer of generated heat to the storage room air stream has been shown to be useful in reducing temperatures in the vicinity of the canister, the results of the ventilation analysis performed by SAI indicate that the storage room air stream will extract almost the same percentage of generated heat from the formation whether or not the free convective aspect of the sleeves is operating. This is fortunate, since it is difficult to predict the free convective performance of the sleeve configuration with a high degree of certainty until a prototype is built and tested.

APPENDIX A

DERIVATION OF EQUATION RELATING BUOYANCY AND AIR FLOW THROUGH THE SLEEVE

In this Appendix a relationship is derived which relates the air flow rate through the sleeve, the buoyant forces on the flow, and the static pressure losses in the air flow channel. In the derivation, the buoyant force on the air flow is taken equal to the difference in weight of the two columns of air and the static pressure drop is related to static pressure loss coefficients and surface friction factors. Two sleeve configurations are considered.

Figure A-1 shows a schematic of the air channel for Configuration 1. Using the dimensions as shown, the values of the pertinent parameters for Figure A-1 are as follows:

$$A_2 = 3.49 \text{ ft.}^2 - \text{cross sectional area of outer air channel}$$

$$P_2 = P_{21} + P_{22} = 20.94 \text{ ft.} - \text{wetted perimeter of outer air channel}$$

$$D_{H2} = 0.67 \text{ ft.} - \text{hydraulic diameter of outer air channel}$$

$$A_1 = 1.40 \text{ ft.} - \text{cross sectional area of inner air channel}$$

$$P_1 = P_{11} + P_{12} = 8.38 \text{ ft.} - \text{wetted perimeter of inner air channel}$$

$$D_{H1} = 0.67 \text{ ft.} - \text{hydraulic diameter of inner air channel.}$$

Assuming an air flow rate of 165 CFM through the sleeve (which corresponds to 100% of the canister generated heat convected to the storage room air),

$$Re_2 = 2.92 \times 10^3 - \text{Reynolds number in outer channel}$$

$$Re_1 = 7.27 \times 10^3 - \text{Reynolds Number in inner channel}$$

Assuming the surface roughness height of the channel, ϵ , to be 0.125 inches, the friction factor for each channel, which is taken from a Moody diagram, is as follows:

$f_2 = 0.054$ - wall surface friction factor for outer channel

$f_1 = 0.052$ - wall surface friction factor for inner channel

For an open cross-sectional area at the base of the concrete sleeve (assuming half of perimeter P_{21} is open channels), the parameters are as follows:

$A_{21} = 1.57 \text{ ft.}^2$ - open cross-sectional area at P_{12}

$C_{21} = 1.13$ contraction loss coefficient at base of sleeve (applied to velocity head in outer channel)

For an open cross-sectional area at the base of the concrete sleeve (assuming half of perimeter P_{12} is open channels), the parameters are as follows:

$A_{12} = 0.87 \text{ ft.}^2$ - open cross-sectional area at P_{12}

$C_{12} = 0.42$ - expansion loss coefficient at base of sleeve (applied to velocity head in inner channel)

$K_{21} = 1.5$ - loss coefficient at base of sleeve due to 90° turn (applied to velocity head in outer channel)

$K_{12} = 1.5$ - loss coefficient at base of sleeve due to 90° turn (applied to velocity head in inner channel)

$K_1 = 1.5$ - loss coefficient due to concrete plug (plug not shown in Figure A-1)

$K_{en} = 0.34$ - loss coefficient at entrance of outer channel

$K_{ex} = 1.0$ - loss coefficient at exit of inner channel

$\frac{f_2 L}{D_{H2}} = 1.612$ - friction loss coefficient in outer channel

$\frac{f_1 L}{D_{H1}} = 1.552$ - friction loss coefficient in inner channel

$$\rho_{a2} = \frac{P}{R(T_{a2}+460)} \text{ lb}_m/\text{ft.}^3 \text{ average density of air in outer channel}$$

$$P = 2117 \text{ lb}_f/\text{ft.}^2 \text{ - static pressure in mine}$$

$$R = 53.3 \text{ lb}_f/\text{ft.}/\text{lb}_m \cdot ^\circ R \text{ - gas constant}$$

$$T_{a2} = ^\circ F \text{ - average air temperature in outer channel}$$

$$\rho_{a1} = \frac{P}{R(T_{a1}+460)} \text{ lb}_m/\text{ft.}^3 \text{ average density of air in inner channel}$$

$$T_{a1} = ^\circ F \text{ - average air temperature in inner channel}$$

$$P_1 = \text{lb}_f/\text{ft.}^2 \text{ - static pressure at top of inner channel}$$

$$P_2 = \text{lb}_f/\text{ft.}^2 \text{ - static pressure at base of inner channel}$$

$$P_3 = \text{lb}_f/\text{ft.}^2 \text{ - static pressure at base of outer channel}$$

$$P_4 = \text{lb}_f/\text{ft.}^2 \text{ - static pressure at top of outer channel}$$

The equations relating the static pressure throughout the channel are as follows:

$$P_4 + \frac{PL}{R(T_{a2}+460)} - (K_{en} + f_2 \frac{L}{D_{H2}} + K_{21} + C_{21}) \left(\frac{CFM}{60 A_2} \right)^2 \frac{\rho}{2g_c} = P_3 \quad (A-1)$$

$$P_1 + \frac{PL}{R(T_{a1}+460)} + (K_{ex} + f_1 \frac{L}{D_{H1}} + K_{12} + K_1 + C_{12}) \left(\frac{CFM}{60 A_1} \right)^2 \frac{\rho}{2g_c} = P_2 \quad (A-2)$$

where

$$\rho = .071 \text{ lb}_m/\text{ft.}^3 \text{ - average density of flow}$$

$$CFM = \text{ft.}^3/\text{min.} \text{ - air flow rate}$$

Since $P_2 \sim P_3$ and $P_1 = P_4$, the following expression remains after one of the above equations is subtracted from the other and the values of the parameters are inserted:

$$(CFM)^2 = 7.58 \times 10^8 \left[\frac{1}{T_{a2}+460} - \frac{1}{T_{a1}+460} \right] \quad (A-3)$$

Figure A-2 shows a schematic of an air channel corresponding to a sleeve in a trench. Using the dimensions as shown, the values of the pertinent parameters for this figure are as follows:

$$A_2 = 18.77 \text{ ft.}^2$$

$$P_2 = P_{21} + P_{22} = 31.47 \text{ ft.}$$

$$D_{H2} = 2.386 \text{ ft.}$$

$$A_1 = 1.745 \text{ ft.}^2$$

$$P_1 = P_{11} + P_{12} = 10.47 \text{ ft.}$$

$$D_{H1} = .67 \text{ ft.}$$

$$\frac{\epsilon}{D_{H2}} = 0.0044$$

$$\frac{\epsilon}{D_{H1}} = 0.016$$

Assuming an air flow rate of 195 CFM through the sleeve (which corresponds to 100% of the canister generated heat convected to the storage room air), the parameters are as follows:

$$Re_2 = 2.2 \times 10^3$$

$$Re_1 = 6.9 \times 10^3$$

$$f_2 = 0.054$$

$$f_1 = 0.052$$

$$C_2 = 65.63 \text{ (applied to the velocity head in the outer channel)}$$

$$C_1 = 0.104 \text{ (applied to the velocity head in the inner channel)}$$

Assuming the entrance loss to the outer channel corresponds to a square edge orifice, the following parameters are obtained:

$$K_{en} = 2.5$$

$$K_{ex} = 1.0$$

$$K_{21} = 1.5$$

$$K_{12} = 1.5$$

$$K_1 = 1.5$$

Substituting these values into Equations A-1 and A-2, and then subtracting the equations results in the following equation:

$$(CFM)^2 = 1.26 \times 10^9 \left[\frac{1}{T_{a2} + 460} - \frac{1}{T_{a1} + 460} \right] \quad (A-4)$$

FIG. A-2 AIR CHANNEL SCHEMATIC – TRENCH

APPENDIX B

CONVECTIVE HEAT TRANSFER COEFFICIENT IN SLEEVE

The expression used for estimating the convective heat transfer coefficient at the surfaces of the sleeve configuration corresponds to that which correlates natural convection of heat from vertical tubes. In the turbulent range where the product of the Grashof number and Prandtl number range between 10^9 and 10^{12} , the following expression is employed:

$$\frac{hL}{K} = 0.13 \left[\left(\frac{L^3 \rho^2}{\mu^2} g \beta \Delta T \right) \left(\frac{C_p \mu}{K} \right) \right]^{1/3} \quad (B-1)$$

where

Nu = Nusselt Number

Gr = Grashof Number

Pr = Prandtl Number

L = effective length of the canister (ft.)

K = thermal conductivity of air (BTU/hr.-ft.-°F)

ρ = density of air (lb_m/ft.³)

g = acceleration of gravity (ft./sec.²)

β = temperature coefficient of volume expansion of air (1/°F)

μ = absolute viscosity (lb_m/ft.-sec.)

ΔT = temperature difference between bulk air and surface (°F)

C_p = specific heat of air (BTU/lb_m-°F)

When computing the product of the Grashof and Prandtl numbers with the properties of air evaluated at 100°F for an effective canister length of 15 feet, a minimum temperature difference of 4.3°F is required between the bulk air and the surface in order for the flow to be turbulent ($Gr \times Pr = 10^9$). Since temperature differences greater than 4.3°F are anticipated (and were confirmed by the analysis) to occur between the air and channel surfaces of the sleeve assembly, Equation B-1 was used to evaluate the

convective heat transfer coefficient at all surfaces of the sleeve assembly.

When evaluating Equation B-1 with the properties of air taken at 100°F, the expression for the heat transfer coefficient reduces to the following:

$$h = 0.21 (\Delta T)^{1/3} \quad (B-2)$$

APPENDIX C
COMPUTATION OF STORAGE ROOM CONVECTIVE HEAT TRANSFER COEFFICIENT

The value of the heat transfer coefficient at the tunnel wall surfaces is dependent on whether the air flow in the tunnel is laminar or turbulent, or if free convection predominates. In this connection, the Reynolds number of the flow in each tunnel segment is evaluated and the appropriate expression for the heat transfer coefficient is applied.

The heat transfer coefficients are calculated in the SES program using the following parameters:

h = convective heat transfer coefficients (BTU/hr.-ft.².-°F)

K = thermal conductivity of air (BTU/hr.-ft.-°F)

T_a = temperature of air (°F)

T_s = wall surface temperature (°F)

Re = Reynolds number

Gr = Grashof number

Pr = Prandtl number

L = axial length over which heat transfer occurs (ft.)

D = diameter of flow passage (hydraulic diameter for cross-section of other than circular) (ft.)

The expression used by the program to evaluate the heat transfer coefficients are as follows:

$$\frac{hD}{K} = 1.86 \left(Re Pr \frac{D}{L} \right)^{1/3} \quad (C-1)$$

for laminar flow;

$$\frac{hD}{K} = 0.022 Pr^{0.6} Re^{0.8} \quad (0.5 \leq Pr \leq 1.0) \quad (C-2)$$

for turbulent flow;

$$h = 0.21 (T_a - T_s)^{1/3} \quad (10^9 \leq Gr \leq 10^{12}) \quad (C-3)$$

when free convection predominates.

The preceeding expressions for the heat transfer coefficient are applied to systems having smooth walls. The coefficient of heat transfer increases with increasing roughness of the tunnel walls, both because of the additional surface area and because of the localized mixing of the gaseous phase at the interface. In order to account for the increase in the turbulent flow heat transfer coefficient due to surface roughness, the following expression is applied:

$$\frac{Nu}{Nu_s} = \left(\frac{f}{f_s} \right)^{0.63} \quad (C-4)$$

where

Nu, Nu_s = Nusselt number for rough and smooth surfaces, respectively
 f, f_s = friction factor for rough and smooth surfaces, respectively.

A wall surface roughness height of 0.25 inches was used in determining the turbulent friction factor, f_s .

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APPENDIX D
EFFECTS OF MOISTURE ON THE DESIGN OF THE MINE VENTILATION SYSTEM

A study was performed to identify potential sources of water in mines and to assess the relative impact on the design of the mine ventilation system. It was established through a literature search that the predominant proportion of moisture gained by the mine ventilating air results from localized evaporation from areas of free water such as pools, water used during mining operations, and leakage through major faults in the strata. The increase in humidity due to water permeation through the strata surface was found, in general, to be insignificant. With respect to the ventilation system and fans in particular, the effects of high humidity do not present any additional problems. With respect to the air filtration system, however, the effects of high humidity and sensible moisture can have detrimental effects on prefilters, HEPA filters and adsorbents.

HEPA filters are highly efficient collectors of solid and liquid particulate matter. Because they can plug rapidly when exposed to high concentrations of liquid, the result can be decreased air flow, increased pressure drop, and possible rupture of the filter pack.

To protect the air filtration system from damage or loss of function due to entrained moisture under normal or upset conditions (when water is introduced for fire protection or when water deluge systems are activated), adequately sized demisters should be provided.