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DESIGN PROCEDURES FOR UNDERGROUND HEAT SINK SYSTEMS

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Walter C. Best and Mounir M. Botros

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This report presents criteria, engineering information and estimation procedures for the disposal of waste heat associated with the generation of power required to supply the needs of hardened defense underground installations. The major emphasis is placed on the temporary disposal of waste heat below ground while the installation is under attack and cannot rely upon aboveground disposal. A series of sample problems are included to illustrate the use of the estimation procedures presented in the report. All of the sample problems are based on the sizing of a heat sink system for an underground nuclear power plant. Under the

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design criteria which were assumed for the sample problems it is shown that the combination ice/water type heat sink concepts provide the most cost effective solutions.

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PREFACE

This report was prepared by John M. Stubstad, Mechanical Engineers, Applied Research Branch, Experimental Engineering Division, U.S. Army Cold Regions Research and Engineering Laboratory; William F. Quinn, Chief, Geotechnical Research Branch, EED, CRREL; Marcus Greenberg, Systems Engineer, Auxiliary Systems Branch, Nuclear Regulatory Commission; and Walter C. Best, Research Nuclear Engineer, and Mounir M. Botros, Mining Engineer, Research and Technology Division, U.S. Army Facilities Engineering Support Agency.

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SECTION 1 - INTRODUCTION

1-01. Purpose and Scope. This report presents criteria, engineering information and procedures for the disposal of waste heat associated with the generation of power required to supply the needs of hardened defense underground installations. The major focus is placed on the temporary disposal of waste heat below ground during the time that an installation is under attack and cannot rely upon aboveground disposal. Such installations may include communications centers, staff headquarters, attack shelters, and other such strategic installations. It is applicable to Division and District Engineers responsible for the location, design, and construction of military underground installations.

1-02. Historical Background. In 1952, the Office, Chief of Engineers, requested the National Bureau of Standards (NBS) to investigate the use of underground reservoirs to absorb the heat dissipated from such equipment as diesel generators, refrigeration and air-conditioning machines and other equipment essential to supporting occupancy in an underground defense installation. The NBS investigation, consisting of several analytical studies followed by experimentation utilizing two field tests, resulted in the development of an empirical equation and some graphs that could be used in estimating, with reasonable accuracy, the amount of heat that could be stored in water and the surrounding rock mass with respect to time. Details of the test and results obtained can be found in National Bureau of Standards Progress Report No. 4740[1] and Report No. 4795[2].

In 1958 the Corps of Engineers assigned to Parsons, Brickenhoff, Hall and McDonald the task of investigating the feasibility of using ice as a heat sink medium. Methods for placing and maintaining ice in a heat sink utilizing an ice-making plant located either above or below ground were considered. It was concluded that the ice-making plant should be located below ground for security reasons. Due to a general lack of technical information, it was recognized that additional data on the behavior of an ice-water mass was needed before ice could be used as a heat sink[3].

To dispose of waste heat from a nuclear power plant located on the Greenland Ice Cap at Camp Century, the Corps of Engineers in 1960 proposed using a heat sink utilizing a hole melted in the ice. The U.S. Army Snow, Ice and Permafrost Research Establishment (SIPRE), using a mathematical analysis, concluded that the scheme was feasible and a sink containing chilled water was formed by melting ice with a portion of the waste heat generated by the PM-2A. The successful construction and operation of this sink are reported in SIPRE Research Report No. 60[4].

In the fall of 1965, while considering the type of heat sink system that should be used in conjunction with an underground nuclear plant,[5,6] the Special Projects Group of OCE requested the U.S. Army Cold Regions Research and Engineering Laboratory (CRREL) to review all heat sink concepts. The CRREL study considered the following heat sink materials: (1) water (particularly chilled), (2) a combination of ice and water, (3) aqueous solutions involving either ethylene glycol or calcium chloride that would depress the freezing point of the solution, and (4) solid materials having high volumetric heat capacities. Although the report concluded that a sink containing ice and water offered the greatest economical potentials, the lack of knowledge about such sinks resulted in the selection of a chilled water sink as the best choice.

In conjunction with the DUCC project in 1967 the Missiles and Protective Structures Branch and the Nuclear Power Division of OCE approached CRREL with a new heat sink concept involving use of a rock tunnel as a condenser[8]. The turbine exhaust steam was to be condensed on the tunnel walls, producing a transient heating of the rock and thus storage of the waste heat. Validation of the concept involved a mathematical study and a field test in an unused portion of a magnetite mine located in Mineville, New York, and owned and operated by the Republic Steel Corporation. During the first few days of the field test warm-up of the rock occurred at a predictable rate when suddenly considerable spalling of the tunnel roof began, retarding the rate of the temperature rise and complicating a comparison between measured and predicted temperatures. However, it was possible to obtain, with reasonable assurance, an overall heat transfer coefficient between an air-steam-vapor mixture and the rock surface. The data also indicated that the mathematical model assumed in making preliminary predictions tended to over-predict the time required to warm the rock surface. The mathematical model, the field test and the problems encountered are documented in several reports[10,11,12,13] published between 1968 and 1971.

To provide guidance in choosing between a centrally located, deeply buried, hardened power plant or remotely located redundant plants the Corps of Engineers Omaha District prepared a report in 1965[15]. The report considered three overpressure levels, four different target configurations and nine weapon patterns, for two basic types of remote plants, diesel and turbine powered generators, and one deeply buried nuclear plant. The report concluded that for the same degree of survivability, one centrally located plant would be more economical to construct than remotely located redundant plants. As a result of this report the U.S. Army Engineer Reactor Group (USAERG) updated a conceptual design for a 1500-kW(e) nuclear plant suitable for such an application[16]. In addition the U.S. Army Engineer Power Group (USAENPG), formerly USAERG, directed CRREL in 1969 to review past heat sink studies to determine the type of heat sink most effective for such an application, giving particular consideration to an ice-water concept.

Their study of the subject indicated that an ice heat sink system would require fewer sinks, resulting in a savings in excavation, especially costly at those depths.

The CRREL study consisted of both a desk study and laboratory scale model experimentation. Details of the dimensional analysis and the scaling of the model parameters are presented in the CRREL report, "Model Ice Heat Sink"[18]. Design of the experiment revealed many problems in simulating prototype operating conditions, e.g. how to introduce the heat load, how to manufacture the ice, and how to maintain it so it will be in readiness at all times. Some of these problems are discussed in a CRREL report[19]. The initial series of experimental tests examined the radial melting of an upright cylindrical solid mass of ice. An annular flow path for the condenser water was selected to provide a stable melting geometry and a large surface area over which melting could occur. In all, seven tests were conducted. The data obtained from these tests validated reasonably well the results predicted by a computer program developed during the desk study. Details of these tests, an analysis of the results, and mention of some of the problems that could be encountered in a full scale prototype are covered in the CRREL report "An Annular Flow Ice-Water Model Heat Sink Study"[20]. This study provided, for the first time, actual engineering data on the flow processes and melt patterns that could occur in an ice-water heat sink and a mathematical method for predicting the relationship between sink outlet temperature, coolant water flow rate and heat rejection rate.

While reviewing results from this study the USAENPG was requested in Oct 1970 by the Navy to review and evaluate their latest heat sink concept for the Sanguine Project[21], an ice-water slurry to be used as the heat sink medium. Although it was recognized that an ice-water slurry will have less latent heat capacity per unit volume than a solid ice cylinder sink, it was thought that as the heated coolant water was introduced at the top of the sink it would tend to stratify and induce a favorable situation by maintaining an essentially constant sink outlet temperature for a relatively long time. A series of experimental tests were conducted by CRREL to assess the influence of ice particle size (surface area) using ice cubes, ice blocks and solid ice cylinders. In all, thirteen tests were conducted. Test results indicated that solid ice cylinder sinks would have a considerably larger ice mass, resulting in a longer period of low temperature operation, increased heat storage capacity, and ultimately a smaller heat sink complex. These experiments were reported in the CRREL report "Experimental Study of Several Heat Sink Concepts"[23].

A study was conducted by Foster-Miller Associates in 1974[24] to (a) conduct a critical review of the prior studies on the ice-water heat sink concepts, (b) extend and improve upon analytical models, (c) assess the available options for flow paths of coolant water within the heat sink, and (d) analyze the interaction between the heat rejection system and the power generation complex. The study concluded that the ice-water concept is feasible and that a coolant water flow path connecting three heat sinks in series would be slightly more efficient than a parallel arrangement. The Foster-Miller study also considered such design features as ice block stability and anchoring, freezing times required to form the ice, and ice cylinder deformation due to elastic and viscous effects.

A major concern raised during the earlier ice-water heat sink studies related to the potential problem in scaling the results of these rather small scale model experiments to a large prototype. Thus another test program was conducted at CRREL involving a larger test model. Results of this test program are reported in the CRREL report "Experimental Scaling Study of an Annular Flow Ice-Water Heat Sink"[25]. Over the range of simulated flow rates tested, it was found that the computer solutions tend to predict on the average coolant water temperatures 1° and 3°F higher than actually measured in the model. It was concluded that the computer predictions could be used as a design basis to select a prototype sink whose actual outlet water temperature would be slightly lower than that predicted. Heat transfer coefficients were correlated with the flow conditions using a Nusselt number based on the height of the ice and a Reynolds number based on an equivalent diameter of the annular space between the ice and the heat sink tank.

1-03 Definitions: The following definitions are provided to clarify certain words and phrases used within this report.

Attack: That period of time during which the installation is subject to bombardment or other hostile action.

Buttoned up: An operational status of the installation in which it is operating in a totally self sufficient manner.

Hardened: A term used to indicate that the item described has been reinforced or designed in such a manner as to preclude damage resulting from a specific intensity of attack. Often used in conjunction with a specific amount of blast induced overpressure that can be withstood without damage.

Heat Sink: That system which is used to absorb the waste heat produced by power generation or other equipment in an underground installation while in the buttoned up mode.

Heat Storage: Term used to describe the ability of the heat sink to absorb waste heat. The use of the word "storage" is not meant to imply that the energy absorbed can at a later date be recovered to produce power.

Installation: An underground facility which may be a command center, communications site, shelter or other vital facility which has been constructed to preclude or reduce the possibility of damage or destruction resulting from attack.

Post Attack: That period of time following attack during which the installation will continue to perform its mission.

Standby: That period of time prior to attack during which the installation will be maintained in a ready condition so that it may be used at a moment's notice. During this period of time the facility is not anticipated to be operating in a self sufficient capacity and may use surface resources for operation.

Thermal Saturation: That condition of the heat sink system wherein the heat sink temperature has increased to the point where it can no longer be used to absorb waste heat from the power generation or other equipment.

Waste Heat: That proportion of the total energy provided to a piece of equipment which does not produce useful work. This term is used to refer to the percentage of energy lost due to the inefficiency of the equipment.

1-04 Nomenclature: The following is a list of symbols used throughout this report. The units indicated are the proper units which should be employed in any equations cited in this report.

a: radius of heat sink chamber, ft

c: specific heat of surrounding rock, Btu/lb°F

c_i: specific heat of ice, Btu/lb°F

c_w: specific heat of water, Btu/lb°F

D_e: equivalent hydraulic diameter, ft

F_a: ice buoyant force, lb

f: friction factor

G: non-dimensional heat storage capacity

g: acceleration of gravity, 32.2 ft/sec^2

h_f: pumping head loss, ft of water

i: subscript referring to ice

k: thermal conductivity of rock, Btu/hr-ft°F

k_i: thermal conductivity of ice, Btu/hr-ft°F

L: length of the ice cylinder, ft

ℓ: length of heat sink reservoir, ft

M_i: mass of ice, lbm

M_T: total mass of ice and water, lbm

M_w: mass of water, lbm

N_{RD}: Reynolds number based on diameter

Q: heat storage capacity, Btu

q_c: cooling load, tons of refrigeration

q_m: maintenance cooling load, tons of refrigeration

q_o: heat rejection rate (constant), Btu/hr

q_p: pumping frictional heat load, Btu/hr

$q_r(T)$: heat rejection rate (function of sink temperature), Btu/hr

q_s: rock cooling load, tons of refrigeration

R: radius, ft

R_e: equivalent radius, ft

R_i: radius of the ice, ft

R_o: nominal radius of the reservoir, ft

T: heat sink temperature, °F

T_∞: ambient temperature of surrounding rock, °F

T_f: final sink temperature, °F

T_o: initial sink temperature, °F

$T_r(R,t)$: temperature of surrounding rock at radius R, time t, °F

T_s: supply water temperature, °F

T_ω: heat sink chamber wall temperature, ^{o}F

t: time, hr

t_c: creep closure time of annulus, sec

t_d: design operating life of the heat sink, hr

t_m: time required to melt the ice, hr

t_p: time required to cool down the reservoir, hr

t_s: length of the stratified flow period, hr

t_t: length of the transition flow period, hr

U_m: mean coolant flow velocity, ft/hr

w: subscript referring to water

W: water flow rate, gpm

y: length along the ice, ft

ΔR: annular gap, ft

ΔT_c: water temperature rise across the condenser, ^{o}F

ρ_i: density of ice, lb/ft^3

ρ_w: density of water, lb/ft^3

λ: latent heat of fusion, Btu/lbm

τ: non-dimensional time

ν: kinematic viscosity of water, ft^2/hr

η: viscosity of ice, lb/ft-sec

SECTION 2 - DESIGN CONSIDERATIONS

2-01 Introduction: This section presents discussions related to the design, construction and operation of an underground heat sink complex. Only those considerations which are unique to the heat sink system are presented; design criteria common to all types of underground facilities are presented in references 15 and 41. Methods for calculating structural loads, column and wall sizes and resistance to attack are presented in references 58, 59, 60, 61 and 62.

2-02 General Considerations. This section presents considerations which are applicable to the majority of underground heat sink systems. Specific considerations for several different types of heat sinks are presented in Section 2-03.

2-02.1 Lined Chambers. The use of internal liners in the tunnels and chambers which compose the heat sink complex will primarily be related to the type of rock at the site, the desired level of hardness of the facility and the anticipated level of attack to which the installation will be subjected.

The use of an internal liner will adversely affect the thermal performance of those heat sinks which depend upon heating of the surrounding rock for heat storage. The liner will reduce the rate at which heat can be conducted into the rock and thus reduce the total heat storage capacity of the sink.

Liners which are in direct contact with the rock surface will have the smallest adverse effect whereas liners separated from the rock surface by an air gap or backfill such as sand will have the greatest. At present no simplified method is available to calculate the influence of liner design on the thermal performance of the heat sink.

2-02.2 Multiple Sinks. The type of rock at the installation site and the anticipated level of overpressure will determine the maximum diameter of the heat sink chamber. This in turn will determine, aside from considerations of redundancy, whether multiple heat sink chambers will be required. If multiple heat sinks are to be employed the minimum distance between the sinks will be related to the type of sink employed as well as the structural characteristics of the rock.

For sinks with heat transfer to the surrounding rock which are maintained at the ambient rock temperature it is necessary to ensure that when they are in operation the temperature gradients induced by adjacent sinks do not overlap to any significant degree. Significant overlapping of the temperature gradients would indicate that both sinks were attempting to reject heat to the same volume of rock and thus the heat storage capacity of the total system would be less than originally predicted. Section 6-04.2 presents a method for calculating the thermal gradient in rock surrounding a cylindrical chamber where the chamber surface is maintained at constant temperature which is different than the ambient rock temperature. To make a first order approximation of the thermal gradient surrounding a sink whose temperature varies with time it is acceptable to assume that this equation can be used with the constant wall temperature set equal to the average temperature of the sink during operation.

While the same requirement that the sinks do not interact thermally during operating also holds true for sub-ambient temperature sinks it is possible that by means of reducing their separation distance to the minimum required to fulfill this requirement the total initial cool down refrigeration load may be reduced. Since the cool down period for the sink will normally be significantly longer than the operating life the gradients resulting from cool down will extend to a greater depth in the surrounding rock than the gradients resulting from operation. Therefore by locating the sinks such that the separation distance is sufficient to prevent thermal interaction during operation, thermal interaction will occur during cool down. The gradient prediction technique presented in Section 6-04.2 should be used to estimate both the cool down and operating gradients. No simplified method is available to determine total refrigeration required when the sinks interact thermally during the cool down period.

2-02.3 Volumetric Changes, Water Reservoir Sinks. Heating of a water reservoir type sink will, for large changes in water temperature, result in a decrease in the water density and thus an increase in the volume required to contain a specific mass of water. The design of the reservoir should include sufficient volume to contain the entire water mass at the final sink temperature.

2-02.4 Volumetric Changes, Ice Reservoir Sinks. The conversion of ice to water during the operation of an ice heat sink will result in a decrease in the total volume required to store the ice and water. Except in situations where the initial percentage of ice is very small, i.e. less than 25% of the total mass of ice and water, and the final sink temperature is very high, on the order of 190°F, a sink which has sufficient volume to contain the initial mass of ice and water will have a sufficient volume throughout the entire operating period.

2-02.5 Water Seepage. As a result of the typical rock fracture that occurs in underground construction consideration must be given to the possible flow of water into or out of the heat sink chamber.

Following excavation in rock it is common for groundwater to flow into the excavated areas. This will require that the installation be designed so that this water can be collected and discharged. When the installation is in the standby mode this water normally would be pumped to the surface and discharged. When the installation is in the buttoned up mode and can no longer discharge this water to the surface an underground reservoir will be required to hold this water. This groundwater reservoir can serve a dual function, however, since waste heat from the power plant can be rejected to it. The use of this water reservoir as a heat sink would allow the normal heat sinks to be reduced in size and thus cost.

If unlined chambers are used in the heat sink complex water may also flow out of the heat sinks into the surrounding rock. This possible loss of water from the heat sinks is an important consideration for the water reservoir sinks since their total heat storage capacity will be reduced if water is lost. Although makeup water can be added to the sinks during the standby mode, once the transition to the buttoned up mode has been made any loss of water will result in reduced capacity and, therefore, a reduced period of operation.

Sealing of the rock surfaces to prevent the flow of water has been given some consideration; however, it is generally acknowledged that most common vapor barrier materials do not have sufficient strength to withstand the imposed hydrostatic pressures that will occur in deep installations. The generally accepted design procedure for deep sites involves the use of an internal liner of steel or concrete with a drainage space between the outer liner surface and the rock surface. This drainage space may be air-filled or, for more support, backfilled with a granular material such as sand. If a backfill is used it is important that good drainage be provided as the presence of water in the backfill will substantially reduce its ability to attenuate shock waves resulting from bombardment.

2-02.6 Water Flow Patterns for Reservoirs. In reservoirs where the predominant reservoir dimension is in the vertical direction it is recommended that the recirculated condenser water be distributed at the top of the reservoir and the coolant water be drawn from the bottom. This flow pattern will assist in the formation of a density stratified flow and ensure that the coldest water in the reservoir is used as source water for cooling of the condensers. Water flow patterns for water reservoirs with the primary dimension in the horizontal direction have not been investigated to any significant degree. However, in general, for this geometry a flow pattern which would tend to provide a high degree of mixing and thus uniform water temperature throughout would be recommended. Water flow patterns for ice reservoirs are discussed in Section 2-03.

2-02.7 Water Quality. One parameter that must be considered in the design of underground water reservoirs is the required quality of the water. Fresh water in contact with rock surfaces for an extended period or with addition of groundwater will exhibit an increase in its dissolved mineral content. Depending upon the local rock strata this water may also become mildly acidic or alkaline in nature. Such changes in the chemical composition of the water can produce corrosion of installed mechanical equipment and ultimately a possible failure of the heat sink system. The proposed site should be analyzed to determine what long term changes may occur in the composition of the water and thus what types of treatment will be required.

2-02.8 Sink Failures. Because when operating in the isolated mode an underground installation is similar to a submarine it is imperative that the facility be designed such that failure of the heat sink containment system will not by itself result in the destruction of the facility. Water reservoirs should be located such that they are on a lower level than the facility or separated from the facility by watertight bulkheads and access doors. Coolant pipes should be located such that breakage of the primary system will not produce flooding of the facility. In addition the primary and backup systems should be designed such that in the event of a complete failure of one system the other may be operated and serviced without difficulty. This may entail the use of separate tunnel networks for the primary and backup systems or providing adequate drainage when they are run in a common tunnel. Using tunnels which are inclined downward away from the installation to the heat sink would assist in containing any possible flooding resulting from pipe failures. Similarly steam pipes should be located so that in the event of failure live steam will not be exhausted into working areas.

2-02.9 Corrosion. Since the heat sink system must remain idle for extended periods but be available for use at a moment's notice all materials used in its construction must be resistant to long term corrosion effects. Valves, flow control equipment, pressure and temperature sensors and so forth must be capable of remaining unused for long periods without suffering adverse effects during systems tests or in buttoned up operation. External surfaces should be similarly protected as humidity levels in non air conditioned tunnels may be high. All parts of the coolant pipe network must be designed so that they may be serviced or replaced without draining any of the water reservoirs. A sufficient quantity of spare parts should be kept on hand so that any item subject to adverse corrosion effects may be replaced immediately.

2-03 Specific Considerations. This section presents specific design considerations for several of the heat sink concepts discussed in Chapter 4.

2-03.1 Tunnel Condenser.

2-03.1a Rock Fall. The introduction of live steam into a rock tunnel can result in considerable thermal distress of the rock surfaces. Temperature induced cracking can result in spalling of the tunnel roof and considerable rockfalls. Unless the falling rock damages the steam distribution piping or blocks the flow of steam or water through the tunnel the occurrence of rock fall in the tunnel will have beneficial effects. As the rock falls from the roof a cooler rock surface will be exposed which in turn will extend the useful life of the heat sink. The rubble piles on the floor of the tunnel will absorb waste heat from the condensed water as it flows to the collection point and thus produce cooler return water. While prediction of the amount of rock fall that will occur is not probable it should be kept in mind that its occurrence will be beneficial.

The use of rock bolts and/or roof braces may reduce or eliminate massive rock falls and should be considered if the tunnel is to remain useable following operation. However, due to the probability of substantial damage resulting from use the tunnel should not be designated as being a primary exit from the installation after completion of mission. At best the tunnel may serve as a secondary escape exit.

Unless absolutely required for structural reasons the tunnel should be unlined. Since the steam manifolds and other installed equipment will have to be protected from falling rock resulting from use, damage resulting from blast induced rock falls or flyrock will have a negligible effect on the performance of the tunnel. Zone 3 or Zone 4 damage as described in reference 59 would not, by itself, be detrimental to the performance of the sink. If a liner is required a steel liner installed as close to the rock surface as possible is recommended. Gaps between the liner and the rock should be filled with materials of high thermal conductivity. Under no circumstances should a liner with a lower thermal conductivity than the surrounding rock be used.

2-03.1b Water Loss. A make up water supply will be required to replace any condensed water which is lost as a result of seepage through the rock. A water reservoir also located underground is the most probable method of supplying any make up water required. Determination of the size of the make up water reservoir will be extremely difficult, however, as this is related to the anticipated rate of water loss. The rate of water loss in turn will be a function of the in situ rock conditions, the method of excavation used to construct the tunnel and the amount of rock fracture that occurs during actual use. It is this third parameter which makes estimation of the water loss difficult as this will require field tests to be performed.

In selecting a location for the tunnel condenser, rock formations which have a high degree of permeability should be avoided. Similarly, rock formations in which the mineral components are not well bonded are not recommended as the possibility of thermally induced rock distress will be higher.

For construction of the tunnel condenser the excavation technique selected should be based on consideration of the amount of rock fracture that will be produced. Efforts should be made to minimize rock fracture and thus reduce the possibility of substantial rock fall or water loss.

2-03.1c Condensate Water Quality. The condensate from the condensing tunnel will require water treatment prior to recirculation to the boiler. Minerals will be dissolved by the hot water as it runs down the sides of the tunnel and these will have to be removed to prevent undesirable buildups of scale in the boiler. Additionally, pH control may also be required, depending upon the type of rock and the materials used in the construction of the steam generator.

2-03.1d Bulkheads. Bulkheads will be required to seal the condensing portion of the tunnel from the remaining portions. The bulkheads must be designed to maintain a positive seal under the anticipated pressure and temperature conditions resulting from operation of the condenser as well as the stresses and displacements resulting from attack.

2-03.2 Chilled Water Sinks.

2-03.2a Initial Cooling. Refrigeration will be required to cool the water to the initial design temperature and to prevent the flux of heat from the surrounding media. This cooling capacity may be provided through the use of a water chilling plant, refrigeration lines installed inside and/or around the reservoir, the addition of ice particles or a combination of those systems.

A system using a water chilling plant would operate as follows. Initially the plant would produce chilled water at or below the design water temperature and begin to fill the reservoir. Since heat will be transferred from the surroundings to the sink some water from the sink would be recirculated back through the plant to maintain the sink temperature. As the reservoir approaches the completely filled point the chilled water plant would gradually increase the amount of recirculated water until the sink is completely filled. Once the sink has been filled the plant would continue to operate using only recirculated water to maintain the correct sink temperature.

A variation on this procedure is to use the water chilling plant to produce the chilled water and an auxiliary refrigeration system using cooling coils mounted around the reservoir to maintain the temperature. The advantage of this type of system is that it would be possible to precool the heat sink chamber and reduce the magnitude of the peak cooling loads.

Another method of cooling the sink initially is through the addition of ice particles to a partially filled sink. Melting of the ice particles will cool the water in the reservoir and at the same time produce more chilled water. Using a heat balance technique the proper amount of ice and warm water needed to produce a filled sink at the desired design temperature can be determined. Cooling of the surrounding rock may be accomplished through the use of additional ice or an auxiliary cooling system as described above.

2-03.2b Maintenance Cooling. Once the sink has attained the desired initial design temperature cooling will be required to prevent heat flux from the surroundings. The amount of cooling capacity required will be high initially but will decrease nonlinearly until it approaches a fairly low constant level after a substantial period of time has elapsed. Methods for predicting the initial and maintenance cooling loads are presented in Chapter 6.

2-03.2c Insulation of the Reservoir. Consideration should be given to the use of insulation to reduce the initial transient cooling loads for those chilled water sinks which do not, to any significant degree, use the surrounding media for heat storage. This type of sink would be found in installations where, because of anticipated blast induced damage, an air gap or loose backfill has been used to shock-isolate the facility from the surrounding rock. Under these conditions the reduced refrigeration costs resulting from insulation may outweigh the cost of additional excavation.

2-03.3 Ice Particle Reservoirs.

2-03-3a Formation and Maintenance. Similar to the procedure used to cool a water sink using ice particles, the initial procedure to be used to form this type of sink involves the addition of water and ice particles to the sink reservoir. However, in this situation it is necessary to add sufficient ice such that when the sink attains the design temperature the correct proportion of ice to total mass has also been attained. Once these two conditions have been satisfied ice would be added to the sink only to replace any ice lost through melting.

One of the problems related to the maintenance of the sink is the determination of the amount of ice that has been melted and therefore must be replaced. Changes in water level, which generally would be a reliable indicator of the amount of ice melted, may be totally misleading as a result of shifts in the ice mass. This problem would be especially pronounced when a very high percentage of ice is employed, as free movement of the overall ice mass will be restricted. In designing this type of sink serious consideration must be given to the use of less than optimum ice density to insure that the design will be physically possible to maintain during standby.

Another problem related to this type of sink is the tendency of the ice particles to coagulate into a large mass. When this occurs mechanical agitation will be required to separate the ice particles. If the ice particles are allowed to coagulate then the overall volume of the ice mass will not decrease significantly as a result of melting. Instead the water level will tend to fall and leave air voids between the ice particles. In this situation it would be impossible to add the required volume of replacement ice without breaking apart the coagulated mass.

2-03.3b Blast Induced Motion. Dynamic motion of the ice mass will occur as a result of passage of blast induced shock waves. Since the ice mass will be free floating the design of the reservoir should include the consideration of possible impact of the ice mass against the walls or top of the reservoir. Inlet and outlet pipes for the reservoir should be designed such that they are protected from or can withstand impact from the ice.

Fracture of the ice resulting from dynamic motion will not adversely affect the thermal performance of this type of ice heat sink. However, the coolant piping network should be designed such that damage to the circulating pumps will not occur if small ice particles are drawn from the sink. Otherwise screens over the outlet pipes will be required to prevent ice chips from being drawn out of the reservoir.

2-03.4 Solid Ice Heat Sinks.

2-03-4a Formation. There are two basic techniques for creating a solid ice heat sink: direct freezing of a water filled reservoir or the freezing of a reservoir filled with ice particles and water. The decision on which system is to be employed should be based on the time allowed to freeze the sink, allowable locations for the cooling coils and equipment costs.

The amount of time that will be allowed to initially freeze the sink will have the greatest impact on the type of freezing system selected. Relatively short freezing periods will require large capacity refrigeration systems operating at very low temperatures. Longer freezing periods will allow the use of lower capacity systems operating at higher temperatures.

Independent of the type of freezing system employed, the cooling load required to freeze the water will exhibit an exponentially decreasing trend. The cooling load will be a maximum initially as ice is first formed. Due to the low thermal conductivity of the ice the rate of heat conduction will rapidly decrease as the thickness of the ice layer increases. If the rate of freezing is not to be reduced substantially, increasingly lower refrigerant temperatures will be required as the thickness of the ice layer increases.

As illustrated by the above discussion the crucial parameter related to the freezing process is the length of the ice conduction path between the freezing front and the cooling coil. Reservoirs in which the length of this conduction path is a maximum, such as when cooling coils are located only along the circumference of the reservoir, will have the longest freezing period. If the circumferential cooling system is augmented by cooling lines within the reservoir the length of the cooling period may be substantially reduced. However, if this system were used the cooling lines within the sink would have to be drained and sealed following freezeup to prevent contamination of the water in the sink in the event of a pipe rupture resulting from blast induced motion of the ice.

The length of the freezing period may also be reduced through the addition of ice. It has been found that the freezing of a reservoir composed of ice particles and water can be predicted using relationships originally derived to predict the freezing of a water filled reservoir. This technique, which is presented in Section 6-11, is based on converting the ice particle mass into an equivalent solid ice mass which is then employed as the initial condition in the equation used to predict the freezing of the water reservoir. A secondary benefit of the use of ice particles is that it will substantially reduce the initial peak refrigeration load and thus allow the use of a lower capacity refrigeration system.

One freezing technique which should be considered for small ice reservoirs is the use of closely packed low temperature ice blocks. If low temperature ice blocks, i.e. $0^{\circ}F$ and below, are stacked in the reservoir to yield a high density, then as chilled water is added to the reservoir the water will freeze as a result of the cooling capacity of the ice blocks. However, for this system to work the blocks must be placed to yield a minimum of unfilled space. Generally this would require the placement of several layers of ice blocks followed by the addition of water. The amount of water added would be restricted to prevent floating of the ice blocks. Once this added water is frozen several additional layers of ice would be added and the procedure repeated. This process would continue until the entire sink was filled. Although this procedure would in some respects simplify the problem of freezing the reservoir the logistics of making and tempering the ice to the correct temperature and then placing it into the reservoir is feasible only for relatively small systems.

2-03.4b Maintenance. Once the reservoir has been frozen cooling will be required to prevent the flux of heat from the surroundings. The cooling requirements for this will in general be an order of magnitude less than the heat sink freezing loads.

If an annular flow type heat sink is to be used a heating capability will also be required to melt the initial annulus and prevent it from refreezing. These requirements are discussed in Section 2-03.5a.

2-03.4c Freezing of Surrounding Rock. A result of the low temperature refrigeration that will be required to freeze the ice sink will be the freezing of the nearby surrounding rock. Like the other types of reservoir heat sinks this results in increasing the heat storage capacity of the rock and therefore increases the total heat storage capacity per unit volume excavated for the overall heat sink complex.

However, the formation of ice in any fissures or cracks in the rock surface will produce other effects. The combination of the low temperatures involved and the formation of ice within the cracks will result in the growth of any cracks in the rock already and possibly produce new cracks. This phenomenon should be considered when analyzing the stability of the excavation under shock loading.

A secondary effect of this phenomenon will be its influence on the flow of water into or out of the reservoir. The formation of ice in the cracks will prevent the flow of groundwater into the complex. However, once the rock thaws during use the rate of water loss out of the reservoir, if it is unlined, may be greater than it was prior to freezing because of the ice induced cracking. Thus possible ice induced distress of the surrounding rock must be thoroughly investigated before the decision to use an unlined solid ice type heat sink can be made.

2-03.4d Stresses Induced During Freezing. Freezing of the reservoir must be accomplished in a manner which will preclude the possibility of trapping liquid water within the ice. The entrapment of water will result in the production of extremely high levels of stress which could lead to failure of the reservoir.

The best method of preventing the entrapment of water is to insure that during the freezing period a free path is maintained to the surface. Vertically installed electric heating cables located at the center of the reservoir can in most situations maintain such a path. Mechanical agitation of the water, such as that produced by an air stream along the centerline of the reservoir, may suffice while at the same time reducing the heat load on the cooling system. It is also recommended that for large reservoirs freezing should commence at the outside lower surfaces to insure that the freezing direction will be upward as well as radial.

For example, if the sink were to be frozen using multiple circumferential cooling coils, initially only those coils at the bottom of the reservoir would be used. Once an ice annulus formed at this location the cooling coils directly above this first set would be activated.

This procedure would be repeated until all coils were activated. In addition the coldest refrigerant should be circulated to the lowest coils first with the warm refrigerant being returned to the compressor after flowing through the uppermost coils. Operation of the cooling system in this manner will produce more uniform loading of the refrigeration plant and help to prevent the formation of an ice cover prior to completion of freezing.

2-03.4e Stresses During Heating. Once the reservoir has been completely frozen a large thermal gradient will exist. Generally the center of the reservoir will be at 32°F while the outer edge may be many degrees below zero. Since maintenance cooling will only be to maintain the ice the sink will gradually equilibrate at a temperature near the ice point. As the outer edges of the ice increase in temperature they will attempt to expand in volume. Since this ice will be confined by the outer reservoir walls and the ice at the center of the reservoir significant stresses will be produced. These stresses will be considerably greater than the nominal unconfined strength of the ice and will result in ice fractures. In addition they can cause fracture of the reservoir walls, whether or not a containment vessel is used and should be included in the design analysis.

2-03.4f Corrosion Protection. Surface coatings used to protect or seal the surfaces of the reservoir must be able to withstand the large temperature variations that will occur without failure. Differential expansion or contraction of the coating and the reservoir walls can produce separation of the coating from the surface and loss of the desired protection. In addition portions of the separated coating may, when the sink is in operation, be drawn into the coolant pipe network and plug control valves, heat exchanger tubes or pressure gages and result in a system failure.

Any surface coating selected for use in this type of heat sink should have a thermal expansion coefficient close to that of the wall material, retain a reasonable degree of ductility at the anticipated lowest temperature during the freezing mode and maintain a strong bond to the wall material through the entire operating temperature range.

2-03.4g Melting Geometries. Three possible melting geometries for a solid ice heat sink have been analytically investigated to date. All three have been developed for sinks in which the ice mass is in an upright cylindrical form although other forms are possible.

The three techniques are 1) the use of multiple vertical flow paths through the ice, 2) melting of the ice along the upper surface only and 3) annular melting of the cylinder. Only the last of these three melting geometries has been examined experimentally. Special considerations related to the latter two geometries are presented in Sections 2-03.5 and 2-03.6.

2-03.5 Special Requirements for an Annular Melting Ice Reservoir.

2-03.5a Creation and Maintenance of the Water Annulus. Once the ice cylinder has been frozen heating will be required to melt the initial annulus and bring the system to an operationally ready condition. If multiple ice reservoirs are to be employed it will not be necessary to melt an annulus for each one. One primary reservoir and one standby should be kept ready for use at all times. The other sinks may have the annulus melted using waste heat from the power plant once the facility is buttoned up.

Melting of the initial annulus can be accomplished by circulating hot refrigerant through the circumferential cooling coils originally employed to freeze the sink. By alternating the flow of hot refrigerant and cold refrigerant a thermal balance may be employed to maintain the annulus.

If this procedure is not possible then an alternative method which can be used is to allow the melted annulus to slowly increase. As the annulus for this sink begins to exceed the desired value an annulus on another sink would be created and the first sink would be refrozen. This procedure would be repeated continually during the standby mode.

Although this second procedure does not rely on trying to maintain a thermal balance over a large surface area and thus would be operationally simpler, the overall size of the reservoirs would have to be larger to accommodate the potential loss of heat storage capacity that would result from maintaining this additional sink in the ready condition while the first is refrozen.

2-03.5b Minimum Annulus Size. The minimum thickness of the water annulus that must be maintained during standby is related to the allowable head pressure loss during operation. The head pressure loss for annular flow is given by equations 2-01a or 2-01b.

$$h_f = \frac{3.91\text{x}10^{-9} fLW^2}{(\Delta R)^3(2R_o - \Delta R)^2} \qquad \text{(2-01a)}$$

or

$$h_f = \frac{9.78\text{x}10^{-10} fLW^2}{R_o^2(\Delta R)^3} \quad \text{for } R_o >> \Delta R \qquad \text{(2-01b)}$$

where h_f = head loss, ft of water

f = friction factor

L = length of the annular passage, ft

W = flowrate, gpm

ΔR = thickness of the annular passage, ft

R_o = outside radius of the annular passage, ft

The friction factor, f, is related to the Reynolds number which can be calculated using either equation (2-02a) or (2-02b).

$$N_{RD} = \frac{2.55W}{\nu(2R_o - \Delta R)} \qquad (2\text{-}02a)$$

or

$$N_{RD} = \frac{1.28W}{\nu R_o} \quad \text{for} \quad R_o >> \Delta R \qquad (2\text{-}02b)$$

where N_{RD} = Reynolds number

ν = kinematic viscosity of water, ft^2/hr

For Reynolds numbers less than $2x10^3$ the friction factor should be calculated using equation (2-03).

$$f = \frac{64}{N_{RD}} \quad \text{for} \quad N_{RD} < 2 \times 10^3 \qquad (2\text{-}03)$$

For Reynolds numbers greater than $2x10^3$ the friction factor can be found using a standard Moody chart for internal pipe flow.

2-03.5c Ice Creep. Once the annulus has been melted creep of the ice cylinder may occur, resulting in closure of the annulus. A first order approximation of the rate of closure may be made using equation (2-04a).

$$\frac{d\Delta R}{dt} = \frac{1}{4\eta}\rho_i g(L-y)R \qquad (2\text{-}04a)$$

where

$\frac{d\Delta R}{dt}$ = rate of closure of the annulus, ft/sec

η = viscosity of ice, lb/ft-sec

ρ_i = density of ice, lb/ft^3

g = acceleration of gravity, 32.2 ft/sec^2

L = total length of the ice cylinder, ft

y = distance from the bottom of the cylinder, ft

R = nominal radius of the ice cylinder, ft

This relationship does not include the effects of the hydrostatic pressure from the water in the annulus and friction between the lower ice surface and bottom of the reservoir. It is applicable to reservoirs where the annular passageway is drained during standby.

The viscosity of ice will vary as a result of stress level, temperature and grain size. For large grained columnar ice at a temperature of 25°F and stresses of 14 to 56 psi the viscosity will generally be in the range

$$7 \times 10^{11} \text{ lb/ft-sec} \leq \eta \leq 5 \times 10^{12} \text{ lb/ft-sec} \qquad (2\text{-}04b)$$

The time required to completely close the annular space at any position y is given by equation (2-04c).

$$t_c = \frac{\frac{d\Delta R}{dt}}{\Delta R} \qquad (2\text{-}04c)$$

where ΔR = size of the ice annulus, ft

t_c = time required to close the annulus, sec.

2-03.5d. Ice Cylinder Stability, Hydrostatic. Introduction of the heated water at the top of the reservoir and removal of the cooled water at the bottom will produce nonuniform radial melting of the ice cylinder. The cylinder will melt most rapidly at the top and least at the bottom. The shape of the cylinder will gradually approach that of a truncated cone with the smaller diameter at the top. This shape will be hydrostatically unstable and will require that the ice be restrained by an anchoring system to prevent damage to the reservoir which would result if the ice "flipped" like an iceberg. This load will be a maximum initially and decrease as melting progresses. The maximum hydrostatic load that the anchor system must withstand is given by equation (2-05).

$$F_a = (\rho_w - \rho_i)\pi R_o^2 L \tag{2-05}$$

where F_a = ice buoyant force, lb

ρ_w = density of water, lb/ft^3

ρ_i = density of ice, lb/ft^3

R_o = nominal ice cylinder radius, ft

L = length of the ice cylinder, ft.

It is recommended that the anchor system be constructed using vertical cables and cross ties to provide some flexibility. In addition to non-uniform melting with respect to height, the melting rate will also vary along the circumference as a result of non-uniform water distribution, and this will produce torsional and bending loads on the anchor systems. The cable system described above will provide the necessary rigidity in the vertical direction while maintaining flexibility with respect to torsion of the cylinder.

2-03.5e Ice Cylinder Stability, Shock Loading. Dynamic motion of the reservoir resulting from blast induced shock waves will induce dynamic motion of the ice cylinder. The anchor system must be designed to withstand any imposed dynamic loads without failure. Dynamically induced ice fracture should also be considered. Should ice fracture occur, ice particles will be accelerated toward the top of the reservoir due to their buoyancy and may produce structural damage resulting from impact. If necessary, the reservoir should be mounted on a shock isolation system to reduce the effect of the anticipated blast induced dynamic loads.

2-03.5f Coolant Water Flow Patterns. The inlet manifold should be designed to provide a uniform flow of water in the annulus. The use of either a single central inlet or a small number of discrete inlets is not recommended as this will produce a non-uniform melting pattern. If possible inlet water should be distributed evenly around the top of the cylinder using a header with multiple spray heads.

Since the rate of melting at the bottom of the cylinder will be much lower than that at the top a single central outlet may be used, provided the ice cylinder is to "float" at all times. If the cylinder is to remain in contact with the bottom of the reservoir during the initial hours of use outlets should be located along the periphery of the bottom of the tank. However, since eventually melting at the bottom of the ice will occur a second series of outlets located towards the center at the bottom of the reservoir should also be available. These outlets would be used once the ice surface had melted past them and insure that the coolant water will flow past the ice surface and not be

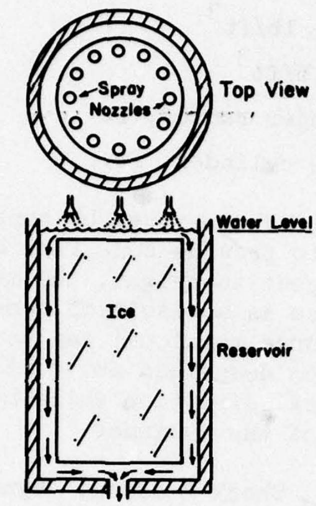

Figure 2-1. Preferred flow pattern for a "floating" ice cylinder.

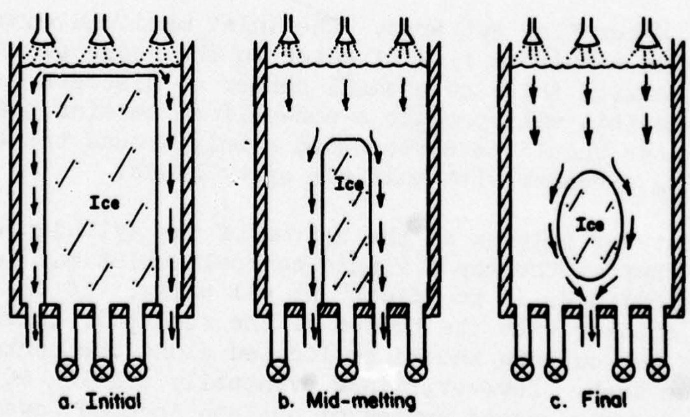

Figure 2-2. Preferred flow pattern for a "seated" ice cylinder.

drawn away to the corner of the reservoir. The preferred flow patterns for a "floating" ice cylinder and a "seated" ice cylinder are illustrated in Figures 2-1 and 2-2.

2-03.6 Special Requirements for Top Melting Solid Ice Reservoirs.

2-03.6a Water Flow Pattern. To eliminate the problems associated with maintaining a water annulus around the ice cylinder during standby an ice heat sink in which melting is confined to the upper surface only may be employed. This will eliminate the requirement of creating and maintaining a water annulus.

In comparison to an annular flow type heat sink, a sink in which melting occurs only along the upper surface will have a lower maximum rate of heat absorption due to its inherently smaller melting surface area. In general this will not be a major limitation because of the relatively high rate at which ice can absorb waste heat, but it can be a factor when using relatively small sinks in conjunction with short duration high thermal loads.

For top melting sinks with a cylindrical geometry the hot inlet water should be introduced at the center, the cooled water drawn out at the periphery. A layer of water covering the upper ice surface should be used to insure that the heated water is not drawn off before the waste heat can be transferred to the ice.

The introduction of the heated water at the center of the ice will result in the formation of a depresssion at this location due to the local higher melting rate. An extendable inlet header which can be lowered into the sink to allow the water discharge point to remain a constant distance from the receding ice surface should be employed. Similarly the return header should also be extendable so that it can remain in close proximity to the ice surface as melting progresses. This type of system is illustrated in Figure 2-3.

Once all the ice has been melted it is recommended that a conventional reservoir flow pattern, i.e. heated water introduced at the top of the reservoir and cool water drawn from the bottom, be employed. During this mode of operation the inlet water should be evenly distributed across the upper surface to produce uniform mixing in the reservoir.

2-04 Operational Considerations. This section presents discussions related to the preparation, maintenance and operation of short term underground heat sink systems.

2-04.1 Preparation. During preparation the heat sink system would be brought to an operational, ready condition. For an ambient temperature

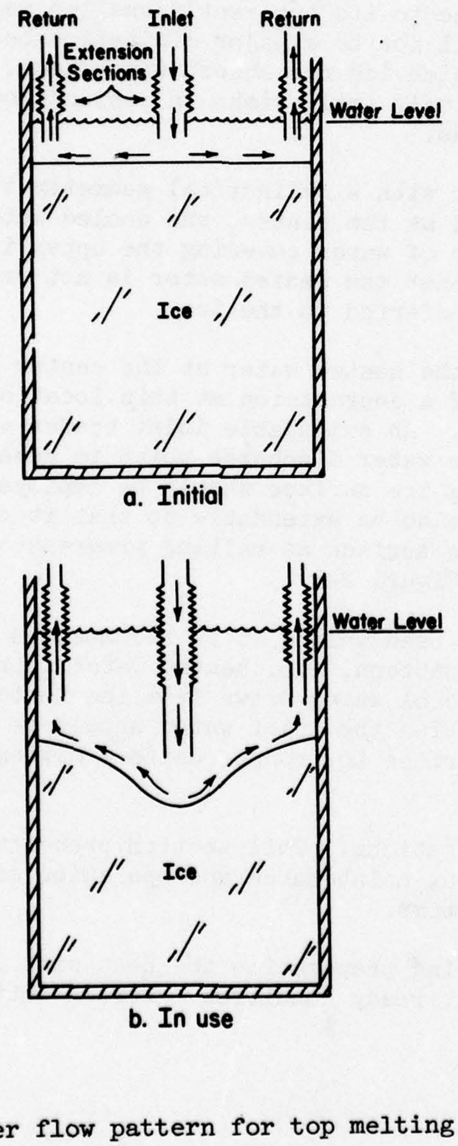

Figure 2-3. Water flow pattern for top melting ice reservoir.

water reservoir this would entail filling of the reservoir with water. For a solid ice heat sink the preparation period would include the filling of the reservoir with water and its subsequent freezing. Similarly for any other sub-ambient temperature heat sink the preparation period would include the time required to attain the desired initial sink temperature.

Normally the ambient temperature heat sink system will not require any substantial preparation to obtain a standby ready condition. Ambient temperature water reservoirs will require only filling and possibly some water treatment to be ready. Other types of ambient sinks, such as the tunnel condenser and the earth heat sink systems, will require little to no post construction preparation before being ready to use.

On the other hand the sub-ambient temperature sinks will require post construction refrigeration before they attain full capacity. The amount of time allowed to initially cool or freeze the heat sink system will determine the total refrigeration capacity required and to a lesser extent which type of cooling systems can be used to accomplish this. For example, circumferential freezing of a large ice type reservoir might require one to two years to complete depending upon the temperature of the refrigerant.

2-04.2 Standby. During standby the heat sink system will be maintained in a condition such that it will be able to absorb the total waste heat load from the power plant within a few minutes following activation. In general the amount of time allowed to convert from standby to operation will be the minimum time required to fill all the pipes and start the circulating pumps.

Solid heat sinks, such as the top melting ice heat sink, will require an auxiliary water reservoir to hold the water required to fill the circulating system at the time of button up. Once the entire system has been filled and is in operation any additional water required can be drawn from the melting ice. An auxiliary water reservoir which is used in this manner during standby can also serve a useful function during operation. After this reservoir has been drained to fill the circulating system any groundwater seepage that collects in the installation may be stored in this reservoir. In addition it can be used as an overflow reservoir to accommodate the increase in volume of water in the heat sink reservoir as a result of increasing temperature. Reuse of the auxiliary water reservoir in this manner would eliminate the requirement that the heat sink reservoir be oversized to allow for volumetric expansion of the water.

2-04.3 System Testing. At periodic intervals the entire power plant and heat sink complex should be brought to full power and operated. This will allow for training of the staff and check out testing of the power system.

In the design of the heat sink complex the total heat storage capacity of the system should be such that at the completion of this type of testing the heat storage capacity of the heat sink system will still be greater than the anticipated requirements for the buttoned mode. This will ensure that even immediately after system testing the installation will still be able to operate at full power for the entire mission should it be necessary.

After system testing of ambient temperature sinks cooling may be required to return the sink to its original maximum capacity. Cooling will be required if the natural rate of cooling, i.e. the rate of heat conduction to the surroundings, is not sufficient to return the sink to its maximum capacity before the next system test is scheduled. Otherwise the sink may be allowed to cool naturally.

An alternative method of restoring an ambient temperature water sink to its presystem testing temperature is to gradually replace the sink water with cooler water from the surface. Provided the surface water is available in sufficient quantity and at a temperature which is below the normal ambient temperature of the sink this method would eliminate the requirement to install refrigeration equipment.

Sub-ambient temperature sinks, such as ice reservoirs, will require refrigeration after system testing to return them to their original maximum capacity. Cooling may be accomplished using the refrigeration system originally employed to freeze the sinks. Cooling loads during refreezing of the sinks should be well below those encountered originally.

In refreezing the annulus of an annular melting sink care should be taken to ensure that liquid water is not trapped by an ice cover. It is recommended that during freezing of the annulus the coldest refrigerant be circulated to the bottom of the sink. This will help induce an upward as well as radial freezing pattern. In addition it is recommended that initially the refrigerant be circulated only to the lowermost cooling coils. Once the lowermost portion of the annulus has been refrozen additional cooling coils above this section can be activated. Using the cooling coils in this manner will assist in preventing the entrapment of liquid water and the high stresses associated with it.

2-04.4 Operation. In heat sink systems in which there is only one actual heat sink the operation of the system is straightforward. The sink is used until the maximum allowable sink temperature has been attained and then the power system must be shut down.

When the heat sink system is composed of several actual heat sinks there are several methods by which they can be used. All the sinks may be used at the same time with the waste heat load divided equally among them. In another method each sink can be used individually so that when one sink has attained its maximum temperature the waste heat load is transferred to a different sink. A variation of this technique is the use of sinks in pairs with the waste heat load divided between two sinks; once they attain their maximum temperature another pair is employed. The basic advantage of using the sinks individually or in pairs is that once the load has been switched the heated sinks will begin to cool again due to heat conduction to the surroundings. Depending upon the final sink temperature, the ambient rock temperature and the amount of time that the other sinks are in use, the sinks used initially may cool sufficiently that they can be used again for a short period of time. Although the length of this secondary period of use will be significantly shorter than the initial period this mode of operation may provide some additional heat storage capacity.

A final mode of operation that should be considered involves multiple switching between several sinks. Initially the heat load would be rejected to one of the sinks until it had obtained an intermediate temperature below its final saturation temperature. At this time the waste heat load would be switched to another sink which would be operated until it also attained this intermediate temperature. One by one each of the remaining sinks would be used in this manner. After all the sinks attained this intermediate temperature the entire process would be repeated, raising each sink to a higher intermediate temperature. In the final step of this process each sink would be used until it had attained its final thermal saturation temperature.

This mode of operation can result in slightly improved overall system efficiency if the power plant waste heat load is a strong function of the heat sink temperature. The procedure of switching from one sink to another can, overall, provide a slightly lower average sink temperature and thus higher power plant efficiency.

2-04.5 Post Attack. For a specified period following attack the installation will remain in the buttoned up mode and continue to perform its mission. It will be during this period that the heat sink system approaches thermal saturation. At the onset of thermal saturation of the heat sinks the power plant and heat sink systems will have to be shut down and the installation personnel will have to return to the surface.

SECTION 3 - WASTE HEAT DISPOSAL REQUIREMENTS

3-01 Waste Heat Sources. The operation of equipment such as power generating units, air conditioning and humidity control systems, electric instruments and even lighting fixtures results in the production of waste heat. The magnitude of the waste heat load these items produce will depend upon their efficiency of operation, their imposed loads and the operating environment.

These last two variables, namely the imposed loading and the operating environment, are influenced by the overall status of the installation. For example, during standby sufficient quantities of cool fresh air may be available from the surface so as to make operation of air conditioning systems unnecessary.

The operating status of the power generation units will depend not only upon the status of the installation but also the type of power units installed. Diesel generators and gas turbine systems, which are capable of rapid starting and can assume full load almost immediately thereafter, would probably be completely shut down during standby. In contrast a nuclear reactor type system would have to be operated at least at a partial load during standby to assure that it could assume the full load at the time of button up. In fact due to its minimal refueling requirements the nuclear reactor might be employed to provide power for the facility in all modes of operation. The selection of this last alternative will depend upon the cost and availability of commercial power, strategic importance of the facility and the power requirements of the electronic equipment installed.

In the following paragraphs a brief synopsis of the impact the different status situations for the installation will have on the waste heat loads is presented. For these discussions it is assumed that commercial power is available and used during standby operation.

3-02. Standby. Installations with diesel or gas turbine generating systems would rely on commercial power completely during this period. Nuclear reactor type systems would operate at partial load with the remainder to be supplied by commercial sources. All other systems, i.e. environmental control, electronics, etc., would be operational. All waste heat produced in the facility during this period would be rejected to an external heat sink medium, the most probable being either surface air or surface water.

3-03. Attack. When in the attack mode the installation will be operating in an isolated condition with all unnecessary connections to the surface severed. The power generation system will be on line supplying all the power requirements of the facility. All waste heat produced within the facility will be rejected to the hardened heat sink system.

3-04. Post Attack. During the post attack period the installation will continue to perform its mission for a specific period of time. The installation will remain in the isolated mode with all power supplied by the hardened power system and all waste heat rejected to the hardened heat sink system.

SECTION 4 - CLASSIFICATION OF HEAT SINKS

4-01. General Categories of Heat Sinks. There are two general categories of heat sinks, the unlimited heat storage capacity type and the finite heat storage capacity type. Sinks which belong to the unlimited heat storage capacity category are able to continuously absorb waste heat independent of their total time in operation. The finite capacity type sinks, on the other hand, have a specific total heat storage capacity that can not be exceeded. In addition during operation they generally experience a change in operating temperature which may affect the rate at which heat can be rejected or the efficiency of the power plant.

4-02. Unlimited Capacity Type Heat Sinks. All heat sink systems which fall within this classification employ either the atmosphere or a large natural body of water, such as a river or large lake, as a heat sink medium. Their heat storage capacity is considered to be unlimited because the waste heat load produced by even the largest underground facility would be small in comparison to the heat load required to produce thermal saturation of the heat sink medium.

The above statement should not be construed to imply that this type of sink can be operated without limitation on the heat rejection rate. The actual heat transfer system used, whether it is a cooling tower, dry fin coil or so forth, will have specific capacity limitations with respect to operating temperatures, imposed heat loads, etc. However, in the thermal design of these systems it is not necessary to consider long term temperature changes of the heat sink medium as a result of the operation of the hardened heat sink system.

A brief discussion of various types of unlimited capacity type heat sinks is presented in the following paragraphs. The advantages and disadvantages of each type of heat sink discussed are summarized in Table 4-1.

4-02.1. Spray Ponds. The simplest type of atmospheric heat sink is the spray pond. The discharge water from the power plant condensers is sprayed through the air from nozzles and then collected in a pond below. The waste heat is transferred from the water droplets to the air via evaporation, radiation and convection. Heat is also transferred from the surface of the pond but at a much lower rate.

The thermal efficiency of this type of system is dependent upon the temperature differential between the air and the water, the rate of air circulation, the relative humidity and the size of the water droplets. Increases in air temperature, relative humidity or the size of the water droplets or a reduction in the rate of air circulation will reduce the heat transfer efficiency of this type of system. Although they are relatively inexpensive to build they are not capable of substantial hardening and require shielding or an intermediate heat exchange loop to prevent radioactive contamination of the facility. The collection pond also will require a relatively large area and a make-up water supply must be available to replace water lost through evaporation.

4-02.2. Dry Finned Coils. The dry finned coil heat sink system also employs the atmosphere as the heat sink medium. The condenser coolant water is circulated through a network of cooling coils whose outer surface is exposed to a positive air flow. The waste heat is transferred from the water to the coil via conduction and convection and from the coil to the atmosphere by conduction, convection and radiation. Cooling fins are installed along the outer surface of the cooling coil to increase the heat transfer surface area and improve efficiency.

Since the condenser water circulates in a closed pipe network radioactive contamination from external sources is reduced. Also the closed system prevents evaporation of the water and eliminates the requirement for make-up water, as in a spray pond system.

Thermal efficiency of these units depends primarily upon the rate of air flow past the cooling coil, the water flow rate through the coil, the temperature difference between the water and the air and the amount of internal and external contamination of the coil surfaces. Increasing the air flow rate or the temperature difference between the air and the water will increase the heat exchanging capacity of the unit. Changes in water flow rate may or may not improve heat transfer, depending upon the flow velocities. Contamination of the heat transfer surfaces, whether internal or external, will reduce the efficiency of the unit. Internal contamination results from the deposition of dissolved minerals in the water along the tube surfaces. This is normally found in systems which are used for prolonged periods or which use high mineral content water for the condenser cooling system. External contamination of the cooling coil results from the deposition of dust and/or debris on the surface of the cooling coil fins. This can be a significant problem when used with a hardened facility and will require protection to prevent contamination resulting from crater throw out.

Dry finned cooling coils generally tend to be more efficient than spray ponds and can be hardened to a higher degree. However, these units are normally not considered to be hardenable to a level compatible with a high hardness facility.

4-02.3. Hardened Cooling Tower. One of the most common methods of rejecting heat to the atmosphere is through the use of a cooling tower. Hot water is passed through the cooling tower where the waste heat is transferred to the air via evaporation and to some extent convection and conduction. Air flow through the cooling tower may be density induced or, as is more common, forced by fans.

The hardened type of cooling tower is similar to commercial designs. The major difference between the two types is that the hardened cooling tower is structurally reinforced to withstand blast effects. Dust control may also be included as accumulation of dust will reduce the efficiency of the unit.

The hardened cooling tower generally is capable of a higher degree of hardness than either a dry finned coil or a spray pond. In addition it is normally more efficient thermally, requiring less cooling air than the other two types of atmospheric heat sinks. This reduced air requirement is important since it can be translated into smaller air passageways and blast valves and thus lower costs.

4-02.4. Locally Available Water. Local water sources such as rivers, lakes, water wells, aquifers or even the ocean for coastal sites can be used for cooling the power plant. Cool water would be drawn from the source, circulated through a heat exchanger and then either returned to the source or wasted at a remote location.

The primary factors affecting the performance of these systems are the temperature of the source water and the water flow rate. High water temperatures will reduce efficiency whereas increasing the coolant flow rate will normally improve it. Water quality can also be an important parameter as corrosion or accumulation of matter in the heat exchanger will reduce efficiency. Shielding may be required within the heat exchanging system to prevent radiation contamination, whether from an internal or external source.

The principal disadvantage of this type of system is the normally low hardness level associated with it. As a result of bombardment an aquifer may be destroyed, a river diverted from its channel, or supply and return piping damaged. Even if the water source is not destroyed but only reduced in flow capacity this could force curtailment of power generation in the hardened facility.

The principal advantages of this type of system are its normally high thermal efficiency, low equipment costs and simplicity of operation. However, hardening of the water source, if possible at all, can entail significant costs.

4-02.5. Summary of Unlimited Capacity Heat Sinks. All unlimited capacity type heat sinks are standard commercial systems which have been adapted for use with an underground facility. Equipment costs tend to be low and the overall system is generally easy to operate. The factors which tend to have the greatest impact on its efficiency are the temperature and flow rate of the cooling medium.

All these systems require some type of connection to the surrounding. For the cooling tower or dry finned coil this may entail air passageways protected by blast valves; for the others it may mean piping networks to the surface. As a result overall hardness levels for these heat sink systems tend to be low, which restricts their use to facilities in the low to moderate hardness range. The salient features of these systems are presented in Table 4-1.

4-03. Finite Capacity Heat Sinks. The greatest disadvantage of the unlimited capacity type heat sinks is their lower hardness capability which results from their interconnection to the surface. To provide a higher hardness level several heat sink concepts have been developed where the sink itself is located underground and does not require connection to an external medium. In this manner the hardness level of the heat sink system can be designed to be equal to that of the facility itself.

However as a result of its underground location and the substantial expense involved in underground excavation the total volume of the sink must be kept to the minimum required. Since all sinks of this category operate on the basic principle of heat storage through a temperature change and/or change in phase of the heat sink medium this means that the total heat storage capability will be finite. Therefore the design of this type of sink requires the calculation of the total heat storage capacity as well as system efficiency and operating conditions.

A brief description of the various types of finite capacity heat sinks is presented in the following paragraphs. The advantages and disadvantages of each concept are summarized in Table 4-2.

4-03.1. Stored Water. Waste heat from the power system can easily be rejected using a change in water temperature. A stored water heat sink, which consists of one or multiple reservoirs of water located underground is used to supply the source water for cooling. During standby the water may be maintained at the ambient temperature or for improved efficiency at a depressed temperature.

In addition the water may be used for once-through cooling and subsequently wasted outside the facility or recirculated back to the sink. In the former mode the sink temperature will remain constant and thus the power system efficiency will remain constant with the heat sink operated until the supply of water is exhausted. In the latter mode the recirculation of the heated water results in an increase in the water temperature of the sink. The sink is used until the water temperature has increased to the point where it can no longer be used for cooling purposes and operation must be terminated. Although the increasing reservoir temperature will reduce the efficiency of the power plant and thus increase the waste heat load the increased volumetric heat storage capacity of this mode of operation more than compensates for the reduced plant efficiency.

Table 4-1

Advantages and Disadvantages of the Unlimited Capacity Type Heat Sinks

	Advantages	Disadvantages
1.	Spray Ponds	
	a. similar to standard commercial type systems	a. low hardness capability
	b. long log of operating experience available	b. requires large surface area
	c. low maintenance and operating costs	c. low overall thermal efficiency
		d. susceptible to contamination from dust and radioactive debris
		e. requires make-up water supply
2.	Dry Finned Coils	
	a. similar to standard commercial type systems	a. low hardness capability
	b. long log of operating experience available	b. high capital costs
	c. minimum make-up water requirements	c. high maintenance and operating costs
	d. affords some shielding from radioactive contamination	d. requires prototype to certify performance
		e. susceptible to reduction in efficiency resulting from dust accumulation on heat transfer surfaces
3.	Cooling Towers	
	a. similar to standard commercial type systems	a. considerable operating and maintenance costs
	b. long log of operating experience available	b. some designs susceptible to losses in efficiency resulting from dust accumulation
	c. generally has highest overall efficiency of ambient air systems	c. could be susceptible to radioactive contamination
	d. capable of hardening to withstand blast effects	d. requires make-up water supply
4.	Locally Available Water Supplies	
	a. similar to standard commercial type systems	a. water source susceptible to damage from ground motion and other blast effects
	b. high thermal efficiencies	b. substantial changes in water temperature or flow rates will reduce efficiency
	c. minimum amount of mechanical equipment required	c. additional hardening of water source generally not practical

4-03.2. Stored Ice. The volumetric heat storage capacity of a water reservoir type sink can be increased substantially if the water is stored in the form of ice. During the initial hours of operation the waste heat melts the ice which results in a low, almost constant temperature source of coolant water. As melting continues the coolant water drawn from the sink will exhibit a gradual increase in temperature until all the ice has been melted. After all the ice has been melted the sink will behave like a stored chilled water reservoir.

The principal advantages of a stored ice reservoir are its higher volumetric heat storage capability and its ability to provide low temperature coolant water during the early hours of operation. Depending upon actual operating temperatures and conditions a stored ice heat sink will normally be one half to two thirds the size of an equivalent heat storage capacity stored water reservoir. The low temperature coolant water generally will result in higher power plant efficiencies and thus ultimately for any specific total period of operation lower waste heat loads and reduced heat storage requirements.

4-03.3. Rock Tunnel Steam Condenser. If the power plant employs steam turbine driven generators the exhaust steam from the turbines may be condensed using the walls of a rock tunnel. The tunnel, which may be specially constructed or one of the passageways used during construction of the facility itself, is sealed off using bulkheads. As the steam is introduced the water vapor will condense along the cooler surfaces of the tunnel and collect along the bottom. The tunnel is maintained at a constant pressure, normally atmospheric, by bleeding noncondensibles (air) out. The condensed steam is drawn out of the tunnel, purified to remove dissolved minerals and then recirculated back to the boiler.

During operation the rejection of the waste heat to the tunnel walls results in an increase in wall temperature. Once the wall surface temperature has reached the dew point (thermal saturation) operation can continue only by either pressurizing the tunnel (raising the dew point) or reducing the rate of heat input.

The principal advantage of this type of system is its potential low construction cost since tunnels made during construction of the facility can be used as the heat sink once the facility is completed. The greatest disadvantages of this concept are that the introduction of live steam into the tunnel can cause rock fall which can damage the steam piping network and that water seepage out of the tunnel may be difficult to assess, thus making it difficult to estimate what the make-up water requirements should be.

4-03.4. Earth Heat Sinks. For installations which are constructed using an open pit mining technique and subsequently buried using backfill the waste heat from the power plant may be rejected to the backfill material. A piping network installed during the backfill operation would be used to circulate the heat transfer fluid, most probably steam, through the backfill.

The use of this type of heat sink would be restricted to installations in the low to moderate hardness level with a total power requirement in the 500 kW range. In addition, since power levels in this range are feasible only with diesel, gas turbine or fuel cell systems special equipment would be required to absorb the waste heat from the power plant coolant and generate steam for use in the piping network.

As in the rock tunnel steam condenser concept the principal advantage of this heat sink concept is its low construction cost since the heat sink complex is not a separate project but part of the construction of the installation itself. The major disadvantages are that 1) its use is restricted to low to moderate hardness level facilities with low total power requirements, 2) at present field tests have not been conducted to certify its performance, and 3) substantial costs can be involved if the required high thermal conductivity backfill must be transported to the site from a remote location.

4-03.5. Summary of the Finite Capacity Heat Sink Concepts. Except for the earth type heat sinks all the finite capacity type heat sinks are capable of substantial hardening and can be used in conjunction with deep sites. In addition all the finite capacity type heat sinks employ a change in temperature and/or phase of the heat sink medium to provide heat storage. Each of these heat sinks will attain a thermal saturation once its total heat storage capacity has been used and this will require termination of operation or, in the case of the rock tunnel steam condenser, a reduction in the rate of waste heat (power) production.

The salient features of each of the finite capacity type heat sink are presented in Table 4-2.

4-04. Other Concepts. During the initial planning phases in the development of a design for an underground facility the consideration of which type of heat sink would be used should not be restricted solely to those concepts mentioned in this report. Local site conditions may at times be employed to provide the necessary heat storage media.

For example, at Camp Century, Greenland, a short term heat sink system was developed using the ice sheet. After a cavity had been melted in the ice sheet the condenser water from the power plant was cooled by circulating it to the cavity. Another example illustrating the use of favorable local conditions to provide a heat sink is a concept developed by Florida utility companies. Using limestone solution cavities which result from the subterranean flow of sea water, the power plant discharge water is pumped into the cavity where it forms a pure

Table 4-2

Advantages and Disadvantages of the Limited Capacity Type Heat Sinks

	Advantages	Disadvantages
1.	Stored Water	
	a. relatively simple operation	a. requires large amount of excavation
	b. minimum amount of mechanical equipment required	b. increasing water temperature will reduce power plant efficiency
	c. capable of substantial hardening	c. has a finite heat storage capacity
	d. minimal maintenance costs	
	e. working systems already installed at some sites	
2.	Stored Ice	
	a. excavation required less than for stored water sink	a. increasing water temperature during final hours of operation results in reduced plant efficiency
	b. provides almost constant low temperature coolant water during initial hours of use	b. high capital costs in mechanical equipment
	c. capable of substantial hardening	c. high maintenance costs
		d. prototype system not yet field tested
		e. substantial period of time involved in initial freezing
		f. has a finite heat storage capacity
3.	Rock Tunnel Steam Condenser	
	a. relatively simple to construct and operate	a. susceptible to damage resulting from ground motion
	b. minimum amount of mechanical equipment required	b. possible damage to tunnel and piping from steam induced rock fall
	c. can use passageways bored for initial construction of facility	c. has a finite heat storage capacity
	d. low maintenance costs	d. requires purification of water prior to reuse
	e. system has been field tested	e. estimating make-up water requirements can be difficult
		f. application restricted to specific types of rock structures
4.	Earth Heat Sinks	
	a. relatively simple to construct	a. not suitable for large power systems
	b. overall size can be less than either stored ice or stored water	b. susceptible to damage from ground motions
	c. anticipated costs lower than stored water or stored ice sink	c. not capable of substantial hardening
	d. low maintenance costs	d. can result in decreasing power plant efficiency with use
		e. capacity can be reduced due to changes in ground water table level
		f. prototype system not yet field tested

water bubble above the seawater. The waste heat is thus transferred to the sea water and the coolant water is recirculated. Although this concept was considered to be of limited economic feasibility for commercial power systems, with some modification of the operating procedure it could provide a low cost method of providing the required heat sink capability for an underground installation.

4-05. Combinations of Heat Sinks. When appropriate, consideration should be given to using several different types of heat sinks to fulfill all the requirements of an underground facility. For example the waste heat from a nuclear power plant may be rejected using a rock tunnel steam condenser while the waste heat from the air conditioners is rejected to an ice heat sink. Although the construction of two types of sinks would seem to be unfeasible economically, eliminating the inherent inefficiency of mechanical refrigeration and thus reducing the size of the power plant may well offset this additional cost.

In a similar manner the liquid fuel stored for diesel and gas turbine power systems can be used to provide supplemental heat storage capacity. Although the total amount of heat that may be rejected to a fuel oil sink is limited by the maximum allowable fuel temperature, a significant amount of waste heat may be stored in this manner.

Another possibility which should not be overlooked in the selection of a heat sink is the use of different heat sinks for different modes of operation of the facility. For example, consider a very hard site which must operate for a significant period of time and at a fairly substantial power level. Normally this would entail the use of a nuclear power plant, and because of the inherent thermal inefficiency of these units a massive heat sink complex. However a smaller and less costly heat sink complex may be constructed using an underground ice reservoir and a hardened cooling tower located in an underground cavern. When the installation is under attack or an imminent threat of attack exists the ice reservoir would be used. The chamber for the cooling tower would be sealed to prevent it from being damaged by blast effects. Once the threat of attack or the attack itself had passed the cooling tower would be used and the ice sink held in reserve for possible future use. In this manner the hardness level of the cooling tower can be increased significantly since massive blast valves, normally considered to be infeasible, could be employed to protect it. Rates of closure of the blast valves would no longer be a problem since they would not have to react in response to bombardment but could be sealed prior to attack.

Of course in this example sizing of the ice reservoir would be dependent upon the anticipated number and length of attacks. But even for a large number of long attacks it would be significantly smaller than a reservoir sized to store the waste heat produced during the entire period the facility would operate in the isolated mode.

4-06. Summary. Selection of a particular heat sink concept for a proposed facility would depend primarily upon the following factors: 1) desired level of hardness, 2) compatibility with the power system, 3) rate of heat rejection, 4) total amount of heat to be rejected, 5) insitu conditions at the proposed site, and 6) economics. The economic analysis should include not only the cost of any mechanical equipment required and initial construction costs, but also all costs associated with preparing the sink, i.e. refrigeration, if required, and maintenance costs when the sink is in the standby mode.

SECTION 5 -- INTEGRATION OF THE HEAT SINK AND POWER PLANT

5-01 Types of Power Plants. Design studies have concluded that, at present, pressurized water nuclear steam turbine generators, diesel engine generators and gas turbine driven generators are the only feasible power plants for use in most underground facilities. Battery or fuel cell power systems are currently considered feasible only for sites with relatively low electrical loads and or rather short periods of operation. The discussions presented in this section will be restricted to the first three types of power plants mentioned.

5-01.1 Plant Characteristics. The primary characteristics considered in the selection of a power plant are its capability to be hardened, size and cost. These three factors are related to one another since to some extent the hardness level of the facility will depend upon its size and the thickness of the structural elements; both of these factors will impact on the total cost.

Other factors which are deemed important in the selection process are the power range over which the system will operate, complexity of the system, maintenance required, system response to changes in loads and the overall reliability of the system.

Another factor which must be considered before a specific type of power plant is selected is the type of heat sink which will be used. The interfacing of the heat sink system to the power plant will affect not only the performance of the power plant but also the total cost of the project since some power plant/heat sink combinations will result in lower overall system efficiency and thus require additional heat storage capacity or plant size.

5-02. Interfacing of the Power Plant and Heat Sink Systems. This section provides discussions related to the interfacing of the power plant to the heat sink system.

5-02.1 Thermal Requirements. The overall efficiency of the power plant will depend upon the temperature of its coolant. In some systems, such as the diesel generator, the plant efficiency is a weak function of coolant temperature with large changes in coolant temperature required to produce a noticeable change in efficiency. On the other hand the efficiency of a steam turbine is a strong function of its coolant temperature since large changes in the coolant (i.e. condenser) temperature produce appreciable changes in plant efficiency.

Since changes in power plant efficiency will affect its net electrical output, rate of fuel consumption and the amount of waste heat produced it is imperative that the design analysis of the power plant consider the effect of increasing heat sink temperature as a result of use.

All power plant systems will have a maximum allowable coolant temperature; this will in turn limit the maximum allowable heat sink temperature. This temperature is called the thermal saturation temperature since when it is attained operation of the power plant must be terminated. In general when comparing different power systems it can be stated that the higher the maximum allowable coolant temperature is the greater the heat storage per unit volume the heat sink will have. However, having a high maximum allowable coolant temperature will not be important if the efficiency of the power plant is low throughout the operating range. This would dictate the use of a large heat sink system and cancel the beneficial aspects of a high maximum coolant temperature.

The minimum allowable coolant temperature for the power plant generally will not be a problem even when using a very low temperature sink. For example if a steam condenser requires 50°F source water and discharges 60°F water but the heat sink provides 40°F, the correct source water temperature can be provided by mixing equal quantities of the 60°F condenser discharge and the 40°F sink water. In this example the correct source water temperature is provided by bypassing one half of the circulating water around the heat sink. By using the option to bypass the sink it is possible to create a return water temperature higher than the sink temperature.

5-02.2 Thermal Stability. The coolant water drawn from any sink will exhibit both slight fluctuations in temperature and generally an overall increasing temperature with respect to time. The slight fluctuations in coolant water temperature should present no problems; however, if the overall increase in temperature of the coolant water is too rapid it is possible that the operators may not be able to make the necessary changes in time, and may produce a system failure. This problem must be faced when density stratified reservoirs are being considered since the sink will exhibit large sudden changes in temperature.

5-02.3 Operating Range. Depending upon the diversity factors and the mission requirements, during the buttoned up mode the power plant may be operating at anywhere between a partial load and a short term overload. The heat sink system should be checked to ensure that it will perform properly and as anticipated under all possible loadings. This is especially important when considering water reservoir type sinks since partial load operation may result in density stratified flow and, if unanticipated, can produce control system problems. Similarly the tunnel condenser and earth heat sink systems may not perform properly when overloaded, even for short periods of time.

5-02.4 Equipment Compatibility. Not all power plants will be directly compatible with any particular type of heat sink. For example the use of a rock tunnel condenser as a heat sink for either a diesel or gas turbine system would require that a secondary steam generation unit be installed to absorb the waste heat from the power plant and reject it to the sink. This will increase the cost and complexity of the system and may also reduce the overall efficiency of the power plant. Therefore unless this type of system would provide additional substantial benefits it would not be considered the most feasible implementation.

As a rule it can be stated that unless significant benefits in efficiency, performance, reliability or hardness are attainable heat sinks which are not directly compatible with the power plant cooling system should not be used.

SECTION 6. HEAT SINK THERMAL ANALYSIS

6-01. General. This section presents techniques for predicting the heat storage capacity of the finite capacity type heat sinks. Techniques for predicting the refrigeration requirements for initial cooling and standby maintenance for the stored chilled water and stored ice type sinks also are included.

To assist in the application of the prediction techniques data forms have been provided in Appendix A. In addition, sample problems illustrating the use of the prediction equations are provided in Section 7.

It should be recognized that the techniques presented herein are only approximations of the gross heat transfer behavior of heat sinks. Their use is recommended only for overall sizing and the development of cost versus benefit relationships to assist in the selection of a heat sink concept. Formulation of a final design must include the consideration of transient heat loads, actual sink configuration, mode of operation and so forth, which is beyond the scope of this manual.

6-02. Stored Water Reservoir, Once-Through Cooling. Waste heat from the power plant may be rejected by drawing water from the heat sink reservoir, circulating it through the condenser or engine jacket one time and then wasting the heated water. Operation in this mode can continue until the reservoir has been completely emptied. Normally the water in the reservoir would be maintained at the ambient temperature of the surrounding rock but stored chilled water may be used to improve power plant efficiency and thus reduce the required heat sink volume. In either situation, however, it can be assumed that during the operating life of the sink the water will remain at its initial temperature.

The total mass of water required for this type of sink is given by:

$$M_w = 8.02\, W \rho_w t_d \qquad (6\text{-}01)$$

where M_w = required mass of water, lb

W = coolant water flow rate, gal/min

ρ_w = density of water, lb/ft^3

t_d = design life of the heat sink system, hr.

If available, the performance relationships for the condenser should be used to determine the coolant water flow rate. Otherwise the coolant water flow rate can be calculated from:

$$W = \frac{0.125\, q_o}{\rho_w c_w \Delta T_c} \qquad (6\text{-}02)$$

where q_o = heat rejection rate (constant), Btu/hr

c_w = specific heat of water, Btu/lb°F

ΔT_c = water temperature rise across the condenser, °F

6-03. Stored Water Reservoir with Recirculation. Recirculation of the coolant water between the heat sink reservoir and the power plant results in an increased total heat storage capacity for the reservoir with respect to a system where once-through cooling is employed. Although recirculation of the heated water back to the reservoir results in an increase in the sink temperature and thus a decrease in the power plant efficiency the net gain in heat storage capacity more than offsets the increase in the power plant waste heat load. Operation of this type of heat sink continues until the sink temperature has increased to a temperature where the water can no longer be used for cooling purposes, this limiting temperature being referred to as the thermal saturation temperature. In sizing this type of sink it is important to insure that thermal saturation will not occur prior to completion of the installation's mission.

Techniques for predicting the behavior of water reservoirs maintained at the ambient temperature of the surrounding rock are presented in this section; prediction techniques for chilled water sinks are presented in Section 6-04. The prediction techniques are presented for sinks which exhibit little to no heat transfer to the surroundings as well as those in which a significant amount of heat transfer to the surrounding rock occurs. In addition prediction equations are presented for sinks where the heat rejection rate remains constant or increases either linearly or exponentially with increasing sink water temperature. These latter two techniques allow an approximation to be made of how changes in the overall system efficiency will affect performance.

6-03.1. Stored Water Reservoirs Without Heat Transfer to Surroundings: This category of stored water reservoirs is typified by the situation where the reservoir is a containment vessel which is not in intimate contact with the surrounding rock. This would occur if the reservoir was located in a large cavern or suspended on shock isolation mounts to prevent damage from motion of the ground resulting from bombardment. Although even in these situations there would be some heat transfer between the reservoir and the surroundings it would normally be at least an order of magnitude smaller than the applied waste heat load. The following techniques are based on the assumption that the sinks behave as well mixed bodies of water. Sinks in which density stratification occurs are discussed in section 6-05.

a. Constant Heat Rejection Rate: If the power plant waste heat load is constant then the required mass of water is given by:

$$M_w = \frac{q_o t_d}{c_w(T_f - T_o)} \qquad (6\text{-}03)$$

where T_f = final sink temperature, °F

T_o = initial sink temperature, °F

The coolant water flow rate can be determined either from the condenser performance relationships or through the use of equation 6-02.

b. Linearly Increasing Heat Rejection Rate: If the power plant waste heat load increases with increasing coolant water temperatures in the following manner:

$$q_r(T) = A_1 T + A_o \qquad (6\text{-}04a)$$

where $q_r(T)$ = temperature dependent waste heat load, Btu/hr

T = sink temperature, °F

A_1 = constant, Btu/hr°F

A_o = constant, Btu/hr

then the mass of water required is given by:

$$M_w = \frac{A_1 t_d}{c_w} \; \frac{1}{ln\left\{\frac{T_f - A_o/A_1}{T_o - A_o/A_1}\right\}} \qquad (6\text{-}04b)$$

Once the required mass of water has been calculated the variation in heat sink temperature as a function of time can be determined using:

$$T = \left\{T_o - \frac{A_o}{A_1}\right\} e^{\left\{\frac{A_1 t}{M_w c_w}\right\}} + \frac{A_o}{A_1} \qquad (6\text{-}04c)$$

where t = time, hr

Using the variation in heat sink temperature with time the variation in the waste heat load and the coolant flow rate with time can be determined.

c. Exponentially Increasing Heat Rejection Rate: If the power plant waste heat load increases exponentially with respect to sink temperature in the following form:

$$q_r(T) = B_o\, e^{B_1 T} \qquad (6\text{-}05a)$$

where B_o = constant, Btu/hr
B_1 = constant, 1/°F

then the required mass of water is given by:

$$M_w = \frac{B_o B_1 t_d}{c_w\left[e^{-B_1 T_o} - e^{-B_1 T_f}\right]} \qquad (6\text{-}05b)$$

Once the mass of water has been determined the variation in sink temperature as a function of time can be calculated using:

$$T = -\frac{1}{B_1}\, \ln \left[e^{-B_1 T_o} - \frac{B_o B_1 t}{M_w c_w}\right] \qquad (6\text{-}05c)$$

The variation in sink temperature with time in turn can be used to determine the variation in heat rejection rate and coolant water flow rate with time.

6-03.2. Stored Water Reservoirs with Heat Transfer to Surroundings: If the reservoir is constructed such that the water has good thermal contact with the surrounding rock then a significant amount of the rejected waste heat may be stored by heating of the rock. An example of this situation is an unlined rock chamber or tunnel which is filled with water.

a. Constant Heat Rejection Rate: If the power plant waste heat load remains constant then the heat storage capacity of a water reservoir with heat transfer to the surrounding rock may be calculated using the nondimensional parameters defined in equations 6-06a, 6-06b, 6-06c and either Figure 6-1 or 6-2:

$$f(\tau, G) = \frac{k(T_f - T_o)\ell}{q_o} \qquad (6\text{-}06a)$$

$$G = \frac{2\pi a^2 \ell \rho c}{M_w c_w} \qquad (6\text{-}06b)$$

$$\tau = \frac{k t_d}{\rho c a^2} \qquad (6\text{-}06c)$$

where
- k = thermal conductivity of the rock, Btu/hr ft °F
- ℓ = length of the reservoir, ft
- a = diameter of the reservoir, ft
- ρ = density of the rock, lb/ft^3
- c = specific heat of the rock, Btu/lb °F

In order to use these nondimensional parameters it is necessary to select dimensions for the reservoir and then calculate the useful life of the heat sink. If the calculated life is less than the desired life these dimensions must be modified and the calculation repeated until the calculated life is equal to the desired design value.

Once the proper sink dimensions have been determined the sink temperature as a function of time can be calculated using the nondimensional parameters and Figure 6-1 or Figure 6-2.

b. Linearly or Exponentially Increasing Heat Rejection Rate: If the power plant waste heat load increases as a result of increasing heat sink temperature the total operating life of the heat sink may be approximated using a modified version of the prediction technique presented in section 6-03.2a.

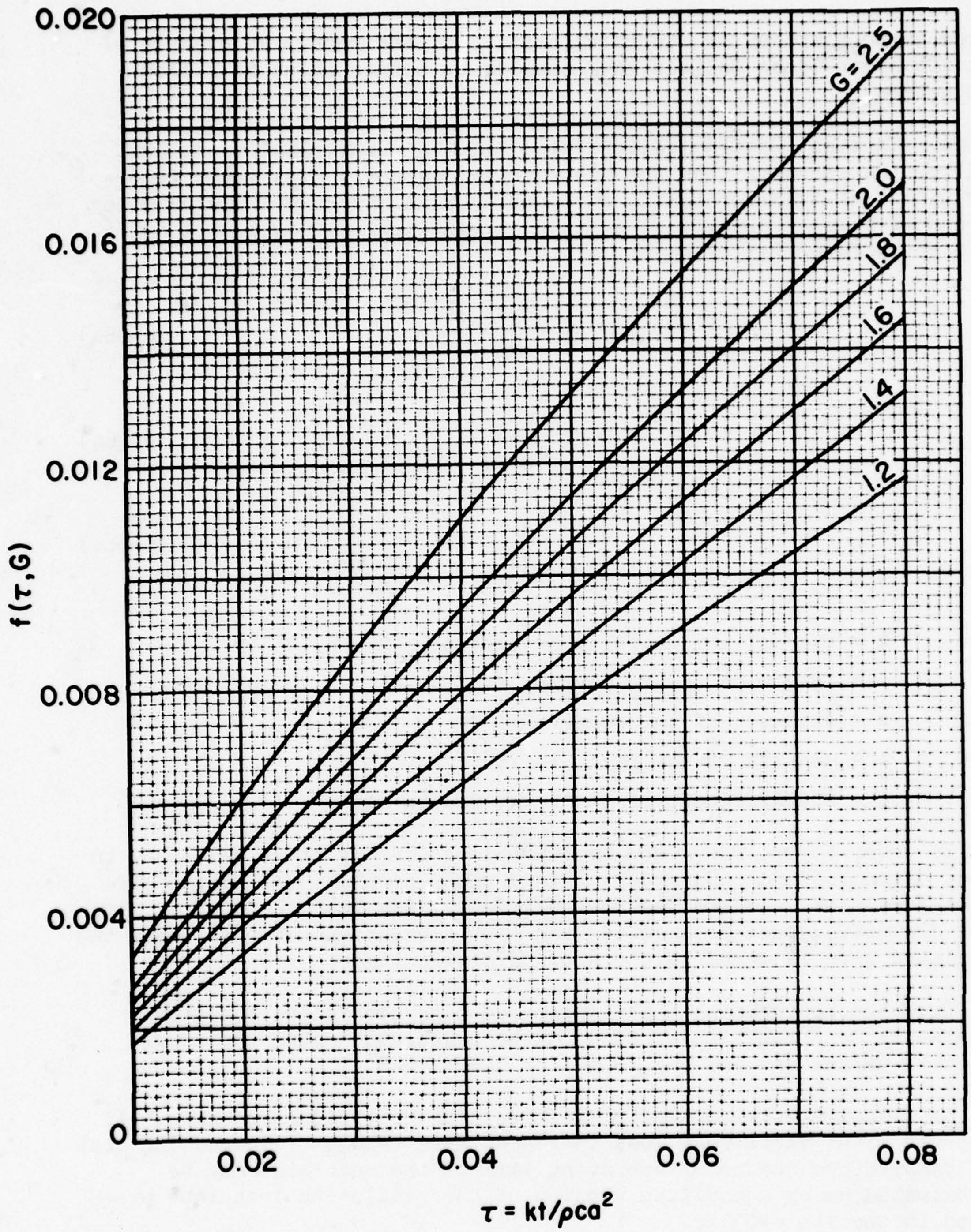

Figure 6-1. Reservoir heat absorption at constant heat rate.

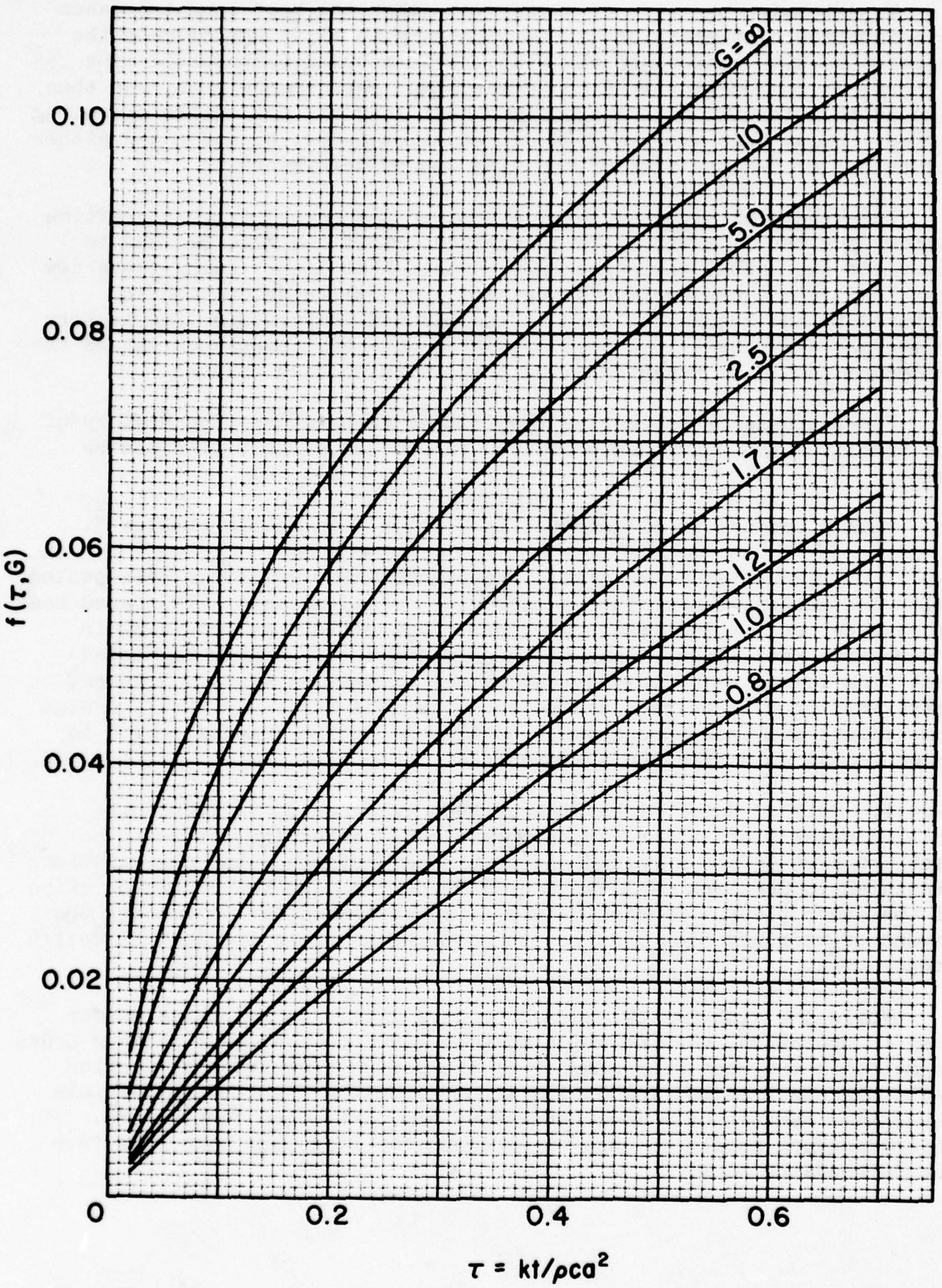

Figure 6-2. Reservoir heat absorption at constant heat rate.

To determine the life of a reservoir when the heat load increases as a function of temperature it is necessary to first approximate the temperature-time history of the sink. This assumed temperature-time history, in conjunction with the power plant performance data, can then be used to determine an average waste heat load for the entire operating period. Using this average load and equation 6-06a, b, and c and either Figure 6-1 or 6-2 the size of the sink can be determined.

Once the sink size has been determined the actual temperature-time history for the sink should be determined. This can then be used to determine the actual variation in the waste heat load. Using these new heat load data a new average waste heat load can be calculated and compared to the assumed value. If the assumed temperature-time history or average waste heat load does not agree with the calculated values the procedure should be repeated using the new calculated values.

For purposes of estimating the initial temperature-time history of the heat sink the equations presented in Section 6-03.1b or c can be used.

This procedure is illustrated in Section 7 in sample problem 5.

6-04. Chilled Water Reservoirs. The use of water which has been cooled below the ambient temperature can provide two advantages: increased heat storage per unit volume and high power plant efficiency resulting in lower waste heat loads. Both of these factors will reduce the total heat sink volume required and thus reduce excavation costs. However, refrigeration will be required to cool the sink initially to its design temperature and then subsequently during the entire standby period to prevent heat gain from the surroundings.

6-04.1. Chilled Water Sinks Without Heat Transfer to Surroundings: Chilled water sinks which can be approximated as having little to no heat transfer to the surroundings should be designed using the equations presented in Sections 6-03.1a, b or c depending upon the characteristics of the power plant. Section 6-10 provides a method for estimating the amount of refrigeration required to cool the water to its initial design temperature.

While the sink may be assumed to have little to no heat transfer when in operation, consideration must be given to long term cooling loads to maintain the heat sink chamber at the design temperature. Section 6-11 presents a method for estimating the cooling required to maintain the inside surface of a rock chamber at any particular temperature. This procedure should be used to estimate the long term heat flux from the surrounding rock.

6-04.2. Chilled Water Sinks with Heat Transfer to Surroundings: For estimating purposes it can generally be assumed that for this type of heat sink system the surrounding rock will have been cooled sufficiently during the sink preparation period so that the rock adjacent to the reservoir will be at the initial design temperature of the water and the prediction techniques presented in Sections 6-03.2 a or b may be employed.

To determine the temperature gradient in the rock as a function of time equation 6-07 should be used in conjunction with Table 6-1.

$$T(R,t) = T_\infty + (T_\omega - T_a)\beta(\frac{R}{a}, \tau) \qquad (6\text{-}07)$$

where $T(T,t)$ = rock temperature at radius R and time t, °F

T_∞ = ambient rock temperature, °F

T_ω = constant surface temperature, °F

$\beta(\frac{R}{a},\tau)$ is from Table 6-1

a = radius of heat sink chamber, ft

$\tau = \dfrac{kt}{\rho c a^2}$

Once the temperature gradient resulting from initial cool down has been determined the temperature gradient that would result from operation of the sink should be calculated. *This gradient may be calculated* using equation 6-08.

$$T(R,t_d) = T_o + (T_m - T_o)\beta(\frac{R}{a},\tau_d) \qquad (6\text{-}08)$$

where T_o = initial heat sink temperature, °F

T_m = mean heat sink temperature during operation, °F

$\tau_d = \dfrac{kt_d}{\rho c a^2}$

t_d = design life of the sink, hr

This gradient should be used to determine the minimum separation distance if multiple sinks are to be built. If the heat sinks are not separated sufficiently thermal interaction between the sinks may occur, resulting in a reduced total heat storage capacity.

Table 6-1. Numerical values for function $\beta(\frac{R}{a},\tau)$. (From reference 56)

	$\frac{R}{a}$													
τ	1.0	1.1	1.2	1.3	1.4	1.5	1.6	1.7	1.8	1.9	2.0	3.0	4.0	5.0
0.001	1.000	0.024	0.000											
0.002	1.000	0.109	0.001	0.000										
0.003	1.000	0.188	0.009	0.000										
0.004	1.000	0.251	0.023	0.001	0.000									
0.005	1.000	0.303	0.042	0.002	0.000									
0.006	1.000	0.345	0.062	0.005	0.000									
0.007	1.000	0.380	0.083	0.010	0.001	0.000								
0.008	1.000	0.410	0.104	0.016	0.001	0.000								
0.009	1.000	0.435	0.124	0.022	0.002	0.000								
0.01	1.000	0.458	0.144	0.030	0.004	0.000	0.000							
0.02	1.000	0.589	0.290	0.117	0.039	0.010	0.002	0.000	0.000					
0.03	1.000	0.652	0.379	0.194	0.087	0.034	0.011	0.003	0.001	0.000				
0.04	1.000	0.691	0.439	0.254	0.133	0.063	0.027	0.010	0.004	0.001	0.000			
0.05	1.000	0.718	0.483	0.302	0.175	0.093	0.046	0.021	0.009	0.003	0.001	0.000		
0.06	1.000	0.739	0.517	0.341	0.211	0.122	0.066	0.033	0.016	0.007	0.003	0.000		
0.07	1.000	0.754	0.544	0.373	0.242	0.149	0.087	0.047	0.024	0.012	0.005	0.000		
0.08	1.000	0.767	0.566	0.400	0.270	0.174	0.106	0.062	0.034	0.018	0.009	0.000		
0.09	1.000	0.778	0.585	0.423	0.294	0.196	0.125	0.077	0.045	0.025	0.013	0.000		
0.1	1.000	0.787	0.601	0.443	0.316	0.217	0.143	0.091	0.055	0.032	0.018	0.000	0.000	
0.2	1.000	0.837	0.691	0.562	0.450	0.355	0.275	0.209	0.156	0.114	0.082	0.001	0.000	
0.3	1.000	0.860	0.733	0.620	0.519	0.430	0.352	0.246	0.229	0.131	0.142	0.006	0.000	
0.4	1.000	0.873	0.758	0.655	0.562	0.479	0.405	0.339	0.282	0.233	0.191	0.015	0.000	0.000
0.5	1.000	0.883	0.776	0.680	0.592	0.514	0.443	0.380	0.323	0.274	0.230	0.027	0.001	0.000
0.6	1.000	0.890	0.789	0.698	0.615	0.540	0.472	0.411	0.356	0.307	0.263	0.040	0.003	0.000
0.7	1.000	0.895	0.800	0.713	0.634	0.562	0.496	0.436	0.382	0.334	0.290	0.054	0.006	0.000
0.8	1.000	0.899	0.808	0.725	0.649	0.578	0.515	0.457	0.405	0.357	0.313	0.068	0.009	0.001
0.9	1.000	0.903	0.815	0.735	0.661	0.594	0.532	0.475	0.424	0.377	0.334	0.082	0.013	0.001
1.0	1.000	0.906	0.821	0.743	0.672	0.606	0.546	0.491	0.440	0.394	0.351	0.095	0.018	0.002
2.0	1.000	0.924	0.854	0.790	0.732	0.677	0.627	0.580	0.536	0.496	0.458	0.194	0.071	0.022
3.0	1.000	0.932	0.870	0.812	0.760	0.711	0.635	0.623	0.583	0.546	0.511	0.256	0.119	0.049
4.0	1.000	0.937	0.879	0.826	0.777	0.731	0.689	0.649	0.612	0.577	0.544	0.300	0.157	0.076
5.0	1.000	0.940	0.886	0.835	0.789	0.746	0.706	0.668	0.633	0.599	0.568	0.332	0.188	0.101
6.0	1.000	0.943	0.890	0.842	0.798	0.757	0.718	0.682	0.648	0.616	0.586	0.357	0.214	0.123
7.0	1.000	0.945	0.894	0.848	0.805	0.765	0.728	0.693	0.661	0.630	0.601	0.378	0.235	0.142
8.0	1.000	0.946	0.898	0.853	0.811	0.773	0.737	0.703	0.671	0.641	0.613	0.395	0.254	0.159
9.0	1.000	0.948	0.900	0.857	0.816	0.779	0.743	0.711	0.680	0.650	0.623	0.409	0.270	0.175
10.	1.000	0.949	0.903	0.860	0.820	0.784	0.749	0.717	0.687	0.658	0.631	0.422	0.284	0.188
20.	1.000	0.956	0.916	0.879	0.845	0.813	0.783	0.756	0.729	0.704	0.681	0.497	0.370	0.277
30.	1.000	0.959	0.922	0.888	0.856	0.827	0.800	0.774	0.750	0.726	0.705	0.534	0.415	0.325
40.	1.000	0.962	0.926	0.894	0.864	0.836	0.810	0.786	0.762	0.741	0.720	0.558	0.444	0.358
50.	1.000	0.963	0.929	0.898	0.869	0.843	0.818	0.794	0.772	0.751	0.731	0.574	0.464	0.381
60.	1.000	0.964	0.931	0.901	0.874	0.848	0.823	0.800	0.779	0.759	0.739	0.588	0.481	0.399
70.	1.000	0.965	0.933	0.904	0.877	0.851	0.828	0.806	0.785	0.765	0.746	0.598	0.494	0.414
80.	1.000	0.966	0.935	0.906	0.879	0.855	0.832	0.810	0.789	0.770	0.752	0.607	0.505	0.427
90.	1.000	0.966	0.935	0.906	0.882	0.857	0.835	0.813	0.793	0.774	0.756	0.614	0.514	0.437
100.	1.000	0.967	0.937	0.909	0.884	0.860	0.838	0.817	0.797	0.778	0.760	0.621	0.522	0.446
200.	1.000	0.970	0.943	0.918	0.895	0.874	0.854	0.835	0.817	0.800	0.784	0.658	0.569	0.500
300.	1.000	0.972	0.946	0.923	0.901	0.881	0.862	0.844	0.827	0.811	0.796	0.677	0.593	0.528
400.	1.000	0.973	0.948	0.926	0.905	0.886	0.867	0.850	0.834	0.819	0.804	0.690	0.609	0.546
500.	1.000	0.974	0.950	0.928	0.908	0.889	0.871	0.855	0.839	0.824	0.810	0.699	0.620	0.559
600.	1.000	0.974	0.951	0.930	0.910	0.891	0.874	0.858	0.842	0.828	0.814	0.706	0.629	0.569
700.	1.000	0.975	0.952	0.931	0.912	0.893	0.877	0.861	0.846	0.831	0.818	0.712	0.636	0.578
800.	1.000	0.975	0.953	0.932	0.913	0.895	0.879	0.863	0.848	0.834	0.821	0.716	0.642	0.585
900.	1.000	0.976	0.954	0.933	0.914	0.897	0.880	0.865	0.850	0.837	0.824	0.720	0.647	0.591
1000.	1.000	0.976	0.954	0.934	0.915	0.898	0.882	0.867	0.852	0.839	0.826	0.724	0.652	0.596

Table 6-1 (cont'd).

	$\frac{R}{a}$													
τ	6.0	6.0	8.0	9.0	10.	20.	30.	40.	50.	60.	70.	80.	90.	100.
0.001														
0.002														
0.003														
0.004														
0.005														
0.006														
0.007														
0.008														
0.009														
0.01														
0.02														
0.03														
0.04														
0.05														
0.06														
0.07														
0.08														
0.09														
0.1														
0.2														
0.3														
0.4														
0.5														
0.6														
0.7	0.000													
0.8	0.000													
0.9	0.000													
1.0	0.000	0.000												
2.0	0.005	0.001	0.000											
3.0	0.018	0.006	0.002	0.000										
4.0	0.034	0.014	0.005	0.002	0.000									
5.0	0.051	0.024	0.010	0.004	0.002	0.000								
6.0	0.067	0.035	0.017	0.008	0.003	0.000								
7.0	0.082	0.046	0.024	0.012	0.006	0.000								
8.0	0.097	0.056	0.032	0.017	0.009	0.000								
9.0	0.110	0.067	0.039	0.022	0.012	0.000								
10.	0.122	0.077	0.047	0.028	0.016	0.000								
20.	0.207	0.153	0.112	0.081	0.057	0.001	0.000							
30.	0.256	0.201	0.157	0.122	0.094	0.004	0.000							
40.	0.290	0.235	0.190	0.153	0.123	0.009	0.000							
50.	0.314	0.260	0.215	0.177	0.146	0.016	0.001	0.000						
60.	0.334	0.281	0.236	0.199	0.167	0.023	0.002	0.000						
70.	0.350	0.297	0.253	0.216	0.184	0.031	0.003	0.000						
80.	0.364	0.312	0.268	0.230	0.198	0.038	0.005	0.000						
90.	0.374	0.323	0.280	0.242	0.210	0.046	0.007	0.001	0.000					
100.	0.385	0.334	0.291	0.254	0.222	0.053	0.010	0.001	0.000	0.000				
200.	0.444	0.397	0.357	0.322	0.291	0.110	0.038	0.011	0.001	0.001	0.000			
300.	0.475	0.430	0.392	0.358	0.328	0.146	0.064	0.026	0.009	0.003	0.001	0.000		
400.	0.495	0.451	0.414	0.382	0.353	0.173	0.086	0.040	0.018	0.007	0.003	0.001	0.000	
500.	0.510	0.468	0.432	0.400	0.372	0.194	0.104	0.054	0.026	0.012	0.005	0.002	0.001	0.000
600.	0.520	0.479	0.444	0.413	0.385	0.210	0.119	0.066	0.035	0.018	0.008	0.004	0.002	0.001
700.	0.530	0.490	0.455	0.424	0.397	0.223	0.132	0.077	0.044	0.024	0.012	0.006	0.003	0.001
800.	0.538	0.498	0.464	0.434	0.407	0.235	0.143	0.087	0.052	0.030	0.016	0.009	0.004	0.002
900.	0.544	0.503	0.471	0.442	0.415	0.245	0.153	0.096	0.059	0.035	0.020	0.011	0.006	0.003
1000.	0.550	0.511	0.478	0.449	0.422	0.254	0.162	0.104	0.066	0.041	0.024	0.014	0.008	0.004

The use of this approximation technique is illustrated in sample problem 6 of Section 7.

6-05. Density Stratified Water Reservoirs. When the gross flow velocity of water through the reservoir is low the heat sink will not behave as a well mixed body of water but will become density stratified. The heated water recirculated from the condensers will form a distinct layer above the cooler water in the sink. As operation of the heat sink continues the lower cooler layer will be drained until it is completely replaced by the hotter recirculated water. When the cool water has been completely drained the sink outlet water temperature will suddenly increase to the temperature of the recirculated water. This behavior is illustrated in Figure 6-3. The sudden rise in temperature will be equal to the temperature difference being maintained across the condenser. The length of the stratification period, t_s, is related to the rate at which the water is recirculated through the condenser. The sloping transition period, of duration t_t, is related to mixing effects that occur at the interface of the cooler and warmer layers.

In a series of scale model tests[20,23,25] it was found that for the scale model, density stratification occurred when the Reynolds number based on diameter was less than 75. In addition it was found that for Reynolds numbers greater than 140 the sink behaved as a body of well mixed water. For the range of Reynolds numbers between 75 and 140 the sink exhibited a combination of these two characteristics.

Equation 6-09a should be used to calculate the Reynolds number based on diameter for reservoirs with a circular cross section. For reservoirs with a noncircular cross section equation 6-09b should be employed.

$$N_{RD} = \frac{5.12W}{R_o \nu} \qquad (6\text{-}09a)$$

where N_{RD} = Reynolds number based on diameter

W = coolant water flow rate, gpm

R_o = radius of the reservoir, ft

ν = kinematic viscosity of water, ft^2/hr

$$N_{RD} = \frac{U_m D_e}{\nu} \qquad (6\text{-}09b)$$

where U_m = mean flow velocity through the reservoir, ft/hr

D_e = equivalent hydraulic diameter, ft

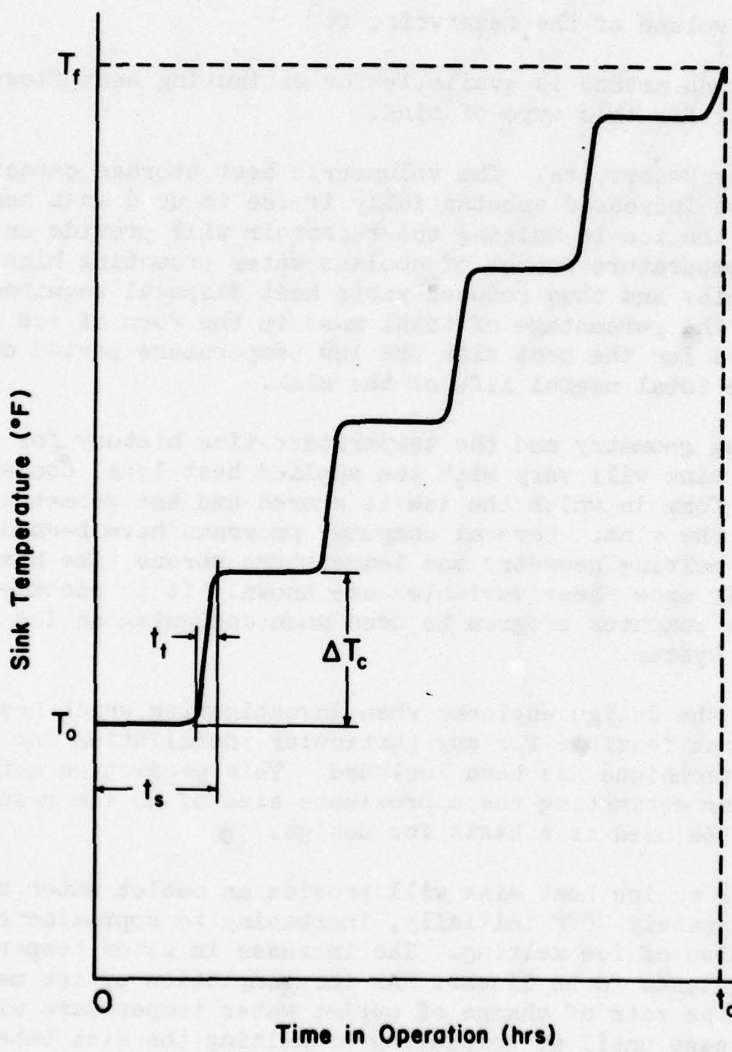

Figure 6-3. Typical behavior of a density stratified water reservoir.

The length of the stratified flow period may be approximated using equation 6-10. It is recommended that the transition period between temperature plateaus, t_t, be estimated as being at least ten percent of the length of the stratified flow period, t_s.

$$t_s = \frac{V_R}{8.02W} \tag{6-10}$$

where t_s = length of the stratified flow period, hr

V_R = volume of the reservoir, ft^3

At present no method is available for estimating heat flows to the surrounding rock for this type of sink.

6-06. Ice-Water Reservoirs. The volumetric heat storage capacity of a reservoir can be increased substantially if ice is used as a heat storage medium. While the ice is melting the reservoir will provide an almost constant low temperature source of coolant water promoting high power plant efficiencies and thus reduced waste heat disposal requirements. Depending upon the percentage of total mass in the form of ice and the operating limits for the heat sink the low temperature period can exceed one half of the total useful life of the sink.

The melting geometry and the temperature-time history for an ice reservoir type sink will vary with the applied heat load, coolant water flow rate, the form in which the ice is stored and the geometrical arrangement of the sink. Several computer programs have been developed to predict the melting geometry and temperature versus time history of an ice reservoir once these variables are known. It is recommended that the appropriate computer program be used when designing an ice reservoir type heat sink system.

To assist the design engineer when investigating which heat sink concepts are most feasible for any particular installation the following approximation technique has been included. This prediction method should only be used for estimating the approximate size of an ice reservoir and should not be used as a basis for design.

In general an ice heat sink will provide an outlet water temperature of approximately 38°F initially, increasing to approximately 46°F at the completion of ice melting. The increase in water temperature during melting tends to be linear. As the completion of ice melting is approached the rate of change of outlet water temperature will gradually increase until at completion of melting the sink behaves like an ordinary stored water reservoir.

As a first order approximation of the behavior it can be assumed that the sink outlet water temperature will increase linearly from 38°F to 46°F at completion of melting and thereafter behave like an ordinary water reservoir.

6-06.1. Constant Heat Rejection Rate: If the heat rejection rate remains constant then the overall useful life of the sink is given by:

$$t_d = \frac{158\ M_i + 14\ M_w}{q_o} + \frac{(M_i + M_w)c_w(T_f - 46)}{q_o} \tag{6-11a}$$

where M_i = initial mass of ice, lb

M_w = initial mass of water, lb

q_o = constant rate of heat rejection, Btu/hr

The time at which melting is completed, t_m, may be calculated using:

$$t_m = \frac{158M_i + 14M_w}{q_o} \tag{6-11b}$$

The sink outlet water temperature as a function of time can be calculated using:

$$T(t) = \frac{8}{t_m} t + 38 \qquad \text{for } t \leq t_m \tag{6-11c}$$

or

$$T(t) = \frac{q_o}{(M_i + M_w)c_w}(t - t_m) + 46 \qquad \text{for } t > t_m \tag{6-11d}$$

where $T(t)$ = sink outlet temperature at time t,°F

6-06.2. Linearly Increasing Heat Rejection Rate: If the waste heat load increases linearly with respect to heat sink water temperature in the form

$$q_r(T) = A_1 T + A_o$$

then the useful life of the ice reservoir is given by:

$$t_d = \frac{158M_i + 14M_w}{42A_1 + A_o} + \frac{c_w(M_i + M_w)}{A_1} \ln\left\{\frac{T_f + A_o/A_1}{46 + A_o/A_1}\right\} \tag{6-12a}$$

The time at which melting is completed is given by:

$$t_m = \frac{158M_i + 14M_w}{42A_1 + A_o} \tag{6-12b}$$

The sink temperature as a function of time is given by:

$$T(t) = \frac{8}{t_m} t + 38 \qquad \text{for } t \le t_m \tag{6-12c}$$

or

$$T(t) = \left[46 - \frac{A_o}{A_1}\right] e^{\left\{\frac{A_1(t-t_m)}{(M_i+M_w)c_w}\right\}} - \frac{A_o}{A_1} \quad \text{for } t > t_m \tag{6-12d}$$

6-06.3. Exponentially Increasing Heat Rejection Rate: If the waste heat load increases exponentially as a function of temperature in the form

$$q_r(T) = B_o e^{B_1 T}$$

then the useful life of the reservoir is given by:

$$t_d = \left[\frac{8B_1}{B_o}\right]\left[\frac{158M_i + 14M_w}{e^{46B_1} - e^{38B_1}}\right] + \frac{c_w[M_i+M_w]}{B_oB_1}\left[e^{-46B_1} - e^{-B_1T_f}\right] \tag{6-13a}$$

The time at which melting is completed is given by:

$$t_m = \left[\frac{8B_1}{B_o}\right]\left[\frac{158M_i+14M_w}{e^{46B_1} - e^{38B_1}}\right] \tag{6-13b}$$

The sink temperature as a function of time is given by:

$$T(t) = \frac{8}{t_m} t + 38 \qquad \text{for } t \le t_m \tag{6-13c}$$

and:

$$T(t) = -\frac{1}{B_1}\ln\left\{e^{-46B_1} - \frac{B_oB_1(t-t_m)}{(M_i+M_w)c_w}\right\} \quad \text{for } t > t_m \qquad (6\text{-}13d)$$

6-07. Density Stratified Ice Reservoirs. Density stratification may occur in ice reservoirs if the gross flow velocity of water through the reservoir is sufficiently small. If the ice reservoir is composed of an upright solid cylinder of ice subjected to annular melting the Reynolds number should be calculated based on an equivalent hydraulic diameter. For this geometry the correct relationship for calculating the Reynolds number is given in equation 6-14.

$$N_{RD} = \frac{16.04W}{\pi\nu(R_o+R_i)} \qquad (6\text{-}14)$$

where R_o = radius of the heat sink reservoir, ft

R_i = average radius of the ice cylinder, ft

The Reynolds number should be calculated for a series of decreasing ice cylinder diameters to determine the influence of ice melting. The criterion for determining if density stratification will occur is identical to that discussed in Section 6-05.

If density stratification will occur the behavior of the reservoir may be approximated by using either equation (6-11b), (6-12b) or (6-13b) to predict the time at which melting has been completed. Once melting has been completed the technique presented in Section 6-05 should be employed to determine the behavior of the sink for the remainder of its useful life.

6-08. Rock Tunnel Steam Condenser. Because of the complex nature of the heat transfer between an air-steam-vapor mixture and a rock surface no simplified prediction technique has been developed yet. A computer program is available from the U.S. Army Cold Regions Research and Engineering Laboratory (CRREL) which can be used to approximate the performance of this type of sink.

6-09. Earth Heat Sinks. The performance of an earth type heat sink is intimately related to the type of backfill employed, location of the water table, size and spacing of the pipe network and operating temperatures.

A computer program is available from the Naval Civil Engineering Laboratory to predict the performance of earth type heat sinks located in unfrozen ground. Heat sink systems utilizing frozen ground can be analyzed using a computer program available from CRREL.

6-10. Water Cooling Loads. The use of either a chilled water system or an ice reservoir system will require refrigeration equipment to cool both the heat sink medium and the surrounding rock. If the sink system is designed so that the reservoir is not in good thermal contact with the surrounding media then the cooling load for the water alone is given by:

$$q_c = \frac{M_t c_w (T_i - T_o)}{12000\ t_p} + \frac{q_p}{12000} \qquad (6\text{-}15)$$

where
q_c = total water cooling load, tons
M_t = total mass of water in the reservoir, lb
T_i = temperature of the source water, ^oF
T_o = design temperature of the sink, ^oF
t_p = length of the cool down period, hr
q_p = frictional pumping heat load, Btu/hr

If the reservoir does have good thermal contact with the surrounding rock then the cooling load for both the water and the surrounding rock during the cool down period can be calculated using the parameters defined by equations 6-16a, b and c and either Figure 6-1 or Figure 6-2.

$$f(\tau,G) = \frac{k(T_\infty - T_o)\ell}{12{,}000\ q_c} \qquad (6\text{-}16a)$$

$$G = \frac{2\pi a^2 \ell \rho c}{M_T c_w} \qquad (6\text{-}16b)$$

$$\tau = \frac{k\ t_p}{\rho c a^2} \qquad (6\text{-}16c)$$

Once the reservoir has attained the desired initial temperature the maintenance cooling load as a function of time may be calculated using:

$$q_m = \frac{k(T_\infty - T_o)\ell I(\tau)}{12{,}000} \qquad (6\text{-}17)$$

where: q_m = maintenance cooling load, tons

k = thermal conductivity of the surrounding rock, Btu/hr ft °F

T_∞ = initial ambient rock temperature, °F

ℓ = length of the reservoir, ft

$\tau = kt/\rho ca^2$

t = time from start of cooling, hr

$I(\tau)$ is from Figure 6-4

6-11. Rock Cooling Loads: If the heat sink reservoir is located in a rock chamber and maintained at a temperature below the ambient but good thermal contact between the reservoir and the rock does not exist then equation 6-18 should be used to determine the amount of cooling required to maintain the chamber interior surface temperature at the desired sink temperature.

$$q_s = \frac{\ell k(T_\infty - T_o)\alpha(\tau)}{1500\pi} \qquad (6\text{-}18)$$

where q_s = rock cooling load, tons

a = diameter of the rock chamber, ft

ℓ = length of the rock chamber, ft

$\tau = kt/\rho ca^2$

T_o = rock surface temperature to be maintained, °F

t = time from the start of cooling, hr

$\alpha(\tau)$ is from Table 6-2

No provision is made in this approximation to include any latent heat loads which might result from freezing water in the rock.

6-12. Ice Freezing: The use of a solid ice type heat sink will require sufficient refrigeration capacity to form the ice. Two methods of creating the solid ice are presented, the freezing of 32°F water and the use of ice blocks to reduce the length of the freezing period and the cooling loads encountered.

6-12.1. Freezing of Chilled Water: If the reservoir is cylindrical in shape and filled with 32°F water with cooling along the outside walls only then the time required to completely freeze the water is given by:

$$t_f = \frac{\rho_i \lambda\, R_o^2}{4k_i(32 - T_\omega)} \qquad (6\text{-}19a)$$

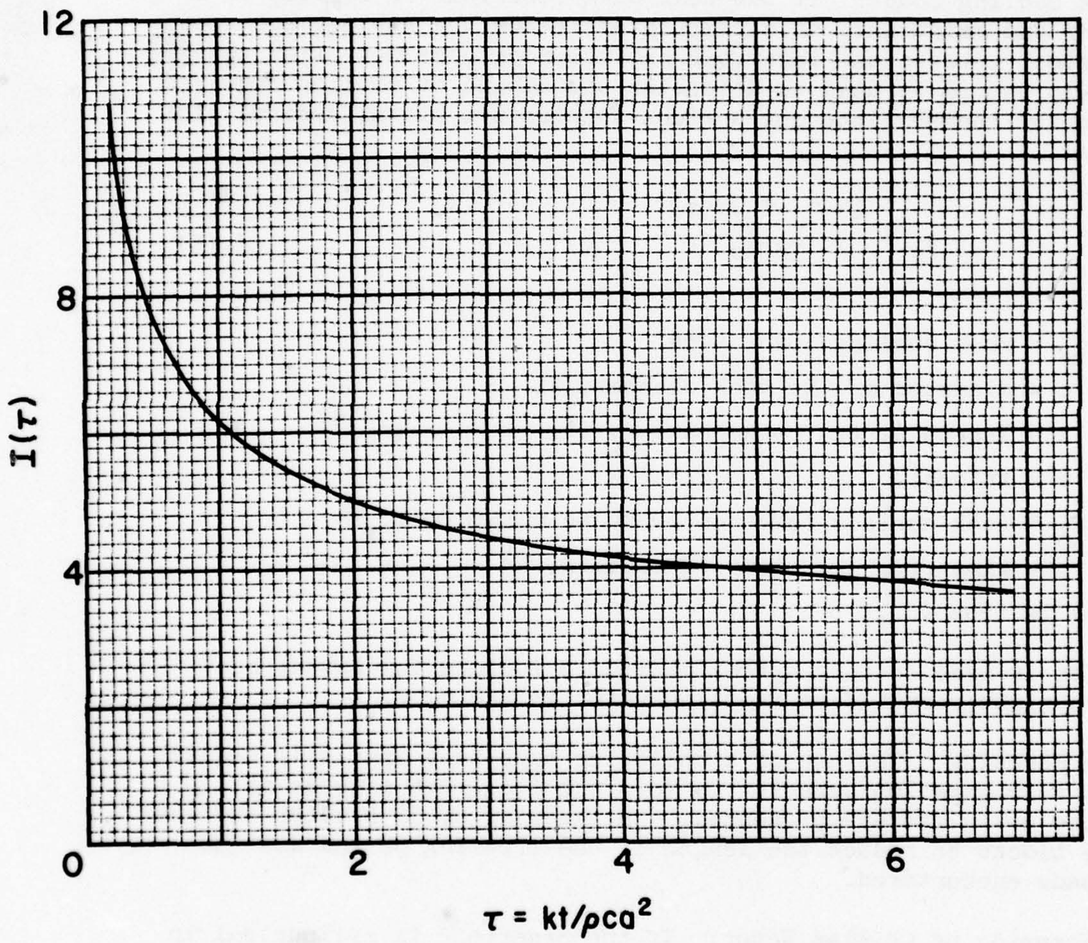

Figure 6-4. Values for function $I(\tau)$ as a function of τ.

Table 6-2. Numerical values for function $\alpha(\tau)$. (From reference 55)

$\alpha(\tau)$

τ	.00	.01	.02	.03	.04	.05	.06	.07	.08	.09
0.0	∞	15.122	11.033	9.218	8.135	7.394	6.846	6.421	6.076	5.790
0.1	5.549	5.340	5.158	4.998	4.854	4.726	4.609	4.503	4.405	4.315
0.2	4.232	4.155	4.083	4.016	3.953	3.894	3.838	3.785	3.735	3.688
0.3	3.643	3.600	3.559	3.520	3.482	3.446	3.412	3.379	3.348	3.317
0.4	3.288	3.259	3.232	3.206	3.180	3.156	3.132	3.109	3.086	3.065
0.5	3.044	3.023	3.003	2.984	2.965	2.947	2.929	2.912	2.895	2.878
0.6	2.862	2.847	2.831	2.816	2.802	2.787	2.773	2.760	2.746	2.733
0.7	2.720	2.708	2.695	2.683	2.671	2.660	2.648	2.637	2.626	2.616
0.8	2.605	2.595	2.584	2.574	2.565	2.555	2.545	2.536	2.527	2.518
0.9	2.509	2.500	2.492	2.483	2.475	2.467	2.459	2.451	2.443	2.435

	0.0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
1.0	2.427	2.357	2.295	2.240	2.191	2.147	2.106	2.069	2.036	2.004
2.0	1.975	1.948	1.923	1.900	1.877	1.856	1.837	1.818	1.800	1.783
3.0	1.767	1.752	1.737	1.723	1.710	1.697	1.684	1.673	1.661	1.650
4.0	1.639	1.629	1.619	1.610	1.600	1.591	1.582	1.574	1.566	1.558
5.0	1.550	1.542	1.535	1.528	1.521	1.514	1.508	1.501	1.495	1.489
6.0	1.483	1.477	1.471	1.465	1.460	1.455	1.449	1.444	1.439	1.434
7.0	1.429	1.425	1.420	1.415	1.411	1.406	1.402	1.398	1.394	1.390
8.0	1.386	1.382	1.378	1.374	1.370	1.367	1.363	1.359	1.356	1.352
9.0	1.349	1.346	1.342	1.339	1.336	1.333	1.329	1.326	1.323	1.320

	0.	1.	2.	3.	4.	5.	6.	7.	8.	9.
10.	1.317	1.290	1.266	1.244	1.225	1.207	1.191	1.176	1.162	1.150
20.	1.138	1.127	1.117	1.107	1.098	1.089	1.081	1.073	1.065	1.058
30.	1.052	1.045	1.039	1.033	1.027	1.022	1.016	1.011	1.006	1.001
40.	0.997	0.992	0.988	0.984	0.980	0.976	0.972	0.969	0.965	0.961
50.	0.958	0.955	0.951	0.948	0.945	0.942	0.939	0.936	0.934	0.931
60.	0.928	0.926	0.923	0.921	0.918	0.916	0.913	0.911	0.909	0.906
70.	0.904	0.902	0.900	0.898	0.896	0.894	0.892	0.890	0.888	0.886
80.	0.884	0.883	0.881	0.879	0.877	0.876	0.874	0.872	0.871	0.869
90.	0.868	0.866	0.865	0.863	0.862	0.860	0.859	0.857	0.856	0.854

	0.	10.	20.	30.	40.	50.	60.	70.	80.	90.
100.	0.853	0.841	0.830	0.820	0.810	0.802	0.794	0.787	0.780	0.774
200.	0.768	0.762	0.757	0.752	0.748	0.743	0.739	0.735	0.732	0.728
300.	0.725	0.721	0.718	0.715	0.712	0.709	0.707	0.704	0.702	0.699
400.	0.697	0.694	0.692	0.690	0.688	0.686	0.684	0.682	0.680	0.678
500.	0.676	0.675	0.673	0.671	0.670	0.668	0.667	0.665	0.664	0.662
600.	0.661	0.659	0.658	0.657	0.655	0.654	0.653	0.652	0.650	0.649
700.	0.648	0.647	0.646	0.645	0.643	0.642	0.641	0.640	0.639	0.638
800.	0.637	0.636	0.635	0.634	0.633	0.632	0.632	0.631	0.630	0.629
900.	0.628	0.627	0.626	0.626	0.625	0.624	0.623	0.622	0.622	0.621
1000.	0.620									

where t_f = time required to freeze the ice, hr

ρ_i = density of ice, lb/ft^3

λ = latent heat of fusion, 144 Btu/lb

R_o = radius of the reservoir, ft

k_i = thermal conductivity of ice, Btu/hr ft°F

T_ω = wall temperature, °F

To calculate the thickness of the ice annulus formed as a function of time, use equation (6-19b):

$$t = \frac{\rho_i \lambda R_o^2}{4k_i(32-T_\omega)} \left[1-\left(\frac{R_a}{R_o}\right)^2 \left\{1-2\ln\left(\frac{R_a}{R_o}\right)\right\}\right] \qquad (6\text{-}19b)$$

where R_a = radius of the ice annulus at time t, ft

In using this equation it is necessary to assume an ice annulus and then calculate the time required to form it. To calculate the cooling load required as a function of time equation (6-19c) should be employed:

$$q_c = \frac{2\pi \ell k_i(32-T_\omega)}{12000 \ln\left(\frac{R_o}{R_a}\right)} \qquad (6\text{-}19c)$$

where q_c = cooling load, tons

ℓ = length of the reservoir

Cooling loads must be calculated with respect to the radius of the ice annulus formed; correlation of the cooling load with time can be made through the use of equation (6-19b).

6-12-2. Freezing using Ice Blocks and Chilled Water: To reduce the length of the freezing period and the cooling load requirements ice particles can be added to the reservoir prior to the start of freezing. For cylindrical reservoirs where a significant amount of ice has been added it has been found that as a first order approximation equations 6-19 a,b, and c can be used to calculate the overall freezing period, rate of freezing and cooling loads with the following modification:

For a preload of a specific mass of ice an equivalent ice annulus should be calculated using equation (6-20a):

$$R_e = \sqrt{R_o^2 - \frac{M_L}{\pi \rho_i \ell}} \qquad (6\text{-}20a)$$

where R_e = equivalent ice annulus radius, ft

M_L = mass of ice preloaded, lb

ℓ = length of the reservoir, ft

This equivalent radius should then be used in equation (6-20b) to determine the time required to freeze this mass of ice:

$$t_e = \frac{\rho_i \lambda R_o^2}{4k_i(32-T_\omega)} \left[1 - \left(\frac{R_e}{R_o}\right)^2 \left\{1 - 2\, ln \left(\frac{R_e}{R_o}\right)\right\}\right] \qquad (6\text{-}20b)$$

where t_e = equivalent time to freeze radius R_e, hr

The length of the total freezing period can then be calulated using equation (6-20c):

$$t_f = \frac{\rho_i \lambda R_o^2}{4k_i(32-T_\omega)} - t_e \qquad (6\text{-}20c)$$

The variation in the cooling load with respect to temperature can then be calculated using equation (6-20d) to calculate hypothetical radii as a function of time and then (2-20e) to calculate the cooling load for this hypothetical radius:

$$t = \frac{\rho_i \lambda R_o^2}{4k_i(32-T_\omega)} \left[1- \left(\frac{R_h}{R_o}\right)^2 \left\{1-2\, \ell n \left(\frac{R_h}{R_o}\right)\right\}\right] -t_e \qquad (6\text{-}20d)$$

where R_h = hypothetical radius, ft, $R_h \leq R_e$

t_e = equivalent time to freeze annulus R_e

$$q_c = \frac{2\pi \ell k_i (32-T_w)}{12000\, ln\left(\frac{R_o}{R_e}\right)} \qquad (6\text{-}20e)$$

SECTION 7 -- SAMPLE PROBLEMS

7-01. Introduction. This section presents a series of sample problems to illustrate the use of the computational procedures presented in Section 6. Table 7-1 summarizes the sample problems and provides a reference guide between each problem and the location in Section 6 where the applicable equations are discussed.

The problems have been arranged in order of increasing difficulty. The level of sophistication of the calculations is equivalent to that which would typically be used for preliminary sizing and concept evaluation. Actual in-depth design calculations would have to be performed in significantly greater detail than the sample problems and thus are beyond the scope of this report.

Table 7-1. Summary of the sample problems.

Problem No.	Type of Heat Sink	Location of the Design Relationships
1	90° water reservoir with once-through cooling	Sect. 6-02
2	40°F water reservoir with once-through cooling	Sect. 6-02
3	90°F recirculated water reservoir with no heat transfer to surroundings	Sect. 6-03.1
4	40°F recirculated water reservoir with no heat transfer to surroundings	Sect. 6-03.1
5	90°F recirculated water reservoir with heat transfer to surroundings	Sect. 6-03.2
6	40°F recirculated water reservoir with heat transfer to surroundings	Sect. 6-03.2
7	50% Ice/50% Water heat sink	Sect. 6-06
8	90% Ice/10% Water heat sink	Sect. 6-06

Detailed information concerning this power system can be found in Appendix A.

7.02. Cost Estimates. Rough cost estimates have been included in the sample problems to illustrate the relative economics of the different types of heat sinks. Appendix B lists the sources used to develop the unit prices utilized in the sample problems. Although considerable effort was expended to develop these cost data, no claim as to their accuracy or current applicability is expressed or implied.

7-03. Power Plant. To provide a common basis for comparison the same power plant is utilized in all the sample problems. The power plant relationships are based on a conceptual design for a nuclear steam turbine power plant developed by the U.S. Army Engineer Reactor Group.

7-04. General Design Criteria. The sample problems have been prepared in accordance with the following assumed general design criteria:

a. The installation must be able to perform its mission for a period of 30 days.

b. The heat sink system shall have a minimum design capacity for a 31.5-day full load operational period. This is to provide for a 30-day mission, a 3% (1-day) safety factor and 0.5 day system testing period.

c. The installation is located at a depth of approximately 4000 feet. The ambient temperature of the surrounding rock is assumed to be 90°F.

d. The sample problem heat sinks shall be required to absorb only the waste heat produced by the power plant. It is assumed that waste heat produced by other equipment such as air conditioning units, electronic equipment and so forth will be rejected to a different heat sink system.

e. The heat sink reservoirs will be in the form of vertical cylinders with hemispherical ends. The ratio of overall length to overall diameter is to be approximately 5:1. The heat sink reservoirs shall not exceed maximum dimensions of 250 feet in length and 50 feet in diameter.

f. The heat sink reservoirs shall be constructed using 2-foot-thick reinforced concrete walls. All interior surfaces shall be coated with two coats of tar-based waterproofing.

g. All heat sink reservoirs, except the ice/water type, shall be completely filled. The upper dome of the ice/water type heat sink reservoirs shall be used to house refrigeration equipment. Appendix C gives the volumetric relationships for the two types of heat sink reservoirs.

h. All equipment required to fill/cool the heat sink system following construction shall be considered to be expendable with no salvage value. The "total" cost of each heat sink system shall include, when appropriate, maintenance costs for a 15-year-long standby period.

i. The overall facility is assumed to have a 15-year useful life following construction and heat sink fill/cool operations. A period of three years between construction completion and beneficial occupancy shall be allowed, when required, to fill/cool the heat sink system.

j. Maintenance staff salaries are assumed to inflate at 5% per annum. Procurement costs for supplies and services are assumed to inflate at 8.3% per annum.

k. Electrical power and water required to fill/cool the heat sink system initially and for subsequent standby maintenance cooling are assumed to be purchased from commercial sources.

l. Contractor contingencies, markup and profit are not included. All costs have been rounded to the nearest $1,000.

7-05. Excavation Costs. Although the sample problems utilize a simplified unit cost factor for excavation of the heat sink reservoirs, a more detailed analysis of excavation costs is provided in Appendix D.

7-06. Unlimited Capacity Type Heat Sinks. In general, unlimited capacity type heat sinks are very dependent upon site conditions and equipment utilized. Therefore, no sample problems for unlimited capacity type heat sinks have been included in this section. However, a sample problem which illustrates a typical design procedure for this type heat sink has been included in Appendix E for reference purposes.

7-07. Sample Problems. The sample problems are presented in the following sections. Each sample problem contains a short discussion of the problem, the engineering calculations required for solution and, where appropriate, a cost estimate. In those problems where several iterations were required before a solution could be found only the first and last iteration are presented in detail. Intermediate iterations are referenced but not included.

7-07.1 Sample Problem 1: Once-Through Cooling Using a 90°F Water Reservoir System. This problem considers a heat sink system where ambient temperature water is drawn from the reservoirs, passed through the steam condenser one time and then wasted. The basic design problem is to determine the minimum volumetric capacity of the water reservoir system that would allow this mode of cooling to be used throughout the mission period.

Since the water from the condenser will not be recirculated back to the water reservoirs the temperature of the water reservoirs will remain constant and equal to the ambient rock temperature of 90°F. Thus, with the heat sink temperature remaining constant at 90°F the following parameters can be determined immediately from Table A-3:

At a 90°F heat sink temperature:

condenser hotwell = 110°F
power plant waste heat load = 27.7×10^6 Btu/hr
condenser coolant flow rate = 4240 gpm
condenser coolant leaving temperature = 103°F
condenser coolant ΔT = 13°F.

Since both the length of the mission and the condenser coolant water flow rate are known, the total minimum water reservoir volume, V_T, can be found by multiplying these two factors. Thus:

$$V_T = (4240 \text{ gpm})(60 \frac{\text{min}}{\text{hr}})(24 \frac{\text{hr}}{\text{day}})(31.5 \text{ days})(0.134 \frac{\text{ft}^3}{\text{gal}})$$

thus $\quad V_T = 25.77 \times 10^6 \text{ ft}^3$.

Since the total volume of the water reservoir system has now been determined, the approximate number of heat sink reservoirs can be determined by dividing the total volume required by the maximum allowable volume of each individual reservoir. Since these are water reservoirs and will be completely filled, the volume of each reservoir is given by (see Appendix C):

$$V_R = \frac{28}{3} \pi R^3$$

where R is the radius of the reservoir, ft.

Since each sink cannot be larger than 50 feet in diameter the maximum reservoir volume is given by:

$$V_R = \frac{28}{3} \pi (25 \text{ ft})^3 = 4.58 \times 10^5 \text{ ft}^3$$

Therefore the number of heat sink reservoirs can be determined from:

$$\text{number of reservoirs} = \frac{V_T}{V_R} = \frac{25.77 \times 10^6 \text{ ft}^3}{4.58 \times 10^5 \frac{\text{ft}^3}{\text{reservoir}}}$$

and thus number of reservoirs = 56.3.

Although some manipulation of the value of "R" could be performed to round out the number of reservoirs to an even total of 57 it is apparent that this excessively high number makes this heat sink system unfeasible.

7-07.2. Sample Problem 2: Once-Through Cooling Using a 40°F Water Reservoir System. This problem is identical to the first except that this system uses water which has been cooled to 40°F rather than water at 90°F. This reduction in the coolant water temperature will increase the efficiency of the power plant and thus reduce the waste heat load. This will in turn reduce the size of the heat sink system. From Appendix A:

At a 40°F heat sink temperature:

condenser hotwell = 60°F
power plant waste heat load = 23.3×10^6 Btu/hr
condenser coolant flow rate = 3345 gpm
condenser coolant leaving temperature = 53.9°F
condenser coolant ΔT = 13.9°F

Thus, using the procedure of problem no. 1, the total minimum heat sink system volume can be calculated by:

$$V_T = (3345 \text{ gpm})(60 \frac{\text{min}}{\text{hr}})(24 \frac{\text{hr}}{\text{day}})(31.5 \text{ days})(0.134 \frac{\text{ft}^3}{\text{gal}})$$

and thus $V_T = 20.33 \times 10^6 \text{ ft}^3$.

Similarly, since the maximum volume of any particular reservoir is given by (see problem 1):

$$V_R = 4.58 \times 10^5 \text{ ft}^3$$

the required number of reservoirs can be found by:

$$\text{number of reservoirs} = \frac{V_T}{V_R} = \frac{20.33 \times 10^6 \text{ ft}^3}{4.58 \times 10^5 \ \frac{\text{ft}^3}{\text{reservoir}}}$$

number of reservoirs = 44.4

Again it can be concluded, as in the first problem, that this concept is not feasible due to the excessive number of heat sinks required.

7-07.3. Sample Problem 3: Recirculated 90°F Water Reservoir with No Heat Transfer to Surroundings. Unlike problems 1 and 2, where the coolant water was wasted after being circulated through the condenser, in this system the heated coolant water is returned to the reservoir and reused. As a result, the temperature of the water in the heat sink will increase as a function of time in operation until it reaches 160°F, at which time the power plant must be shut down (see Appendix A).

In the statement of the problem it was assumed that no heat transfer between the heat sink reservoirs and the surrounding rock would occur. This model represents a heat sink system wherein direct thermal contact between the reservoir and the surrounding media does not exist.

The basic design problem for this example is to determine the minimum heat sink volume that will keep the cooling water from attaining a temperature of 160°F prior to completion of the mission.

The basic parameters of this problem are that initially the sink temperature will be 90°F and at completion of mission the temperature will be 160°F. Using the exponential approximation of the power plant waste heat load (i.e. equation A-10) the heat load as a function of heat sink temperature is:

$$q_r = 20.125\text{x}10^6 \; e^{3.512\text{x}10^{-3} T_{sink}}$$

Since this is in the format of:

$$q_r(T) = B_o e^{B_1 T}$$

equation 6-05b, namely

$$M_w = \frac{B_o B_1 t_d}{c_w \left[e^{-B_1 T_o} - e^{-B_1 T_f} \right]}$$

can be utilized to determine the required mass of water. For this calculation the values of the various parameters are:

$B_o = 20.125\text{x}10^6$ Btu/hr

$B_1 = 3.512\text{x}10^{-3}$ 1/°F

$T_i = 90°F$

$T_f = 160^oF$

$c_w = 1.0$ Btu/lb^oF

$t_d = 31.5$ days = 756 hours.

Thus:

$$M_w = \frac{(20.125\times10^6\ \text{Btu/hr})(3.512\times10^{-3}\ 1/^oF)(756\ \text{hr})}{\left(1.0\ \text{Btu/lb}^oF\right)\left[\left(e^{-(90^oF)(3.521\times10^{-3}\ 1/^oF)} - e^{-(160^oF)(3.512\times10^{-3}\ 1/^oF)}\right]}$$

and therefore:

$$M_w = 3.363\times10^8\ \text{lbm}.$$

Since at 90^oF $\rho_w = 62.1$ lb/ft^3 the total reservoir volume is given by

$$V_T = \frac{M_w}{\rho_w} = \frac{3.363\times10^8\ \text{lb}}{62.1\ \text{lb/ft}^3}$$

and therefore $V_T = 5.415\times10^6\ \text{ft}^3$.

Since the maximum volume of any one heat sink reservoir (see problem no. 1) is given by:

$$V_R = 4.58\times10^5\ \text{ft}^3$$

the number of heat sink reservoirs required can be calculated using:

$$\text{number of reservoirs} = \frac{V_T}{V_R} = \frac{5.415\times10^6\ \text{ft}^3}{4.58\times10^5\ \text{ft}^3/\text{reservoir}}$$

and therefore:

$$\text{number of reservoirs} = 11.8.$$

It is relatively easy to demonstrate that if 12 reservoirs are used then since

$$V_T = 12\ V_R$$

and

$$V_R = \frac{28}{3}\pi R^3$$

each reservoir must have a radius of 24.9 ft to provide the required 5.415×10^6 ft^3 of water storage capacity. Thus 12 reservoirs, each 49.8 ft in diameter by 249 ft long (see Appendix C), filled with 90°F water will provide the required heat storage capacity.

Using equation 6-05c, namely

$$T = -\frac{1}{B_1} \ln \left[e^{-B_1 T_o} - \frac{B_o B_1 t}{M_w c_w} \right]$$

the variation in heat sink temperature as a function of time in operation can be determined. Substituting into this relationship yields:

$$T = -\frac{1}{3.512 \times 10^{-3}\ 1/°F} \ln \left[e^{-(90°F)(3.512 \times 10^{-3}\ 1/°F)} - \frac{(20.125 \times 10^6\ Btu/hr)(3.512 \times 10^{-3}\ 1/°F)t}{(3.363 \times 10^8\ lbs)(1.0\ Btu/lb°F)} \right]$$

and thus

$$T = (-284.7°F) \ln \left[0.729 - 2.10 \times 10^{-4}\ t\ 1/hr\right]$$

where t is the time in operation in hours. Using this relationship the variation in heat sink temperature as a function of time in operation is presented below. These data have also been plotted on Figure 7-1. Using the information provided in Appendix A, Figure 7-1 also gives the required coolant water flow rate as a function of time.

Time in Operation (days)	Heat Sink Temperature (°F)
0	90.0
2.5	95.0
5.0	100.0
7.5	105.2
10.0	110.4
12.5	115.8
15.0	121.2
17.5	126.8
20.0	132.4
22.5	138.2
25.0	144.1
27.5	150.1
30.0	156.2
31.5	160.0

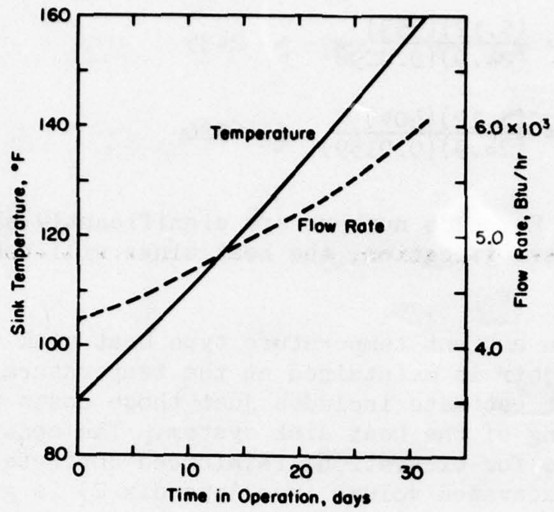

Figure 7-1. Performance of the sample problem 3 heat sink system.

Since it is possible that with all sinks in operation simultaneously the water flow in the heat sink would be density-stratified, it is worthwhile to calculate Reynolds numbers for the minimum and maximum anticipated coolant water flow rates. Using either Figure 7-1 or Appendix A the minimum and maximum coolant flow rates are, respectively:

minimum: $G = 4240$ gpm when $T_{sink} = 90°F$

maximum: $G = 6045$ gpm when $T_{sink} = 160°F$

Assuming that the coolant water is divided equally among the sinks, then each sink will receive one twelfth of the total flow. Thus the minimum and maximum coolant water flow rates for each heat sink are:

minimum: $W_R = 353$ gpm

maximum: $W_R = 504$ gpm.

Using equation 6-09a, namely

$$N_{RD} = \frac{5.12\ W}{R\nu}$$

and that:

at 90°F, $\nu = 0.0298\ ft^2/hr$

and at 160°F, $\nu = 0.0159\ ft^2/hr$

then

$$\text{minimum } N_{RD} = \frac{(5.12)(353)}{(24.9)(0.0298} \approx 2435$$

$$\text{maximum } N_{RD} = \frac{(5.12)(504)}{(24.9)(0.0159)} \approx 6520$$

Thus, since these Reynolds numbers are significantly above those required for density stratification, the heat sinks will behave as well mixed bodies of water.

Because this is an ambient temperature type heat sink (i.e. the water within the reservoir is maintained at the temperature of the surrounding rock) the cost estimate includes just those costs related to construction and filling of the heat sink system. The construction costs include the costs for excavation, reinforced concrete and waterproofing. The total excavated volume (see Appendix C) is given by:

$$V_E = (12)(\frac{28}{3})(\pi)(24.9+2)^3 \text{ ft}^3$$

thus

$$V_E = 6.85\text{x}10^6 \text{ ft}^3 = 2.54\text{x}10^5 \text{ yd}^3.$$

The required volume of reinforced concrete (see Appendix C) is given by:

$$V_c = (12)(\frac{28}{3})(\pi)[(24.9 + 2)^3 - (24.9)^3] \text{ ft}^3$$

$$V_c = 1.42\text{x}10^6 \text{ ft}^3 = 5.25\text{x}10^4 \text{ yd}^3.$$

The internal surface area of the heat sink reservoirs is given by:

$$A = (12)(2.)(\pi)(24.9)^2 \text{ ft}^2$$

or

$$A = 4.68\text{x}10^5 \text{ ft}^2.$$

The total volume of water required is $5.42\text{x}10^6$ ft^3, which is equivilent to $4.05\text{x}10^7$ gallons.

The estimated cost for this heat sink is shown in Table 7-2. The unit costs for each item can be found in Appendix B.

Table 7-2: Cost estimate for sample problem 3.

Item No.	Item Description	Unit Cost	Quantity	Item Cost	Subtotals
A	CONSTRUCTION				
1	Excavation	$ 75/yd^3	2.54×10^5 yd^3	$19,100,000.	
2	Reinforced concrete, in place	120/yd^3	5.25×10^4 yd^3	6,300,000.	
3	Waterproofing	0.31/ft^2	4.68×10^5 ft^2	145,000.	
					$25,454,000
B	PREPARATION				
1	Water	$0.04/100 gal.	4.05×10^7 gal	$ 16,000.	
					$ 16,000
				TOTAL COST	$25,561,000

7-07.4. Sample Problem 4: Recirculated 40°F Water Reservoir with No Heat Transfer to Surroundings. Except for the initial water temperature this heat sink system is identical to the one in problem 3. Thus the same procedure is used to determine the minimum required heat sink size. However, since this system must be cooled to a temperature below the normal ambient, refrigeration requirements for initial and maintenance cooling must be considered.

Therefore, if, as in problem 3, the exponential form of the approximated power plant waste heat load,

$$q_r = 20.125\times10^6 \, e^{3.512\times10^{-3} T_{sink}} \text{ Btu/hr}$$

is used, then from equation 6-05b,

$$M_w = \frac{B_o B_1 t_d}{c_w \left[e^{-B_1 T_o} - e^{-B_1 T_f} \right]}$$

with
$B_o = 20.125\times10^6$ Btu/hr
$B_1 = 3.512\times10^{-3}$ 1/°F
$t_d = 31.5$ days = 756 hr
$c_w = 1.0$ Btu/lb°F
$T_o = 40°F$
$T_f = 160°F$

it can be shown that the required mass of water, M_w, is

$$M_w = 1.788 \times 10^8 \text{ lb.}$$

Thus, since at 40°F $\rho_w = 62.4 \text{ lb/ft}^3$ the total reservoir system volume, V_T, is given by:

$$V_T = \frac{M_w}{\rho_w} = \frac{1.788 \times 10^8 \text{ lb}}{62.4 \text{ lb/ft}^3}$$

and thus $V_T = 2.87 \times 10^6 \text{ ft}^3$.

Since the volume of each heat sink reservoir is given by

$$V_R = \frac{28}{3} \pi R^3$$

it can be easily shown that seven, reservoirs each with a radius of 24.1 ft, will provide the required volume. Thus seven reservoirs 48.2 ft in diameter by 241 ft long with an initial temperature of 40°F will provide the required heat storage capacity.

Using equation 6-05c, namely

$$T = -\frac{1}{B_1} \ln \left[e^{-B_1 T_o} - \frac{B_o B_1 t}{M_w c_w} \right]$$

the variation in heat sink temperature as a function of time in operation can be determined. Upon substitution of the appropriate constants from the preceding calculations it can be shown that this equation reduces to:

$$T = -287.4 \ln [0.869 - 9.49 \times 10^{-3} t] \text{ °F}$$

where t is the time in operation in days. Using this relationship the following data, which present the heat sink system temperature as a function of time, were developed:

Time in Operation (days)	Heat Sink Temperature (°F)	Time in Operation (days)	Heat Sink Temperature (°F)
0	40.0	17.5	100.4
2.5	47.9	20.0	110.1
5.0	56.0	22.5	120.3
7.5	64.3	25.0	130.8
10.0	72.9	27.5	141.7
12.5	81.8	30.0	153.0
15.0	90.9	31.5	160.0

These results are plotted on Figure 7-2. From these results and Appendix A, Table A-3, the variation in coolant water flow rate as a function of time has also been determined and is plotted on Figure 7-2. Thus, during the operating period the coolant water flow rate will have to be increased as shown on Figure 7-2 from an initial rate of 3345 gpm to 6045 gpm by the end of the mission.

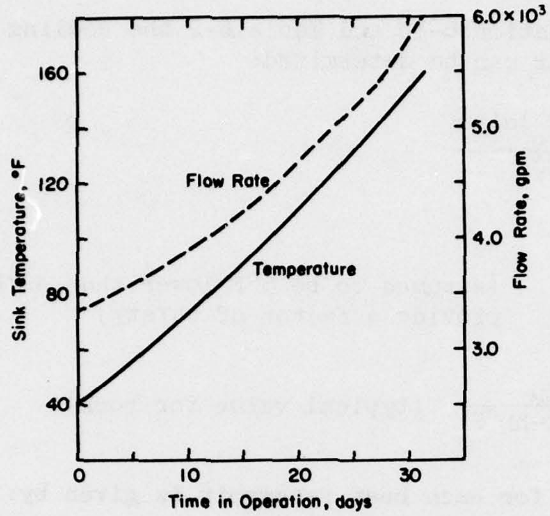

Figure 7-2. Performance of the sample problem 4 heat sink system.

Using the minimum coolant flow rate of 3345 gpm the potential for density stratification can be examined. For a seven-reservoir heat sink system, the minimum flow rate per reservoir, W_R, is:

$$\text{minimum } W_R = \frac{3345 \text{ gpm}}{7} = 478 \text{ gpm.}$$

Thus, since at 40°F ν = .0601 ft^2/hr, equation 6-09a,

$$N_{RD} = \frac{5.12\ W}{R\nu}$$

yields a Reynolds number of:

$$N_{RD} = \frac{(5.12)(478)}{(24.1)(.0601)} = 1690.$$

Since the Reynolds number is well above the range in which density stratification occurs it can be concluded that the reservoirs will behave as well mixed bodies of water.

Since the heat sink system is to have an initial temperature of 40°F and the rock surrounding the facility has an ambient temperature of 90°F, refrigeration will be required to cool the system and maintain it at 40°F during the standby mode. The initial cooling of the heat sink will involve cooling of the surrounding rock and the water within the reservoirs. The magnitude of the rock cooling load will be examined first.

Through the use of equation 6-18 and Table 6-2 the cooling load for each heat sink reservoir can be determined:

$$q_s = \frac{\ell k(T_\infty - T_o)\alpha(\tau)}{1500\pi}$$

and

$T_\infty = 90°F$

$T_o = 35°F$ (assumed to be 5°F lower than 40°F to provide a factor of safety)

$\ell = 241$ ft

$k = 1.46 \frac{Btu}{ft\text{-}hr°F}$ (typical value for rock)

then the rock cooling load for each heat reservoir is given by:

$$q_s = 4.11\ \alpha(\tau) \quad \text{, tons}$$

Since the function $\alpha(\tau)$ is tabulated in Table 6-2 versus the variable τ, the relationship:

$$\tau = \frac{kt}{\rho c a^2}$$

must be used to determine the rock cooling load as a function of time. Using:

$k = 1.46 \frac{Btu}{lb\ hr°F}$

$\rho = 186 \frac{lb}{ft^3}$ (typical value for rock)

$c = 0.2 \frac{Btu}{lb°F}$ (typical value for rock)

$a = 24.1$ ft

substitution into the above equation yields:

$$\tau = 6.76 \times 10^{-5} t$$

where t is the time, in hours, after the start of cooling. Table 7-3 has been prepared using the two relationships above and data from Table 6-2. This table illustrates the variation in the rock cooling load as a function of time after the start of cooling.

Table 7-3 shows that the cooling load is very high initially and then decreases exponentially. If all sinks were to be cooled simultaneously the initial cooling load would be seven times the first entry of Table 7-3 or approximately 435 tons. However, with simultaneous cooling of all the reservoirs the total load decreases rapidly. Within 12 months the total rock cooling load would be only 83 tons. Since this method of cooling would require the installation of a relatively large refrigeration plant which, during most of its operating life, would be used at a relatively low partial load, it does not provide the most efficient utilization of the equipment.

If, on the other hand, the start of cooling of each reservoir is staggered with respect to the others a more uniform loading can be developed. Table 7-4 provides the total cooling load for the heat sink system using staggered starts.

In this mode the start of cooling of any particular reservoir is not allowed to occur until the total peak load would not exceed approximately 120 tons. Note that although the delay period between any two consecutive reservoir cooling starts increases, all sinks will have experienced at least seven months of cooling prior to the completion of the preparation period (i.e. the 36th month).

Figure 7-3 illustrates the variation in the total rock cooling load that will occur using this staggered start procedure. Note that during the first three years of operation the total rock cooling load will vary from 60 to 120 tons; this variation is significantly lower than the variation that would occur if simultaneous cooling was employed.

The water cooling load can be readily calculated using equation 6-15. If it is assumed that frictional pump heating will be negligible, then the water cooling load is given by

$$q_c = \frac{M_t c_w (T_1 - T_o)}{12,000\ t_p}$$

where $M_t = 1.788 \times 10^8$ lb

$c_w = 1.0$ Btu/lb°F

$T_1 = 80°F$ (assumed)

$T_o = 40°F$ (heat sink design temperature)

$t_p = 3$ years (26,280 hr).

Table 7-3. Reservoir rock cooling loads for sample problem number 4.

Elapsed Time (months)	τ	$\alpha(\tau)$	Cooling load q_s (Btu/hr)	Cooling load q_s (Tons)
"0" (6 days)	.01	15.12	7.45×10^5	62.1
1	.05	7.39	3.64×10^5	30.3
2	.10	5.55	2.73×10^5	22.8
3	.15	4.73	2.33×10^5	19.4
4	.19	4.32	2.13×10^5	17.7
5	.24	3.95	1.95×10^5	16.2
6	.29	3.69	1.82×10^5	15.1
7	.34	3.48	1.71×10^5	14.3
8	.39	3.32	1.64×10^5	13.6
9	.44	3.18	1.57×10^5	13.1
10	.49	3.07	1.51×10^5	12.6
11	.54	2.97	1.46×10^5	12.2
12	.58	2.90	1.43×10^5	11.9
13	.63	2.82	1.39×10^5	11.6
14	.68	2.75	1.35×10^5	11.3
15	.73	2.68	1.32×10^5	11.0
16	.78	2.63	1.30×10^5	10.8
17	.83	2.57	1.27×10^5	10.6
18	.88	2.53	1.25×10^5	10.4
19	.93	2.48	1.22×10^5	10.2
20	.97	2.45	1.21×10^5	10.1
21	1.0	2.43	1.20×10^5	10.0
22	1.1	2.36	1.16×10^5	9.7
23	1.1	2.36	1.16×10^5	9.7
24	1.2	2.30	1.13×10^5	9.4
25	1.2	2.30	1.13×10^5	9.4
26	1.3	2.24	1.10×10^5	9.2
27	1.3	2.24	1.10×10^5	9.2
28	1.4	2.19	1.08×10^5	9.0
29	1.4	2.19	1.08×10^5	9.0
30	1.5	2.15	1.06×10^5	8.8
31	1.5	2.15	1.06×10^5	8.8
32	1.6	2.11	1.04×10^5	8.7
33	1.6	2.11	1.04×10^5	8.7
34	1.7	2.07	1.02×10^5	8.5
35	1.7	2.07	1.02×10^5	8.5
36	1.8	2.04	1.00×10^5	8.4
39	1.9	2.00	9.86×10^4	8.2
42	2.0	1.98	9.75×10^4	8.1
48	2.3	1.90	9.36×10^4	7.8
54	2.6	1.84	9.06×10^4	7.6
60	2.9	1.78	8.77×10^4	7.3
66	3.2	1.74	8.57×10^4	7.1
72	3.5	1.70	8.37×10^4	7.0
84	4.1	1.63	8.03×10^4	6.7

Table 7-3. (Con't)

Elapsed Time (months)	τ	$\alpha(\tau)$	Cooling load q_s (Btu/hr)	Cooling load q_s (Tons)
96	4.7	1.57	7.73×10^4	6.5
108	5.3	1.53	7.54×10^4	6.3
120	5.8	1.50	7.39×10^4	6.2
132	6.4	1.46	7.19×10^4	6.0
144	7.0	1.43	7.04×10^4	5.9
156	7.6	1.40	6.90×10^4	5.8
168	8.2	1.38	6.80×10^4	5.7
180	8.8	1.36	6.70×10^4	5.6
192	9.4	1.34	6.60×10^4	5.5
204	9.9	1.32	6.50×10^4	5.4
216	11.0	1.29	6.35×10^4	5.3

Table 7-4: Total rock cooling loads for sample problem number 4.

Time	Cooling loads, tons							
	Sink	Sink	Sink	Sink	Sink	Sink	Sink	Total
(months)	#1	#2	#3	#4	#5	#6	#7	
0	62.1							62.1
1	30.3	62.1						92.4
2	22.8	30.3	62.1					115.2
3	19.4	22.8	30.3					72.5
4	17.7	19.4	22.8					59.9
5	16.2	17.7	19.4	62.1				115.4
6	15.1	16.2	17.7	30.3				79.3
7	14.3	15.1	16.2	22.8				68.4
8	13.6	14.3	15.1	19.4				62.4
9	13.1	13.6	14.3	17.7	62.1			120.8
10	12.6	13.1	13.6	16.2	30.3			85.5
11	12.2	12.6	13.1	15.1	22.8			75.8
12	11.9	12.2	12.6	14.3	19.4			70.4
13	11.6	11.9	12.2	13.6	17.7			67.0
14	11.3	11.6	11.9	13.1	16.2			64.1
15	11.0	11.3	11.6	12.6	15.1			61.6
16	10.8	11.0	11.3	12.2	14.3			59.6
17	10.6	10.8	11.0	11.9	13.6	62.1		120.0
18	10.4	10.6	10.8	11.6	13.1	30.3		86.8
19	10.2	10.4	10.6	11.3	12.6	22.8		77.9
20	10.1	10.2	10.4	11.0	12.2	19.4		73.3
21	10.0	10.1	10.2	10.8	11.9	17.7		70.7
22	9.7	10.0	10.1	10.6	11.6	16.2		68.2
23	9.7	9.7	10.0	10.4	11.3	15.1		66.2
24	9.4	9.7	9.7	10.2	11.0	14.3		64.3
25	9.4	9.4	9.7	10.1	10.8	13.6		63.0
26	9.2	9.4	9.4	10.0	10.6	13.1		61.7
27	9.2	9.2	9.4	9.7	10.4	12.6		60.5
28	9.0	9.2	9.2	9.7	10.2	12.2		59.5
29	9.0	9.0	9.2	9.4	10.1	11.9	62.1	120.7
30	8.8	9.0	9.0	9.4	10.0	11.6	30.3	88.1
31	8.8	8.8	9.0	9.2	9.7	11.3	22.8	79.6
32	8.7	8.8	8.8	9.2	9.7	11.0	19.4	75.6
33	8.7	8.7	8.8	9.0	9.4	10.8	17.7	73.1
34	8.5	8.7	8.7	9.0	9.4	10.6	16.2	71.1
35	8.5	8.5	8.7	8.8	9.2	10.4	15.1	69.2
36	8.4	8.5	8.5	8.8	9.2	10.2	14.3	67.9
39	8.2	8.4	8.4	8.5	8.8	9.7	12.6	64.6
42	8.1	8.2	8.2	8.2	8.5	9.4	11.6	62.2
48	7.8	7.8	7.8	8.2	8.2	8.8	10.2	58.8
54	7.6	7.6	7.6	8.1	8.1	8.4	9.4	56.8
60	7.3	7.3	7.6	7.6	7.8	8.1	8.8	54.5
72	7.0	7.0	7.0	7.1	7.3	7.6	8.1	51.1
84	6.7	6.7	6.7	7.0	7.0	7.1	7.6	48.8
96	6.5	6.5	6.5	6.7	6.7	6.7	7.1	46.7
120	6.2	6.2	6.2	6.3	6.3	6.5	6.7	44.4
144	5.9	5.9	5.9	6.0	6.0	6.2	6.2	42.1
168	5.7	5.7	5.7	5.7	5.8	5.9	6.0	40.5
192	5.5	5.5	5.5	5.5	5.6	5.7	5.8	39.1
216	5.3	5.3	5.3	5.3	5.4	5.5	5.6	37.7

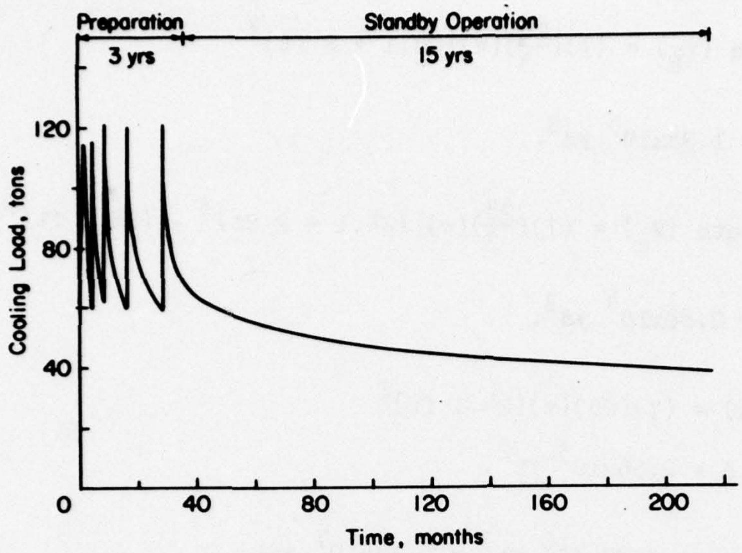

Figure 7-3. Total rock cooling load for sample problem 4.

Thus after substitution q_c is calculated to be:

$$q_c = \frac{(1.788 \times 10^8 \text{ lb})(1.0 \frac{\text{Btu}}{\text{lb}^\circ\text{F}})(80^\circ\text{F}-40^\circ\text{F})}{(12{,}000 \frac{\text{Btu/hr}}{\text{ton}})(26{,}280 \text{ hr})} = 22.7 \text{ tons.}$$

Thus the preparation cooling loads for this heat sink system can be satisfied through the use of a 125-ton refrigeration unit to cool the surrounding rock and a 25-ton refrigeration unit to cool the water.

Figure 7-3 also illustrates the amount of refrigeration that will be required to maintain the heat sink system at the desired design temperature during the standby mode. Note that the standby mode commences at the 36th month and continues for 15 years. During this time the total refrigeration load is initially approximately 70 tons and slowly decreases to a final value of slightly less than 40 tons. Thus a portion of the 125-ton refrigeration plant used to initially cool down the heat sink reservoirs can also be utilized to provide maintenance cooling during the standby period.

The cost estimate for this heat sink system must include the cost of construction, the cost of preparing (i.e. cooling) the heat sinks and the cost of maintaining the heat sinks during the standby period.

Since the procedure for estimating the construction cost for this sample problem is identical to the procedure used for sample problem 3, only the basic calculation will be provided here. Problem 3 should be referred to for the details of the procedure.

1. Excavated volume $(V_E) = (7)(\frac{28}{3})(\pi)(24.1 + 2\text{ ft})^3$

$$V_E = 1.35\text{x}10^5 \text{ yd}^3.$$

2. Volume of concrete $(V_c) = (7)(\frac{28}{3})(\pi)[(24.1 + 2\text{ ft})^3 - (24.1\text{ ft})^3]$

$$V_c = 2.88\text{x}10^4 \text{ yd}^3.$$

3. Surface area (A) = $(7)(20)(\pi)(24.1\text{ ft})^2$

$$A = 2.56\text{x}10^5 \text{ ft}^2.$$

4. Quantity of water = $2.87\text{x}10^6 \text{ ft}^3 = 2.14\text{x}10^7$ gal.

5. Refrigeration, preparation cooling.

Preparation cooling will require one 125-ton refrigeration system to cool the surrounding rock and one 25-ton refrigeration unit to cool the water in the heat sink reservoirs. From visual inspection of Figure 7-3 it is estimated that the 125-ton plant will operate at an estimated average load of approximately 90 tons. It is estimated that the 25-ton plant will operate at a full load of 25 tons for the three-year preparation period. From the refrigeration plant performance chart (curve B of Figure F-3), at an evaporator temperature of -20°F approximately 2.6 HP will be required to provide each ton of refrigeration. Thus, the average refrigeration loads of 90 and 25 tons can be converted into average electrical loads through the use of:

electrical consumption = (90+25 tons)(2.6 HP/ton)(0.746 kW/HP) = 223 kW.

Since both of these plants will be operated 24 hours per day during the first three years the total amount of electrical power used each year can be calculated using

electrical power/year = (223 kW)(24 hr/day)(365 day/yr) = $1.95\text{x}10^6$ kW-hr/yr.

Salaries and supplies to support the operation of the refrigeration plant for the first three years can be determined using the information in Appendix F.

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DESIGN PROCEDURES FOR UNDERGROUND HEAT SINK SYSTEMS.(U)
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6. **Refrigeration, maintenance cooling.**

Since this heat sink system must be maintained at a subambient temperature during the standby period, refrigeration will be required after completion of the preparation cooling period. From Figure 7-3 it can be seen that this maintenance cooling load will be approximately 68 tons at the start of the maintenance period (i.e. the 36th month) and will slowly decrease to approximately 38 tons after 15 years. It is assumed that approximately 75 tons of the 125-ton system originally used to cool the surrounding rock during the preparation phase will be used to maintain the heat sink system during the standby period. Using Figure 7-3 and a visual estimation procedure it is also assumed that during this standby period the system will operate at an average load of approximately 45 tons. Thus the electrical consumption for the plant will be:

electrical consumption = (45 tons)(2.6 HP/ton)(0.746 kW/HP) = 87 kW

and the total average consumption per year will be:

electrical power/year = (87 kW)(24 hr/day)(365 day/yr) = $7.62\text{x}10^5$ kW-hr/yr.

The costs for salaries, supplies and equipment overhauls are given in Appendix F.

The costs for this sample problem are summarized in Table 7-5.

7-07.5. Sample Problem 5: Recirculated 90°F Water Reservoir with Heat Transfer to Surroundings. This sample problem is similar to the heat sink system considered in sample problem 3 except that in this problem it is assumed that the reservoirs are constructed in such a way that heat transfer will occur through the reservoir walls to the surrounding rock. For simplicity it is assumed that the thermal properties of the reservoir walls (i.e. concrete) are similar to the thermal properties of the surrounding rock.

The technique employed to predict the performance of this type of reservoir is discussed in Section 6-03.2. This technique relies on the use of several non-dimensional parameters, described by equations 6-06a, b and c, and the use of Figure 6-1 or 6-2. The calculation of these non-dimensional parameters requires the use of a constant average heat rejection rate. Since the power plant under consideration does not have a constant heat rejection rate, that is, the heat rejection rate increases with heat sink temperature, an average heat rejection rate will have to be approximated. Using a trial and error solution process several iterations will be required to size the heat sink system and allow refinement of the approximated "average" heat rejection rate.

Table 7-5. Cost estimate for sample problem no. 4.

Item No.	Item Description	Unit Cost	Quantity	Inflation Factor	Item Cost	Sub-totals
A	Construction					
1	Excavation	\$ 75/yd^3	1.35×10^5 yd^3	-	\$10,125,000	
2	Reinforced concrete	120/yd^3	2.88×10^4 yd^3	-	3,456,000	
3	Waterproofing	0.31/ft^2	2.56×10^5 ft^2	-	79,000	
						\$13,660,000
B	Preparation					
1	Water	\$.04/100 gal	2.14×10^7 gal	-	\$ 9,000	
2	Refrigeration unit	\$1850/ton	25 ton	-	46,000	
3	Refrigeration unit	\$1850/ton	125 ton	-	231,000	
4	Salary	\$$1.2 \times 10^5$/yr	3 years	1.051	378,000	
5	Supplies	\$225/ton-yr	[3 years, 150 tons	1.085	110,000	
6	Electricity	\$.035/kW-hr	[1.95×10^6 kW-hr/yr, 3 years	1.085	222,000	
						\$ 996,000
C	Maintenance					
1	Electricity	\$.035/kW-hr	[7.62×10^5 kW-hr/yr, 15 years	2.142	\$ 857,000	
2	Salary	\$$1.2 \times 10^5$/yr	15 years	1.563	2,813,000	
3	Supplies	\$225/ton-yr	[15 years, 75 tons	2.142	542,000	
4	Overhaul	\$250/ton-overhaul	[5 overhauls, 75 tons	1.853	174,000	
						\$ 4,386,000
					Total Cost	\$19,042,000

As an initial starting point to determine the "average" heat rejection rate the results of sample problem 3 will be utilized. Note that since that heat sink system would operate within the same temperature range (90°F to 160°F) and have the same period of operation, its average heat rejection rate would probably be very similar to the rate in this sample problem.

From sample problem 3 the total mass of water for the heat sink system was 3.363×10^8 lb and the heat sink temperature change was from 90°F to 160°F, so the total quantity of heat rejected to that heat sink system can be calculated using:

$$Q = mc_w \Delta T .$$

Thus the total amount of heat rejected to the sample problem 3 heat sink system is

$$Q = (3.363 \times 10^8 \text{ lb})(1.0 \frac{\text{Btu}}{\text{lb}^\circ\text{F}})(160\text{-}90^\circ\text{F}) = 2.35 \times 10^{10} \text{ Btu} .$$

Since the assumed operating period was 31.5 days, the "average" heat rejection would then be:

$$q_a = \frac{2.35 \times 10^{10} \text{ Btu}}{(31.5 \text{ days})(24 \text{ hr/day})} = 31.1 \times 10^6 \text{ Btu/hr} .$$

The non-dimensional parameters which must be calculated are:

$$f(\tau,G) = \frac{k(T_f - T_o)\ell}{q_o}$$

$$G = \frac{2\pi a^2 \ell \rho c}{M_w c_w}$$

$$\tau = \frac{kt_d}{\rho c a^2} .$$

Using $k = 1.45$ Btu/hr-ft°F   typical values for rock

$c = 0.2$ Btu/lb°F

$\rho = 186$ lb/ft^3

$c_w = 1.0$ Btu/lb°F

$T_f = 160$°F

$T_o = 90$°F

$q_o = 31.1 \times 10^6$ Btu/hr

it should be apparent that these non-dimensional parameters cannot be calculated without first assuming a size for the heat sink system. Thus the procedure that will be employed will be to assume a heat sink reservoir system and then, through the use of these parameters and Figures 6-1 and 6-2, calculate the operating period, t_d. This calculated period will then be compared with the desired period of 31.5 days and the assumed heat sink system will be revised accordingly.

As a starting point a heat sink system consisting of 12 reservoirs 50 ft in diameter by 250 ft long will be assumed. Thus

$$f(\tau,G) = \frac{(1.45 \frac{Btu}{hr\ ft^oF})(160\text{-}90^oF)(12)(250\ ft)}{31.1x10^6\ Btu/hr}$$

$$f(\tau,G) = 9.79x10^{-3}.$$

Similarly, from Appendix C, the mass of water, M_w, is given by

$$M_w = \frac{28}{3} n\ \pi r^3 \rho_w$$

where n = number of reservoirs

Thus $$M_w = (\tfrac{28}{3})(12)(\pi)(25\ ft)^3(62.4\tfrac{lb}{ft^3}) = 3.43x10^8\ lb$$

Therefore:

$$G = \frac{2\pi(25\ ft)^2(12)(250\ ft)(186\ lb/ft^3)(0.2\ Btu/lb^oF)}{(3.43x10^8 lb)(1.0\ Btu/lb^oF)}$$

and thus G = 1.28.

Using $f(\tau,G) = .0098$ and $G \approx 1.3$, from Figure 6-2

$$\tau = .061$$

Since $$\tau = \frac{kt_d}{\rho ca^2}$$

$$t_d = \frac{\tau\rho ca^2}{k}$$

and therefore $$t_d = \frac{(.061)(186\ lb/ft^3)(0.2\ Btu/lb^oF)(25\ ft)^2}{1.45\ Btu/hr\text{-}ft^oF}$$

or $t_d = 978$ hr = 40.8 days.

Note that this calculated design life exceeds the desired design life of 31.5 days by a significant degree. This means that the assumed 12-reservoir system is too large and a smaller system can be used.

If, as a second attempt, a heat sink system composed of nine reservoirs each 50 ft in diameter by 250 ft long is used, then following the calculation procedure outlined above yields (details omitted):

$$f(\tau,G) = 7.34\text{x}10^{-3}$$

and

$$G = 1.28.$$

Thus from Figure 6-2 using $f(\tau,G) = .0073$ and $G=1.3$ yields

$$\tau = .0435$$

and thus $t_d = 698 \text{ hr} = 29.1 \text{ days}.$

Since this assumed system provides an operating period which is too short a slightly larger system will be required.

Reiterating using a heat sink system composed of 10 reservoirs 49.5 ft in diameter by 248 ft long yields:

$$f(\tau,G) = \frac{(1.45 \text{ Btu/hr ft}^\circ\text{F})(160\text{-}90^\circ\text{F})(10)(248 \text{ ft})}{(31.1\text{x}10^6 \text{ Btu/hr})}$$

$$f(\tau,G) = 8.09\text{x}10^{-3}$$

$$M_w = (10)(\frac{28}{3})(\pi)(24.75 \text{ ft})^3(62.4 \frac{\text{lb}}{\text{ft}^3}) = 2.77 \times 10^8 \text{ lb}$$

$$G = \frac{2\pi\ (24.75 \text{ ft})^2(10)(248 \text{ ft})(186 \text{ lb/ft}^3)(0.2 \text{ Btu/lb}^\circ\text{F})}{(2.77\text{x}10^8 \text{ lb})(1.0 \text{ Btu/lb}^\circ\text{F})}$$

$$G = 1.28.$$

And thus from Figure 6-2 for $f(\tau,G) = .0081$ and $G=1.3$,

$$\tau = .0495$$

and therefore $$t_d = \frac{(.0495)(186 \text{ lb/ft}^3)(0.2 \text{ Btu/lb}^\circ\text{F})(24.75 \text{ ft})^2}{(1.45 \text{ Btu/hr ft}^\circ\text{F})}$$

$$t_d = 778 \text{ hr} = 32.4 \text{ days}.$$

Note that while this heat sink system would appear to be oversized by 0.9 day it is based on the assumed average heat rejection rate of 31.1×10^6 Btu/hr. To check the performance of this heat sink system under the actual heat rejection load the following procedure may be employed.

The parameters τ and G are functions of time, geometry and thermal properties only. Thus it is possible to calculate their variation with respect to elapsed time independent of the actual heat rejection rate. Through the use of Figure 6-2, these two parameters can also be used to determine the variation in the parameter $f(\tau,G)$ as a function of time. Table 7-6 presents the results of these calculations.

The parameter $f(\tau,G)$ provides a relationship between the temperature change and the average heat rejection rate, namely:

$$f(\tau,G) = \frac{k(T_f - T_o)\ell}{q_o} .$$

Table 7-6: Variation in $f(\tau,G)$ with time (G=1.3) for sample problem 5.

Time (days)	τ	$f(\tau,G)$
6.5	.01	.0019
8.2	.0125	.00225
9.8	.015	.0027
11.4	.0175	.0031
13.1	.02	.0035
14.7	.0225	.0040
16.4	.025	.0044
18.0	.0275	.0048
19.6	.03	.0052
21.3	.0325	.0056
22.9	.035	.0059
24.5	.0375	.0063
26.2	.04	.0067
27.8	.0425	.0071
29.4	.045	.0075
31.1	.0475	.0079

If the straight line approximation of the heat rejection rate (see Appendix A) is used, a second relationship between the average waste heat load and the heat sink temperature can be developed. Namely, since

$$Q_R = 1.005x10^5\ T_{sink} + 18.85x10^6 \quad Btu/hr$$

the average waste heat load between two temperatures, T_{sink} and T_o, is given by:

$$q_o = 1.005x10^5 \left[\frac{T_{sink}+T_o}{2}\right] + 18.85x10^6 \quad Btu/hr$$

where $\frac{T_{sink}+T_o}{2}$ represents the mean temperature.

Since the initial heat sink temperature, T_o, is 90°F, then, using the appropriate values for k and ℓ, the following equations can be developed:

$$f(\tau,G) = \frac{(1.45)(T_{sink}-90)(10)(248)}{q_o}$$

and $$q_o = 1.005x10^5 \left[\frac{T_{sink}}{2} + 45\right] + 18.85x10^6.$$

If this second relationship is substituted into the first and then rearranged to solve for T_{sink}, the resulting relationship is:

$$T_{sink} = \frac{3.236x10^5 + 23.37x10^6\ f(\tau,G)}{3596 - 5.025x10^4\ f(\tau,G)},\ °F.$$

Using this relationship and the data in Table 7-6 the temperature of the heat sink system as a function of time can be determined. Table 7-7 presents the results of these calculations.

Table 7-7. Heat sink temperature versus time for sample problem 5.

Time (days)	$f(\tau,G)$	T_{sink} (°F)
6.5	.0019	105.1
8.2	.00225	108.0
9.8	.0027	111.8
11.4	.0031	115.1
13.1	.0035	118.5
14.7	.0040	122.9
16.4	.0044	126.4
18.0	.0048	129.9
19.6	.0052	133.5
21.3	.0056	137.1
22.9	.0059	139.9
24.5	.0063	143.6
26.2	.0067	147.3
27.8	.0071	151.1
29.4	.0075	155.0
31.1	.0079	158.9

As Table 7-7 indicates, this heat sink system will attain a temperature of approximately 160°F at the end of the mission period. Therefore, it can be concluded that this system, composed of 10 reservoirs which are 49.5 ft in diameter by 248 ft long, will provide the correct heat storage capacity.

Figure 7-4 has been prepared using the data presented in Table 7-7. The curves for waste heat load and coolant water flow rate have been determined by correlating the temperature-time history with the data provided in Appendix A.

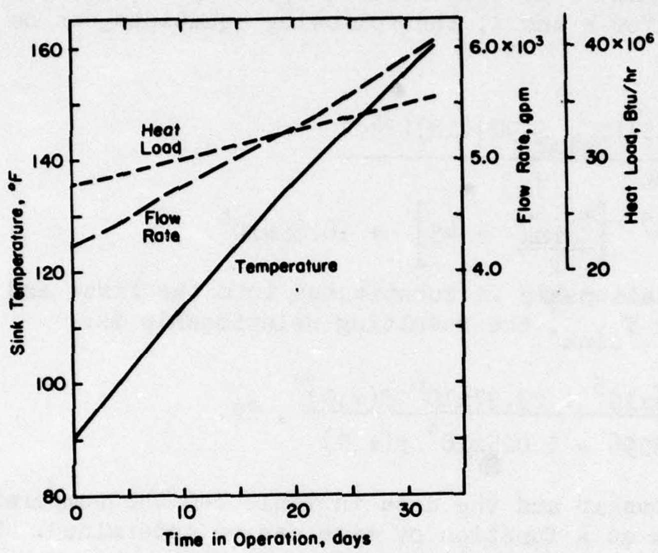

Figure 7-4. Performance of the sample problem 5 heat sink system.

Since this heat sink system relies on heat transfer (i.e. heat storage) in the surrounding rock media it is necessary to determine the depth of penetration of the thermal gradient around each heat sink to assure that sufficient separation is provided. The thermal gradient in the surrounding rock can be determined through the use of equation 6-07, namely

$$T(R,t) = T_\infty + (T_w - T_o)\beta(\frac{R}{a},\tau)$$

since $$\tau = \frac{kt}{\rho c a^2} .$$

Then using the data provided earlier,

$$\tau = \frac{(1.45 \frac{Btu}{hr ft\,^oF})(31.5 \text{ days})(24 \text{ hr/day})}{(186 \text{ lb/ft}^3)(0.2 \text{ Btu/lb}^oF)(24.75 \text{ ft})^2}$$

$\tau = .048.$

Using Table 6-1 with $\tau = 0.05$ yields:

$\frac{R}{a}$	$\beta(\frac{R}{a},\tau)$
1.0	1.00
1.1	0.718
1.2	0.483
1.3	0.302
1.4	0.175
1.5	0.093
1.6	0.048
1.7	0.021
1.8	0.009
1.9	0.003
2.0	0.001

And thus using $T_\infty = 90^oF$

$T_w = 160^oF$

$a = 24.75$ ft

the variation in rock temperature at the end of the operational period would be:

$\frac{R}{a}$	R (ft)	T(R, 31.5 days) (^oF)
1.00	24.75	160.0
1.1	27.2	140.3
1.2	29.7	123.8
1.3	32.2	111.1
1.4	34.7	102.3
1.5	37.1	96.5
1.6	39.6	93.4
1.7	42.1	91.5
1.8	44.6	90.6
1.9	47.0	90.2
2.0	49.5	90.1

These results have been plotted in Figure 7-5. Note that the 5% change occurs at a radius of approximately 38 ft. Thus it would be recommended that the heat sink reservoirs be located so that the center

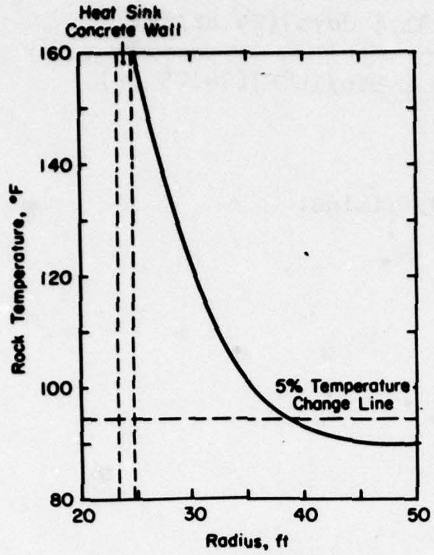

Figure 7-5. Temperature gradient in the surrounding rock at the end of the mission, sample problem 5.

to center distance was at least 76 ft. It should be noted that this calculation procedure provides a rather conservative distance since it assumes that the sink would operate at a temperature of 160°F throughout the mission period. This provides a greater degree of penetration of the thermal gradient than actually would occur since in practice the "average" sink wall temperature would be lower.

Since the procedure for estimating the cost of this sample problem heat sink system is identical to the procedure used for sample problem 3 no details of the calculations are provided. The results of the cost estimation procedure for this sample problem are presented in Table 7-8.

Table 7-8. Cost estimate for sample problem no. 5.

Item No.	Item Description	Unit Cost	Quantity	Item Cost	Subtotals
A	Construction				
1	Excavation	$75/yd^3	2.08x10^5 yd^3	$15,590,000	
2	Reinforced concrete	$120/yd^3	4.03x10^4 yd^3	5,160,000	
3	Waterproofing	$0.31/ft^2	3.85x10^5 ft^2	120,000	
					$20,870,000
B	Preparation				
1	Water	$.04/100 gal	3.33x10^7 gal	$ 13,000	
					$ 13,000
			Total Cost		$20,883,000

7-07.6. Sample Problem 6: Recirculated 40°F Water Reservoir with Heat Transfer to Surroundings. This problem is similar to sample problem 5 except that for this heat sink system the initial sink temperature is below the ambient temperature of the surrounding rock. Thus the solution procedure is similar to that used for the previous sample problem.

To estimate an average waste heat load, the results of sample problem 4 will be utilized, since both that problem and this one have the same temperature limits of operation.

From sample problem 4 the total mass of water contained in that heat sink system was 1.788×10^8 lb of water; thus the total heat storage capacity would be:

$$Q = mc_w \Delta T = (1.788 \times 10^8 \text{ lb})(1.0 \frac{\text{Btu}}{\text{lb}^\circ\text{F}})(160\text{-}40^\circ\text{F})$$

or

$$Q = 2.15 \times 10^{10} \text{ Btu.}$$

And since the operational period was 31.5 days the average heat load would be:

$$q_a = \frac{2.15 \times 10^{10} \text{ Btu}}{(31.5 \text{ day})(24 \text{ hr/day})} = 28.4 \times 10^6 \text{ Btu/hr.}$$

For the first iteration a heat sink system consisting of five reservoirs 50 ft in diameter by 250 ft long will be examined. Thus, using equations 6-06 a, b and c with

$$k = 1.45 \text{ Btu/hr-ft}^\circ\text{F}$$

$$\rho = 186 \text{ lb/ft}^3$$

$$c = 0.2 \text{ Btu/lb}^\circ\text{F}$$

$$f(\tau,G) = \frac{k(T_f - T_o)\ell}{q_o} = \frac{(1.45 \text{ Btu/hr-ft}^\circ\text{F})(160\text{-}40^\circ\text{F})(5)(250 \text{ ft})}{28.4 \times 10^6 \text{ Btu/hr}}$$

$$f(\tau,G) = 7.66 \times 10^{-3}.$$

Since:

$$M_w = (5)(\frac{28}{3})\pi(25 \text{ ft})^3(62.4 \text{ lb/ft}^3) = 1.43 \times 10^8 \text{ lb water}$$

then

$$G = \frac{2\pi a^2 \rho c \ell}{M_w c_w} = \frac{(2)(\pi)(25 \text{ ft})^2(186 \text{ lb/ft}^3)(0.2 \text{ Btu/lb}^\circ\text{F})(5)(250 \text{ ft})}{(1.43 \times 10^8 \text{ lb})(1.0 \text{ Btu/lb}^\circ\text{F})}$$

$$G = 1.28$$

and thus using Figure 6-2 with $f(\tau,G) = .0077$ and $G=1.3$

$$\tau = .0460.$$

Since $$\tau = \frac{kt_d}{\rho ca^2}$$

then $$t_d = \frac{(.0460)(186\ \text{lb/ft}^3)(0.2\ \text{Btu/lb}^\circ\text{F})(25\ \text{ft})^2}{(1.45\ \text{Btu/hr ft}^\circ\text{F})}$$

or $$t_d = 738\ \text{hr} = 30.7\ \text{day}$$

Since the assumed heat sink system would attain its saturation temperature prior to the completion of the mission period a larger system must be considered. If a system composed of six reservoirs 47.5 ft in diameter by 238 ft long is considered, then

$$f(\tau,G) = \frac{(1.45\ \text{Btu/hr ft}^\circ\text{F})(160\text{-}40^\circ\text{F})(6)(238\ \text{ft})}{(28.4\text{x}10^6\ \text{Btu/hr})} = .0087$$

$$M_w = (6)(\tfrac{28}{3})(\pi)(23.75\ \text{ft})^3(62.4\ \text{lb/ft}^3) = 1.47 \times 10^8\ \text{lb}$$

$$G = \frac{(2)(\pi)(186\ \text{lb/ft}^3)(23.75\ \text{ft})^2(0.2\ \text{Btu/lb}^\circ\text{F})(6)(238\ \text{ft})}{(1.47\text{x}10^8\ \text{lb})(1.0\ \text{Btu/lb}^\circ\text{F})} = 1.28.$$

And from Figure 6-2 for $f(\tau,G) = .0087$ and $G=1.3$,

$$\tau = .0530$$

and thus since $$\tau = \frac{kt_d}{\rho ca^2}$$

$$t_d = \frac{(.0530)(186\ \text{lb/ft}^3)(0.2\ \text{Btu/lb}^\circ\text{F})(23.75\ \text{ft})^2}{(1.45\ \text{Btu/hr ft}^\circ\text{F})}$$

$$t_d = 767\ \text{hr} = 32.0\ \text{days}$$

As with sample problem 5, this potential heat sink system must be checked to determine the variation in sink temperature as a function of time. Again, the procedure utilized is to determine the parameters τ and G at a number of points in time and use Figure 6-2 to determine the variation in $f(\tau,G)$. Using the relationship for $f(\tau,G)$ and the straight line approximation for the waste heat load the variation in heat sink temperature is determined as follows.

Since

$$f(\tau,G) = \frac{k(T_{sink}-T_o)\ell}{q_o} = \frac{(1.45)[T_{sink}-40](6)(238)}{q_o}$$

$$f(\tau,G) = \frac{2071[T_{sink}-40]}{q_o}$$

but between the temperatures T_{sink} and 40°F, the average heat rejection rate is given by

$$q_a = 1.005\text{x}10^5 \left[\frac{T_{sink}+40}{2}\right] + 18.85\text{x}10^6.$$

Substituting this second relationship into the first and solving for T_{sink} yields:

$$T_{sink} = \frac{8.28\text{x}10^4 + 20.86\text{x}10^6\ f(\tau,G)}{2071 - 5.025\text{x}10^4\ f(\tau,G)}.$$

Since G=1.3 and τ is given by

$$\tau = \frac{(1.45\ \text{Btu/hr-ft°F})\ t_d}{(180\ \text{lb/ft}^3)(0.2\ \text{Btu/lb°F})(23.75\ \text{ft})^2}$$

or $\quad t = 6.91\text{x}10^{-5}\ t_d$ where t_d is in hr

then Table 7-9 can be been developed using these relationships for τ and T_{sink} and Figure 6-2. As the data indicate, this heat sink system will attain thermal saturation just after completion of the mission and thus has sufficient heat storage capacity.

The data presented in Table 7-9 have been plotted in Figure 7-6. The data for the waste heat load and coolant water flow rate operated in Figure 7-6 were developed by correlating the temperature-time history of this heat sink system with the power plant performance data presented in Appendix A.

Table 7-9. Variation in heat sink temperature with time for sample problem 6.

Time (days)	τ	$f(\tau,G)$	T_{sink} (°F)
6.0	.01	.0019	62.0
7.5	.0125	.0023	66.9
9.0	.015	.0027	71.9
10.6	.0175	.0031	77.0
12.1	.02	.0035	82.2
13.6	.0225	.0040	88.9
15.1	.0250	.0044	94.4
16.6	.0275	.0048	100.0
18.1	.03	.0052	105.7
19.6	.0325	.0056	111.5
21.1	.035	.0059	116.0
22.6	.0375	.0063	122.1
24.1	.04	.0067	128.3
25.6	.0425	.0071	134.7
27.1	.045	.0075	141.2
28.6	.0475	.0079	147.9
30.1	.05	.00825	153.9
31.7	.0525	.0086	160.0

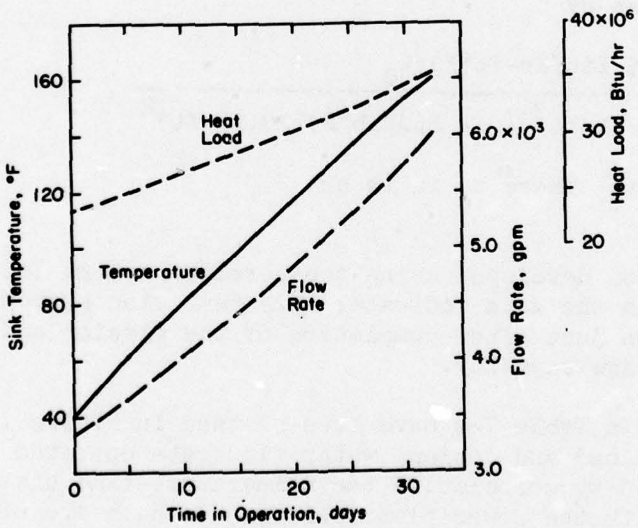

Figure 7-6. Performance of the sample problem 6 heat sink system.

Since this heat sink system will have an initial temperature below the ambient temperature of the surrounding rock, cooling will be required. As in sample problem 4 this cooling requirement can be considered to be a two-stage operation. During the first three years the heat sink system must be cooled to bring it down to its initial temperature of 40°F. Thereafter cooling must be supplied to maintain the system at that temperature.

The initial cooling load can be calculated using an inverse of the procedure employed to calculate the performance of the heat sink under a heat load. Namely, since the size of the heat sink and the length of the cooling period are known, equations 6-06 a, b and c, in conjunction with Figures 6-1 and 6-2, can be used to calculate the required average cooling load.

To calculate the cooling loads it will be assumed that two heat sinks are cooled each year. Thus at the end of the three-year preparation period all six heat sinks will have been brought down to the proper initial temperature.

Thus, since one sink must be cooled each six months, using equation 6-06c:

$$\tau = \frac{kt}{\rho c a^2}$$

where t = 183 days (4380 hrs).

Yields, upon substitution for the other values,

$$\tau = \frac{(1.45 \frac{\text{Btu}}{\text{fthr}^\circ\text{F}})(4380 \text{ hr})}{(186 \frac{\text{lb}}{\text{ft}^3})(0.2 \frac{\text{Btu}}{\text{lb}^\circ\text{F}})(23.75 \text{ ft})^2}$$

$$\tau = 0.30.$$

Thus for G=1.3 and $\tau = 0.30$, Figure 6-1 yields

$$f(\tau,G) = 0.0365.$$

And thus since from equation 6-06a:

$$f(\tau,G) = \frac{k(T_\infty - T_f)\ell}{q}$$

$$q = \frac{(1.45 \frac{Btu}{hr ft ^{o}F})(90-40^{o}F)(237.5 \text{ ft})}{(.0365)}$$

$$q = 4.73 \times 10^{5} \frac{Btu}{hr} = 39.4 \text{ tons.}$$

Once each heat sink reservoir has attained its initial temperature of 40°F, cooling must continue to be provided to prevent heat flow from the surrounding rock from warming the heat sinks. This maintenance cooling requirement can be calculated using equation 6-17:

$$q_m = \frac{k(T_\infty - T_o)\ell I(\tau)}{12{,}000}$$

when $k = 1.45$ Btu/hr ft°F

$T_\infty = 90^{o}F$

$T_o = 40^{o}F$

$\ell = 237.5$ ft

$\tau = kt/\rho ca^2$

and $I(\tau)$ is from Figure 6-4.

Thus
$$\tau = \frac{(1.45 \frac{Btu}{hr ft ^{o}F})t}{(186 \frac{lb}{ft^3})(0.2 \frac{Btu}{lb ^{o}F})(23.75 \text{ ft})^2}$$

or $\tau = 6.91 \times 10^{-5}\, t \qquad 1/hr$

and
$$q_m = \frac{(1.45 \frac{Btu}{hr ft ^{o}F})(90-40^{o}F)(237.5 ft)\, I(\tau)}{12{,}000 \frac{Btu}{hr\text{-}ton}}$$

or $q_m = 1.43\, I(\tau)$ tons

Table 7-10 provides the *maintenance* cooling loads for one heat sink reservoir using selected time intervals. In this table the time refers to the elapsed time following the start of maintenance cooling. As the data in this table indicate, the maintenance cooling load decreases exponentially with time. It will be assumed that for times greater than 12 years the maintenance cooling load will remain constant at a load of 5 tons.

Table 7-10. Maintenance cooling loads for one heat sink reservoir, sample problem 6.

Time (months)	τ	$I(\tau)$	q_m (tons)
6	.30	9.15	13.1
9	.45	8.05	11.5
12	.60	7.35	10.5
15	.75	6.85	9.8
18	.90	6.45	9.3
24	1.19	5.85	8.4
30	1.49	5.45	7.8
36	1.79	5.20	7.5
42	2.09	5.00	7.2
48	2.39	4.80	6.9
54	2.69	4.65	6.7
60	2.99	4.50	6.5
66	3.29	4.38	6.3
72	3.59	4.28	6.1
78	3.88	4.20	6.0
84	4.18	4.12	5.9
90	4.48	4.05	5.8
96	4.78	3.99	5.7
102	5.08	3.92	5.6
108	5.38	3.88	5.6
114	5.68	3.82	5.5
120	5.98	3.77	5.4
126	6.27	3.74	5.4
132	6.57	3.69	5.3
138	6.87	3.65	5.2
144	7.17		5.1 (est)

Table 7-11 provides the total maintenance cooling for the entire heat sink system using phased loading. Note that for any particular heat sink, maintenance cooling does not occur until after that sink has been cooled to the desired initial temperature of 40°F.

Using the data provided in Table 7-11 and the results of the prior calculation of the initial cooling load, the total cooling requirements for this heat sink system can be determined. Specifically, a cooling rate of 39.4 tons is required to cool each heat sink reservoir to the desired initial temperature of 40°F. Since this cooling rate was calculated based on sequential cooling of the six reservoirs for a period of six months each, this implies a constant cooling load of 39.4 tons for the three-year preparation period. This cooling load, in conjunction with the cooling loads indicated in Table 7-11, provides the total system cooling load.

Table 7-11. Total maintenance cooling loads for sample problem 6.

Time (months)	Heat Sink #1	Heat Sink #2	Heat Sink #3	Heat Sink #4	Heat Sink #5	Heat Sink #6	Total
0	0	0	0	0	0	0	0
6	13.1	0	0	0	0	0	13.1
9	11.5	0	0	0	0	0	11.5
12	10.5	13.1	0	0	0	0	23.6
15	9.8	11.5	0	0	0	0	21.3
18	9.3	10.5	13.1	0	0	0	32.9
21	8.9	9.8	11.5	0	0	0	30.2
24	8.5	9.3	10.5	13.1	0	0	41.4
27	8.2	8.9	9.8	11.5	0	0	38.4
30	7.8	8.5	9.3	10.5	13.1	0	49.2
33	7.7	8.2	8.9	9.8	11.5	0	46.1
36	7.5	7.8	8.5	9.3	10.5	13.1	56.7
39	7.4	7.7	8.2	8.9	9.8	11.5	53.5
42	7.2	7.5	7.8	8.5	9.3	10.5	50.8
45	7.1	7.4	7.7	8.2	8.9	9.8	49.1
48	6.9	7.2	7.5	7.8	8.5	9.3	47.2
54	6.7	6.9	7.2	7.5	7.8	8.5	44.6
60	6.5	6.7	6.9	7.2	7.5	7.8	42.6
66	6.3	6.5	6.7	6.9	7.2	7.5	41.1
72	6.1	6.3	6.5	6.7	6.9	7.2	39.7
78	6.0	6.1	6.3	6.5	6.7	6.9	38.5
84	5.9	6.0	6.1	6.3	6.5	6.7	37.5
90	5.8	5.9	6.0	6.1	6.3	6.5	36.6
96	5.7	5.8	5.9	6.0	6.1	6.3	35.8
102	5.6	5.7	5.8	5.9	6.0	6.1	35.1
108	5.6	5.6	5.7	5.8	5.9	6.0	34.6
114	5.5	5.6	5.6	5.7	5.8	5.9	34.1
120	5.4	5.5	5.6	5.6	5.7	5.8	33.6
126	5.4	5.4	5.5	5.6	5.6	5.7	33.2
132	5.3	5.4	5.4	5.5	5.6	5.6	32.8
138	5.2	5.3	5.4	5.4	5.5	5.6	32.4
144	5.1	5.2	5.3	5.4	5.4	5.5	31.9
150	5.0	5.1	5.2	5.3	5.4	5.4	31.4
156	5.0	5.0	5.1	5.2	5.3	5.4	31.0
162	5.0	5.0	5.0	5.1	5.2	5.3	30.6
168	5.0	5.0	5.0	5.0	5.1	5.2	30.3
174	5.0	5.0	5.0	5.0	5.0	5.1	30.1
180	5.0	5.0	5.0	5.0	5.0	5.0	30.0

Using this procedure Figure 7-7 has been prepared to illustrate the total system cooling load as a function of time. Note that the cooling load increases during the initial years until it reaches a maximum of approximately 90 tons six months prior to the start of the

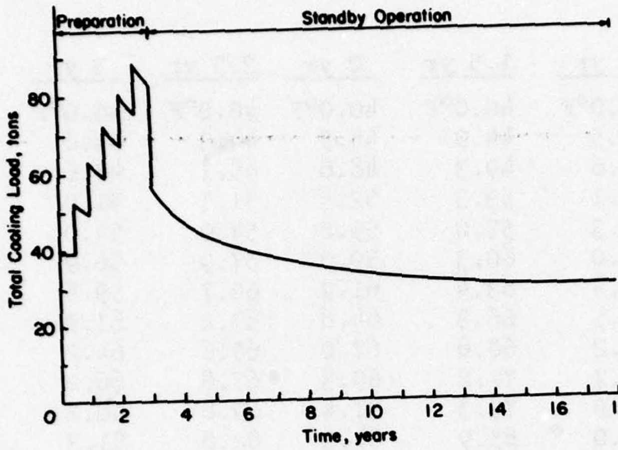

Figure 7-7. Total cooling load for sample problem 6.

standby period. During the standby period the cooling load decreases nonlinearly, eventually approaching a cooling load of 30 tons asymptotically.

The temperature gradient in the surrounding rock can be calculated through the use of equation (6-07) and Table 6-1. The rock temperature, as a function of radius and time, is given by

$$T(R,t) = T_\infty + (T_W - T_\infty)\beta(\frac{R}{a},\tau)$$

where

$$T_\infty = 90^oF$$
$$T_W = 40^oF$$
$$a = 23.75 \text{ ft}$$
$$\tau = \frac{kt}{\rho c a^2} .$$

From the previous calculations

$$\tau = 6.91x10^{-5}t \qquad 1/hr.$$

Thus

$$T(R,t) = 90 + (90-40)\beta(\frac{R}{a},\tau) \qquad ^oF.$$

Table 7-12 has been prepared using the above relationships and Table 6-1. The data illustrate the penetration of the cooling gradient during the period from six months to 18 years after cooling has begun. These same data are also presented graphically in Figure 7-8.

Table 7-12: Rock temperatures during cooling for sample problem 6.

R/a	Radius (feet)	0.5 yr	1 yr	1.5 yr	2 yr	2.5 yr	3 yr	18 yr
1.0	23.75	40.0°F	40.0°F	40.0°F	40.0°F	40.0°F	40.0°F	40.0°F
1.1	26.1	47.0	45.5	44.9	44.5	44.3	44.0	42.5
1.2	28.5	53.4	50.6	49.3	48.6	48.1	47.6	44.8
1.3	30.9	59.0	55.1	53.3	52.4	51.7	51.0	46.9
1.4	33.3	64.1	59.3	57.0	55.8	54.9	54.0	48.9
1.5	35.6	68.5	63.0	60.3	59.0	57.9	56.9	50.7
1.6	38.0	72.4	66.4	63.4	61.9	60.7	59.5	52.4
1.7	40.4	75.7	69.5	66.3	64.6	63.2	61.9	54.0
1.8	42.8	78.6	72.2	68.8	67.0	65.6	64.2	55.4
1.9	45.1	81.0	74.7	71.2	69.3	67.8	66.2	56.9
2.0	47.5	82.9	76.9	73.3	71.4	69.8	68.2	58.2
3.0	71.3	89.7	88.0	85.9	84.3	82.8	81.3	68.5
4.0	95.0	90.0	89.9	89.4	88.6	87.8	87.0	75.4
5.0	118.8	90.0	90.0	90.0	89.7	89.4	89.1	80.2
6.0	142.5	90.0	90.0	90.0	90.0	89.9	89.8	83.5
7.0	166.3	90.0	90.0	90.0	90.0	90.0	90.0	85.8
8.0	190.0	90.0	90.0	90.0	90.0	90.0	90.0	87.3
9.0	213.8	90.0	90.0	90.0	90.0	90.0	90.0	88.3
10.0	237.5	90.0	90.0	90.0	90.0	90.0	90.0	89.0

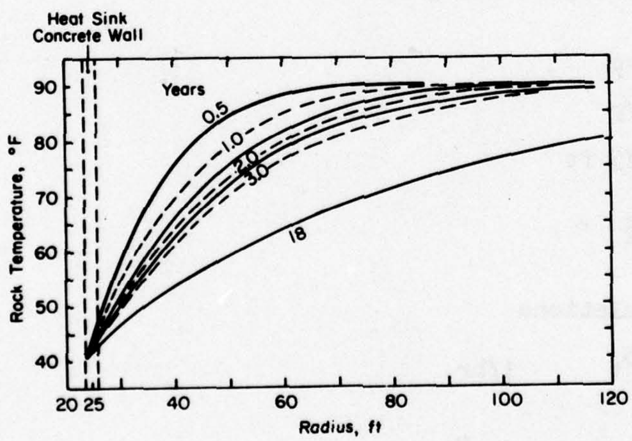

Figure 7-8. Thermal gradient in rock as a function of time.

To calculate the thermal gradient in the rock during operation equation 6-08 should be used, namely:

$$T(R,t_d) = T_o + (T_m - T_o)\beta(\frac{R}{a},\tau_d) .$$

To determine the mean wall temperature, T_m, during operation of the heat sink, it will be assumed that it is equal to the heat sink temperature when the power plant is at its average load condition. In the previous calculations an average waste heat load of 28.4×10^6 Btu/hr was assumed during sizing of the heat sink. The same average waste heat load will be assumed for these calculations.

From Appendix A, a waste heat load of 28.4×10^6 Btu/hr will occur when the heat sink temperature is approximately 104°F. Thus this temperature will be assumed to be equal to the mean wall temperature during operation. Therefore

$$\tau = \frac{kt}{\rho c a^2} = \frac{(1.45 \frac{Btu}{hr ft {}^\circ F})(30\ day)(24\ hr/day)}{(186 \frac{lb}{ft^3})(0.2 \frac{Btu}{lb {}^\circ F})(23.75\ ft)^2}$$

or $\tau = 0.05$.

Thus

$$T(R,t_d) = 40^\circ F + (104 - 40^\circ F)\beta(\frac{R}{a}, 0.05).$$

Using Table 6-1, Table 7-13 has been prepared. These data have also been plotted in Figure 7-9. This figure also contains a line indicating where the ambient temperature has increased five percent. This line intersects the thermal gradient line at a radius of approximately 39 ft. This indicates that the heat sink reservoirs should be located so that their center-to-center distance is at least 78 ft to insure that they do not thermally interact during the operating period.

(It should be noted that in predicting the penetration of this thermal gradient it is assumed that the surrounding rock has a uniform initial temperature of 40°F. In actuality the surrounding rock will have a thermal gradient similar to the ones depicted in Figure 7-8. Thus this procedure should be used only to provide a first order approximation of the minimum separation distances.)

Since the procedure utilized to determine the construction costs for this sample problem is identical to the procedures used in the preceding problems the calculation details will be omitted. The estimation procedures for the refrigeration costs, however, are provided below.

Table 7-13. Thermal gradient in surrounding rock at the end of the operating period, sample problem 6.

R/a	R (feet)	$\beta(\frac{R}{a},\tau)$	$T(R,t_d)$ (°F)
1.0	23.75	1.00	104.0
1.1	26.1	0.718	86.0
1.2	28.5	0.483	70.9
1.3	30.9	0.302	59.3
1.4	33.3	0.175	51.2
1.5	35.6	0.093	46.0
1.6	38.0	0.046	42.9
1.7	40.4	0.021	41.3
1.8	42.8	0.009	40.6
1.9	45.1	0.003	40.2
2.0	47.5	0.001	40.1

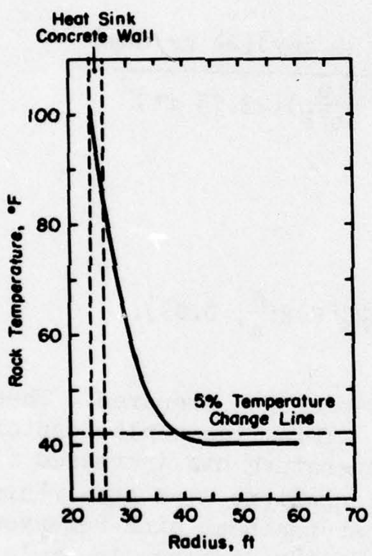

Figure 7-9. Temperature gradient in surrounding rock following operation of the heat sink, sample problem 6.

As indicated by Figure 7-7, a refrigeration plant with a capacity of approximately 90 tons would provide sufficient cooling during the preparation period. Using a visual estimation technique it is estimated that during these first three years this refrigeration plant would operate at an average load of approximately 70 tons.

If an evaporator temperature of -20°F is assumed, then from Appendix F, curve B of Figure F-3, approximately 2.6 HP will be required for each ton of refrigeration. Thus the average annual electrical consumption during the first three-year period would be:

electrical consumption: = (70 tons)(2.6 HP/ton)(0.746 kW/HP) = 136 kW

and therefore for 24-hour-per-day operation:

electrical power/year = (136 kW)(24 hr/day)(365 day/yr) = $1.19\text{x}10^6$ kW-hr/yr.

The salary and supply requirements to support the plant are calculated using the procedures discussed in Appendix F.

Using Figure 7-7 to also approximate the maintenance cooling requirements, it is estimated that a refrigeration plant with a capacity of 60 tons will be required. This capacity will be provided through the utilization of part of the refrigeration system originally installed to cool the sinks during the first three years. In addition it is estimated that this plant will operate at an average load of approximately 45 tons during the 15-year standby period.

Thus the electrical power required during standby will be:

electrical consumption = (45 tons)(2.6 HP/ton)(0.746 kW/HP) = 87 kW.

Thus for 24-hour-a-day operation the average annual consumption will be

electrical power/year = (87 kW)(24 hr/day)(365 day/yr) = $7.62\text{x}10^5 \frac{\text{kW-hr}}{\text{yr}}$.

Salary, supply and overhaul costs for the maintenance cooling period are estimated using the procedures described in Appendix F.

Table 7-14 summarizes the cost estimate for this sample problem.

7-07.7. Sample Problem 7: Ice Block and Water Heat Sink: This sample problem considers a heat sink system where fifty percent of the heat sink mass is composed of ice blocks and fifty percent is chilled (32°F) water.

If the exponential approximation for the power plant waste heat load is used, namely

$$q_r(T) = B_o e^{B_1 T}$$

where $B_o = 20.125\text{x}10^6$ Btu/hr

$B_1 = 3.512\text{x}10^{-3}$ 1/°F

Table 7-14. Cost estimate for sample problem 6.

		Unit Cost	Quantity	Inflation Factor	Item Cost	Subtotals
A	Construction					
1	Excavation	$75/yd^3$	1.11×10^5 yd^3	-	\$ 8,325,000	
2	Reinforced concrete	$120/yd^3$	2.38×10^4 yd^3	-	2,856,000	
3	Waterproofing	$0.31/ft^2$	2.13×10^5 ft^2	-	66,000	
						\$11,247,000
B	Preparation					
1	Water	\$.04/100 gal	1.77×10^7 gal	-	\$ 7,000	
2	Refrigeration unit	\$1850/ton	90 tons	-	\$ 167,000	
3	Salary	\$$1.2\times10^5$/yr	3 years	1.051	\$ 378,000	
4	Supplies	\$225/ton-yr	3 years, 90 tons	1.085	\$ 66,000	
5	Electricity	\$.035/kw-hr	1.19×10^6 kw-hr/yr, 3 years	1.085	\$ 136,000	
						\$ 754,000
C	Maintenance					
1	Electricity	\$.035/kw-hr	7.62×10^5 kw-hr/yr, 15 yrs	2.142	\$ 857,000	
2	Salary	\$$1.2\times10^5$/yr	15 years	1.563	\$ 2,813,000	
3	Supplies	\$225/ton-yr	15 years, 60 tons	2.142	\$ 434,000	
4	Overhaul	\$250/ton-overhaul	60 tons-5 overhauls	1.853	\$ 140,000	
						\$ 4,244,000
					Total cost	\$16,245,000

then equation (6-13a) can be used to predict the design life of the heat sink. Thus

$$t_d = \left[\frac{8B_1}{B_o}\right]\left[\frac{158\,M_i + 14M_w}{e^{46B_1} - e^{38B_1}}\right] + \frac{c_w[M_i + M_w]}{B_o B_1}\left[e^{-46B_1} - e^{-B_1 T_f}\right]$$

where T_f = final sink temperature

M_i = mass of ice

M_w = mass of water

c_w = specific heat of water.

Since the final sink temperature will be 160°F and the specific heat of water is 1.0 Btu/lb°F substitution into the above equation and simplifying yields:

$$t_d = \frac{(1.396x10^{-9})(158\ M_i + 14\ M_w)}{e^{.1616} - e^{.1335}} + (1.415x10^{-5})(M_i + M_w)(e^{-.1616} - e^{-.5619}).$$

Since one-half of the initial total heat sink mass, M_T, will be ice and one-half will be water, the following relationship can be developed:

$$M_i = 0.5\ M_T$$
$$M_w = 0.5\ M_T$$
$$M_i = M_w.$$

Using this last relationship and substituting into the expression for sink design life yields, after simplification,

$$t_d = (4.363x10^{-8})(172\ M_i) + (3.972x10^{-6})(2\ M_i).$$

But since a design life of 31.5 days (7560 hr) is required, then

$$7560 = 7.50x10^{-6}\ M_i + 7.944x10^{-6}\ M_i$$

or: $$M_i = \frac{7560}{1.545x10^{-5}} = 4.893x10^7\ \text{lb}.$$

Thus the heat sink system must contain at least $4.893x10^7$ lb of ice and $4.893x10^7$ lb of water to provide the proper heat storage capability. The total volume of the heat sink system can be rapidly calculated using

$$V_T = \frac{M_i}{\rho_i} + \frac{M_w}{\rho_w}$$

where ρ_i = density of ice (57.0 lb/ft^3)

and ρ_w = density of water (62.4 lb/ft^3).

Upon substitution this yields:

$$V_T = \frac{4.893x10^7\ \text{lb}}{57.0\ \text{lb/ft}^3} + \frac{4.893x10^7\ \text{lb}}{62.4\ \text{lb/ft}^3}$$

or $$V_T = 1.643x10^6\ \text{ft}^3.$$

Since it has been assumed that the upper domes of these heat sink reservoirs will not be filled, the volume of each reservoir is given by:

$$V_R = \frac{26}{3} \pi R^3.$$

If a total of three heat sink reservoirs is assumed, the radius of each may be calculated using:

$$3V_R = 3\left(\frac{26}{3} \pi R^3\right) = 1.643 \times 10^6 \text{ ft}^3$$

and thus $R = 27.2$ ft.

However, this radius violates the assumed design criteria, which require a radius less than 25 ft. Reiterating using four reservoirs yields:

$$4V_R = 4\left(\frac{26}{3} \pi R^3\right) = 1.643 \times 10^6 \text{ ft}^3$$

or $R = 24.71$ ft

which is acceptable. Thus the heat sink system would consist of four reservoirs 49.4 ft in diameter by 247 ft long.

The temperature-time history of this heat sink system can be predicted through the use of equations 6-13b, c and d. The time at which all the ice will be melted, t_m, is given by:

$$t_m = \left[\frac{8B_1}{B_o}\right]\left[\frac{158\, M_i + 14\, M_w}{e^{46B_1} - e^{38B_1}}\right].$$

Thus, since

$$B_1 = 20.125 \times 10^6 \text{ Btu/hr}$$
$$B_1 = 3.512 \times 10^{-3} \text{ 1/°F}$$
$$M_i = M_w = 4.893 \times 10^7 \text{ lb}$$

$$t_m = \frac{(8)(20.125 \times 10^6)}{3.512 \times 10^{-3}}\left[\frac{158 + 14}{e^{.1616} - e^{.1335}}\right](4.893 \times 10^7)$$

and therefore

$$t_m = 367.1 \text{ hr} = 15.3 \text{ days}.$$

Thus the sink outlet temperature during the first 15 days is given by equation 6-13c, namely

$$T_{sink}(t) = \frac{8}{t_m} t + 38$$

and therefore $$T_{sink}(t) = \frac{8}{15.3} t + 38 \quad {}^{o}F$$

or $$T_{sink}(t) = 0.523t + 38 \quad {}^{o}F \quad \text{for } t \leq t_m$$

where t is in days.

Once all the ice has been melted the outlet temperature of the sink can be predicted through the use of equation 6-13d, namely

$$T_{sink}(t) = -\frac{1}{B_1} \ln \left\{ e^{-46B_1} - \frac{B_o B_1 (t-t_m)}{(M_i+M_w)c_w} \right\}.$$

Therefore, upon substitution of the appropriate constants this becomes

$$T_{sink}(t) = -\frac{1}{3.512x10^{-3}} \ln \left\{ e^{-46(3.512x10^{-3})} - \frac{(20.125x10^6)(3.512x10^{-3})(t-367.1)}{(4.893x10^7 + 4.893x10^7)(1.0)} \right\}$$

(note: in this equation t and t_m must have units of hr).

And thus, upon simplification this becomes:

$$T_{sink}(t) = 2.85x10^2 \ln \{.8508 - 7.22x10^{-4} (t-367.1)\} \text{ for } t>t_m .$$

Using this and the previously developed relationship, Table 7-15 has been prepared to illustrate the heat sink temperature as a function of time in operation.

Figure 7-10 provides a graphical representation of the heat sink temperature as a function of time in operation. This figure also contains the power plant waste heat load and the required coolant water flow rate data. These curves were developed by correlating the heat sink temperature with the data provided in Appendix A.

Table 7-15. Temperature-time history of the sample problem 7 heat sink system.

Time (days)	Heat Sink Temperature (°F)
0	38.0
1	38.5
3	39.6
5	40.6
7	41.7
9	42.7
11	43.8
13	44.8
15	45.8
17	56.1
19	68.4
21	81.2
23	94.7
25	108.8
27	123.7
29	139.3
31	155.9
31.5	160.2

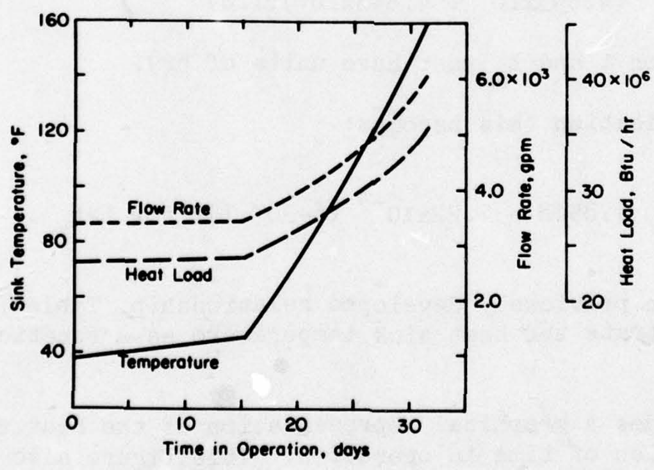

Figure 7-10. Performance of sample problem 7 heat sink system.

The preparation requirements for this heat sink system include manufacture of the ice particles, chilling of the water, and cooling of the rock surrounding the sink reservoirs. Maintenance requirements consist only of long term cooling of the surrounding rock.

It is assumed that ice machines capable of producing 2000 lb of ice per day will be used to manufacture the ice (see Appendix F). Thus the minimum number of ice machines required can be calculated using:

$$\text{no. of ice machines} = \frac{4.893\text{x}10^7 \text{ lb}}{(3 \text{ years})(365 \frac{\text{day}}{\text{yr}})(2000 \text{ lb/day/machine})}$$

$$\text{no. of ice machines} = 22.3.$$

Since there are four heat sinks and some melting of the ice is bound to occur, a total of 24 ice machines (i.e. 6 machines/sink) will be used.

If it is assumed that the water to be used to fill the reservoirs will have an initial temperature of 80°F then the water cooling load is given by:

$$q_c = \frac{(4.893\text{x}10^7 \text{ lb})(1.0 \frac{\text{Btu}}{\text{lb}^\circ\text{F}})(80\text{-}32^\circ\text{F})}{(3 \text{ years})(365 \frac{\text{day}}{\text{yr}})(24 \frac{\text{hr}}{\text{day}})(12{,}000 \text{ Btu/hr/ton})}$$

or $q_o = 7.5$ tons.

To provide some surplus capacity a 10-ton water chiller will be used.

The rock cooling load can be calculated through the use of equation 6-18, namely:

$$q_s = \frac{\ell k(T_\infty - T_o)\alpha(\tau)}{1500\pi} \text{ , tons}$$

where $\tau = \dfrac{kt}{\rho c a^2}$.

Using the following data:

ℓ = 247 ft

k = 1.45 Btu/hr ft°F

T_∞ = 90°F

T_o = 32°F

ρ = 186 lb/ft^3

c = 0.2 Btu/lb°F

a = 24.7 ft

and substituting and simplifying yields:

$\tau = 6.389 \times 10^{-5} t$ where t is the time in hours

and

$q_s = 4.408\ \alpha(\tau)$ where q_s is in tons of refrigeration.

Using these two relationships and Table 6-2 the cooling load for each reservoir as a function of time can be determined. Table 7-16 presents the results of these calculations. This table also contains

Table 7-16. Rock cooling loads for sample problem 7.

t (days)	τ	$\alpha(\tau)$	$q_s(t)$ (tons)	$q_T(t)$ (tons)
6.5	.01	15.122	66.7	266.8
13.0	.02	11.033	48.6	194.5
19.6	.03	9.218	40.6	162.5
32.6	.05	7.394	32.6	130.4
65.2	.10	5.549	24.5	97.8
97.8	.15	4.726	21.0	83.9
130.5	.20	4.232	18.7	74.6
195.7	.30	3.643	16.1	64.2
260.9	.40	3.288	14.5	58.0
326.2	.50	3.044	13.4	53.7
489.2	.75	2.660	11.7	46.9
652.3	1.00	2.427	10.9	43.6
978.5	1.50	2.147	9.5	37.9
1305	2.00	1.975	8.7	34.8
1631	2.50	1.856	8.2	32.7
1957	3.00	1.767	7.8	31.2
2609	4.00	1.639	7.2	28.9
3262	5.00	1.550	6.8	27.3
4892	7.50	1.406	6.2	24.8
6523	10.00	1.317	5.8	23.2

the total cooling load for all the sinks, q_T, assuming that they are cooled simultaneously. The total rock cooling load for all the sinks is also illustrated in Figure 7-11. As both Table 7-16 and Figure 7-11 illustrate, the rock cooling load decreases exponentially from its maximum value of approximately 270 tons. Within two months after the start of cooling the load will have decreased to less than one-half of its initial value. Thereafter the load continues to decrease but at a rate which itself is decreasing.

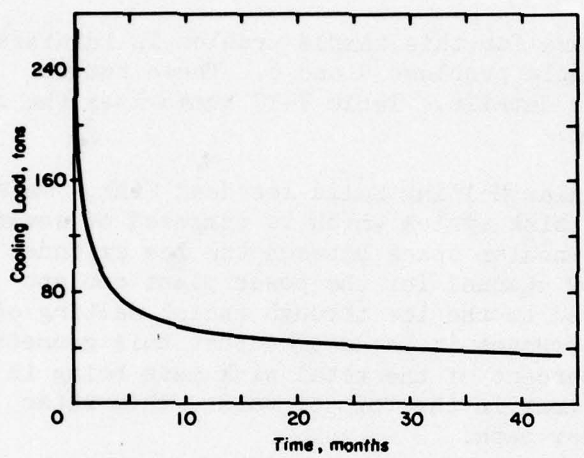

Figure 7-11. Total rock cooling load for sample problem 7.

Because of this high rate of change of the rock cooling load all equipment will be sized based on anticipated average loads rather than anticipated peak loads. This procedure recognizes that load averaging techniques, such as those employed in sample problem 4, can be utilized to reduce the installed plant capacity without affecting performance.

To determine the average cooling load that will be encountered during any particular time period the following procedure is employed. First the area under the cooling load curve (Figure 7-11) for that time interval is calculated. This area represents the total amount of cooling which must be supplied by the refrigeration plant during that interval*.

Using this procedure the average cooling rate for the first three years can be calculated as follows. The area beneath the cooling curve between the limits of time = 0 months and time = 36 months is equal to 1978 ton-months. Dividing this figure by the length of the period (i.e. 36 months) produces an average load of approximately 55 tons.

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*Note: Since 1 ton is equivalent to 12,000 Btu/hr, the area beneath the cooling curve, which is the product of cooling rate (tons) multiplied by time (months), will have units of Btu. Subsequent division by the length of the period (time) again will produce a cooling rate (tons or Btu/hr) representing the average cooling load during that period.

If a similar technique was employed for the third through the eighteenth years it could be shown that the area beneath the cooling curve is equal to 4790 ton-months. Thus, since this period is 180 months long the average load would be equal to 4790 divided by 180 or approximately 27 tons.

To provide sufficient surplus cooling capacity an 80-ton refrigeration plant will be selected for use during the first three years of cooling. At the end of this period, 40 tons of this system would be deactivated and the remaining 40 tons would be used to provide maintenance cooling.

The cost estimation procedure for this sample problem is identical to the procedure utilized in sample problems 4 and 6. These sample problems should be consulted for details. Table 7-17 summarizes the cost estimate for this sample problem.

7-07.8. Sample Problem 8: Annular Melting Solid Ice Heat Sink. This sample problem considers a heat sink system which is composed of several solid ice cylinders. Using an annular space between the ice cylinder and the reservoir wall as a flow channel for the power plant coolant water, the waste heat is rejected to the ice through radial melting of the ice cylinder. For design purposes it is assumed that this geometric arrangement will result in 90 percent of the total sink mass being in the form of ice and only 10 percent in the form of water, this water being located in the annular flow path.

The thermal analysis of this heat sink system is identical to the procedure employed in the previous sample problem. Thus, using the exponential approximation for the power plant waste heat load and substituting the appropriate values into equation 6-13a (see sample problem 7 for details) yields:

$$t_d = \frac{(1.396x10^{-9})(158M_i+14M_w)}{e^{.1616} - e^{.1335}} + (1.415x10^{-5})(M_i+M_w)(e^{-.1616} - e^{-.5619}).$$

Since 90 percent of the total sink mass, M_T, will initially be ice, then:

$$M_i = 0.90\ M_T$$

and

$$M_w = 0.10\ M_T.$$

Using these two relationships it can easily be shown that:

$$M_i = 9M_w.$$

Table 7-17. Cost estimate for sample problem no. 7.

	Unit Cost	Quantity	Inflation Factor	Item Cost	Subtotals
A **CONSTRUCTION**					
1 Excavation	$\$75/yd^3$	$8.27 \times 10^4\ yd^3$	-	\$ 6,203,000	
2 Reinforced concrete	$\$120/yd^3$	$1.71 \times 10^4\ yd^3$	-	2,052,000	
3 Waterproofing	$\$.31/ft^2$	$1.53 \times 10^5\ ft^2$	-	47,000	
					\$ 8,302,000
B **PREPARATION**					
1 Water	\$.04/100 gal	1.17×10^7 gal	-	\$ 6,000	
2 Ice machines	\$10,000 ea	24	-	240,000	
3 Refrigeration units	\$1850/ton	80 tons	-	148,000	
4 Salary	$\$1.2 \times 10^5$/yr	3 years	1.051	378,000	
5 Supplies, ice machines	\$500 /unit-yr	[3 years, 24 units	1.085	39,000	
6 Supplies, refrigeration units	\$225/ton yr	[80 tons, 3 years	1.085	59,000	
7 Electricity, total	\$.035/kw-hr	[2.63×10^6 $\frac{kwhr}{yr}$, 3 years	1.085	300,000	
					\$ 1,170,000
C **MAINTENANCE**					
1 Electricity	\$.035/kw-hr	[15 yr, 6.8×10^5 $\frac{kwhr}{yr}$	2.142	\$ 765,000	
2 Salary	$\$1.2 \times 10^5$/yr	15 yr	1.563	2,813,000	
3 Supplies	\$225/ton-yr	[40 ton, 15 yr	2.142	289,000	
4 Overhauls	\$250/ton-overhaul	[40 ton, 5 overhauls	1.853	93,000	
					\$ 3,960,000
				Total Cost	\$13,432,000

Thus, substituting for M_i in the relationship for the heat sink design life, t_d, yields, after simplification:

$$t_d = 6.265 \times 10^{-5}\ M_w + 3.972 \times 10^{-5}\ M_w.$$

However, since the sink must be capable of operating for a period of 31.5 days, then:

$$t_d = 31.5 \text{ days} = 7560 \text{ hr.}$$

Therefore, substituting for t_d and solving for M_i yields:

$$M_i = \frac{7560}{(6.265x10^{-5} + 3.972x10^{-5})}$$

or $$M_i = 7.383x10^6 \text{ lb.}$$

Therefore, using the relationships between M_i, M_w and M_T yields:

$$M_i = 9\ M_w = 6.645x10^7 \text{ lb}$$

and

$$M_T = M_i + M_w = 7.383x10^7 \text{ lb.}$$

To determine the volume of the heat sink system the relationship:

$$V_T = V_i + V_w$$

where

$$V_i = \frac{M_i}{\rho_i}$$

and $$V_w = \frac{M_w}{\rho_w}$$

can be utilized. Substituting the appropriate values for M_i and M_w from above and using

$$\rho_w = 62.4 \text{ lb/ft}^3$$
$$\rho_i = 57.0 \text{ lb/ft}^3$$

yields:

$$V_T = \frac{6.645x10^7 \text{ lb}}{57.0 \text{ lb/ft}^3} + \frac{7383x10^6 \text{ lb}}{62.4 \text{ lb/ft}^3}$$

or $\qquad V_T = 1.284 \times 10^6 \ ft^3.$

If a total of three ice reservoirs is assumed, then using the relationship

$$V_R = \frac{26}{3} \pi R^3$$

yields (note that for the ice heat sinks it is assumed that the upper dome of the reservoir is used for equipment)

$$(3)(\frac{26}{3} \pi R^3) = 1.284 \times 10^6 \ ft^3.$$

Therefore, solving for R yields:

$$R = 25.05 \ ft.$$

Although this calculated radius does exceed the maximum assumed radius by 0.05 ft it will be accepted for the purpose of this sample problem.

The temperature-time history of this heat sink system can be calculated through the use of equations 6-13b, 6-13c and 6-13d. Equation 6-13b, namely,

$$t_m = \left[\frac{8 B_1}{B_o}\right] \left[\frac{158 M_i + 14 M_w}{e^{46B_1} - e^{38B_1}}\right]$$

can be used to determine the time at which melting of the ice will have been completed. Substituting the appropriate values into each of the parameters of this relationship yields:

$$t_m = \frac{(8)(3.512 \times 10^{-3})}{(20.125 \times 10^6)} \quad \frac{[(158)(6.645 \times 10^7) + (14)(7.383 \times 10^6)]}{[e^{(3.512 \times 10^{-3})(46)} - e^{(38)(3.512 \times 10^{-3})}]}$$

or

$$t_m \approx 463 \ hr.$$

Therefore, using this value for t_m and substituting the appropriate values into equations 6-13c and 6-13d yields the following:

$$T(t) = \frac{8}{463}t + 38 \qquad \text{for} \quad t \leq 463 \text{ hr}$$

and

$$T(t) = \frac{-1}{(3.512\text{x}10^{-3})} \ln \left\{ e^{-46(3.512\text{x}10^{-3})} - \frac{(20.125\text{x}10^{6})(3.512\text{x}10^{-3})(t-463)}{(6.645\text{x}10^{7} + 7.353\text{x}10^{6})(1.0)} \right\}$$

for $t > 463$ hr.

Simplifying this second relationship yields:

$$T(t) = -284.74 \ln \left[1.294 - 9.573\text{x}10^{-4} t \right] \qquad \text{for } t > 463 \text{ hr.}$$

Using this and the preceding relationship, Table 7-18 has been prepared, illustrating the variation in the heat sink temperature as a function of time. These data also have been plotted in Figure 7-12. The data for power plant waste heat load and coolant water flow rate were developed by correlating the heat sink temperatures provided in Table 7-18 with the data provided in Appendix A.

Table 7-18. Temperature-time history for sample problem 8 heat sink system.

Time (days)	Time (hr)	T_{sink} (°F)
0	0	38.0
2	48	38.8
4	96	39.7
6	144	40.5
8	192	41.3
10	240	42.1
12	288	43.0
14	336	43.8
16	384	44.6
18	432	45.5
19.3	463	46.0
20	480	51.5
22	528	67.6
24	576	84.7
26	624	102.9
28	672	122.4
30	720	143.2
31.5	756	159.9

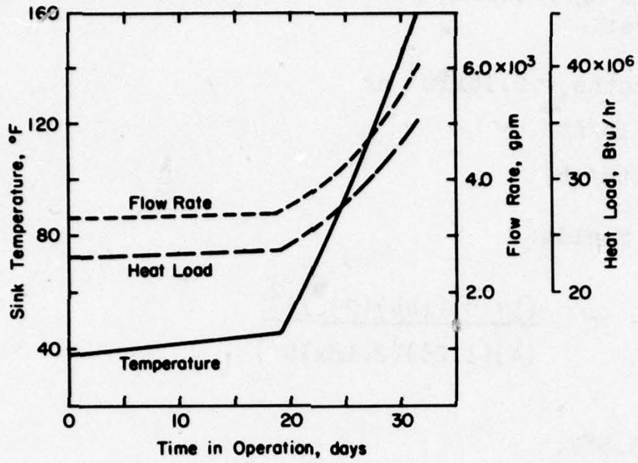

Figure 7-12: Performance of the sample problem 8 heat sink system.

During the preparation period of this heat sink system three distinct types of cooling loads will be encountered: initial cooling of the water in the sink to 32°F, cooling to freeze the water, and cooling of the surrounding rock to prevent unnecessary heat flow from the surroundings. Each of these cooling loads is analyzed in the following sections.

Assuming that radial freezing will be used the cooling load related to freezing of the water in the heat sink reservoirs depends on the length of the freezing period and the reservoir wall temperature. Equation 6-19b, namely

$$t = \frac{\rho_i \lambda R_o^2}{4k_i(32-T_\omega)} \left[1-\left(\frac{R_a}{R_o}\right)^2 \left\{1-2\,ln\left(\frac{R_a}{R_o}\right)\right\}\right]$$

provides the relationship between these two parameters and the radius of the frozen annulus. For complete freezing the radius of the frozen annulus, R_a, will be zero and this equation reduces to:

$$t = \frac{\rho_i \lambda R_o^2}{4k_i(32-T_\omega)} \quad .$$

If a 30-month-long freezing period is assumed, then using the following parameters:

$$t = 30 \text{ months} = 2.16\text{x}10^4 \text{ hr}$$

$$\rho_i = 57.0 \text{ lb/ft}^3$$

$$\lambda = 144 \text{ Btu/lb}$$

and solving for T_ω yields:

$$T_\omega = 32 - \frac{(57.0)(144)(25.1)^2}{(4)(1.26)(2.16\text{x}10^4)}$$

or $$T_\omega = -15.5°\text{F}.$$

Substituting this back into equation 6-19b yields:

$$t = 2.16\text{x}10^4 \left[1 - \left(\frac{R_a}{R_o}\right)^2 \left\{1 - 2\, ln\left(\frac{R_a}{R_o}\right)\right\}\right].$$

From this relationship the growth of the ice annulus as a function of time can be determined. To determine the cooling loads during this process, equation 6-19c, namely

$$q_c = \frac{2\pi \ell k_i (32 - T_\omega)}{12{,}000\; ln\left(\frac{R_o}{R_a}\right)}$$

must be employed. Using the previously cited constants and ℓ=226 ft (note that the upper dome is not used for ice storage), this becomes:

$$q_c = \frac{7.08}{ln\left(\frac{R_o}{R_a}\right)}$$

Using this relationship and the previous relationship between time and the ice annulus it is possible to simultaneously determine the variation of both the ice annulus and the cooling load during the freezing period. Table 7-19 presents the results of this computation procedure. The variation in heat sink cooling as a function of time is also illustrated in Figure 7-13. Note that the very high initial

Table 7-19. Per sink cooling loads for freezing the sample problem 8 heat sinks.

Radius of ice annulus R_a (ft)	Time (month)	Cooling load q_c (tons)
24.5	.03	290
24.0	.11	156
23.5	.24	106
23.0	.41	80
22.75	.50	71
22.5	.63	64
22.0	.89	53
21.8	1.00	50
21.5	1.19	45
21.05	1.50	40
20.5	1.91	35
20.4	2.00	34
20.0	2.32	31
19.8	2.50	30
19.25	3.00	26
19.0	3.27	25
18.75	3.50	24
18.3	4.00	22
18.0	4.36	21
17.85	4.50	21
17.45	5.00	19
17.05	5.50	18
16.65	6.00	17
16.0	6.90	16
15.95	7.00	15
15.6	7.50	15
15.2	8.00	14
15.0	8.34	14
13.0	11.48	11
11.0	14.89	8
9.0	18.43	7
7.0	21.94	5
5.0	25.23	4
3.0	28.05	3
1.0	29.96	2
0.5	30.2	2
0.1	30.3	1

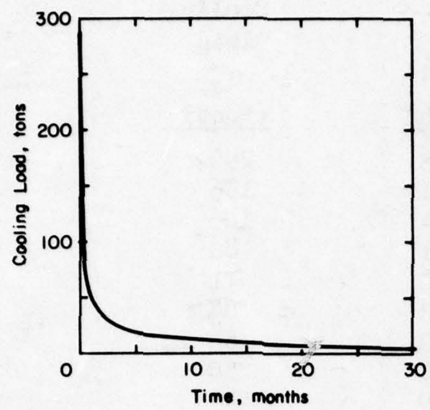

Figure 7-13. Per sink cooling loads to freeze the sample problem 8 heat sink reservoirs.

load tends to decrease significantly during the first few months of operation. Thereafter the cooling load continues to decrease but at a reduced rate.

Since this heat sink system would employ three ice reservoirs, if the reservoirs were to be frozen simultaneously the maximum ice cooling load would be three times the maximum value of Table 7-19 or approximately 870 tons. However, if the sinks are cooled in a sequential manner the maximum ice freezing cooling load can be reduced. Since approximately 30 months will be required to freeze each heat sink the cooling procedure shown in Table 7-20 has been assumed.

In preparing Table 8-20 the start of rock cooling for reservoirs no. 2 and no. 3 has been time-phased to commence 1.5 months prior to the filling of the reservoirs to allow each reservoir to be preccoled.

Using this procedure, all the water required by this heat sink system must be chilled to 32°F within 6 months. Assuming that the initial water temperature will be 80°F, the water cooling load will be:

$$Q = M_T c_w \Delta T = (7.383\text{x}10^7 \text{ lb})(1.0 \frac{\text{Btu}}{\text{lb}^\circ\text{F}})(80\text{-}32^\circ\text{F}) \quad \text{or}$$

$$Q = 3.544\text{x}10^9 \text{ Btu.}$$

Table 7-20. Assumed cooling procedure for preparation of the sample problem 8 heat sink system.

Time from the start of the preparation period (months)		Action to be taken
0	a.	commence filling reservoir no. 1 with chilled water
	b.	commence rock cooling for reservoir no. 1
0.5	a.	commence rock cooling for reservoir no. 2
2.0	a.	complete filling reservoir no. 1 with chilled water
	b.	commence freezing of reservoir no. 1
	c.	commence filling reservoir no. 2 with chilled water
2.5	a.	commence rock cooling for reservoir no. 3
4.0	a.	complete filling reservoir no. 2 with chilled water
	b.	commence freezing of reservoir no. 2
	c.	commence filling reservoir no. 3 with chilled water
6.0	a.	complete filling reservoir no. 3 with chilled water
	b.	commence freezing of reservoir no. 3
32.0	a.	complete freezing of reservoir no. 1
34.0	a.	complete freezing of reservoir no. 2
36.0	a.	complete freezing of reservoir no. 3

Converting this into a cooling rate yields

$$q = \frac{Q}{\text{time}} = \frac{3.544\text{x}10^9 \text{ Btu}}{(6 \text{ month})(30 \frac{\text{day}}{\text{month}})(24 \frac{\text{hr}}{\text{day}})} = 8.204\text{x}10^5 \frac{\text{Btu}}{\text{hr}}$$

or using the relationship that one ton of cooling is equal to 12,000 Btu/hr the cooling rate is

$$q = \frac{8.204\text{x}10^5 \text{ Btu/hr}}{12{,}000 \text{ Btu/hr/ton}} = 68.4 \text{ tons.}$$

Thus a 70-ton-capacity water chilling system will provide sufficient cooling capability to fill the reservoirs with chilled water.

The final cooling requirement which must be determined is related to the cooling of the rock surrounding each reservoir. In estimating this cooling load it should be remembered that since a reservoir wall temperature of -15°F is required to freeze the heat sinks, the walls of the sink chambers must also be cooled to a temperature of at least -15°F during the ice freezing period.

The rock cooling load* is calculated through the use of equation 6-18, namely

$$q_s = \frac{\ell k \, (T_\infty - T_o)\alpha(\tau)}{1500\pi}$$

where
$\ell = 226$ ft
$k = 1.45$ Btu/hr ft°F
$T_\infty = 90°F$
$T_o = -15°F$
$\tau = kt/\rho c a^2$
$\rho = 186$ lb/ft^3
$c = 0.2$ Btu/lb°F
$\alpha(\tau)$ = is from Table 6-2.

Upon substitution and simplification this yields:

$\tau = 6.19 \times 10^{-5} t$ where t is in hours

and

$q_s = 7.30\alpha(\tau)$ where q_s is in tons

Using these two relationships and Table 6-2 the rock cooling load as a function of time has been calculated. Table 7-21 presents the results of these calculations. These data are also presented graphically on Figure 7-14.

*Note: It is assumed that there will be no latent heat load during cooling of the surrounding rock. This assumption implies that none of the fractures or crevices in the rock surrounding each reservoir will contain a significant amount of water.

Table 7-21. Per sink rock cooling loads for sample problem 8.

Time (months)	(years)	τ	α(τ)	q_s (tons)
.22	.02	.01	15.122	110
.45	.04	.02	11.033	81
.90	.08	.04	8.135	59
1.57	.13	.07	6.421	47
2.02	.17	.09	5.790	42
3.14	.26	.14	4.854	35
4.04	.34	.18	4.405	32
4.93	.41	.22	4.083	30
5.61	.47	.25	3.894	28
6.05	.50	.27	3.785	28
6.95	.58	.31	3.600	26
8.07	.67	.36	3.412	25
8.97	.75	.40	3.288	24
11.9	.99	.53	2.984	22
15.0	1.3	.67	2.760	20
18.2	1.5	.81	2.595	19
21.1	1.8	.94	2.475	18
24.7	2.1	1.1	2.357	17
29.2	2.4	1.3	2.240	16
33.6	2.8	1.5	2.147	16
47.1	3.9	2.1	1.948	14
71.8	6.0	3.2	1.737	13
98.7	8.2	4.4	1.600	12
119.	9.9	5.3	1.528	11
157.	13.1	7.0	1.429	10
179.	14.9	8.0	1.386	10
202.	16.8	9.0	1.349	10
215.	17.9	9.6	1.329	10

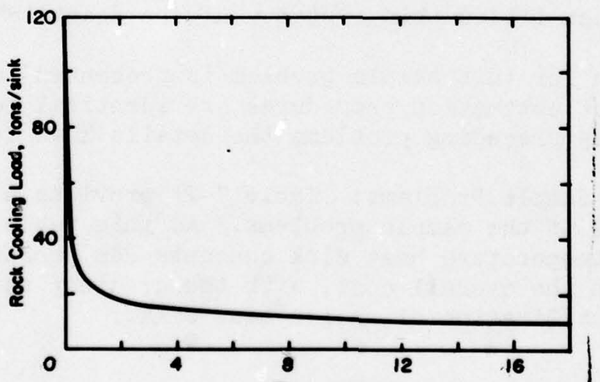

Figure 7-14. Per sink rock cooling loads for sample problem 8.

Using the operational procedure presented in Table 7-20 and the appropriate data from Table 7-19 and Table 7-21 the total rock and ice freezing cooling loads for this heat sink system as a function of time have been determined (Table 7-22). The variation in this cooling load during the first three years of operation (i.e. the preparation period) is also presented graphically on Figure 7-15. It should be noted that while this figure does indicate that several substantial peak loads will be encountered, the magnitude of these peak loads will be significantly lower than the peak load that would be encountered if the reservoirs were frozen simultaneously.

The total refrigeration plant capacity for this sample problem has been determined using the procedures employed in sample problems 4 and 7. These procedures rely on the use of load averaging to reduce the total installed tonnage of the refrigeration plant. The details of this estimation procedure are presented in sample problem 4.

Using this type of analysis a 250-ton capacity will be required to freeze the heat sinks and cool the surrounding rock. During the first year of operation this plant will operate at an average load of approximately 166 tons. During the second and third years this plant would operate at an average load of only 76 tons.

For the maintenance cooling period (i.e. the third through eighteenth years) a 50-ton refrigeration plant would be sufficient. This plant would operate at an average load of approximately 35 tons. This cooling requirement will be provided through the use of a portion of the 250-ton plant required for ice freezing and rock cooling. The remaining 200 tons of this plant would be deactivated at the end of the preparation period.

In addition to the cooling requirements cited above, a 70-ton water chilling system will be required during the first six months of the preparation period to fill the reservoirs with chilled water. Once all the reservoirs have been filled this system would be deactivated.

The cost estimate for this sample problem is presented in Table 7-23. Because the cost estimation procedures are identical to the procedures utilized in the preceding problems the details have been omitted.

7-08. Summary of the Sample Problems: Table 7-24 provides a brief summary of the results of the sample problems. As this table demonstrates, all the low temperature heat sink concepts can produce substantial reductions in the overall cost, with the greatest savings realized through the utilization of an ice heat sink.

Table 7-22. Total cooling loads for the sample problem no. 8 heat sink system.

	Reservoir Cooling Loads (tons)						
	Sink #1		Sink #2		Sink #3		
Time (months)	Rock	Ice Freezing	Rock	Ice Freezing	Rock	Ice Freezing	Total cooling load (tons)
0	110	0	0	0	0	0	110
0.5	74	0	110	0	0	0	184
1.0	57	0	74	0	0	0	131
1.5	48	0	57	0	0	0	105
2.0	42	290	48	0	0	0	380
2.5	38	71	42	0	110	0	261
3.0	36	50	38	0	74	0	198
3.5	33	40	36	0	57	0	166
4.0	32	34	33	290	48	0	437
4.5	30	30	32	71	42	0	205
5.0	29	26	30	50	38	0	173
5.5	28	24	29	40	36	0	157
6.0	27	22	28	34	33	290	434
6.5	27	21	27	30	32	71	208
7.0	26	19	27	26	30	50	178
7.5	26	18	26	24	29	40	163
8.0	25	17	26	22	28	34	152
8.5	25	16	25	21	27	30	144
9.0	24	15	25	19	27	26	136
9.5	24	15	24	18	26	24	131
10.0	23	14	24	17	26	22	126
11	23	12	23	15	25	19	117
12	22	11	23	14	24	17	111
14	21	9	21	11	23	14	99
16	20	8	20	9	22	11	90
18	19	7	19	8	21	9	83
20	19	6	19	7	20	8	79
22	18	5	18	6	19	7	73
24	17	5	17	5	19	6	69
26	17	4	17	5	18	5	66
28	16	4	16	4	17	5	62
30	16	3	16	4	17	4	60
32	16	3	16	3	16	4	58
34	16	0	16	3	16	3	54
36	16	0	16	0	16	3	51
48	14	0	15	0	15	0	44
60	14	0	14	0	14	0	42
72	13	0	13	0	13	0	39
84	13	0	13	0	13	0	39
96	12	0	12	0	12	0	36
108	12	0	12	0	12	0	36
120	11	0	11	0	11	0	33
132	11	0	11	0	11	0	33
144	10	0	10	0	10	0	30
156	10	0	10	0	10	0	30
168	10	0	10	0	10	0	30
180	10	0	10	0	10	0	30
192	10	0	10	0	10	0	30
204	10	0	10	0	10	0	30
216	10	0	10	0	10	0	30

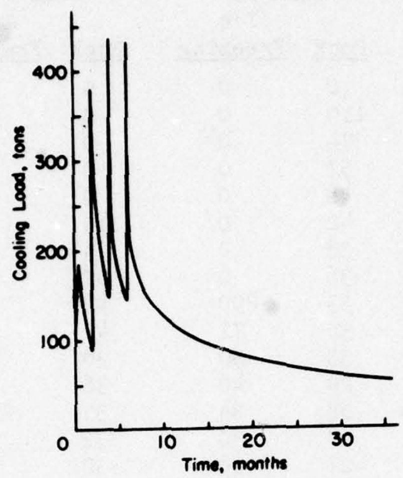

Figure 7-15. Total rock and ice freezing cooling loads during the preparation period for sample problem 8.

Table 7-23. Cost estimate for sample problem 8.

		Unit Cost	Quantity	Inflation Factor	Item Cost	Subtotals
A	**CONSTRUCTION**					
1	Excavation	$75/yd^3	6.48×10^4 yd^3	-	$4,860,000	
2	Reinforced concrete	$120/yd^3	1.33×10^4 yd^3	-	1,596,000	
3	Waterproofing	$.31/ft^2	1.19×10^5 ft^2	-	37,000	
						$ 6,493,000
B	**PREPARATION**					
1	Water	$.04/100 gal	8.85×10^6 gal	-	$ 4,000	
2	Water chiller	$1850/ton	70 ton	-	130,000	
3	Refrigeration units	$1850/ton	250 ton	-	463,000	
4	Salary	$\$1.2 \times 10^5$/yr	3 years	1.051	378,000	
5	Supplies, water chiller	$225/ton year	[0.5 year, 70 ton		8,000	
6	Supplies, refrigeration	$225/ton-yr	[3 years, 250 ton	1.085	183,000	
7	Electricity, chiller	$.035/kw-hr	8.8×10^5 kw-hr	-	30,000	
8	Electricity, refrigeration units (1st yr)	$.035/kw-hr	[1 year, 3.69×10^6 $\frac{\text{kw-hr}}{\text{yr}}$	-	129,000	
9	Electricity, refrigeration units (2nd & 3rd yrs)	$.035/kw-hr	[2 years, 1.69×10^6 $\frac{\text{kw-hr}}{\text{yr}}$	1.085	128,000	
						$ 1,453,000
C	**MAINTENANCE**					
1	Electricity	$.035/kw-hr	[15 years, 7.8×10^5 $\frac{\text{kw-hr}}{\text{yr}}$	2.142	$ 876,000	
2	Salary	$\$1.2 \times 10^5$/yr	15 years	1.563	2,813,000	
3	Supplies	$225/ton-yr	[50 ton, 15 years	2.142	361,000	
4	Overhauls	$250/ton-overhaul	[50 ton, 5 overhauls	1.853	116,000	
						$ 4,166,000
					Total Cost	$12,112,000

Table 7-24. Summary of the Sample Problems.

Sample Problem Number	Type of Heat Sink	Number of Reservoirs	Total Cost
1	Once through cooling using 90°F water	57	not feasible
2	Once through cooling using 40°F water	45	not feasible
3	Recirculated 90°F water with no heat transfer to surroundings	12	$25,561,000
4	Recirculated 40°F water with no heat transfer to surroundings	7	$19,042,000
5	Recirculated 90°F water with heat transfer to surroundings	10	$20,883,000
6	Recirculated 40°F water with heat transfer to surroundings	6	$16,245,000
7	Ice block and water	4	$13,432,000
8	Solid ice	3	$12,112,000

Figure 7-16, which provides a breakdown of the costs for each of the feasible heat sink concepts, illustrates the area in which the savings are realized. Note that while the preparation and maintenance costs for various low temperature heat sink systems are roughly equivalent, the construction cost varies significantly.

It should be remembered that the relative cost of each of the various heat sink systems is extremely dependent upon the assumed cost factors. Figure 7-17 has been prepared to illustrate the influence of the cost of excavation on the various sample problem heat sink systems. Note that if the cost of excavation were to be less than $25 per cubic yard the ice heat sink system might not be the most cost effective.

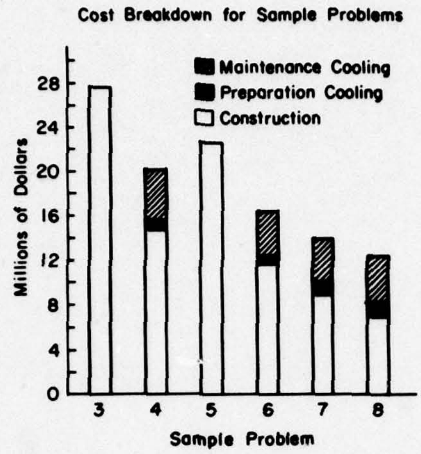

Figure 7-16. Comparison of the cost for each of the feasible sample problem heat sink systems.

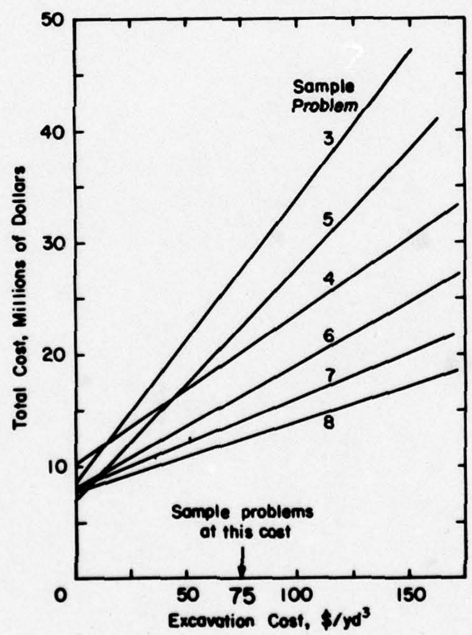

Figure 7-17. Influence of the excavation cost on the cost of the various feasible sample problem heat sink systems.

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Appendix A. Power Plant and Steam Condenser Performance Relationships

A-01. Introduction. This appendix provides the performance relationships and operating conditions for the power plant and steam condenser system used in the sample problems. These relationships and operating conditions are based on information provided in references 16 and 24.

A-02. Power System Description. The power system used in the sample problems is a pressurized water nuclear reactor (PWR) coupled to a steam turbine driven generator. The nuclear reactor, which utilizes two feedwater heaters, is a 13.1 MW(t) unit. The generator has a net output of 2112 gross kW(e). Approximately 500 kW(e) of this output is required to support the power plant and an additional 112 kW(e) is required for pumping of the plant coolant to the heat sink system. The remaining 1500 kW(e) of the gross output is the net export to the underground installation.

A-03. Power System Operating Conditions. It is assumed that the flow of coolant water through the condenser will be throttled so that the condenser hotwell temperature is 20°F greater than the temperature of the coolant water entering the steam condenser. It is also assumed that the power system has been designed so that at a condenser hotwell temperature of 180°F the turbine back pressure will be such that power generation must be terminated.

From these two assumptions it can be concluded that the heat sink temperature must not exceed a temperature of 160°F during the mission period. Thus for this power system the heat sink will have attained "thermal saturation" when the heat sink temperature is 160°F. Since the sample problems include the consideration of ice type heat sinks then the overall ranges of possible heat sink and condenser hotwell temperatures are given by the expressions:

$$34°F \leq T_{sink} \leq 160°F \tag{A-1}$$

$$54°F \leq T_{cond} \leq 180°F \tag{A-2}$$

Using data provided in reference 16 the waste heat produced by the power plant and the overall power plant efficiency as a function of temperature are tabulated in Table A-1. Figures A-1 and A-2 display this information in a graphical format.

Table A-1. Waste heat and plant efficiency of the sample problem power plant.

Condenser Hotwell Temperature °F	Heat Sink Temperature* °F	Power Plant Waste Heat Btu/hr	Power Plant Efficiency %
54	34	22.81×10^6	22.5
80	60	24.72×10^6	21.3
100	80	26.68×10^6	20.1
120	100	28.49×10^6	19.2
140	120	30.71×10^6	18.1
160	140	32.75×10^6	17.2
180	160	35.54×10^6	16.1

* Assuming hotwell is maintained at a temperature 20°F greater than the heat sink.

A-04. Power Plant Numerical Approximations. Since a number of the prediction techniques presented in section 6 require a mathematical relationship for the waste heat load produced by the power plant as a function of heat sink temperature a numerical approximation for the data presented in Table A-1 is required. This section provides two approximation techniques which can be utilized to fulfill this requirement.

A-04.1. Least Squares Straight Line Fit. If a relationship of the format:

$$y = mx + b \qquad \text{(A-3)}$$

is utilized where the variable y represents the waste heat load produced by the power plant and x represents the heat sink temperature, then using the data presented in Table A-1 the coefficients m and b may be calculated, for a least squares straight line fit, from:

$$m = \frac{N\sum xy - \sum x \sum y}{N\sum x^2 - (\sum x)^2} \qquad \text{(A-4)}$$

$$b = \frac{\sum y \sum x^2 - \sum x \sum xy}{N\sum x^2 - (\sum x)^2} \qquad \text{(A-5)}$$

where N = number of data points
x = independent variable
y = dependent variable
$\sum$ = summation of

For the data provided in Table A-1, using equations A-3 and A-4 yields:

$$m = 1.005 \times 10^5 \ \frac{\text{Btu}}{\text{hr}^{\circ}\text{F}}$$

$$b = 18.85 \times 10^6 \ \frac{\text{Btu}}{\text{hr}}$$

Thus the least squares straight line approximation for the power plant waste heat load as a function of heat sink temperature is:

$$q_r(T_{sink}) = 1.005 \times 10^5 \ T_{sink} + 18.85 \times 10^6 \ \frac{\text{Btu}}{\text{hr}} \qquad \text{(A-6)}$$

A-04.2. Least Squares Exponential Fit. Since, as Figure A-1 indicates, the relationship between the waste heat load and the heat sink temperature is curvilinear, a curve fitted to an exponential format may be more suitable. If a format such as:

$$q_r(T_{sink}) = A_1 \ e^{(A_2 T_{sink})} \qquad \text{(A-7)}$$

is utilized then the constants A_1 and A_2 may be determined by the following procedure. Taking the natural logarithm of equation A-7 yields:

$$\ln q_r = A_2 \ T_{sink} + \ln A_1 \qquad \text{(A-8)}$$

and using y to represent the term in q_r and b to represent the term $\ln A_1$ it can be seen that this equation has the same format as the linear relationship of equation A-3. Thus after converting the data points for q_r into the form $\ln q_r$, equations A-4 and A-5 can be used to calculate the values of m and b for a least squares fit where:

$$m = A_2 \qquad \text{(A-9a)}$$

and

$$b = \ln A_1 \qquad \text{(A-9b)}$$

Using the above cited relationship and some algebraic manipulation it can be shown that the least squares exponential fit to the data presented in Table A-1 is given by:

$$q_r(T_{sink}) = 20.125x10^6 \, e^{(3.512x10^{-3} \, T_{sink})} \text{ Btu/hr} \qquad \text{(A-10)}$$

A-04.3. Accuracy of the Approximations. Table A-2 provides a comparison between the actual power plant waste heat load and the waste heat loads predicted by the two numerical approximations developed in the preceding sections. While both approximations can be seen to be very accurate the exponential format provides the better overall accuracy. This can be attributed to the fact that the actual waste heat load tends to be related to the heat sink temperature in an exponential manner.

Table A-2. Comparison of the actual to the approximated power plant waste heat loads.

Heat Sink Temperature °F	Actual Power Plant Waste Heat Load MBtu/hr	Straight Line Approximated Heat Load MBtu/hr	Exponential Approximated Heat Load MBtu/hr
34	22.81	22.27 (-2,4%)	22.68 (-0.6%)
60	24.72	24.88 (+0.6%)	24.85 (+0.5%)
80	26.68	26.89 (+0.8%)	26.65 (-0.1%)
100	28.49	28.90 (+1.4%)	28.59 (+0.4%)
120	30.71	30.91 (+0.7%)	30.67 (-0.1%)
140	32.75	32.92 (+0.5%)	32.91 (+0.5%)
160	35.54	34.93 (-1.7%)	35.30 (-0.7%)

Note: figures in parentheses represent the percent deviation between the approximated and actual waste heat loads.

A-05. Steam Condenser Performance Relationships. To determine the coolant water flow rates and condenser coolant exiting temperature, performance relationships for the condenser system are required. From reference 24 the condenser performance relationships which will be used for these sample problems are:

$$q_r = 501 \, Wc_w \, (T_h - T_{in})(1 - e^{(-68.7/\sqrt{W})}) \qquad \text{(A-11)}$$

$$T_{out} = T_h - (T_h - T_{in}) \, e^{(-68.7/\sqrt{W})} \qquad \text{(A-12)}$$

where T_h = hotwell temperature, ^oF

T_{in} = condenser coolant entering temperature, ^oF

T_{out} = condenser coolant leaving temperature, ^oF

W = coolant flow rate, gpm

c_w = specific heat of water $(= 1.0 \frac{Btu}{lb^oF})$

Since the condenser entering temperature is identical to the heat sink temperature and it has been assumed that the coolant water flow rate will be throttled to maintain the hotwell at a temperature 20°F higher than that of the heat sink, then using the relationship:

$$T_h = T_{sink} + 20^oF \tag{A-13}$$

equations A-11 and A-12 can be simplified to:

$$q_r = 10020W\left(1 - e^{(-68.7/\sqrt{W})}\right) \tag{A-14}$$

$$T_{out} = T_{sink} + 20\left(1 - e^{(-68.7/\sqrt{W})}\right) \tag{A-15}$$

Using data from Table A-1 for q_r as a function of heat sink temperature and an iterative solution technique equation A-14 can be used to determine coolant water flow rates as a function of heat sink temperature. From this equation A-15 can then be used to determine the condenser coolant leaving temperature. Table A-3 summarizes the results of these calculations for the sample problem power plant. Figure A-3 presents these data in a graphical format.

Table A-3. Performance data for the sample problem steam condenser.

Heat Sink Temparature °F	Condenser Hotwell Temperature °F	Power Plant Waste Heat Load MBtu/hr	Condenser Coolant Flow Rate gpm	Condenser Coolant Leaving Temperature °F	ΔT Across Condenser °F
34	54	22.8	3250	48.0	14.0
40	60	23.3	3345	53.9	13.9
50	70	24.1	3505	63.7	13.7
60	80	24.7	3625	73.6	13.6
70	90	25.8	3845	83.4	13.4
80	100	26.7	4025	93.2	13.2
84	104	27.2	4135	97.1	13.1
90	110	27.7	4240	103.0	13.0
100	120	28.5	4410	112.9	12.9
110	130	29.7	4675	122.7	12.7
120	140	30.7	4905	132.5	12.5
130	150	31.8	5155	142.3	12.3
140	160	32.8	5375	152.2	12.2
150	170	34.2	5720	161.9	11.9
160	180	35.5	6045	171.7	11.7

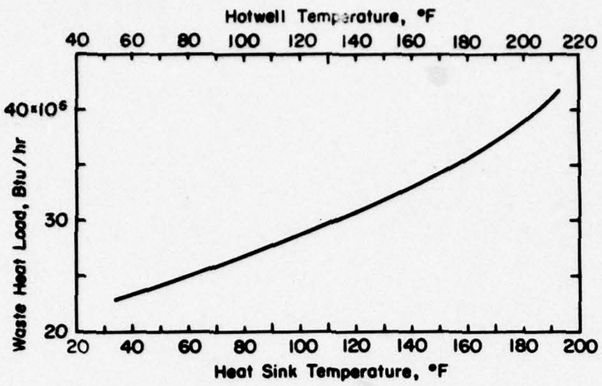

Figure A-1. Power plant waste heat load as a function of the hotwell or the heat sink temperature.

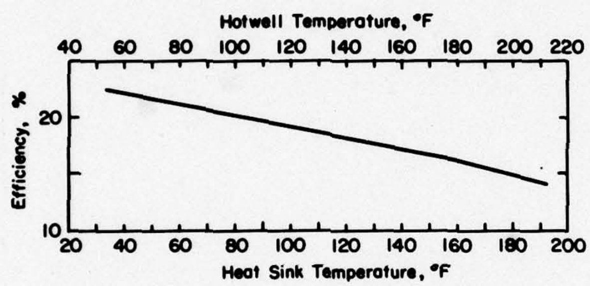

Figure A-2. Power plant efficiency as a function of the hotwell or the heat sink temperature.

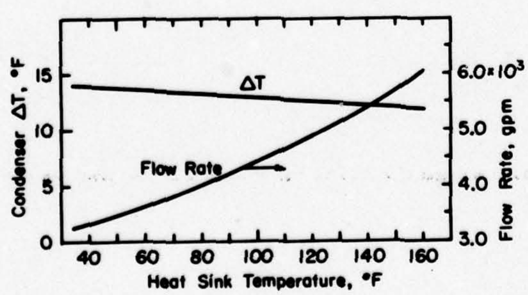

Figure A-3. Condenser temperature differential and coolant water flow rate as a function of the heat sink temperature.

Appendix B. Sources for Cost Data

B-01. Introduction. This appendix provides a listing of the sources used to develop the cost data employed in the sample problems. However, no certification as to its accuracy or applicability is expressed or implied.

Also contained in this appendix are a series of the inflation factor tables which have been used to generate the inflation factors for the sample problems. In preparing these tables it has been assumed that the rate of inflation will remain constant over the stated period and is compounded annually. Again, no certification as to the accuracy or applicability of these inflation data is expressed or implied.

B-02. Cost Data Sources. Table B-1 provides a serialized listing of the sources used to develop the unit cost for the sample problems.

Table B-1. Cost Data Sources

	Item	Source
a.	Excavation costs	Telephone conversations with the Omaha District, CoE and the USA Waterways Experiment Station
b.	Concrete costs, Water-proofing costs	Building Construction Cost Data, 1976 edition, published by Robert Snow Means Company, Inc. Duxbury, Mass.
c.	Water	Current water cost for CRREL
d.	Refrigeration units, (central plant)	Comparative Analysis of Packaged and Central Station Refrigeration by Lucien St. Onge, ASHRAE Journal, Oct 1970
e.	Ice machines	Telephone conversation with several manufacturers
f.	Electricity	Current industrial rate for CRREL
g.	Salary and supplies for refrigeration systems	Current costs for CRREL central plant system
h.	Inflation rates	EIRS Bulletin 76-02, Proposed draft revision to AR 415-17 dated 16 Jan 76

B-03. Inflation Factors. Tables B-2 and B-3 provide the inflation factors for the assumed inflation rates of 5% and 8.3%, respectively. The definition of each of the factors listed on these two tables as well as the formulas employed to calculate them are discussed in the following paragraphs.

B-03.1. Compounded Inflation Factor. The compounded inflation factor provides the numerical factor used to convert costs in the base year to costs in a later project year for the assumed inflation rate. This factor is determined through the use of the standard compounding relationship:

$$\text{future cost} = (1 + r)^n \cdot \text{present cost}$$

where

r = decimal inflation rate
n = number of compounding periods.

Using a numerical example to illustrate this, from Table B-2 the compounded inflation factor for the fifth year is 1.216. Thus if an item were to cost $100 in the base year, then assuming a 5% inflation rate the cost in the fifth year would be:

$$\text{5th year cost} = (1.216) \cdot (\$100) = \$121.60$$

B-03.2. Cumulative Inflation Factor. The cumulative inflation factor represents the sum of the compounded inflation factors from the base year up to and including the year of interest. In mathematical terms this may be expressed as:

$$\text{cumulative inflation factor} = \sum_{n=1}^{n=k} (1+r)^n$$

where r and n are defined in the same manner as above and k represents the number of years of interest. Using a numerical example, again from Table B-2 it can be easily shown that the cumulative inflation factor for the fourth year (i.e. 4.311) is equal to the sum of the compounded inflation factors for the first, second, third and fourth years.

The cumulative inflation factor is normally employed when it is required to determine the total cost of a fixed* annual expenditure under a given inflation rate. Again using data from Table B-2, if an expenditure of $100 must be made in each of the first four years of a project then the total expenditure would be:

$$\text{cumulative expenditure} = (4.311) \cdot (\$100) = \$431.10$$

B-03.3. Average Annual Inflation Factor. The average annual inflation factor represents the average value of the cumulative inflation factor over a fixed period of years. Mathematically, this may be expressed as:

$$\text{average annual inflation factor} = \frac{1}{k} \sum_{n=1}^{n=k} (1+r)^n$$

*In this context, a fixed expenditure refers to an expenditure fixed only in regard to a "constant" dollar spending power.

This factor would be utilized as follows. Suppose an annual expenditure of $100 must be made for the first six years. The average annual expenditure, assuming a 5% inflation rate and using Table B-2, would be:

average annual expenditure = (1.134) · ($100) = $113.40/year

To convert this to a total expenditure for the entire period it is only necessary to multiply this factor by the total number of years. That is:

total expenditure = ($113.40/year) · (6 years) = $680.40

Except for a small error due to rounding off, this result is identical to the result that would have been determined if the cumulative inflation factor for the sixth year (6.803) and the annual cost ($100) had been employed.

The use of the average annual inflation factor can simplify calculations when expenditures are made at a frequency less than annually. For example, if an expenditure of $100 had to be made only during the first, third and fifth project years then the average annual inflation factor, assuming a 5% inflation rate, would be:

$$\text{average annual inflation factor} = \frac{1.000 + 1.103 + 1.216}{3} = 1.106$$

Therefore, the average annual cost for each year that the expenditure was made would be:

average annual cost = (1.106) · ($100) = $110.60/year

And finally the total cost would be:

total expenditure = ($110.60/year) · (3 years) = $331.90

Table B-2. Inflation factor table for a 5% annual inflation rate.

Year	Compounded inflation factor	Cumulative inflation factor	Average annual inflation factor
1 (base year)	1.000	1.000	1.000
2	1.050	2.050	1.025
3	1.103	3.153	1.051
4	1.158	4.311	1.078
5	1.216	5.527	1.105
6	1.276	6.803	1.134
7	1.340	8.143	1.163
8	1.407	9.550	1.194
9	1.477	11.027	1.225
10	1.551	12.578	1.258
11	1.629	14.207	1.292
12	1.710	15.917	1.326
13	1.796	17.713	1.363
14	1.886	19.599	1.400
15	1.980	21.579	1.439
16	2.079	23.658	1.479
17	2.183	25.841	1.520
18	2.292	28.133	1.563
19	2.407	30.540	1.607
20	2.527	33.067	1.653

Table B-3. Inflation factor table for an 8.3% annual inflation rate.

Year	Compounded inflation factor	Cumulative inflation factor	Average annual inflation factor
1 (base year)	1.000	1.000	1.000
2	1.083	2.083	1.042
3	1.173	3.256	1.085
4	1.270	4.526	1.132
5	1.376	5.902	1.180
6	1.490	7.392	1.232
7	1.614	9.006	1.287
8	1.747	10.753	1.344
9	1.892	12.645	1.405
10	2.050	14.695	1.470
11	2.220	16.915	1.538
12	2.404	19.319	1.610
13	2.603	21.922	1.686
14	2.819	24.741	1.767
15	3.054	27.795	1.853
16	3.307	31.102	1.944
17	3.581	34.683	2.040
18	3.879	38.562	2.142
19	4.201	42.763	2.251
20	4.549	47.312	2.366

Appendix C. Volumetric Relationships for the Heat Sink Reservoirs

C-01. Introduction. This appendix provides the volumetric relationships for the heat sink reservoirs based on the assumptions employed in the sample problems.

C-02. Reservoir Geometry. Figure C-1 presents the overall geometry of the heat sink reservoirs employed in the sample problems. The reservoirs have been assumed to be vertical cylinders with hemispherical ends. It has also been assumed that, based on a structural analysis, the walls will be reinforced concrete with a thickness of 2 feet. It is also assumed that for structural reasons the ratio of the length of the reservoir, L, to the internal radius, R, should be 8 to 1.

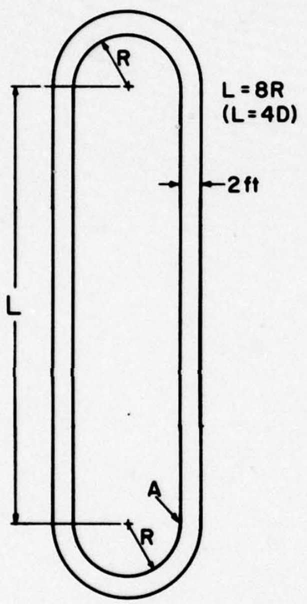

Figure C-1. Basic geometry of the heat sink reservoirs.

C-03. Water and Ice Reservoirs. For the sample problems it will be assumed that the heat sink reservoir will be completely filled if it is to be a water type reservoir. However, if the reservoir is to contain a mixture of ice and water or just ice then it will be assumed that the upper dome will not be filled. This volume will be reserved to hold refrigeration or other equipment which will be needed to chill or maintain the heat sink.

C-04. Volumetric Relationships. Based on the information provided in the preceding sections the following relationships can be easily developed:

1. **Reservoir volume, V_R:**

$$V_R = \frac{28}{3} \pi R^3 \quad \textbf{(completely filled reservoir)}$$

$$V_R = \frac{26}{3} \pi R^3 \quad \textbf{(upper dome not filled)}$$

2. **Excavated volume, V_E:**

$$V_E = \frac{28}{3} \pi (R+2)^3$$

3. **Volume of concrete, V_c**

$$V_c = \frac{28}{3} \pi [(R+2)^3 - R^3]$$

4. *Internal Surface Area,* A_s

$$A_s = 20 \pi R^2$$

5. **Overall Internal Length,** ℓ

$$\ell = 10R$$

Appendix D. Excavation Costs

D-01. Introduction. In the sample problems a flat rate cost for rock excavation of $75 per cubic yard has been employed. This was done to simplify the analysis of the construction cost of the heat sink reservoirs. However, it should be recognized that the excavation cost will be influenced by a number of factors. In the following sections the results of a study on excavation costs has been presented.

D-02. Excavation and Shaft Sinking Cost, prepared by Walter C. Best and Mounir M. Botros.

Table of Contents

1. Toughness of Rocks

KIND OF ROCK	TOUGHNESS Lime Stone = 1
Fresh diabase	3.0
Pyroxen quartzite	2.7
Sandstone	2.6
Altered diabase	2.4
Fresh basalt	2.3
Hornblende schist	2.1
Diorite	2.1
Hornblende granite	2.1
Rhyolite	2.0
Quartzite	1.9
Biotite gneiss	1.9
Altered basalt	1.7
Feldspathic sandstone	1.7
Gabbro	1.6
Chert	1.5
Calcereous Sandstone	1.5
Granite	1.5
Slate	1.2
Granite gneiss	1.2
Andesite	1.1
Lime Stone	1.0
Mica Schist	1.0
Dolomite	1.0
Biotite granite	1.0
Hornblende gneiss	1.0

2. Weight of Rocks

Material	Wt/ft^3, lb		Ft^3/ton		$Tons/yd^3$	
	In place	Broken	In Place	Broken	In place	Broken
Dolomite	160		12.5		2.16	1.30
Gneiss	168	96	11.9	20.8	2.27	1.30
Granite	170	97	11.8	20.6	2.30	1.31
Lime Stone Areas	154	--	13.0	----	2.08	----
Quartz	165	94	12.1	21.3	2.23	1.27
Sand Stone	151	86	13.2	23.3	2.08	1.16

3. Drilling

Drill used in shaft sinking are: hand churn, single or double hammer, and piston or hammer machine drill, hand churn drill in the hands of energetic workmen may be advantageous in soft rock. Hand hammer drilling is best applicable with low priced unskilled labor.

Machine drill repairs

The practice of overhauling machine drill at the shop after each round has resulted in very low repair costs. At Pyne shaft Birmingham, Alabama, (1918-1918) $1.14 per ft of shaft.

3A. Cost of Machine Drilling

Comparisons (a) wages of drill crew;

(b) Proportion of wages of power plant crew
(c) Fuel
(d) Drill sharpening
(e) Repair and renewals
(f) Oil and water
(g) Interest on plant
(h) Depreciation of plant
(i) Proportion of general expense including taxes
(j) Erecting, dismantling, and moving plant

3B. Factor Affecting Speed of Drilling

(a) Character of rock as hardness, stickiness, seams, sludge, and dust forming qualities
(b) Time for changing bits
(c) Time for taking down, moving, and setting up machine
(d) Depth of hole
(e) Direction of hole
(f) Diameter of hole
(g) Use of air or water or both, in the hole
(h) Shape of bit
(i) Quality of blacksmithing
(j) Percentage of time lost for blasting, breakdowns, delays
(k) Size, weight, and type of drill and mounting
(l) Air or steam press at the drill
(m) Skill of crew

3C. Average Time Drilling Vertical Holes
(Tripod Mounted Drill)

KIND OF ROCK	LM	S	Sd	Gr	Tr
Length of shift hr.	10	10	10	10	10
Air pressure, lb per sq in	70	70	70	80	70
Diameter drill cylinder, in	3.25	3.25	3.25	3.25	3.25
Diameter starting bit, in	2.5	2.5	2	3.5	2.5
Diameter finishing bit, in	1.75	1.5	1.25	1.25	2
Depth of hole, ft	12	6	12	20	6
Drilling first 2 ft, min	9	10.5	8	12	14
Cranking out, removing bit min.	1	3.5	1.25	1	1.5
Cleaning out hole, min	1	3	1.25	1	1.5
Putting in new bit cranking, min	1	2.5	1	1.5	1.5
Drilling second 2 ft, min	13	10.5	1	1.5	14
Drilling last 2 ft, min	12	10.5	6	11	14
Moving machine, setting up, min	15	35	12	11	36
Ft drilled per shift	--	--	96	48	36

Note: Lm = Limestone, S = Sandstone (hard) Sd = Sandstone (soft),
Tr = Trap (diabase) Gr = Granite

3D. **Following are comparative costs of 6 and 9 inch drill at Tilden Pit, Cleveland Cliffs Iron Co., using Bucyrus Armstrong 29T, 9 in. bit (E+M Jor. Nov. 1937)**

	6 in bit	9 in bit
No of holes	14	14
Total footage drilled	1416	1377
Avg depth of hole	105	100
Spacing of holes	15 x 22	20 x 30
Burden per ft of hole cu yd	12.2	22.2
Total tons blasted	35,400	62,000
Total explosive, lb	9,300	20,040
Tons per lb explosive	3.8	3.04
Drilling cost per ton	$0.074	$0.0334
Operating cost/ft	$1.95	$1.69
Drilling rate, ft in 8 hr	13.7	15.6

NY Trap Rock Corp., on Hudson River, average of 4 years operation in 4 quarries based on 5000 to 35000 ft drilled per year.

KIND OF ROCK	AVERAGE DEPTH OF HOLE, FT	DIAMETER OF HOLE IN	SPACING FT.	FT/HR ELAPSED TIME	COST PER FT
Dolomite, hard	70	8	17x24	2.03	$1.13
Limestone, soft	35	6	14x18	2.75	0.64
Limestone, medium	110	10	15x27	3.11	1.06
Basalt, hard	120	8	22x30	1.4	2.12

4. Blasting: Spacing of holes, charges and results of churn drill, Blasts original.

KIND OF ROCK	CHARACTER OF WORK	NO. OF HOLES	HOLE (in)	AVG DEPTH (ft)	AVG. DIS. APART FROM FACE (ft)	AVE. DIS. APART OF HOLES (ft)	TOTAL EXPLOSIVE USED (lb)	GRADE OF EXPLOSIVE (%)	KIND OF EXPLOSIVE	ROCK BLASTED (yd^3)	EXPLOSIVES (lb)
E	Crushed stone	4	5-5/8	66	--	--	5,500	--	A	20,000	0.275
E	Crushed stone	8	5	50	12	12	1,200	40	A	--	--
E	Cement quarry	12	6	204	32	16.5	7,700	--	B	47,000	0.483
							15,000	--	A		
E	Cement quarry	8	6	65	20	20	4,000	40	A	--	--
E	[Open pit iron mine]	--	6	20	15	15	65(b)	40	A	--	--
E	R R Basalt	8	6	48	19	18	3,300	40	A	5,720	0.578
E	Cement Quarry	9	6	62	32	20	[1,800	40]	A	12,370	0.349
							2,500]	60]			
								60]			
E	Cement quarry	12	8	140	30	30	[1,700	--	B]	54,000	0.55
							1,600	--	A]		
E	Hard RR basalt]	8	6	95	33	28	[2,200	--	B]		
							3,350	60	A]	2,700	0.249
							1,250	40	A]		
E	Lime quarry	3	6	100	24	17	[1,200	60]	A	5,720	0.315
							600	40]			
E	Cement quarry	9	6	52.5	36	20	[1,700	60]	A	13,660	0.309
							2,500	40]			
F	Hard granite	16	6	115	16	32	18,597	--	C	34,000	0.5
G	RR thru cut	578	4	25-40	f	f	[1,200	60]	B	35,000	0.814
							27,295	35]			

A = Dynamite B = Gelatine C = Nitramon
E = Limestone F = Sandstone G = Basalt
(b) = per hole

5. Charging and firing explosives. Priming is the placing of a detonator, electric blasting cap or blasting cap attached to fuse in a dynamite cartridge or placing an electric squib in a cartridge of blasting powder or pellet powder.

Wiring for electric blasting

There are three general methods, series, parallel, and parallel series connection.

Electric firing may be done with a blasting machine, or power or lighting circuit.

6. Shaft sinking in Rock.

Example: Ross shaft, Homestake Mining Co., So. Dak (28) 6 compt, 14 ft by 19 ft 3 in. outside steel sets, designed for 5200 ft depth; sets installed at end of 1934 for 3242 ft. Sunk from surface, 137 ft; raised full size 250 ft from 800 level, but wt. of broken rock crushed the timbering; shaft was raised elsewhere with 6 by 6 ft pilot raises in center of shaft area, then enlarged to size. Steel sets (6 ft centers); plates and dividers, 6 in 25 lb H - beams; posts, 3.5 by 3 by 3/8 in angles; the 2 skipways laced with 14 gage galvanized corrugated steel; ladders and sollars of steel; all steel specified to contain 0.20-0.25% copper. Upper 308 ft of shaft concreted solidly outside of sets, at cost of $46.27 per ft., total of 150 ft concreted below in sections of 1 to 3 sets, elsewhere the shaft walls were gunited. Costs per ft of 3241.5 ft of steel supported shaft (1433-34).

EXCAVATION	PILOT RAISES	ENLARGING	TOTAL
Mining Labor	$9.19	$13.77	$22.96
Explosives	2.73	3.11	5.84
Air and drills	3.20	2.54	5.74
Timbering	4.24	0.67	4.91
Trackage	0.34	0.56	0.90
Pipe	0.49	0.25	0.74
Hoist and hoisting	1.82	1.26	3.08
Haulage	0.85	2.87	3.72
Elec Supplies	--	0.05	0.05
Miscellaneous	1.13	1.43	2.56
Surveying	0.12	--	0.12
TOTAL	$24.11	$26.51	$50.62

STEEL SUPPORT	SHAPES	CORRUG LACING	TOTAL
Invoice Cost	$21.13	$ 6.35	$27.48
Unloading, etc	2.08	---	2.08
Installing			
Labor	6.92	1.06	7.98
Hoist Labor	2.15	0.11	2.26
Power	0.33	---	0.33
Air and drills	0.12	---	0.12
Misc. Supplies	3.21	0.13	3.34
Clips for fastening	--	0.39	0.39
TOTAL	$35.94	$ 8.04	$43.98
Sollars, loaders, railings	$ 2.52		
Shaftdoor, guides, chairs	3.50		
Surveying	$ 0.32		
TOTAL PER FT.	$50.32		

TOTAL COST	
Excavation (above)	$50.62
Steel Supports (above)	50.32
Concreting and guniting	11.40
Stations	5.59
Piping and wiring	13.82
General Construction	0.66
TOTAL PER FT.	$132.31

(Based on 1934 dollars)

Cost Components

7-1. For convenience, COSTUN considers tunneling to consist of twelve operations. A cost component is the cost of one of these operations. For the purpose of the present study, the following components have been selected:

1. Excavation Setup
2. Excavation
3. Muck Loading
4. Muck Transport
5. Muck Hoisting
6. Muck Disposal
7. Supports
8. Lining Formwork
9. Lining
10. Grouting
11. Pumping
12. Air Conditioning

All cost components have been subdivided into labor, equipment and material subcomponents.

8-1. LABOR: The labor subcomponent is the cost of the labor force required for the construction operation. Unless otherwise noted, the labor force for any construction operation is considered to be on duty continuously throughout the progress of the work, whether or not that labor crew is actually working at all times. The size of the labor crew has been determined according to standard crew make-up in the tunneling industry.

8-2. EQUIPMENT: The equipment is like the labor crew, usually on duty continuously, whether or not the equipment is actually in use at all times. Therefore, the equipment subcomponents cost of depreciation of the purchase cost is largely dependent upon the total time the equipment is on the project. Depreciation is calculated on a straight line basis over the estimated useful life of the particular equipment selected for the work, less any salvage value that it might have. Other equipment costs are spare and replacement parts.

8-3. MATERIAL: The material subcomponent consists of all materials installed in the work, such as cement, and of all expendable materials used to service and operate the equipment, such as lubricants and fuels.

FACTORS AFFECTING COSTS

9. The project dependent factors that affect the cost of a tunnel system fall into three categories:

- those relating to site characteristics;
- those relating to design requirements; and
- those relating to construction methods.

FACTORS RELATING TO SITE CHARACTERISTICS

9-1. These factors include the characteristics of the medium through which the tunnel system is constructed, and hence, have a major influence upon the tunnel system selected.

9-1.A. RQD: RQD or Rock Quality Designation is a system for classifying the degree of coherence of rock. The RQD is a continuous variable, expressed as a percent, and may vary from 0 to 100. Thus, the RQD system eliminates the need for a system based on discontinuous rock classes. Underground headings in rock with RQD's less than 25 are considered to require soft ground tunneling techniques. Therefore, all segments having rock with an RQD less than 25 require the method of excavation to be either soft ground or cut-and-cover.

RQD is a modified core recovery classification that is based on counting only those pieces of sound rock in the core which are longer than four inches. In determining lengths of sound rock, breaks in the core caused by handling and drilling are ignored. Correlation between RQD and Terzaghi's rock classification system and loads on rock tunnels is shown on Table D-1.

9-1-B. ROCK STRENGTH: Rock strength is the unconfined compressive strength of a specimen of intact rock within the rock mass. It should be noted that no values of RQD and rock strength are mutually exclusive; a high RQD may occur with a low rock strength (as in an unjointed unbedded soft shale) or a low RQD may occur with a high rock strength (as in crushed quartzite).

9-1-C. GOVERNING SHEAR STRENGTH: The governing shear strength is the material strength that controls its behavior. For soils, this is described by certain combinations of friction and cohesion. For rocks, it may be the rock strength, or in the case of rock the low RQD, the strength of the material filling the joints. The joint material may cause the rock mass to behave like a soil.

All materials whose governing shear strength is approximately equal to or less than the imposed in-situ stress should be considered soft ground. Conversely, all materials whose governing shear strength is much greater than the imposed in-situ stress should be considered rock.

9-1-D. UNIT WEIGHT OF SOIL: Unit weight of soil is the saturated soil unit weight in pounds per cubic foot. This is a required user input except for rock tunnels. *If no value is input for soft ground* or cut-and-cover tunnels, use 120 pcf.

9-1-E. Phi: Phi (Ø) is the angle of internal friction that governs the behavior of cohesionless soils, or of rocks with RQD's less than 25, and is a required user input. If foundation data are not available, clean sands and gravels may be assumed to be characterized by Ø in the range of 29 to 45 degrees and cohesion equal to zero.

9-1-F. COHESION: Cohesion is the undrained shear strength of a cohesive soil in psf. If foundation data are not available, the cohesion of a normally consolidated clay can be estimated from Figure D-1 by using the appropriate soil plasticity index and soil unit weight. If the plasticity index is not known, it may be assumed at an average value of 35, which gives cohesion = .25ΣOH.

9-1-G. GROUNDWATER ELEVATION: Groundwater elevation refers to the average elevation of the groundwater for tunnel segments and to elevation at the shaft for shaft segments. The groundwater elevation is a required user input. If the groundwater table (GWT) is below the tunnel, the exact elevation specified is immaterial as long as it is specified below the tunnel. If the GWT is not input, COSTUN will assume it is at

RQD (%)	JOINT SPACING Cm.	JOINT SPACING In.	DESCRIPTION	TERZAGHI CLASSIFICATION DESCRIPTION		ROCK LOAD Initial	ROCK LOAD Final
		24					
	50		Excellent	Hard	Hard & intact	0	0
					Stratified or Schistose	0	0.25B
90		12		Massive, moderately jointed		0	0.5B
	20		Good				
80				Moderately blocky & seamy		0	0.25B to 0.35C
		6					
70			Fair	Very blocky, seamy, and shattered		0 to 0.6C	0.35C to 1.1C
60							
50	10	4					
			Poor				
40							
30							
20			Very Poor	Completely crushed, gravel and sand and squeezing and swelling rocks		0.54C to 1.2C	0.62C up to 250
10							
	5	2					
		1					

C = 2B for circular and horseshce C = 1.5B for baskethandle

Table D-1: Correlation between RQD and Terzaghi's rock classification system.

elevation 0. The groundwater elevation for shaft must coincide with the elevation of a shaft segment boundary nodal point, unless the groundwater elevation is above or completely below the shaft.

A GWT above the ground surface can be input. This would cover the condition in which a tunnel passes beneath a lake or where the ground water is under artesian pressure.

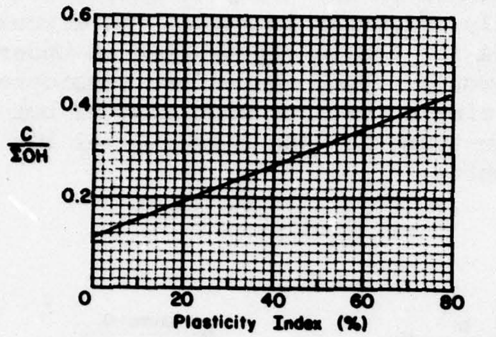

Figure D-1. Cohesion of normally consolidated clays.

Approximate cohesion is given by: $\frac{C}{\Sigma OH}$ = 0.11 + C.004 P.I. where
C = cohesion, psf

ΣOH = Summation of weight of soil layers using submerged unit soil weight below water table and total weight above.

P.I. = Plasticity index. If no data are available, use P.I. = 35.

FACTORS RELATING TO DESIGN REQUIREMENTS

9-2. Tunnel and Shaft Type.

Type refers to the classification of the tunnel or shaft according to the construction approach best suited to the ground conditions. The two construction approaches are underground heading and cut-and-cover heading. Each approach may be used to construct tunnels or shafts in rock or soft ground. The following table shows all the combinations of the above considered by CONTUN.

LOCATION	UNDERGROUND HEADING				CUT-AND-COVER HEADING			
Axis	Tunnel		Shaft		Tunnel			Shaft
Geology	Soft Ground	Rock	Soft Ground	Rock	Soft Ground	Rock	Mixed	All
Term used in text	Soft Ground Tunnel	Rock Tunnel	Soft Ground Shaft	Rock Shaft	Cut-and-cover Tunnel			Cut-and-Cover shaft

It is suggested that cut-and-cover be selected whenever the mid-height of the tunnel is less than two tunnel widths below the ground surface. A cut-and-cover tunnel may be constructed in a cross section consisting of all rock, all soil, or soil above rock.

9-2A. SHAPE: Shape is the geometry of the shaft or tunnel. For soft ground or rock tunnels, three shapes, as shown on Figure D-2, are considered. These shapes are circle, horseshoe, and basket handle. Only the circular shape is considered in the study of shafts constructed as underground headings entirely or partly in rock. The circular and square shapes are considered for shafts constructed as underground headings entirely in soft ground. Both shapes are considered for all cut-and-cover shafts. The single level or double level box shapes are considered for cut-and-cover tunnels. The single level box may consist of any number of multiple units.

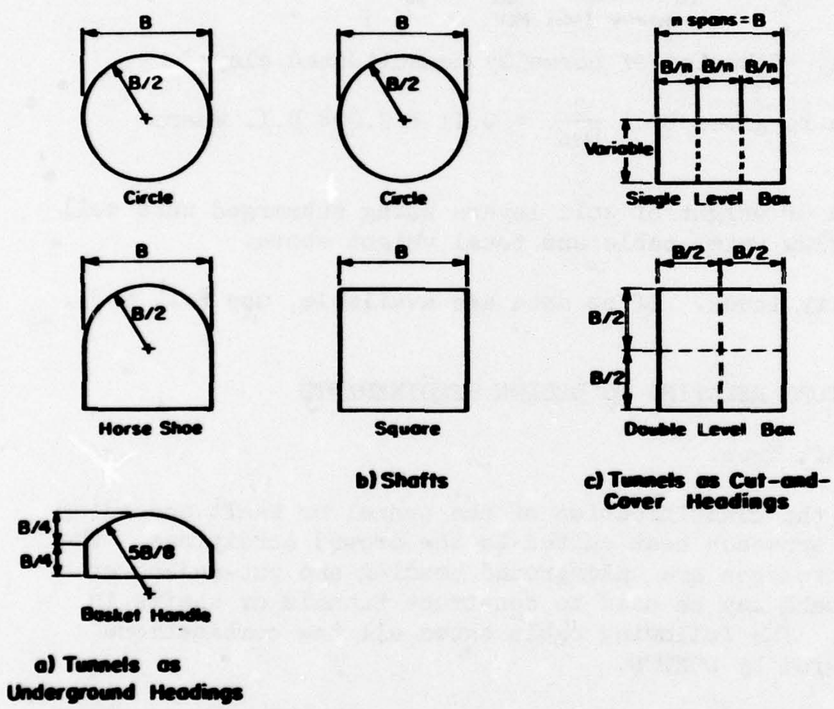

Figure D-2. Tunnel and shaft shapes.

9-2-B. SIZE: Size is the characteristic dimension, B, of the tunnel or shaft, as shown in Figure D-2. The characteristic finished dimension may vary between 10 feet and a maximum of 40 feet.

9-2-C. SLOPE: Slope is the slope of the centerline of the shaft or tunnel. All shafts are considered to be vertical. Tunnel slopes varying between plus 26% and minus 26% are considered.

9-3. FACTORS RELATING TO CONSTRUCTION METHODS. Six major categories of construction are treated as variables. These variables, three of which are required user input and three of which are optional user input are:

- Construction work week;
- Soft ground stabilization method;
- Excavation method;
- Muck transport method;
- Advance rate. (while the advance rate is technically not a method, it is a characteristic measure of construction methods not otherwise identified;)
- Lining, and in soft ground and cut-and-cover segments, supports.

Appendix E. Sample Cooling Tower Design

E-01. Introduction. This appendix presents a typical design procedure for one type of unlimited capacity heat sink system, a package cooling tower unit. The design charts and relationships presented in this section have been reproduced through the courtesy of the Baltimore Aircoil Company, Inc.

The design procedure will consider the selection of a cooling tower which would be used in conjunction with the power plant described in Appendix A. It will be assumed that the cooling tower will be located in an underground chamber with air intake and exhaust tunnels to the surface.

E-02. Design Conditions. To proceed with the design calculations climatic data for the site are required. The following weather data have been assumed for this sample problem:

Air Design Data:	10%:	80°F db,	74°F wb
	5%:	93°F db,	76°F wb
	2-1/2%:	96°F db,	77°F wb
	1%:	100°F db,	78°F wb

This type of data can be found in TM5-785 (AFM 88-8), Engineering Weather Data, for a number of locations throughout the world.

E-03. Modifications of the Design Data. Passage of the intake air through rock tunnels will result in changing both its wet bulb and dry bulb temperature. For this sample problem it will be assumed that these effects will be small and thus may be neglected. However, TM 5-855-4, Heating and Air Conditioning of Underground Installations, which provides techniques for determining these effects, should be consulted for a real problem.

Since the performance of package type cooling towers depends significantly on the wet bulb temperature, a wet bulb temperature of 80°F will be used in the design calculations.

E-04. Design Cooling Load. With an 80°F wet bulb temperature, it will be assumed that the coolant water returned to the condenser will be 5°F warmer or 85°F. Thus from the data provided in Appendix A the following can be determined:

$$q_r = 27.13 \times 10^6 \text{ Btu/hr}$$

$$G = 4120 \text{ gpm}$$

$$T_{in} = 85°F$$

$$T_{out} = 98.1°F$$

E-05. Design Calculations. Using the data provided above the following can be determined*

$$\text{Range} = 98.1^{\circ}F - 85^{\circ}F = 13.1^{\circ}F$$

$$\text{Approach} = 85^{\circ}F - 80^{\circ}F = 5^{\circ}F$$

$$\text{Tower Load} = \frac{4120 \text{ gpm} \times 500 \times 13.1}{15000} \approx 1800 \text{ tons}$$

Therefore, from Chart 1 of Figure E-1, using a 13.1°F range with a 5°F approach and at an 80°F wb the tower load correction factor is approximately 1.05. Therefore the corrected tower load is given by:

Corrected Tower Load = (1.05)(1800 tons) = 1890 tons.

A corrected tower load of 1900 tons will be used in all subsequent calculations.

E-06. Unit Selection. Using Chart 2 of Figure E-2 and a corrected tower cooling load of 1900 tons and a coolant water flow rate of 4120 gpm it can be seen that this lies within the recommended operating range of a VST 2190 B unit. And thus from the engineering data (Figure E-3), a coolant air flow rate of 426,000 cfm will be required.

If it was desired to employ three cooling units rather than one then the following procedure would be employed. The corrected tonnage and the coolant water flow rate would be divided equally among all three to yield:

corrected tonnage = 640 tons/unit
coolant water flow rate = 1375 gpm/unit

Using these data and Chart 1 of Figure E-2, it can be seen that three VST-730 B units will be required. From the engineering data of Figure E-3 it can be determined that each unit would require 142,000 cfm of cooling air.

The advantage of using multiple units is that the overall security of the cooling system can be increased. If only one cooling unit is employed then destruction of that unit would require that the power plant be shut down. However, using multiple units, if one unit is destroyed the power plant could still be operated but at a reduced power level. Thus the multiple units provide an inherent redundancy not attainable with a single unit system.

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* see reference 59 for details.

Entering Water Temp ________ °F   Leaving Water Temp ________ °F
Leaving Water Temp ________ °F   Design Wet Bulb Temp ________ °F
Range ________ °F   Approach ________ °F

Load = $\dfrac{\_\_\_\_\_\_\text{ gpm} \times 500 \text{ (Constant)} \times \_\_\_\_\_\_ {}^\circ\text{F Range}}{15{,}000 \text{ Btu/hr. ton}}$ = ________ tons

Corrected Load = ________ tons × ________ = ________ Corrected Tons

Selection Example:

GIVEN: To cool 2175 GPM of water from 103°F to 85°F with 76°F wet bulb.

1. Determine Range
   Range = Water on 103°F − Water off 85°F = 18°
2. Determine Approach
   Approach = Water off 85°F − Wet Bulb 76°F = 9°
3. Calculate Tower Ton Load
   Load = $\dfrac{2175 \text{ GPM} \times 500 \text{ (Constant)} \times 18^\circ \text{ Range}}{15{,}000 \text{ BTU/HR/TON}}$ = 1305 Tons
4. Determine Correction Factor
   Enter the Wet Bulb Correction Section of Chart 1 on the 18° Range line as shown by the dotted line. From the intersection of the 18° Range line and the 76° Wet Bulb curve, project a straight line into the Approach Section to intersect the 9° Approach curve. From this point, extend a line horizontally into the capacity Multiplier Factor Section, intersecting the 18° Range line to obtain the correction factor. The factor is 0.65.
5. Determine Corrected Load
   Multiply the tower ton load by the correction factor
   0.65 × 1305 = 848
6. Select Cooling Tower
   Referring to chart 2, using the GPM and corrected tonnage, Unit VLT-900B should be selected.

NOTE: Always select a unit above the intersection point of the GPM and Corrected Tonnage lines.

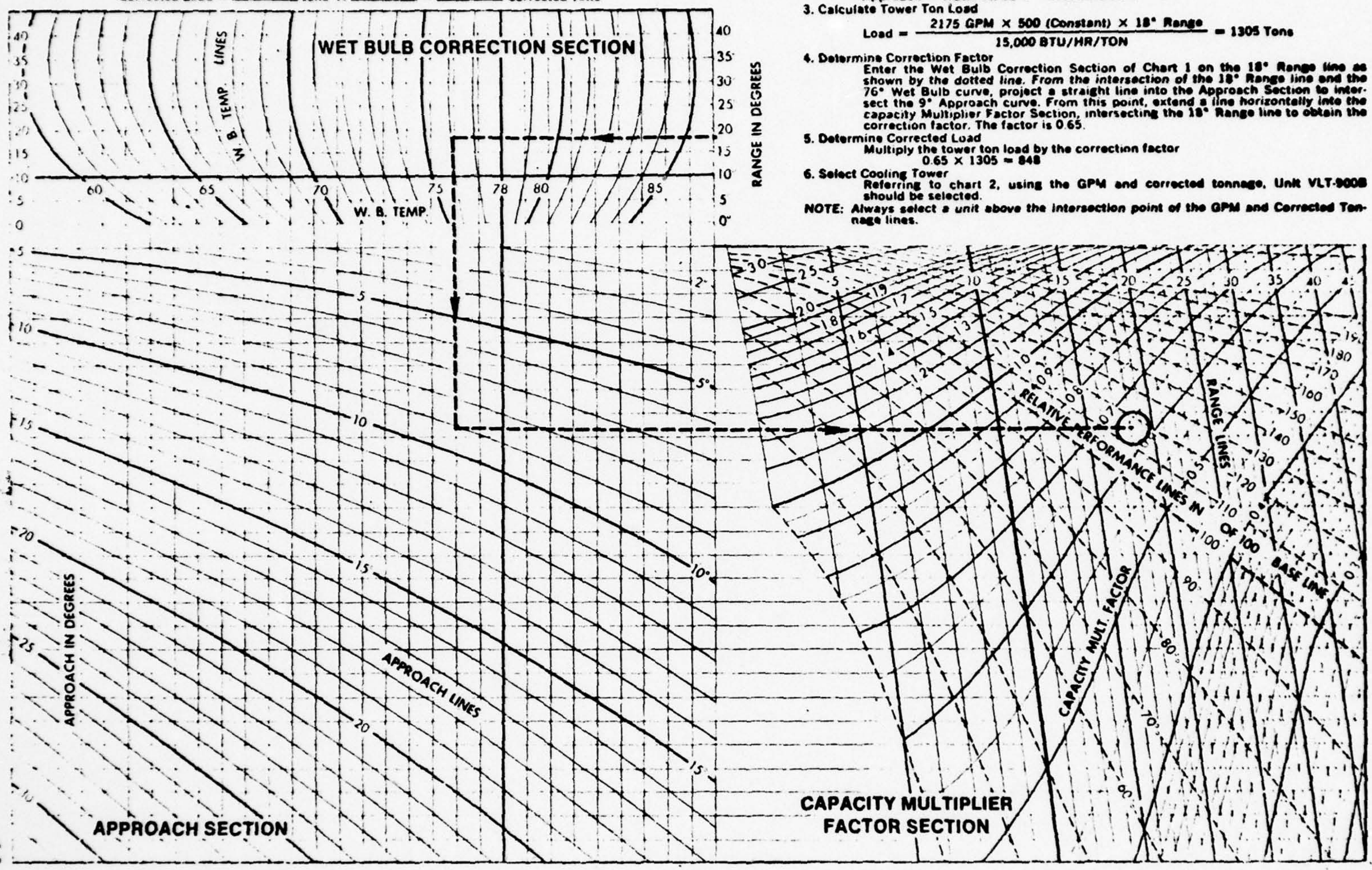
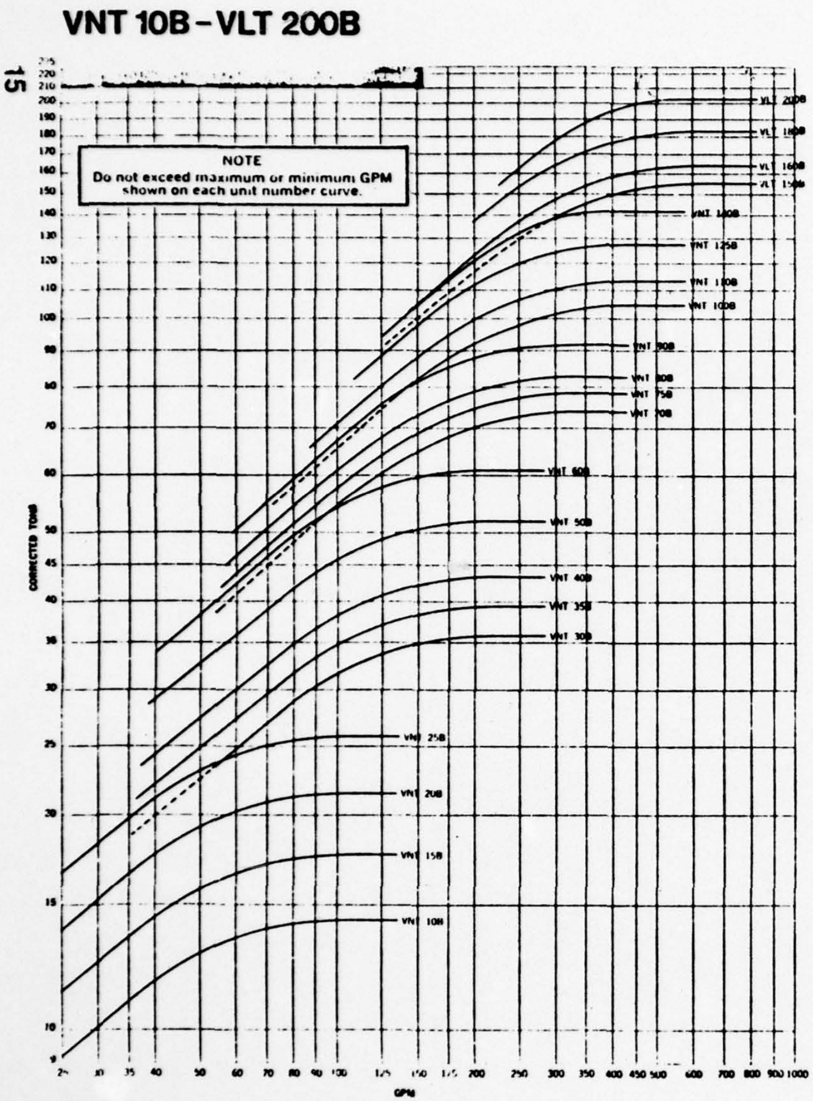


Figure E-1. Cooling tower data (courtesy of Baltimore Aircoil Company, Inc.).



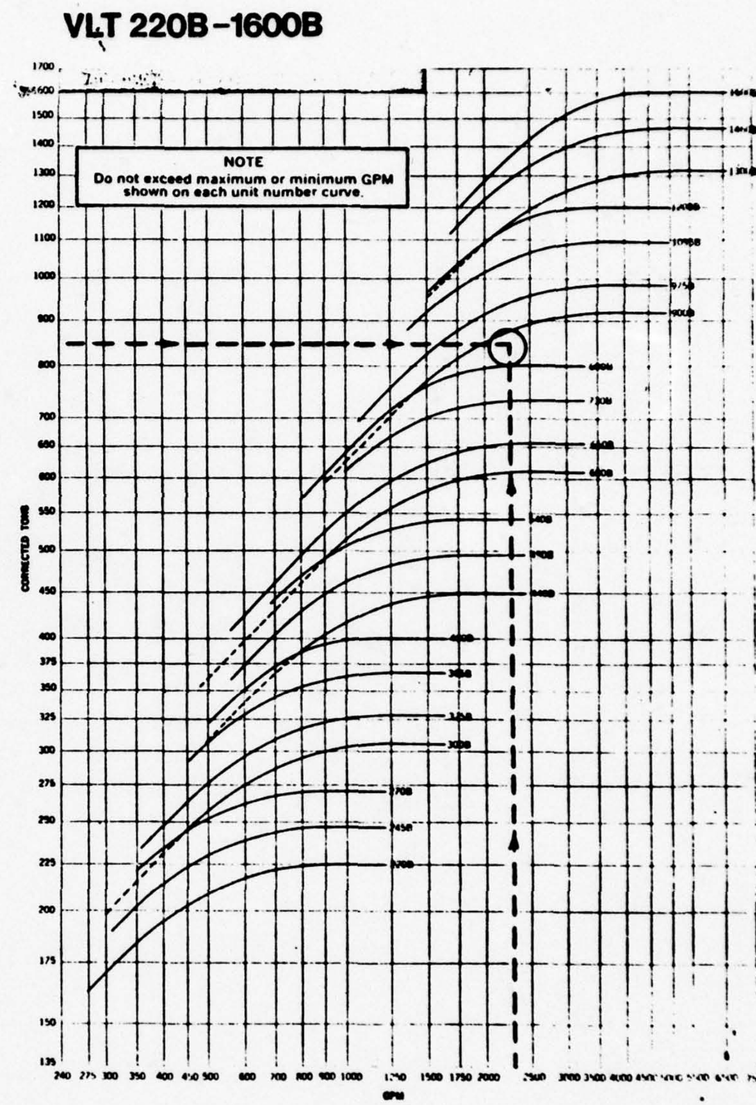


Figure E-2. Cooling tower data (courtesy of Baltimore Aircoil Company, Inc).

Engineering Data for V-Line Cooling Towers

MODEL NO.	WEIGHTS (LBS.) Approx. Shpg.	Approx. Oper.	Heaviest Section (Pan)	CFM	FAN MOTOR HORSEPOWER 0" ESP	1/8" ESP	1/4" ESP	3/8" ESP	1/2" ESP	DIMENSIONS A	B	H	Water Outlet Conn.	Make-up Conn. Size
VNT-10B	820	940	640	3720	¾	1	1	1	1½	5' 8⅜"	2' 7¾"	6' 5⅜"	3"	1"
VNT-15B	840	980	660	4710	2	3	3	3	3	5' 8⅜"	2' 7¾"	6' 5⅜"	3"	1"
VNT-20B	880	1000	660	4890	1½	2	2	2	2	5' 8⅜"	2' 7¾"	6' 5⅜"	3"	1"
VNT-25B	910	1030	690	5900	3	5	5	5	5	5' 8⅜"	2' 7¾"	6' 5⅜"	3"	1"
VNT-30B	1270	1550	930	9600	3	5	5	5	5	5' 8⅜"	2' 7¾"	6' 5⅜"	3"	1"
VNT-35B	1320	1600	980	8390	2	3	3	3	3	5' 8⅜"	2' 7¾"	6' 5⅜"	3"	1"
VNT-40B	1360	1640	990	9500	3	5	5	5	5	5' 8⅜"	2' 7¾"	6' 5⅜"	3"	1"
VNT-50B	1470	1750	990	11700	5	7½	7½	7½	7½	5' 8⅜"	2' 7¾"	6' 5⅜"	3"	1"
VNT-60B	1580	1840	1040	11920	7½	10	10	10	10	7' 2⅜"	4' 1¾"	7' 11⅜"	3"	1"
VNT-70B	2050	2520	1380	17170	7½	10	10	10	10	6' 3⅛"	3' 2½"	7' 0⅛"	4"	1"
VNT-75B	2090	2560	1380	17400	7½	10	10	10	10	6' 3⅛"	3' 2½"	7' 0⅛"	4"	1"
VNT-80B	2120	2590	1400	17700	10	10	10	15	15	6' 3⅛"	3' 2½"	7' 0⅛"	4"	1"
VNT-90B	2180	2650	1400	17600	10	15	15	15	15	7' 2⅜"	4' 1¾"	7' 11⅜"	4"	1"
VNT-100B	2470	3180	1740	23000	10	15	15	15	15	6' 3⅜"	3' 2½"	7' 0⅜"	4"	1"
VNT-110B	2620	3330	1800	23500	15	20	20	20	20	6'3⅜"	3' 2½"	7' 0⅜"	4"	1"
VNT-125B	3250	3580	1800	24630	15	20	20	20	20	7' 9⅛"	4' 8"	8' 6⅛"	4"	1"
VNT-140B	3580	3790	1800	26800	15	20	20	20	20	9' 11⅝"	6' 11"	10' 8⅝"	4"	1"
VLT-150B	5100	6220	3680	32500	15	20	20	20	20	8' 10½"	2' 9⅝"	9' 9"	6"	2"
VLT-160B	5450	6570	3680	33100	15	20	20	20	20	10' 4½"	4' 3⅝"	11' 3"	6"	2"
VLT-180B	5580	6680	3930	35300	20	25	25	25	25	10' 4½"	4' 3⅝"	11' 3"	6"	2"
VLT-200B	5870	6990	3930	38400	20	25	25	25	25	11' 10½"	5' 9⅝"	12' 9"	6"	2"
VLT-220B	6600	8460	4520	46700	2-10	2-15	2-15	2-15	2-15	8' 10½"	2' 9⅝"	9' 9"	8"	2"
VLT-245B	7050	8910	4520	48100	2-10	2-15	2-15	2-15	2-15	10' 4½"	4' 3⅝"	11' 3"	8"	2"
VLT-270B	7250	9110	4660	52300	2-15	2-15	2-20	2-20	2-20	10' 4½"	4' 3⅝"	11' 3"	8"	2"
VLT-300B	7740	10300	5310	66200	2-15	2-20	2-20	2-20	2-20	8' 10½"	2' 9⅝"	9' 9"	8"	2"
VLT-325B	7880	10440	5420	69800	2-20	2-25	2-25	2-25	2-25	8' 10½"	2' 9⅝"	9' 9"	8"	2"
VLT-365B	8500	11060	5420	71000	2-20	2-25	2-25	2-25	2-25	10' 4½"	4' 3⅝"	11' 3"	8"	2"
VLT-400B	9140	11700	5420	76800	2-20	2-25	2-25	2-25	2-25	11' 10½"	5' 9⅝"	12' 9"	8"	2"
VLT-440B	12490	15960	8750	93400	4-10	4-15	4-15	4-15	4-15	8' 10½"	2' 9⅝"	9' 9"	8"	2"
VLT-490B	13390	16860	8750	96200	4-10	4-15	4-15	4-15	4-15	10' 4½"	4' 3⅝"	11' 3"	8"	2"
VLT-540B	13770	17240	9000	104600	4-15	4-15	4-20	4-20	4-20	10' 4½"	4' 3⅝"	11' 3"	8"	2"
VLT-600B	14690	19400	10330	132400	4-15	4-20	4-20	4-20	4-20	8' 10½"	2' 9⅝"	9' 9"	10"	2-2"
VLT-650B	14960	19670	10600	139600	4-20	4-25	4-25	4-25	4-25	8' 10½"	2' 9⅝"	9' 9"	10"	2-2"
VLT-730B	16200	20910	10600	142000	4-20	4-25	4-25	4-25	4-25	10' 4½"	4' 3⅝"	11' 3"	10"	2-2"
VLT-800B	17480	22190	10600	153600	4-20	4-25	4-25	4-25	4-25	11' 10½"	5' 9⅝"	12' 9"	10"	2-2"
VLT-900B	21680	28890	15400	198600	6-15	6-20	6-20	6-20	6-20	8' 10½"	2' 9⅝"	9' 9"	2-10"†	2-2"
VLT-975B	22270	29300	15700	209400	6-20	6-25	6-25	6-25	6-25	8' 10½"	2' 9⅝"	9' 9"	2-10"†	2-2"
VLT-1095B	24140	31170	15700	213000	6-20	6-25	6-25	6-25	6-25	10' 4½"	4' 3⅝"	11' 3"	2-10"†	2-2"
VLT-1200B	26060	33090	15700	230400	6-20	6-25	6-25	6-25	6-25	11' 10½"	5' 9⅝"	12' 9"	2-10"†	2-2"
VLT-1300B	29380	38890	20600	279200	8-20	8-25	8-25	8-25	8-25	8' 10½"	2' 9⅝"	9' 9"	2-10"	2-2"
VLT-1460B	31870	41380	20600	284000	8-20	8-25	8-25	8-25	8-25	10' 4½"	4' 3⅝"	11' 3"	2-10"	2-2"
VLT-1600B	34430	43940	20600	307200	8-20	8-25	8-25	8-25	8-25	11' 10½"	5' 9⅝"	12' 9"	2-10"	2-2"

†One 10" connection may be used if flow does not exceed 3000 GPM.

Notes:

Spray Pressure: Approximately 5 psig at inlet header on all units.

All connections 6" and smaller are MPT. Connections 8" and larger are beveled-for-welding.

Standard connection arrangements are as shown. Variations can be furnished on special order. Consult B.A.C. representative for details.

When the cooling tower is installed with a remote sump, the tower should be specified with an oversized bottom outlet. The make-up valve and strainer normally furnished with the unit are omitted.

For indoor applications, the room may be used as a plenum with ductwork attached to the discharge only. If inlet ductwork is required, an enclosed fan section must be specified. Consult B.A.C. representative for details.

Standard fan motor locations are as shown. Single fan-side models VNT-10B through VLT-200B have one fan motor. Motors are located on both sides of double fan-side models VLT-220B through VLT-1600B.

Consult B.A.C. representative for available accessories and normal shipping breakdown.

Figure E-3. Cooling tower engineering data (courtesy of Baltimore Aircoil Company, Inc.).

Appendix F. Refrigeration Data

F-01. Introduction. This appendix provides information on the costs for procurement and operation of centralized refrigeration systems. It should be recognized that these data represent only rough estimating values and have been included only to illustrate the overall analysis procedure. No certification as to the accuracy or applicability of these data is expressed or implied.

F-02. Refrigeration System Capacity Calculations. In this report and the sample problems refrigeration loads have been calculated in units of tons of refrigeration. For this purpose one ton of refrigeration is defined as a cooling capacity of 12,000 Btu/hr.

In the sizing of the refrigeration units for the sample problems it has been assumed that a one ton capacity refrigeration unit can provide one ton of cooling capacity. In an actual system this one to one relationship need not be true. The influence of operating conditions such as the evaporator temperature and the condensing temperature can result in significant differences between the standard rated capacity of a refrigeration unit and its actual cooling capability in an operating system.

However, this level of sophistication was not included in the sample problems for two basic reasons. First, the actual relationship between ton rating and cooling capacity will be dependent upon the type of equipment selected and the actual loading conditions. Thus, each sample problem would have to consider a wide range of both equipment and operating procedures. These additional data would tend to obscure the basic purpose of the sample patterns which is to illustrate the conceptual design procedures for several different types of heat sink systems.

The second reason for not including these considerations is both practical and fundamental. A number of standard reference texts on the design and operation of refrigeration systems are readily available. Thus providing detailed discussions on refrigeration system design would be both repetitious and beyond the scope of this report.

F-03. Central Refrigeration System Procurement Costs. Figure F-1 provides a range of costs for central plant refrigeration systems. Although this figure does indicate that a substantial range in unit costs does occur, a median value can be assumed to provide a rough estimate.

F-04. Electrical Consumption for Central Refrigeration Systems: Figure F-2 provides conversion relationships which can be used to convert the ton rating of a central refrigeration system into an electrical horsepower requirement. This horsepower requirement can then be converted

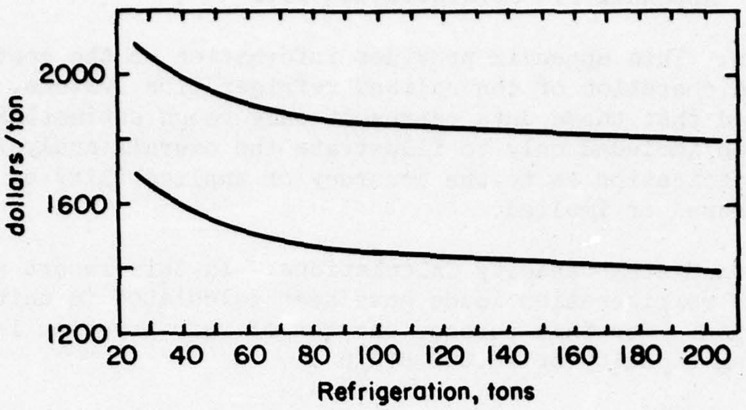


Figure F-1. Approximate costs for central plant refrigeration systems. (courtesy of the ASHRAE Journal).

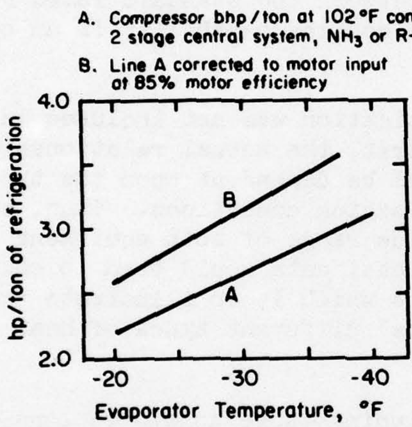


Figure F-2. Power requirements for central plant refrigeration systems (courtesy of the ASHRAE Journal).

into an electrical power requirement (i.e. kilowatts) through the use of a standard conversion factor.

F-05. Operating Costs for Central Refrigeration Systems. The operational cost factors which have been used in the sample problems have been developed from cost data related to the operation of the CRREL central refrigeration system. This system, which has a rated capacity of 200 tons (nominal) and can provide refrigerated brine at -35°F and -73°F, is staffed and operated on a 24-hour-per-day basis. Since this system is utilized for long term research studies and must provide a high degree of reliability, its design and operation probably provides a close approximation to the type of refrigeration systems which would be used to cool and maintain low temperature heat sinks.

F-05.1. Salary Requirements for the Operating Staff. The salary requirements for the operation of a central plant will be dependent directly upon the size of the staff. Excluding the consideration of extremely large cooling systems, the size of the staff will generally be independent of the installed plant capacity. It will be dependent significantly, however, on the number of hours the system is in operation. For 24-hour-per-day operation a staff of approximately 9 will be required. This staff will include a foreman, several refrigeration mechanics and the plant operators. The cost for such a staff will be around $120,000 per year.

F-05.2. Operating Supplies. Supplies will be required to perform normal preventative maintenance and to effect minor repairs. In general the annual expenditures for supplies will increase with the installed tonnage of the refrigeration system. An estimating factor of $250 per ton per year of operation has been used in the sample problems to account for the normal annual procurement of these supplies.

F-05.3. Overhaul Costs. Periodic major maintenance or overhaul will be required for any refrigeration plant which is to be operated for a substantial period of time. Such periodic maintenance normally involves a temporary shutdown of the plant and may include replacement of seals, bearings, valves, insulation and so forth. For the sample problems it has been assumed that this periodic major maintenance will be performed at three year intervals. A cost factor of $225 per ton per overhaul has been associated with this maintenance to account for the procurement of replacement parts and supplies.

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