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Heat Pipe Cooling System for Underground, Radioactive Waste Storage Tanks

University of California

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Heat Pipe Cooling System for Underground, Radioactive Waste Storage Tanks

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CONTENTS

HEAT PIPE COOLING SYSTEM FOR UNDERGROUND, RADIOACTIVE WASTE STORAGE TANKS

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ABSTRACT

An array of 37 heat pipes inserted through the central hole at the top of a radioactive waste storage tank will remove 100 000 Btu/h with a heat sink of 70°F atmospheric air. Heat transfer inside the tank to the heat pipe is by natural convection. Heat rejection to outside air utilizes a blower to force air past the heat pipe condenser.

The heat pipe evaporator section is axially finned, and is constructed of stainless steel. The working fluid is ammonia. The finned pipes are individually shrouded and extend 35 ft down into the tank air space. The hot tank air enters the shroud at the top of the tank and flows downward as it is cooled, with the resulting increased density furnishing the pressure difference for circulation. The cooled air discharges at the center of the tank above the sludge surface, flows radially outward, and picks up heat from the radioactive sludge. At the tank wall the heated air rises and then flows inward to complete the cycle.

INTRODUCTION

Cooling of a radioactive waste storage tank by heat pipes was investigated analytically. Heat pipes were inserted from above into the air space above the heat-generating sludge at the bottom of the tank. A letter from Rockwell International, reproduced as Appendix A, was used as a basis of the design. The configuration is shown in Fig. 1. This is a final report on Task 1.

DESIGN CONSTRAINTS AND ASSUMPTIONS

The following design constraints were observed:

- Upper surface temperature of the dry radioactive sludge was not to exceed 200°F.
- Heat pipes were to be inserted only through existing openings in the dome. These openings are one of 42 in. diam. and several 12 in. and 4 in. diam.
- Tank is to be sealed.
- Maximum length of heat pipe inside tank is equal to 40 ft.
- 100 000 Btu/h heat removal maximum, 60 000 Btu/h minimum.[1]
- Radiation level in tank 1000 rad/h.
- Tank diameter 75 ft.

The following assumptions were made:

- Because of low heat transfer coefficients of air to heat pipe, an extended surface (fins) was required on heat pipes.
- Heat pipe o.d. is 2 in. with 0.095 in. walls[2] (based on commercial steel tubing).
- Three working fluids were considered: water, ammonia, and methanol.
- Axial fins in the evaporator to be same material as heat pipes for ease of fabrication, and this material had to be compatible with the working fluid (carbon steel or stainless steel for methanol or ammonia, copper for water).
- A circular shroud surrounds each finned evaporator for maximum pumping efficiency. This shroud can be continued vertically downward past the end of the heat pipe to act as a downcomer to augment pumping if the heat exchange function of the heat pipe is completed at a shorter length than the pumping requirement.
- All heat pipes to be inserted vertically through the central 42-in.-diam opening in the dome.
- A fixed, insulated tube of maximum diameter is inserted through the 42 in. opening to reduce heat leakage from tank air to cooled, pumped air inside shrouds.
- Heat pipes are in equilateral triangular spacing. This results in possible arrays of 7, 19, 37, or 61 heat pipes.
- Electrical power available for condenser section blower if necessary.[1]

- Maximum daily averaged outside air temperature for design conditions 70°F.
- Friction factor f = 0.05, to be used with equation; pressure drop = f*L/D*velocity head.
- Shroud is not thermally attached to fins, and does not contribute to heat transfer. It is 0.01 in. thick.
- Reynolds number of air being cooled by heat pipe to exceed 2300, so the air is definitely turbulent (avoiding the transition region).
- Utilize maximum flow area for air being cooled, consistent with need for a flange to seal and support heat pipes on perforated plate in the 42 in. opening.
- Heat transfer coefficient from sludge surface to air = 0.75 Btu/h-ft^2-°F, corresponding to natural convection.[3] This gives a temperature drop from sludge to air of 30°F.
- Thermal conductivity of fins = 26.0 Btu/h-ft°F for carbon steel, 9.4 Btu/h-ft°F for stainless steel, and 225.0 Btu/h-ft°F for copper.
- Inlet and exit pressure losses 0.5 and 1.0 velocity heads, respectively.
- Other detailed assumptions in program listing, Appendix B.

DESIGN

The internal design of the heat pipe is discussed first, then the external design.

Heat Pipe Internal Design

The thermal transport requirements within the heat pipe are not critical to its design in this application. The heat pipe will serve as the major structural element of each cooling subassembly. A 2-in. pipe with a 0.095 in. wall thickness was chosen.[2] This may be the minimum diameter for which attachment of axial fins to the exterior is feasible. The heat pipe length is 55 ft: the evaporator is 35 ft, the adiabatic section is 10 ft and the condenser is 10 ft. Since the heat pipe material was specified to be stainless steel, either methanol or ammonia is a suitable working fluid. Water is acceptable from a performance standpoint but requires the heat pipe to be fabricated from copper. Water also has a potential freezing problem during shipment and storage which is disadvantageous.

Because heat is added at the bottom and removed from the top, the heat pipe internal design is simplified. A wickless configuration can be used with either a knurled or grooved interior to ensure liquid distribution around the inside wall. Since the preferred configuration utilizes 37 heat pipes and the total heat load is 29.3 kW, the load per heat pipe is 0.8 kW. A 2-in.-o.d. heat pipe provides a vapor flow area of 1660 mm^2, which gives vapor velocities well below the sonic limit for any of the working fluids. This makes axial temperature gradients negligible and the heat pipe performance will be limited by entrainment of the liquid by the counterflowing vapor. Table I lists performance characteristics for each of the working fluids considered. The entrainment limit is dependent on the wavelength[4] generated by the internal surface of the heat pipe. For small wavelengths, created by closely spaced grooves, the entrainment limit increases; however, the finer grooves are costly and require more fluid inventory in the heat pipe. A maximum wavelength of 6.35 mm, corresponding to four threads per inch, was chosen and used to calculate the entrainment limit for each working fluid. All of the entrainment limits are significantly above the 0.8 kW power requirement.

Also of concern is the thickness of the liquid layer in the condenser. Since the proposed working fluids have a poor thermal conductivity, any buildup of liquid on the inside of the condenser wall could lead to a significant radial temperature gradient. Table I shows both the liquid layer thickness and the temperature difference across the layer.[5] In all cases the low power transport in the heat pipe contributes to thin liquid layers and small temperature gradients.

The vapor pressure of each working fluid at 200°F is also shown in Table I. Complete failure of the heat rejection system will result in maximum temperature. Although ammonia has the highest working pressure, it is still below the allowable pressure of 2100 psi for the 2-in. stainless steel pipe.

With the exception of the freezing problem for water, any of the three working fluids considered are acceptable from heat pipe performance considerations, therefore, chemical stability in the radiation environment is the primary consideration in choosing the working fluid. It is unlikely that any of the fluids will be completely stable and some decomposition will occur. The design should, therefore, include the capability for in situ replacement of the working fluid. This can be accomplished with a fill tube and shut-off valve installed at the condenser end of each heat pipe. In addition, temperature monitoring of the condenser is necessary to determine the presence of non-condensible gas in the heat pipe.

TABLE I
COMPARISON OF HEAT PIPE WORKING FLUIDS

Working Fluid	Vapor Pressure @200°F, (kPa)	Entrainment Limit @ 90°F (kW)	Liquid Layer Thickness (mm)	Radial ΔT in Condenser (C)
Water	12	6.4	1.85×10^{-2}	0.2
Ammonia	780	23.9	4.57×10^{-2}	0.3
Methanol	39	4.9	3.60×10^{-2}	0.6

Heat Pipe External Design--Evaporator

The heat transfer conditions external to the heat pipe are critical to the performance of the proposed system and are therefore discussed here in considerable detail.

A computer program, TANKL3, was written to make a parametric study of the evaporator external design. The program listing is given in Appendix B with comments. Parametric studies were made with copper, carbon steel, and stainless steel materials. Stainless steel was then chosen as the desired material, based on corrosion resistance. The following are evaporator conditions and dimensions for the proposed design, using stainless steel fins and heat pipe. The heat pipe external design features are shown in Figs. 2 and 3.

The computer printout from which this design was selected is shown in Table III. Fin thickness, number of fins, and downcomer temperature were varied. The design chosen has an acceptable length and Reynolds number. Other choices are possible.

TABLE II

HEAT PIPE EVAPORATOR DESIGN

Heat removal (Btu/h)	100 000
Tank air temperature (°F)	170
Heat pipe temperature (°F)	80
Shroud exit air temperature (°F)	85
Number of heat pipes	37
Heat pipe o.d. (in.)	2.0
Fin thickness (in.)	0.05
Number of fins	8
Min. evaporator length (ft)	31.3
Reynolds number	2 600
Fin effectiveness	0.69
Shroud o.d (in.)	5.37
Heat pipe centerline spacing (in.)	6.0

Heat Pipe External Design--Condenser

In the evaporator, the large (85°F) drop in air temperature provided the density difference required to pump the air and, in turn, required a low (80°F) heat pipe temperature. The heat pipe is too cool to permit natural convection cooling to outside air at 70°F. A moderate wind will not help enough, either, so a forced draft is necessary. An electrically-driven blower sends cooling air across the heat pipes having helical fins. This part of the design has not been calculated in detail, but the following characteristics will apply to the design.

- Helical fins with close fin spacing give a large heat transfer area in a short length so the condenser length will not be excessive.
- The cooling system will function at lower than design capacity when the outside air exceeds 70°F. Because of the large heat capacity of the sludge, tank, and coupled soil, the temperature of the sludge can be drawn down in the cooler months, and warm up in July

TABLE III
HEAT PIPE EXTERNAL DESIGN-EVAPORATOR

TANKL3, HEAT PIPE TO COOL WASTE TANK
LONGITUDINAL FINS + SHROUD

NOMENCLATURE, IN ORDER OF APPEARANCE IN PRINTOUT

HT=HEAT REMOVED FROM TANK (1000 BTU/HR)
THP=HEAT PIPE TEMPERATURE (DEG F)
NHP=NUMBER OF HEAT PIPES
HPOD=O.D. OF HEAT PIPE (IN.)
SOD=O.D. OF SHROUD (IN.)
F=ASSUMED FRICTION COEFFICIENT
TAI=TANK AIR TEMPERATURE (DEG F)
VHLI=INLET VELOCITY HEADS LOSS (LBF/SQ FT)
VHLO=OUTLET " " " " "
KFIN=FIN THERMAL CONDUCTIVITY (BTU/(HR FT DEG F))

FT=FIN THICKNESS (IN.)
NF=NUMBER OF FINS PER HEAT PIPE
TAO=DOWNCOMER AIR TEMPERATURE (DEG F)
RE=REYNOLDS NUMBER IN PASSAGE
H=HEAT TRANSFER COEFFICIENT (BTU/HR SQ FT DEG F)
ETA=FIN EFFICIENCY
ETAW=WEIGHTED HEAT TRANSFER EFFICIENCY
LEV=HEAT PIPE EVAPORATOR LENGTH (FT)
LTOT=EVAPORATOR+DOWNCOMER LENGTH (FT)

HT	THP	NHP	HPOD	SOD	F	TAI	VHLI	VHLO	KFIN
100	80	37	2.00	5.37	.050	170	.5	1.0	9.4

FT	NF	TAO	RE	H	ETA	ETAW	LEV	LTOT
.04	8	95	2975	1.6	.63	.70	21.4	31.7
.05	8	95	2989	1.6	.67	.73	20.3	30.8
.06	8	95	3004	1.6	.71	.76	19.5	30.2
.04	10	95	2628	1.7	.62	.68	17.7	29.9
.05	10	95	2642	1.7	.67	.72	16.7	29.1
.06	10	95	2657	1.7	.70	.74	16.1	28.6
.04	12	95	2353	1.7	.62	.66	15.0	28.7
.05	12	95	2367	1.7	.66	.70	14.2	27.9
.06	12	95	2381	1.7	.70	.73	13.6	27.5
.04	16	95	1947	1.8	.61	.64	11.5	27.1
.05	16	95	1959	1.8	.65	.68	10.8	26.5
.06	16	95	1972	1.8	.68	.71	10.2	26.2
.04	8	90	2789	1.5	.64	.71	25.5	29.1
.05	8	90	2803	1.5	.68	.74	24.3	28.2
.06	8	90	2816	1.5	.72	.77	23.4	27.7
.04	10	90	2464	1.6	.63	.69	21.1	27.2
.05	10	90	2477	1.6	.68	.72	20.0	26.4
.06	10	90	2491	1.6	.71	.75	19.2	25.9
.04	12	90	2206	1.6	.63	.67	18.0	25.9
.05	12	90	2219	1.6	.67	.71	17.0	25.2
.06	12	90	2232	1.6	.71	.74	16.2	24.8
.04	16	90	1825	1.7	.62	.65	13.7	24.3
.05	16	90	1837	1.7	.66	.69	12.9	23.6
.06	16	90	1849	1.7	.69	.72	12.2	23.4
.04	8	85	2625	1.5	.65	.71	32.8	32.8
.05	8	85	2638	1.5	.69	.75	31.3	31.3
.06	8	85	2651	1.5	.73	.78	30.1	30.1
.04	10	85	2319	1.5	.64	.70	27.1	27.1
.05	10	85	2331	1.5	.69	.73	25.7	25.7
.06	10	85	2344	1.5	.72	.76	24.7	25.1
.04	12	85	2076	1.5	.64	.68	23.0	25.0
.05	12	85	2089	1.6	.68	.72	21.8	24.2
.06	12	85	2101	1.6	.71	.75	20.9	23.8
.04	16	85	1718	1.6	.63	.66	17.6	23.2
.05	16	85	1729	1.6	.67	.70	16.5	22.5
.06	16	85	1740	1.7	.70	.73	15.7	22.2

and August when the average monthly temperatures exceed 70°F (77.5 and 75.3°F, respectively),[3] compared to the annual average of 53.5°F. As the air temperature drops, the forced cooling can be reduced.

RADIATION DAMAGE

The heat pipe evaporators are in a gamma radiation environment of about 1000 rad/h. This will decompose the working fluid to some extent. Methanol is expected to suffer more damage than water or ammonia. Conservative calculations indicate a moderate decomposition rate. The noncondensible gases will be carried by the working fluid vapor to the top of the heat pipe where they will gradually reduce the condenser area. For this reason, the heat pipe has a valve at the top for bleeding gases and replenishing working fluid. Ammonia, having a vapor pressure above atmospheric, is easier to bleed and replenish than water.

The diffusion of hydrogen gas through the heat pipe wall is very slow, and presents no problem to the tank. Periodic venting of the tank to remove hydrogen will not affect the operation of the cooling system.

CONCLUSIONS

The analysis presented herein demonstrates the feasibility of using an array of finned heat pipes to remove waste heat from underground radioactive waste storage tanks. The design philosophy concentrates on utilizing existing technology combined with adequate performance margins with the goal of minimizing development time and reducing cost.

The design is based on passive thermal components inside the tank with tank penetration through existing dome ports. In addition, uncertainties associated with degradation of the heat pipe working fluid in the radiation environment are overcome by including on-site fluid replacement capability. Boundary temperatures were selected based on worst case hot conditions. If minimum temperature limits are established, these can be met by controlling fan power and, hence, the thermal conductance to the external environment.

Commercially available materials and sizes were selected wherever possible. This should greatly reduce costs especially where a small number of units are required.

REFERENCES:

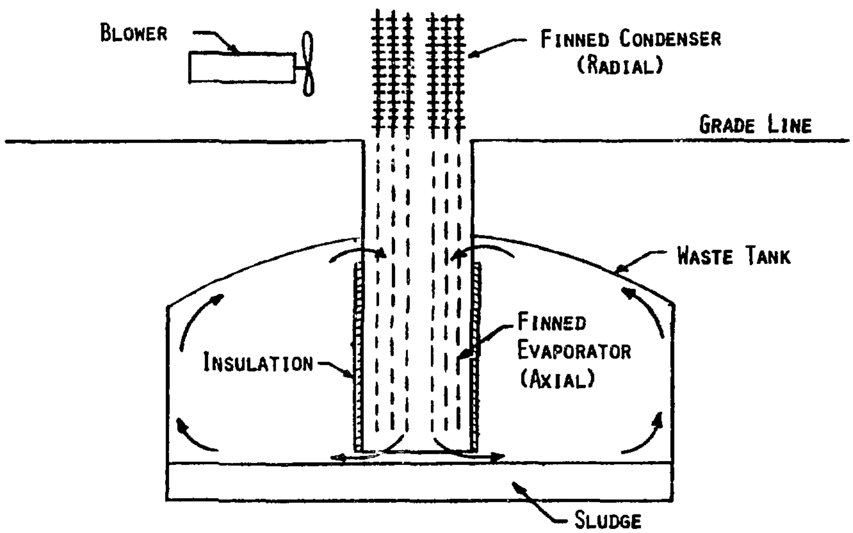
1. S. Bath, Rockwell International, Personal Communication, July 1979.

2. G. Grover, Q-DOT Corporation, August 1979.

3. F. Kreith, Principles of Heat Transfer, International Textbook Co., Scranton, PA, 1965.

4. J. E. Kemme, "Vapor Flow Considerations in Conventional and Gravity-Assist Heat Pipes," Los Alamos Scientific Laboratory report, LA-UR-75-2308.

5. P. D. Dunn and D. A. Reay, Heat Pipes, Pergamon Press, Oxford, 1976.



ARROWS DENOTE AIR CIRCULATION WITHIN THE TANK

Fig. 1. Cooling concept for waste storage tank.

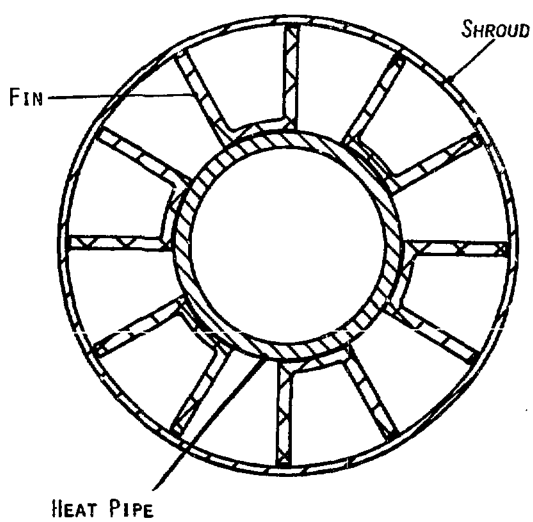


Fig. 2. Heat pipe cross section.

APPENDIX A
SYSTEM DESCRIPTION

The radioactive waste storage tank system can be described in the following major sections:

Storage Tank:

A radioactive waste storage tank consists of a steel shell (3/8 in. thick) with a 6 in. thick concrete vault around it. The tank is buried underground with at least 6 ft layer of soil over the center of the dome. There are several risers (openings) in the tank dome, used for processing and monitoring purposes. These risers vary from four inches to twelve inches in diameter. There is a 42-in. diam opening in the center of the dome. The inside diameter of a storage tank is 75 ft. The center to center distance between two storage tanks is 102 ft. The height of the tank at the center is 46 ft.

Burial Ground

The tanks are buried in the ground. The ground (Hanford soil) consists of sand and gravel with very low moisture content. The thermal conductivity of average Hanford soil is 0.25 BTU/hr ft ^{o}F. This was determined experimentally.

Waste Sludge

The radioactive waste sludge consists of mainly the metal oxides, salts of sodium and heavy metal ions. The sludge contains most of radionuclides present in the tank. The radiation level in the sludge is in the order of 1000 rads/h. The thickness of the sludge at the tank bottom varies between four feet to six feet. The total heat generated by the sludge is about 80,000 BTU/h. The average thermal conductivity of the dry sludge was determined to be 0.25 BTU/h ft ^{o}F.

Pertinent Data:

i.d. of the storage tank	75 ft
Depth of soil at the dome center	6 ft
Height of the tank at dome center	46 ft
Depth of heat generating sludge at the tank bottom	4-6 ft
Depth of the water table	200 ft
Total heat generated in the sludge	80,000 BTU/h
Highest allowable temp at the sludge surface	$200^{o}F$
Radiation level in the tank (approximation)	1000 Rads/h
Thermal conductivity of the soil	0.25 BTU/h ft ^{o}F
Thermal conductivity of the sludge	0.25 BTU/h ft ^{o}F
Size of openings in the tank dome	4-42 in.

Program Definition:

The objective of this program is to determine the technical and economical feasibility of an air thermosyphon for cooling an underground waste storage tank with 60,000 to 100,000 BTU/h heat generation rate.

Scope of Work

Task-1: Perform thermal analysis and energy balance for a thermosyphon system on an underground waste storage tank to determine it's feasibility. Parametric study to determine the lowest sludge surface temperature and duct size for the air flow will be made.

Task-2: This task requires that a conceptual design report be prepared which includes cost estimates, prototype test unit design, and environmental assessment.

Tast-3: Analyze prototype test data to scale up the thermosyphon system for an actual tank.

Progress Reporting:

A technical report (in letter form) will be needed at the completion of each task.

Key Personnel:

Charles Anderson

Coyne Prenger

Joe Kemme

Technical Liaison

S. S. Bath

Milestone Completion Dates:

Complete Task 1	July 31, 1979
Complete Task 2	August 31, 1979

Task 3 will be completed within two months after the data is made available.

LASL Identification
No. LP-2049 TANKL 3

APPENDIX B

This program calculates heat transfer, friction pressure drop and density increase for air being cooled by the heat pipe. It calculates incrementally until the specified air discharge temperature is reached. If the pressure is negative at this point, a downcomer addition is calculated.

```
$ FTN (I=TANKL3,GO)
      PROGRAM TANKL3(TTY,INPUT,OUTPUT=TTY)
C     HEAT PIPE TO COOL WASTE TANK, LONGITUDINAL FINS + SHROUD
      DIMENSION FTX(10),NFX(10),DELX(10)
      REAL H,HT,KAY,KFIN,MU,LFIN,LTOT,LINC,LEV,LD,HPOD
C     FT IS FIN THICKNESS (IN.)
      DATA(FTX(I),I=1,3)/.04,.05,.06/
C     NF IS NUMBER OF FINS PER HEAT PIPE
      DATA(NFX(I),I=1,4)/8,10,12,16/
C     DEL IS HEAT PIPE- AIR OUTLET TEMPERATURE (DEG F)
      DATA (DELX(I),I=1,3)/15.,10.,5./
C     TO PRINT NOMENCLATURE, SET NPRINT=1
      NPRINT=1
      PI=3.14159
C     TOTAL TANK HEAT (BTU/HR)
      PWR=1.0E5
C     HEAT RELEASE (1000 BTU/HR)
      HT=PWR/1000.
C     HEAT PIPE TEMPERATURE (DEG F)
      THP=80.
C     NUMBER OF HEAT PIPES
      NHP=37
C     HEAT PIPE OUTER DIAMETER (IN.)
      HPOD=2.
C     SHROUD OUTER DIAMETER (IN.)
      SOD=5.37
C     SHROUD THICKNESS (IN.)
      SHRT=.01
C     AIR INLET TEMPERATURE (DEG F)
      TAI=170.
C     LENGTH OF INCREMENT (FT)
      LINC=.01
C     INLET PRESSURE LOSS, VELOCITY HEADS
      VHLI=.5
C     OUTLET PRESSURE LOSS, VELOCITY HEADS
      VHLO=1.
C     THERMO PROPS OF AIR AT 150 F (FRAAS + OZISIK)
C     SPECIFIC HEAT (BTU/(LBM*DEG F))
      CP=.241
C     THERMAL CONDUCTIVITY (BTU/(HR*FT*DEG F))
      KAY=.0173
C     VISCOSITY (LB/(HR*FT))
      MU=.0493
```

```
C       PRANDTL NUMBER
        PN=.687
C       ASSUMED FRICTION COEFFICIENT (DARCY-WEISSBACH)
        F=.05
C       FIN THERMAL CONDUCTIVITY (BTU/(HR*FT*DEG F))
        KFIN=9.4
        PRINT 2000
        PRINT 2010
        IF(NPRINT.EQ.1) GO TO 2200
    900 CONTINUE
        IHT=IFIX(HT)
        ITHP=IFIX(THP)
        ITAI=IFIX(TAI)
        PRINT 2020
        PRINT 2030,IHT,ITHP,NHP,HPOD,SOI,F,ITAI,VHLI,VHLO,KFIN
        PRINT 2100
C       START CALCULATION
        DO 1200 L=1,3
        DEL=DELX(L)
        DO 1200 M=1,4
        NF=NFX(M)
        DO 1200 N=1,3
        FT=FTX(N)
C       AIR OUTLET TEMPERATURE (DEG F)
        TAO=THP+DEL
C       FIN HEIGHT (IN.)
        FH=(SOD-HPOD)/2.-SHRT
C       TOTAL AIR FLOW RATE (LBM/HR)
        AFT=PWR/(CP*(TAI-TAO))
C       AIR FLOW RATE PER HEAT PIPE (LBM/S)
        AF=AFT/(NHP*3600.)
C       NUMBER OF COOLING PASSAGES PER HEAT PIPE
        NH=NF
C       AIR FLOW RATE PER PASSAGE (LBM/S)
        AFP=AF/NH
C       AIR PASSAGE INNER DIAMETER=HEAT PIPE O.D. (IN.)
        DIF=HPOD
C       AIR PASSAGE INNER ARC WIDTH (IN.)
        WAPI=PI*HPOD/NF-FT
   3100 CONTINUE
C       INITIAL VALUE OF STATION (FT)
        X=0.
C       AIR PASSAGE OUTER DIAMETER (IN.)
        DOF=SOD-2.*SHRT
C       AIR PASSAGE OUTER ARC WIDTH (IN.)
        WAPO=PI*DOF/NF-FT
C       AIR PASSAGE PERIMETER FOR FRICTION (IN.)
        PAP=WAPI+WAPO+2.*(FH-FT/2.)
C       AIR PASSAGE PERIMETER FOR HEAT TRANSFER (IN.)
        PHT=WAPI+2.*(FH-FT/2.)
C       INCREMENT HEAT TRANSFER AREA (SQ FT)
        AHTI=PHT*LINC/12.
C       AIR PASSAGE FLOW AREA (SQ FT)
        AAP=PI/4.*((DOF**2-DIF**2)/NF-FT*FH-WAPI*FT/2.)/144.
```

```
C     EQUIVALENT DIAMETER (FT)
      DEQ=4.*AAP/PAP*12.
C     REYNOLDS NUMBER
      RE=AFP*DEQ*3600./(AAP*MU)
C     HEAT TRANSFER COEFFICIENT (BTU/(HR SQ FT DEG F))
      H=.023*KAY*RE**.8*PN**.4/DEQ
C     FIN EFFICIENCY (ECKERT-DRAKE H+MT)
      EM=(2.*H/(KFIN*FT/12.))**.5
      ELEF=FH/12.
      ETA=TANH(EM*ELEF)/(EM*ELEF)
C     WEIGHTED HEAT TRANSFER EFFICIENCY (HEAT PIPE+FIN)
      ETAW=(ETA*FH+WAPI/2.)/(FH+WAPI/2.)
C     INLET CONDITIONS
C     INLET DENSITY (LBM/CU FT)
      RHI=39.7/(460.+TAI)
C     INLET VELOCITY (FT/S)
      VAI=AFP/(RHI*AAP)
C     INLET VELOCITY HEAD (LBF/SQ FT)
      VHI=RHI*VAI**2/64.4
C     INLET PRESSURE LOSS (LBF/SQ FT)
      DPI=VHI*VHLI
C     SET INITIAL GAGE PRESSURE (LBF/SQ FT)
      PG=0.
C     KEEP RUNNING TRACK OF GAGE PRESSURE
C     TOTAL PRESSURE JUST PAST INLET
      PG=PG-DPI
C     TAKE EXIT VELOCITY HEAD LOSS NOW TO SIMPLIFY BOOKKEEPING
C     OUTLET CONDITIONS
C     OUTLET DENSITY (LBM/CU FT)
      RHO=39.7/(460.+TAO)
C     OUTLET VELOCITY (FT/SEC)
      VAO=AFP/(RHO*AAP)
C     OUTLET VELOCITY HEAD (LBF/SQ FT)
      VHO=RHO*VAO**2/64.4
C     OUTLET PRESSURE LOSS (LBF/SQ FT)
      DPO=VHO*VHLO
      PG=PG-DPO
C     INITIAL AIR TEMPERATURE (DEG F)
      TA=TAI
C     STEP ALONG INCREMENTS UNTIL AIR IS COOLED TO TAO
  950 CONTINUE
C     VELOCITY HEAD AT START OF INCREMENT (LBF/SQ FT)
      VH=VHI*TA/TAI
C     FRICTION PRESSURE DROP IN INCREMENT (LBF/SQ FT)
      DPF=F*VH*LINC/DEQ
C     DENSITY AT START OF INCREMENT (LBM/CU FT)
      RH=39.7/(460.+TA)
C     BUOYANT DRIVING PRESSURE FOR INCREMENT (LBF/SQ FT)
      DPB=LINC*(RH-RHI)
C     GAGE PRESSURE AT END OF INCREMENT (LBF/SQ FT)
      PG=PG+DPB-DPF
C     AIR TEMPERATURE DROP IN INCREMENT (DEG F)
      DT=(TA-THP)*H*AHTI*ETAW/(CP*AFP*3600.)
C     AIR TEMPERATURE AT END OF INCREMENT (DEG F)
```

```
      TA=TA-DT
C     LENGTH ALONG HEAT PIPE (FT)
      X=X+LINC
      IF(TA.GT.TAO) GO TO 950
 1000 CONTINUE
C     LENGTH OF EVAPORATOR (FT)
      LEV=X
C     GAGE PRESSURE AT END OF EVAPORATOR (LBF/SQ FT)
      PGE=PG
      IF(PGE.GT.0) GO TO 1050
C     GET LENGTH OF DOWNCOMER TO RECOVER PGE
C     DOWNCOMER DENSITY (LBM/CU FT)
      RHD=39.7/(460.+TAO)
C     DIAMETER OF DOWNCOMER (FT)
      DDC=(SOD-FT)/12.
C     GROSS BUOYANT PRESSURE PER FOOT (LBF/SQ FT)
      DPB=RHD-RHI
C     DOWNCOMER VELOCITY (FT/SEC)
      VDC=AF*4./(RHD*PI*DDC**2)
C     DOWNCOMER VELOCITY HEAD (LBF/SQ FT)
      VHDC=RHD*VDC**2/64.4
C     FRICTION PRESSURE DROP PER FOOT (LBF/SQ FT)
      DPF=F*VHDC/DDC
C     NET BUOYANT PRESSURE DROP PER FOOT (LBF/SQ FT)
      DPN=DPB-DPF
C     DOWNCOMER LENGTH (FT)
      LD=-PG/DPN
      GO TO 1070
 1050 LD=0.
 1070 CONTINUE
C     TOTAL LENGTH EVAPORATOR + DOWNCOMER (FT)
      LTOT=LEV+LD
      ITAO=IFIX(TAO)
      IRE=IFIX(RE)
      PRINT 2110,FT,NF,ITAO,IRE,H,ETA,ETAW,LEV,LTOT
 1200 CONTINUE
      GO TO 2600
 2000 FORMAT(/*TANKL3, HEAT PIPE TO COOL WASTE TANK*)
 2010 FORMAT(* LONGITUDINAL FINS + SHROUD*)
 2020 FORMAT(/4X*HT*,2X*THP*,1X*NHP*,2X*HPOD*,4X*SOD*,5X*F*,
     12X*TAI*,1X*VHLI*,1X*VHLO*,3X*KFIN*)
 2030 FORMAT(I6,I5,I4,F6.2,F7.2,F6.3,I5,2F5.1,F7.1)
 2100 FORMAT(/4X*FT*,2X*NF*,2X*TAO*,5X*RE*,4X*H*,2X*ETA*,1X*ETAW*,
     13X*LEV*,2X*LTOT*)
 2110 FORMAT(F6.2,I4,I5,I7,F5.1,2F5.2,F6.1,F6.1)
 2200 CONTINUE
      PRINT 2280
      PRINT 2290
      GO TO 900
 2280 FORMAT(/*NOMENCLATURE, IN ORDER OF APPEARANCE IN PRINTOUT*
     1//4X*HT=HEAT REMOVED FROM TANK (1000 BTU/HR)*
     1/3X*THP=HEAT PIPE TEMPERATURE (DEG F)*
     1/3X*NHP=NUMBER OF HEAT PIPES*
     1/2X*HPOD=O.D. OF HEAT PIPE (IN.)*
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     1/3X◆SOD=O.D. OF SHROUD (IN.)◆
     1/5X◆F=ASSUMED FRICTION COEFFICIENT◆
     1/3X◆TAI=TANK AIR TEMPERATURE (DEG F)◆
     1/2X◆VHLI=INLET VELOCITY HEADS LOSS (LBF/SQ FT)◆
     1/2X◆VHLO=OUTLET     "        "     "     "    "    ◆
     1/2X◆KFIN=FIN THERMAL CONDUCTIVITY (BTU/(HR FT DEG F))◆)
2290 FORMAT(/◆    FT=FIN THICKNESS (IN.)◆
     1/4X◆NF=NUMBER OF FINS PER HEAT PIPE◆
     1/3X◆TAD=DOWNCOMER AIR TEMPERATURE (DEG F)◆
     1/4X◆RE=REYNOLDS NUMBER IN PASSAGE◆
     1/5X◆H=HEAT TRANSFER COEFFICIENT (BTU/HR SQ FT DEG F)◆
     1/3X◆ETA=FIN EFFICIENCY◆
     1/2X◆ETAW=WEIGHTED HEAT TRANSFER EFFICIENCY◆
     1/3X◆LEV=HEAT PIPE EVAPORATOR LENGTH (FT)◆
     1/2X◆LTOT=EVAPORATOR+DOWNCOMER LENGTH (FT)◆)
2600 CONTINUE
     STOP
     END
```