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TUNNEL TEMPERATURE STUDY

D. Shuman

SSC Central Design Group

Conventional Facilities

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Tunnel Temperature Profile — D. Shuman

Tunnel air and cryostat shell temperature functions are explored in this note. Except in unusual situations, the cryostat shell temperature equals that of the surrounding air. During operation, no outside air is pulled into the tunnel, however, air is kept circulating around the tunnel to distribute generated heat. Temperature is a function of both the heat generated in the tunnel and the circulation velocity. During maintenance, fresh air is pulled into the tunnel at the exit/vent shafts and exhausted at the service area shafts. If this air is at a different temperature from that of the tunnel it will have to travel a certain distance before it equilibrates with the tunnel wall.

1. Incoming Air Temperature Decay Length, £=263 meters

As air (at an initial outside temperature) travels into the tunnel, it loses or gains heat from the tunnel walls (which are essentially at a constant temperature equal to that of the surrounding rock) until the the temperature difference between the air and the wall is zero. This temperature difference follows an exponential decay function when plotted against distance from the air entry point, and the decay length is defined as the distance along the tunnel from the air entry point at which the temperature difference has decayed to 37% (1/e) of its original value. For the case of little or no heat input (from cables, etc.) into the tunnel air, the decay length is computed to be 263 meters, using the following tunnel parameters as follows:

1. Tunnel cross section radius, r = 1.5 meters
2. Average tunnel air speed, v = 0.5 m/s
3. Thermal conductivity of rock, k = 3.5 Watt/meter °C.
4. Convection heat transfer coeff. thru wall, h = 1.6 Watt/sq. meter °C.

The decay length formula and value for h are derived in the appendix. Assumptions are that wall temperature remains constant over time and length. Since air is to be pulled into the tunnel only for maintenance, and the thermal time constant of the wall temperature is on the order of a year, this assumption holds true. See the Appendix, Wall Temp. Profile, for a discussion of this.

Daily temperature changes of the earth do not penetrate further than 3-4 ft. and annual summer/winter temperature differences are not seen at depths of more than 60-70 ft. Earth temperature at SSC depths has been estimated to be 66°F for SSC depths (from Texas proposal, Volume 7).

2. Cryostat shell Temperature

Cryostat shell temperature is essentially that of the surrounding air. Heat loss to the 80K shield is negligible; the thermal time constant is fairly short, and the support stands do not act as thermal short circuits to tunnel wall or floor (assumptions checked in appendix).

2.1. Thermal time constant

Cryostat shell thermal time constant is calculated, with the above assumptions, in the appendix to be 4 hours. This is the time it takes for the cryostat shell to respond to a sudden temperature change of the ambient air such that the temperature difference between shell and ambient reaches 37% of the initial difference. Faster than normal air flows and/or condensation/evaporation on the cryostat shell will reduce this time constant. Conversely, insulating the cryostat shell on the outside will increase the time constant.

2.2. Installation and servicing considerations

For structural reasons, it may be desirable to bolt the magnets rigidly to their support stands or footings. The main ring dipole cryostat shell expands/contracts a distance of 0.0058" between the support feet (540" foot spacing) for each °C of temperature change. It is assumed that the magnets would be allowed to come to thermal equilibrium before being aligned and bolted rigidly to their support stands. Therefore, the above mentioned cryostat shell thermal expansion

properties will primarily be of concern where large air temperature excursions may occur (near exit/vent shafts) and to a lesser extent the entire tunnel (generated heat fluctuations and niches). Fresh air will be pulled into the tunnel only for installation and maintenance. The CDR specifies a 100 kW heater at each exit/vent shaft, but no cooling provisions. The heater is sufficient to raise the temperature 24 °C, which should be adequate for servicing in most winter conditions. Therefore the support stands need no contraction cabability (unless the capability for withstanding cryogen leaks is to be incorporated). For summer conditions where servicing is to be carried out on a continuous basis for several days or more, magnet supports in these areas must either be designed to allow for shell expansion, or the air can be heated/cooled and/or dehumidified as it enters the tunnel. It will require 32 tons of refrigeration (≈32 kW power) plus 38 kW heating to bring outside air at 85 °F and 60 % RH to 65 °F and 60 % RH (average summer conditions). See appendix for other conditions.

Rigid magnet supports without (full) expansion/contraction capability might be feasible under the following conditions:

For short servicing periods, or for day/night air temperature averages which are close to the rock temperature of 66°F, magnet cryostats near the air entry point could be covered with insulation if servicing periods were kept short. For high average day/night temperatures, the tunnel could be ventilated at night when the air is coolest, then this tunnel air could be circulated during the day, without fresh air intake (hot). Two days worth of fresh air can be stored in the tunnel if it is assumed that service is occurring at only one point in the tunnel (e.g. a magnet changeout). The tunnel is 280,000 ft long so if air velocity is 1.5 ft/sec, it will take 51 hours for air to circulate around the entire colllider tunnel. Similarly, it will take 5.5 hours to charge the tunnel with fresh air (from each exit/vent shaft), unless higher than normal air velocity is used.

Air temperatures at the current SSC site can range from 112°F to -8°F (which is a +46°, -74° F temperature swing from 66°F). Average daily temperatures (computed as an average of the monthly average maximum and minimum) are above 65°F 6 months of the year, and above 80°F for 3 months, reaching 85°F in July. This would represent the maximum temperature of an insulated cryostat shell (thermal time constant on the order of a day). This information is from the DEIS.

A note of caution: Any scenario which relies on air temperature control to limit cryostat expansion has an inherent risk, in that emergency situations such as a tunnel cave-in, fire, medical emergency, cryogen spill, etc. may require continuous ventilation for long periods of time, and it would be a shame to have realign, or possibly even replace large numbers of magnets and/or magnet stands due to this one-time occurrence. It is therefore suggested that the magnet/magnet stand assembly be designed to be damage resistant, if not re-alignment resistant, to the maximum temperature excursion that may occur. Alternatively, ventilation intakes can be provided with full heat/cool/dehumidification capability with backup or emergency power.

3. <u>Air temperature rise</u>

Air temperature rise occurs from both distributed heat sources such as cables and lights, and from lumped sources such as electronics niches. Shown on the next page is a chart showing various tunnel air heat sources, the estimated percentage of heat which finds its way into the tunnel air and their combined effects on air temperature. Lumped sources are averaged along the tunnel for this analysis. The totals occur only between service areas E and F and between G and H, as can be seen from the 35 kV feeder (main distributed heat source) diagram on page 12. Not all the heat generated from these items will find its way into the tunnel air, as some items such as lights and cables are mounted directly to the ceiling and some heat transfer through direct conduction occurs. There is also a laminar boundary sublayer or film of air next to the tunnel wall which does not mix with the bulk tunnel air, however this layer is only ≈ 3-4 mm thick which is negligible.

MAXIMUM TUNNEL HEAT LOADS

CATEGORY ITEM	UNIT POWER DISS. kW			UNIT SEPAR. meter	POWER DISS.. watt/meter			% HT ->AIR	WATT/METER --> AIR			TUNNEL AIR TEMP. RISE °C		
	AVG	PK	SRVC		AVG	PK	SRVC		AVG	PK	SRVC	AVG	PK	SRVC
TUNNEL TOTAL					25.8	44.3	41.6		24.6	41.3	38.8	1.6	2.8	2.6
TUNNEL DISTRIBUTED TOTAL					12.6	30.4	24.4		11.3	27.3	21.9			
35 KV 3Ø 500MCM CABLE A					10	24.2	10	90	9	21.8	9			
35 KV 3Ø 500 MCM CABLE B					2.5	6	2.5	90	2.25	5.4	2.25			
13.8 KV 3Ø 500MCM CABLE					0.1	0.18	0.18	90	0.09	0.16	0.16			
480V 3Ø 250MCM CABLE (magnet section warmup)					0	0	11.7	90	0	0	10.5			
120/240V CABLE					0	0		90	0	0	0			
TRANSPORTER BUS					0	0		90	0	0	0			
HELIUM LEAD RETURN PIPE					0	0	0	90	0	0	0			
CONDENSATION FROM AIR					0	0	0	100	0	0	0			
EVAPORATION FROM WALL SEEPAGE					0	0	0	100	0	0	0			
TUNNEL LUMPED TOTAL					13.2	13.9	17.2		13.2	13.9	16.9			
LIGHTS			0.1	30.5	0	0	3.26	90	0	0	2.95			
NICHE ELECTRONICS SUBTOTAL	2.54	2.54	2.54	228	11.1	11.1	11.1	100	11.1	11.1	11.1			
ELECTRONICS	2.5	2.5	2.5											
SERVICE TRANS. 480-120/240	0.04	0.04	0.04											
POWER ALCOVE ADDITIONAL SUBTOTAL	2.16	2.9	2.9	1031	2.11	2.82	2.82	100	2.11	2.82	2.82			
TRANS., 13.8kV-480V,500KVA (1.5%)	2.16	2.9	2.9											

3.1. Distributed heat sources

Assuming steady state conditions, and a location several temp decay lengths into the tunnel; any heat flow into the main airstream will cause a rise in air temperature until heat flow (out) through wall convection matches the inflow. This steady state temperature rise of bulk tunnel air is simply:

$$q_{in} = hA(T_{air} - T_{wall}) = h(2\pi r)\Delta T \quad \text{or}$$

$$\left(\frac{q}{l}\right)\cdot\left(\frac{1}{\Delta T}\right) = 2\pi r h$$

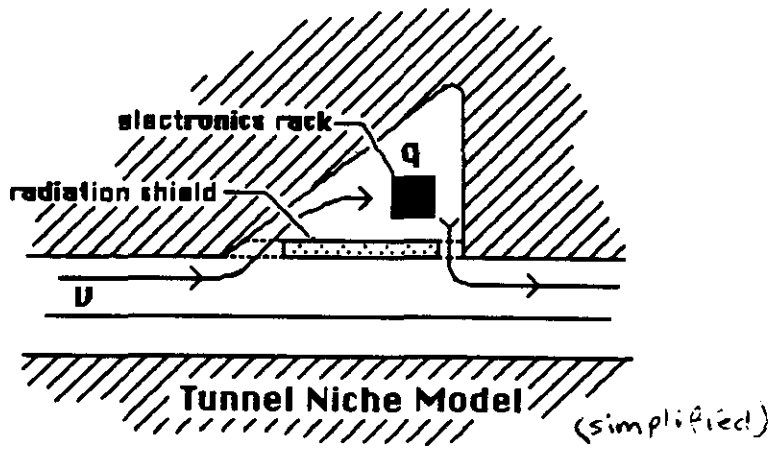
$$= 2\pi(1.5 \text{ meters})(1.6 \frac{\text{Watt}}{\text{m}^\circ\text{C}})$$

$$= 15 \frac{\text{W}}{\text{m}} \text{ heat into tunnel air to produce a } 1^\circ\text{C air temp rise}$$

Heating values for these sources are derived in the Appendix under Tunnel Heat Sources.

3.2. <u>Lumped heat sources</u>

Current thinking is to locate most of these sources in tunnel niches or alcoves, as shown below. These niches occur every cell, (228 meters) and contain electronics and power supplies for correction elements, beam position monitors, radiation detectors, etc. Some niches also contain step-down power transformers and distribution panels. It is shown in the appendix that these niches, if enclosed, are not self-cooling and require active heat removal. We assume here that part of the tunnel air is diverted into the niche or alcove to cool these heat sources and the return flow back into the tunnel is well mixed into the main air stream and does not impinge directly on the cryostat shells. These niche fans might also be used to keep tunnel air circulating during operation, as is done at Fermilab, which is necessary to avoid hot air recirculation back into the niches. Shaft fans are only used for tunnel intake/exhaust. (Note that other methods of cooling these lumped sources may obviate the need to keep tunnel air circulating during operation, at least for heat removal purposes).



The temperature rise of the tunnel air immediately downstream of the niche is then a simple energy balance on a tunnel control volume of air. The formula and an example for a typical electronics niche is:

$$\dot{Q} = q = \rho c_p \frac{d(Vol)}{dt} \Delta T$$

$$\Delta T = \frac{q}{\rho c_p va} = q(kW)\left(\frac{1}{1.005\frac{kJ}{kg}\,1.177\frac{kg}{m^3}\,0.5\frac{m}{s}\,\pi\left(1.5m^2\right)}\right)$$

$$\Delta T\,(^\circ C) = q(kW)(.239) = 2.5(.239) = 0.6\ ^\circ C$$

4. Appendix

Assumptions used are as follows:

1. Tunnel diameter = 3 meters
2. Average air velocity = 0.5 m/sec
3. Tunnel rock or cement wall heat conductivity, k = 3.5 W/m°K
4. Dry tunnel walls, and no condensation from humid air.

All numbered formulas and values (other than those derived here, or otherwise noted) are from J.P. Holman, Heat Transfer, 4th ed.

4.1. Nomenclature

a = cross sectional area of tunnel

β = volume coeff. of expansion (1/T)

α = thermal diffusivity ($k/\rho c$)

c, c_p = heat capacity at constant pressure

Gr = Grashof #

Gr^* = modified Grashof #($Gr_x Nu_x$)

k = thermal conductivity

h = surface heat transfer coefficient (convection

ν = kinematic viscosity

ρ = density

Pr = Prandtl #

r = tunnel cross-section radius

Re = Reynold's #

T_w = Temperature at tunnel wall

q = dQ/dt

Q = heat energy

4.2. Convection heat transfer coefficient, h:

1. Reynolds #, air flowing through tunnel:

$$Re = \frac{dv}{\nu} \quad \text{where} \quad d = \text{tunnel dia.} = 3 \text{ m}$$

$$v = \text{avg. air velocity} = 0.5 \frac{m}{s}$$

$$\nu = \text{air kinematic viscosity} = 16.84 \times 10^{-6} \frac{m^2}{2} \quad \text{(table A-5, Holman)}$$

$$Re = \frac{3(0.5)}{16.84 \times 10^{-6}} = 89{,}000 \quad \text{(indicates fully turbulent flow)}$$

2. Tunnel wall surface heat transfer coefficient, h:

$$Nu = .023\, Re^{.8} Pr^{.4} \qquad \text{(6-4, Holman)}$$

$$Pr = .708\ ; \quad k = .0262 \frac{W}{m°C} \qquad \text{(table A-5, Holman)}$$

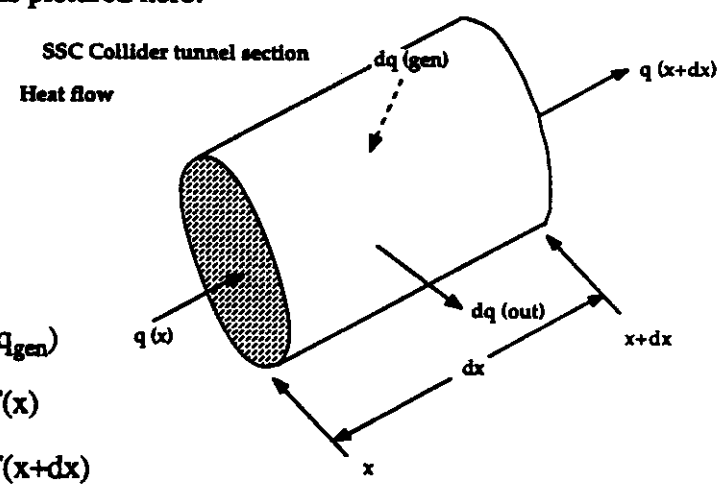
$$Nu = .023\,(89{,}000)^{.8}(.708)^{.4}$$

$$Nu = 182 = \frac{hd}{k}$$

$$h = \frac{k\,Nu}{d} = \frac{.0262 \frac{W}{m\,°C}(182)}{3m} = 1.6 \frac{W}{m^2 °C}$$

4.3. <u>Air -wall ΔT decay length :</u>

Consider a section of tunnel as pictured here:



Heat balance:

$$q(x) = q(x+dx) + (dq_{out} - dq_{gen})$$

$$q(x) = c_p \rho va\, T(x)$$

$$q(x+dx) = c_p \rho va\, T(x+dx)$$

$$dq_{out} = 2\pi rh\,(T_{air} - T_{wall})dx$$

$$dq_{gen} = \left(\frac{q_{gen}}{l}\right)dx; \quad \text{where } \frac{q_{gen}}{l} = \text{heat/unit length produced from cables, etc.}$$

$$c_p \rho va\, T(x) = c_p \rho va\, T(x+dx) + \left[2\pi rh(T_a - T_w) - \frac{q_{gen}}{l}\right]dx$$

$$\text{or } \frac{dT}{dx} = \frac{T(x+dx) - T(x)}{dx} = \frac{2\pi rh(T_a - T_w) - \frac{q_{gen}}{l}}{c_p \rho va}$$

let $\theta = T_a - T_w$; $A = \frac{2\pi rh}{c_p \rho va}$; and $B = \frac{\frac{q_{gen}}{l}}{c_p \rho va}$

then $d\theta = dT_a$

and $\frac{d\theta}{dx} = A\theta + B$

or $\frac{d\theta}{\theta - \frac{B}{A}} = -A\,dx$

integrating both sides gives:

$$\ln\left(q\frac{B}{A}\right) + C_1 = -Ax + C_2$$

exponentiating both sides gives:

$$C_1\left(q\frac{B}{A}\right) = C_2 e^{-Ax}$$

for x = 0, and $\frac{B}{A} = 0$ (no heat generation):

$$\theta_{x=0} = \frac{C_2}{C_1} = T_i - T_w \quad (T_i = \text{initial air temp})$$

or: $\theta - \frac{B}{A} = (T_i - T_w)e^{-Ax}$

$$\therefore\ T_a(x) = T_w + (T_i - T_w)e^{-Ax} + \frac{B}{A}$$

Temp decay length (no heat generation) :

let $\frac{T_a - T_w}{T_i - T_w} = \frac{1}{e}$

then $\frac{1}{e} = e^{-Ax}$

$$\ln\left(\frac{1}{e}\right) = -Ax$$

$$x = A^{-1} = \frac{c_p \rho vr}{2h} = \frac{(1.005)\frac{kJ}{kg^oC}(1.117)\frac{kg}{m^3}(0.5)\frac{m}{s}(1.5)m}{2(.0016)\frac{kW}{m^2\,^oC}} = 263 \text{ meters}$$

Note that this expression appears to be a direct function of air velocity; however , the quantity h is also a nearly direct function of air velocity (v to the 0.8 power); as a result only a small increase in decay length results from a large increase in velocity (Δv to the 0.2 power).

We can make the assumption that both the wall temperature and surface heat transfer coefficient through the wall (h), stay fairly constant over time. The wall temperature assumption is shown to be true elsewhere in the appendix. The heat transfer coefficient is only a weak function of temperature difference, and is best assumed constant for this analysis. If present, humidity or tunnel seepage would have an appreciable effect on wall temperature. The tunnel is expected to be dry, however, and will be lined if water is present (from conversation with Dave Goss). Dehumidification is expected to be provided for condensation control and estimated requirements are calculated in the last section of this appendix.

4.4. Cryostat shell thermal time constant

1. Lumped mass criterion:

$$\frac{h}{k}\left(\frac{V}{A}\right) \leq 0.1 \text{ where:}$$

h = convection heat transfer coeff. $= 1.6 \dfrac{W}{m^2\,°C}$

k = conduction " " " $= 55 \dfrac{W}{m\,°C}$ (low carbon steel)

V = volume, cryostat shell
A = area, cryostat shell
t = shell thickness = .635 cm

for a thin-walled cylinder of radius r, length l and thickness t :

$$\frac{V}{A} = \frac{2\pi rtl}{2\pi rl} = t \quad \text{and} \quad \frac{ht}{k} = \frac{1.6(.00635)}{55} = .00018 \leq .01$$

$\therefore$ use lumped mass analysis (negligible thermal gradient across thickness)

2. Time constant, using lumped mass solution:

$$\tau = \frac{c\rho V}{hA} = \frac{c\rho t}{h} \qquad \text{(4-5; Holman)}$$

where: c = heat capacity, steel $= .465 \dfrac{kJ}{kg\,°C}$

ρ = density, steel $= 7833 \dfrac{kg}{m^3}$

$$\tau = \frac{.465\left(\frac{kJ}{kg\,°C}\right)7833\left(\frac{kg}{m^3}\right).00635\text{ m}}{1.6\left(\frac{\frac{J}{s}}{m^2\,°C}\right).001\left(\frac{kJ}{J}\right)} = 14445 \text{ s} = 4 \text{ hours}$$

3. <u>Support stand thermal resistance (relative to air):</u>
For what length of cryostat is the conduction thermal resistance (to the stand) equal to (10x) the convection thermal resistance (to tunnel air)?

L = magnet half length = 8.2 m
t = cryostat thickness = .25" = .00635 m
d = cryostat dia. = 24" = 0.61 m

$$k = 54 \frac{W}{m\,°C} \quad \text{(table A-2, Holman)}$$

$$h = 2.22 \frac{W}{m^2\,°C} \quad \text{(as computed below)}$$

cryostat convection heat transfer coefficient, h:

$$Nu = C(Gr\,Pr)^m \qquad \text{(7-25, Holman)}$$

$$Gr = \frac{g\beta(\Delta T)\,d^3}{\nu^2} = \frac{9.81\left(\frac{1}{300}\right)10\,(.61)^2}{\left(16.84 \times 10^{-6}\right)^2} = 183 \times 10^6 \qquad \text{(7-21, Holman)}$$

$$Pr = .708 \qquad \text{(table A-5, Holman}$$

$$Gr\,Pr = 183 \times 10^6$$

and C = 0.53; m = .25 (table 7-1, Holman)

$$Nu = 0.53(183\,E6)^{.25} = 62 = \frac{hd}{k}$$

$$k = 0.026 \frac{W}{m\,°C} \quad \text{(table A-5)}$$

$$h = \frac{62(.026)}{.61} = 2.22 \frac{W}{m^2\,°C}$$

therefore:

$$R_k = \frac{\Delta x}{ka} = \Delta x\left(\frac{1}{k\pi dt}\right) = 10\,R_h = 10\frac{1}{ha} = 10\frac{1}{h\pi dL}$$

$$\text{or} \quad \Delta x = 10\frac{kt}{hL} = \frac{54(.00635)}{2.22(8.2)} = .019 \text{ m}$$

Thus even for a very short cryostat connected to a thermal ground (if the stand could even be assumed as one), the thermal resistance to air is still an order of magnitude less, therefore cryostat shell temperature comes to equilibrium with the air only.

4.5. Tunnel wall temperature

Temperature response chart

Shown below is a chart showing surface temperature change as a function of time for a cylindrical cavity in an infinite solid (Heisler, Temperature Response Charts).

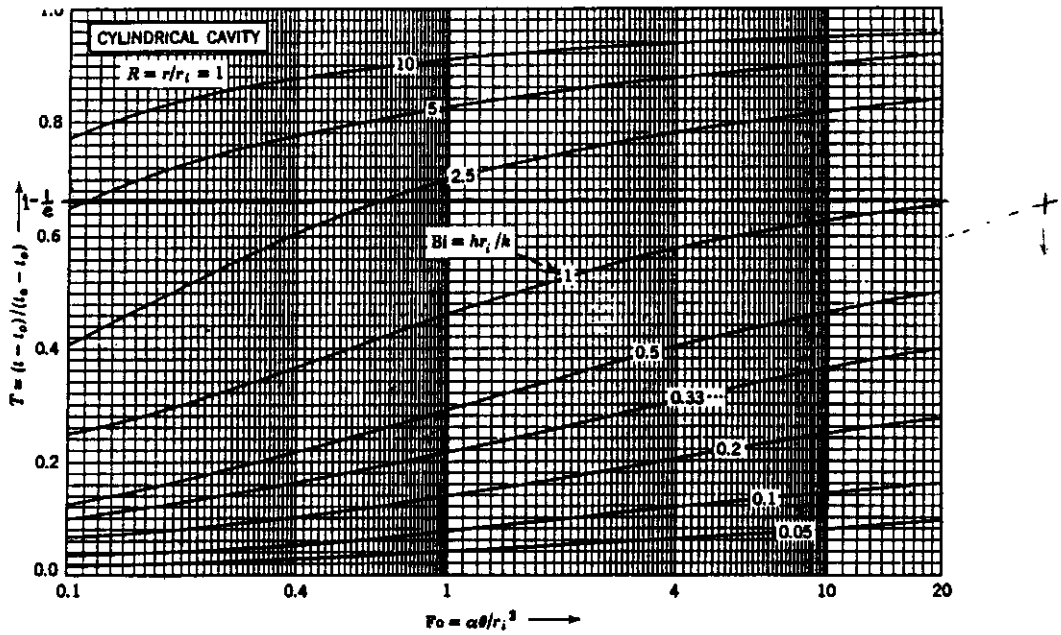


Chart 34. Temperature response of an infinite solid surrounding a cylindrical cavity, $r \geq r_i$, after sudden exposure to a uniform-temperature convective environment t_a at $r = r_i$, (E, E').

$$Bi = \frac{1.6\frac{W}{m^2\,°C}\,1.5\text{ m}}{3.5\frac{W}{m\,°C}} = 0.69$$

Fo = Fourier # > 20 (from graph)

$$\theta = \text{time} = \frac{Fo\, r_i^2}{\alpha} > \frac{20\,(1.5\text{m})^2}{1.4 \times 10^{-6}\frac{m^2}{s}} = 3.2 \times 10^7 \text{ sec} = 372 \text{ days}$$

4.6. Niche heating

enclosed niche; convection cooling to niche wall; fan cooling

Assume an enclosed cubic niche 3x3x3 meters, with sufficient fans on the electronics to thoroughly mix the heat and air. Assume further that the walls remain at constant temperature (65°F). The air temperature rise at steady state conditions is:

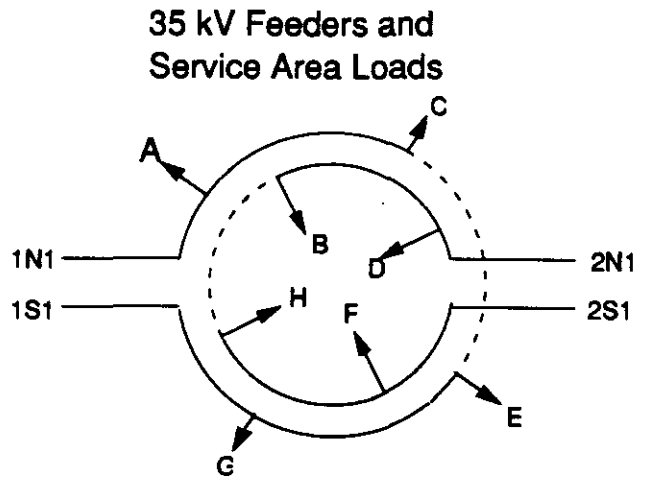
$$q = ha\Delta T \text{ or } \Delta T = \frac{q}{hA} = \frac{2500\ \text{W}}{2\frac{\text{W}}{\text{m}^2\,°\text{C}}\ 6(3\text{x}3)\ \text{m}^2} = 23°\ \text{C} = 41.6°\ \text{F}$$

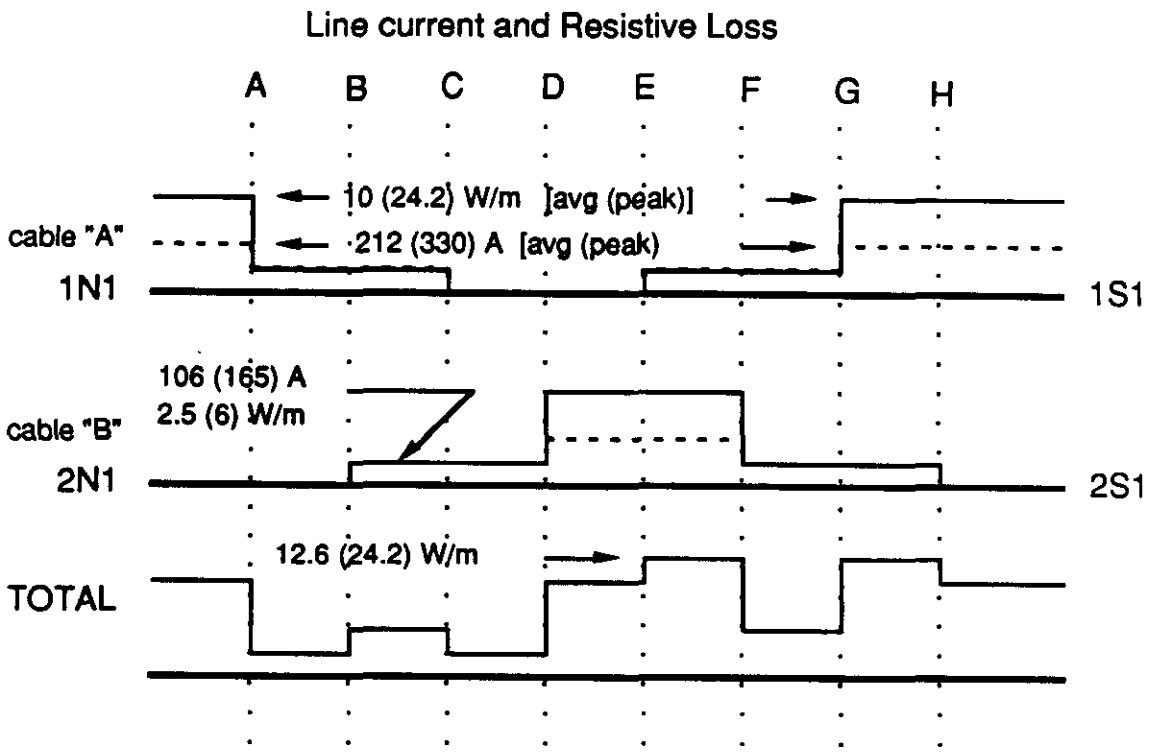
The air temperature is thus at least 65+41 = 106 °F and possibly higher as wall temperature will slowly rise over time. In addition, the electronics themselves run appreciably hotter than the cooling air. It is expected that these temperatures are too high (for reliability) and that the niche heat must be actively removed.

4.7. <u>Tunnel heat sources</u>

Cable current and resistive losses (I^2R) are shown as a function of tunnel location in the graph below, along with averaged niche and other tunnel losses. Since air will be kept circulating through the tunnel at ≈0.5 meter/sec, point heat sources (niche electronics racks, transformers) and sinks (helium lead flow pipes under certain conditions) will average out in a relatively short distance and can probably be treated as one continuous heat source/sink.

1.





2. Cable heating

These 35 kV 500 MCM triplex cables are the dominant cable heating source in the tunnel. Full Current heating values for each cable (in the near cluster tunnel sector) turn out to be 10.0 Watt/meter for average loads, and 24.2 Watt/meter during peak power demand. Even so, this represents negligible air temperature rise. Of more interest is the temperature of the cable jacket, for safety reasons during servicing.

2.1. Power usage per sector (from SSC-N-282, Conventional Systems Parameters)

P = 5.7 MW (avg) ; 8.91 MW (peak)

line current per sector: I = 106 A (avg) ; 165 A (peak)

at sectors F2,F3,F7, and F8: I = 212 A (avg) ; 330 A (peak)

2.2. Delivered power ; line current:

$$P = \sqrt{3}\, EI \cos\phi$$

E = load voltage

I = line current

$\cos\phi$ = power factor

or

$$I = \frac{P}{\sqrt{3}\, E \cos\phi}$$

2.3. Cable resistance:

$$R\ (500\text{MCM conductor, } 20\,^\circ\text{C}) = \rho\left(\frac{L}{A}\right) = \frac{.66\text{x}10^{-6}\frac{\Omega}{\text{in}}(39.37)\frac{\text{m}}{\text{in}}}{\left(500 \text{ x } 10^3\right)\frac{\pi}{4}(.001")^2} = 66.2\,\frac{\mu\Omega}{\text{m}}$$

temperature correction to 50 °C :

$$R = R_0(\alpha(\Delta T)+1) = 66.2\text{E-}6(.0039(30)+1) = 1.117(66.2\text{E-}6) = 74\,\frac{\mu\Omega}{\text{m}}$$

2.4. Feeder heating

$$q = 3(I^2R)$$

for cable "A", near cluster, avg operating cond.= $3(212)^2 74\text{E-}6 = 10.0\,\frac{\text{W}}{\text{m}}$

cable "A", near cluster, peak operating = $3(330)^2 74\text{E-}6 = 24.2\,\frac{\text{W}}{\text{m}}$

3. <u>Cable surface temperature</u>

$$q = ha\Delta T \text{ or } \Delta T = \frac{q}{h(\pi d)}$$

$$d = \text{cable dia.} = 4.3'' = .11\text{m}$$

$$q = 23.3\,\frac{W}{m}$$

to find h, first assume $\Delta T = 10\,°C$; then:

$$Nu = C(Gr\,Pr)^m \qquad \text{(7-25, Holman)}$$

$$Gr = \frac{g\beta(\Delta T)\,d^3}{\nu^2} = \frac{9.81\left(\frac{1}{300}\right)10\,(.11)^2}{\left(16.84 \times 10^{-6}\right)^2} = 1.4 \times 10^7 \qquad \text{(7-21, Holman)}$$

$$Pr = .708 \qquad \text{(table A-5, Holman}$$

$$Gr\,Pr = 1.0 \times 10^7$$

and $C = 0.53$; $m = .25$ (table 7-1, Holman)

$$Nu = 0.53(1.0E7)^{.25} = 30 = \frac{hd}{k}$$

$$k = 0.027\,\frac{W}{m\,°C} \qquad \text{(table A-5)}$$

$$h = \frac{30(.027)}{.11} = 7.4\,\frac{W}{m^2\,°C}$$

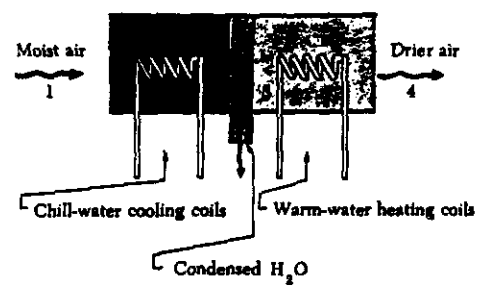
$$\therefore\ \Delta T = \frac{q}{ha} = \frac{23.3}{7.4\,\pi(.11)} = 9.2\,°C \ \ (\Delta T \text{ assumption correct})$$

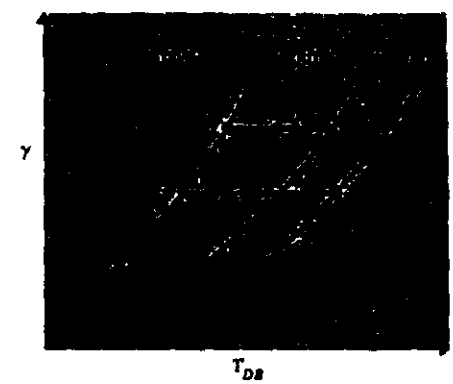
This represents exterior temperature rise for a cable mounted horizontally in open still air. For a ceiling or side mounted cable, less convection, but more conduction heat loss will occur and the temperature is estimated not to differ substantially.

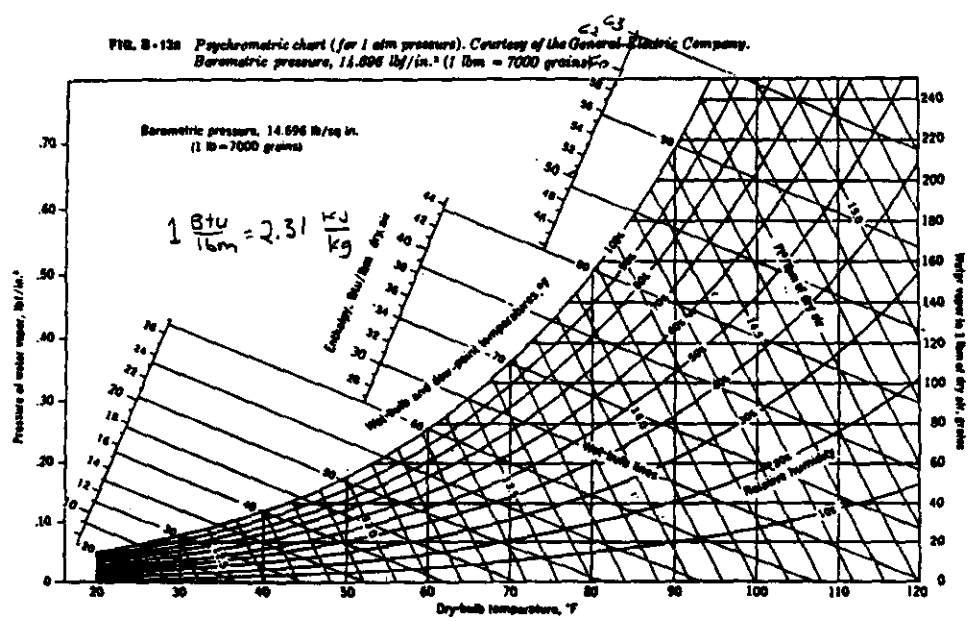
4.8. Dehumidification requirements

The DEIS states that average monthly maximum dewpoint temperatures are 66-67 °F for July and August. However it is not uncommon for higher dewpoints to occur, and Dave Goss has estimated 105 °F (70 % Relative Humidity) as a likely maximum. This is equivalent to a 95 °F dewpoint. 80% RH (65 °F) is probably the highest humidity that the electronics and cables in the tunnel should see (dewpoint = 59 °F). Power and heating needs are calculated here, using a typical refrigeration cycle of cooling the air to condense out the water, then reheating slightly to lower the RH. Swamp coolers won't work due to high RH. Dessicant systems may be an alternative.

Process and psychrometric chart







$$\text{air mass flow rate, } \dot{m} = \rho v a = 1.177\frac{kg}{m^3}(0.5)\frac{m}{s}\pi\,(1.5m)^2 = 4.16\frac{kg}{s}$$

Outside air 85 °F; 60% RH to tunnel air at 65 °F; 80% RH

$$h^*_1 = 35\,\frac{\text{Btu}}{\text{lbm}}\ ;\quad h^*_3 = 26\,\frac{\text{Btu}}{\text{lbm}}\quad \text{(from fig B-13a, Holman)}$$

$$h^*_1 - h^*_3 = 25.5 - 37.5 = -11\,\frac{\text{Btu}}{\text{lbm}} = -25.4\,\frac{\text{kJ}}{\text{kg}}$$

$$25.4(4.16) = 106\text{ kW} = 22.3\text{ tons refrig.}$$

$$\text{mech. power req.} \approx 1\,\frac{\text{hp}}{\text{ton}}\quad \text{and electr. power req.} \approx 1\,\frac{\text{kW}}{\text{hp}}$$

$$\therefore\ \text{electr. pwr. req.} = 22\text{ kW}$$

$$h^*_4 = 27\,\frac{\text{Btu}}{\text{lbm}}\ ;\ h^*_4 - h^*_3 = 1.5\,\frac{\text{Btu}}{\text{lbm}} = 3.5\,\frac{\text{kJ}}{\text{kg}} = 14\text{ kW reheat}$$

Outside air 85 °F; 60% RH to tunnel air at 65 °F; 60% RH

$$h^*_1 = 35\,\frac{\text{Btu}}{\text{lbm}};\quad h^*_3 = 21\,\frac{\text{Btu}}{\text{lbm}}$$

$$h^*_1 - h^*_3 = 21 - 37.5 = -15.5\,\frac{\text{Btu}}{\text{lbm}} = -36\,\frac{\text{kJ}}{\text{kg}}$$

$$36(4.16) = 149\text{ kW} = 31.5\text{ tons refrig.} = 31.5\text{ kW power req'd.}$$

$$h^*_4 - h^*_3 = 25 - 21 = 4\,\frac{\text{Btu}}{\text{lbm}} = 9.2\,\frac{\text{kJ}}{\text{kg}} = 38\text{ kW reheat}$$

Outside air 95 °F; 60% RH to tunnel air at 65 °F; 80% RH

$$h^*_1 = 46\,\frac{\text{Btu}}{\text{lbm}};\quad h^*_3 = 26\,\frac{\text{Btu}}{\text{lbm}}$$

$$h^*_1 - h^*_3 = 25.5 - 46 = -20.5\,\frac{\text{Btu}}{\text{lbm}} = -47\,\frac{\text{kJ}}{\text{kg}}$$

$$47(4.16) = 197\text{ kW} = 42\text{ tons refrig.} = 42\text{ kW power req'd}$$

Outside air 105 °F; 70% RH to tunnel air at 65 °F; 80% RH

$$h^*_1 = 63\,\frac{\text{Btu}}{\text{lbm}}\quad h^*_3 = 26\,\frac{\text{Btu}}{\text{lbm}}$$

$$h^*_1 - h^*_3 = 25.5 - 63 = -37.5\,\frac{\text{Btu}}{\text{lbm}} = -87\,\frac{\text{kJ}}{\text{kg}}$$

$$87(4.16) = 362\text{ kW} = 76\text{ tons refrig.} = 76\text{ kW power req'd}$$