UNIVERSITY OF CALIFORNIA
University of California Observatories/Lick Observatory
UCO/Lick Observatory Technical Reports
No. 51

(Also Keck Tech Note #290)

KECK TELESCOPE
HIGH RESOLUTION SPECTROGRAPH
THERMAL MODELS

JACK OSBORNE
BRUCE BIGELOW

Santa Cruz, California
October 1989

Revised March 1990

UNIVERSITY OF CALIFORNIA

Lick Observatory Technical Reports

No. 51

(Also Keck Tech Note #290)

Keck Telescope High Resolution Spectrograph Thermal Models

Jack Osborne

Bruce Bigelow

Santa Cruz, California

October 1989

Revised March 1990

HIRES

LICK
OBSERVATORY

Abstract

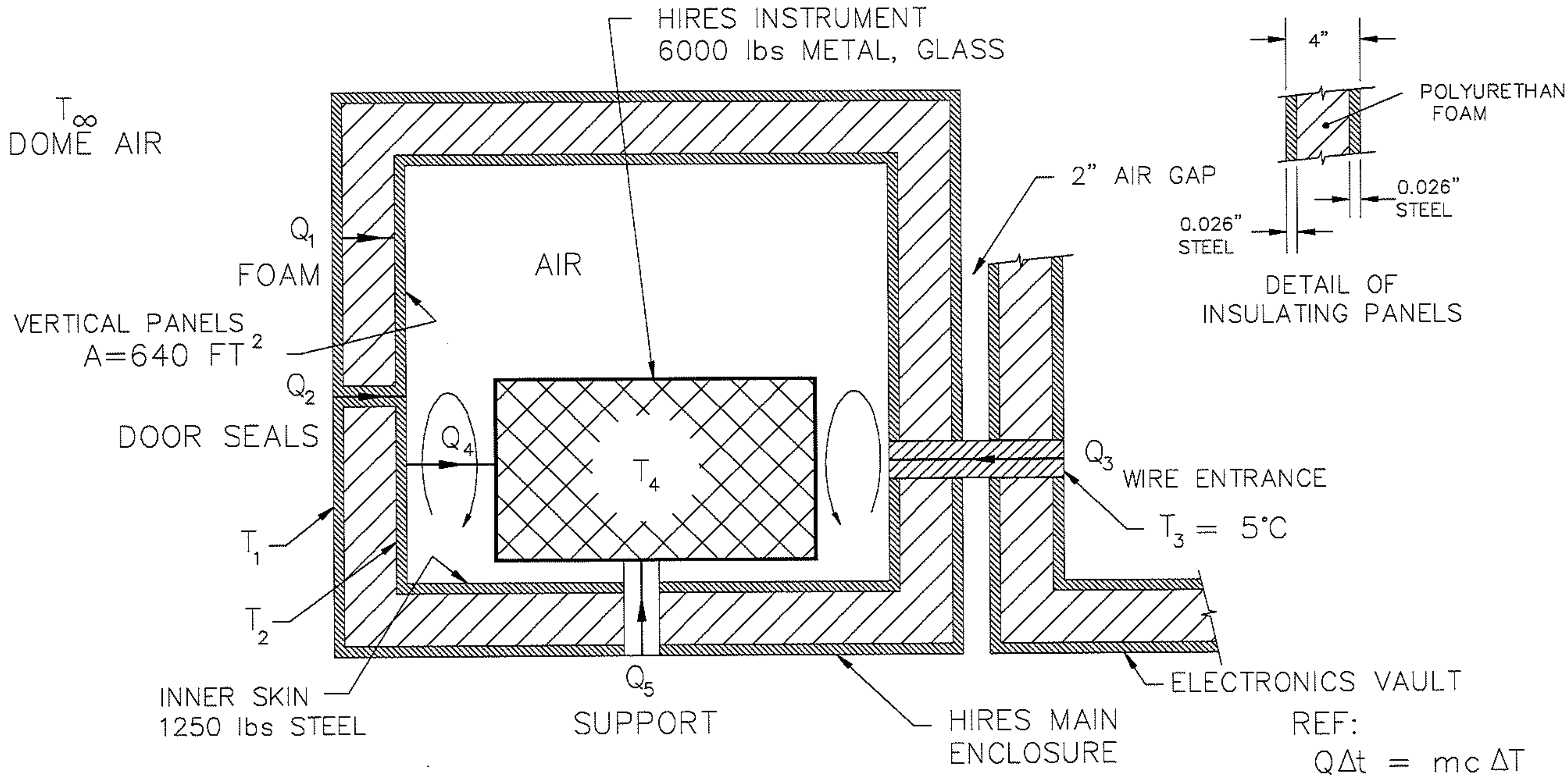
We present 4 cases:

I. A sealed housing around the HIRES instrument
II. A "bubble" of dome air entering the sealed housing
III. Transient heat source:
   1. An astronomer for 1 hour
   2. All motors running during a typical setup.
IV. No enclosure at all.

We conclude that Case I is an acceptable design, that Case II is a tolerable condition, that Case III should be minimized or avoided and Case IV is un-acceptable.

For Case I, the time constant for an increase of 1°C to the dome air temperature (instantaneous) is 9 hours.

We list the assumptions made and include a scale drawing of the actual housing.



REF:

$$Q\Delta t = mc\,\Delta T$$

and, $$Q = \frac{kA\Delta T}{\Delta x}$$

$Q_1 =$ 65Btu/hr
$Q_2 =$ 54Btu/hr
$Q_3 =$ 13Btu/hr
$Q_4 =$ 132Btu/hr△
$Q_5 =$ 8Btu/hr

CASE I, SEALED HOUSING

1. $T_\infty = T_1 = T_2 = T_4 = T_5 = 0°C$
2. T_∞ CHANGES TO 1°C

...THEN T_4 CHANGES TO 1°C IN 9.5 HOURS

KECK/HIRES
HOUSING THERMAL MODEL
CASE I: SEALED
j.c. 10 11 89
H6402.B

CASE I:

1. $T_\infty = T_1 = T_2 = T_4 = T_5 = 0°C$
2. T_∞ AND T_1 CHANGE TO 1°C VERY QUICKLY.

HEAT FLOW RATES:

1. POLYURETHANE FOAM:

$Q_1 = (T_2 - T_1) k_1 A_1 / \Delta x_1$

$k_1 = 0.01$ Btu/hr–ft–°F
$\Delta T = 1.8°F$
$A_1 = 1200$ ft^2
$\Delta x_1 = 4''$ (0.33ft)

$Q_1 = 65$ Btu/hr x 0.29 = 19 Watts

2. STEEL DOOR FRAMING:

$Q_2 = n(T_2 - T_1) k_2 A_2 / \Delta x_2$

$k_2 = 31$ Btu/hr–ft–°F
$\Delta T = 1.8°F$
$A_2 = 0.04$ ft^2
$\Delta x_2 = 6''$ (0.5ft)
$n = 8$ joints

$Q_2 = 54$ Btu/hr x 0.29 = 16 Watts

3. COPPER WIRES:

$Q_3 = n(5° - T_1) k_3 A_3 / \Delta x_3$

$k_3 = 223$ Btu/hr–ft–°F
$\Delta T = 9°F$
$\Delta x_3 = 10''$
$A_3 = 0.00008$ in^2
$n = 9457$ WIRES (30 ga)

$Q_3 = 13$ Btu/hr x 0.29 = 4 Watts

4. AIR INSIDE ENCLOSURE:

$Q_4 = h\ A\ (T_2 - T_1)$

$\Delta T = 1.8°F$
$A = 640$ ft^2
$h = 0.23$ Btu/hr–ft^2–°F

$Q_4 = 262$ Btu/hr (Greater than $Q_1 + Q_2 + Q_3$ so set $= Q_1 + Q_2 + Q_3$)

$Q_4 = 132$ Btu/hr x 0.29 = 38 Watts

H6407.B 3–19–90 j.o.

5. SUPPORT ELEMENTS:

$Q_5 = (T_4 - T_1)k_5 A_5/\Delta x_5$

$k_5 = 31$ Btu/hr–ft–°F

$\Delta T = 1.8°F$

$A_5 = 0.07\ ft^2$ (2"ø x 3)

$\Delta x_5 = 0.5$ ft

$Q_5 = 8$ Btu/hr x 0.29 = 2.3 Watts

HEAT FLOW TIMES:

1. OUTER–TO–INNER WALL:

$Q_T = Q_1 + Q_2 + Q_3 = 132$ Btu/hr = 39 Watts

ASSUME Q_T CONSTANT.

USE $Q_T \Delta t = m_2 c_2 \Delta T$ WHERE Δt = time

$$\Delta t = \frac{m_2 c_2 \Delta T}{Q_T}$$

$m_2 = 1250$ lbs

$c_2 = 0.1$ Btu/lb–°F, STEEL

$\Delta T = 1.8°F$

$$\Delta t = \frac{(1250)(0.1)(1.8)}{(132)} = 1.7 \text{ HOURS}$$

2. HEAT FLOW TO INSTRUMENT:

$Q_T = Q_4 + Q_5 = 140$ Btu/hr = 41 Watts

ASSUME Q_T CONSTANT.

$$\Delta t = \frac{m_4 c_4 \Delta T}{Q_T}$$

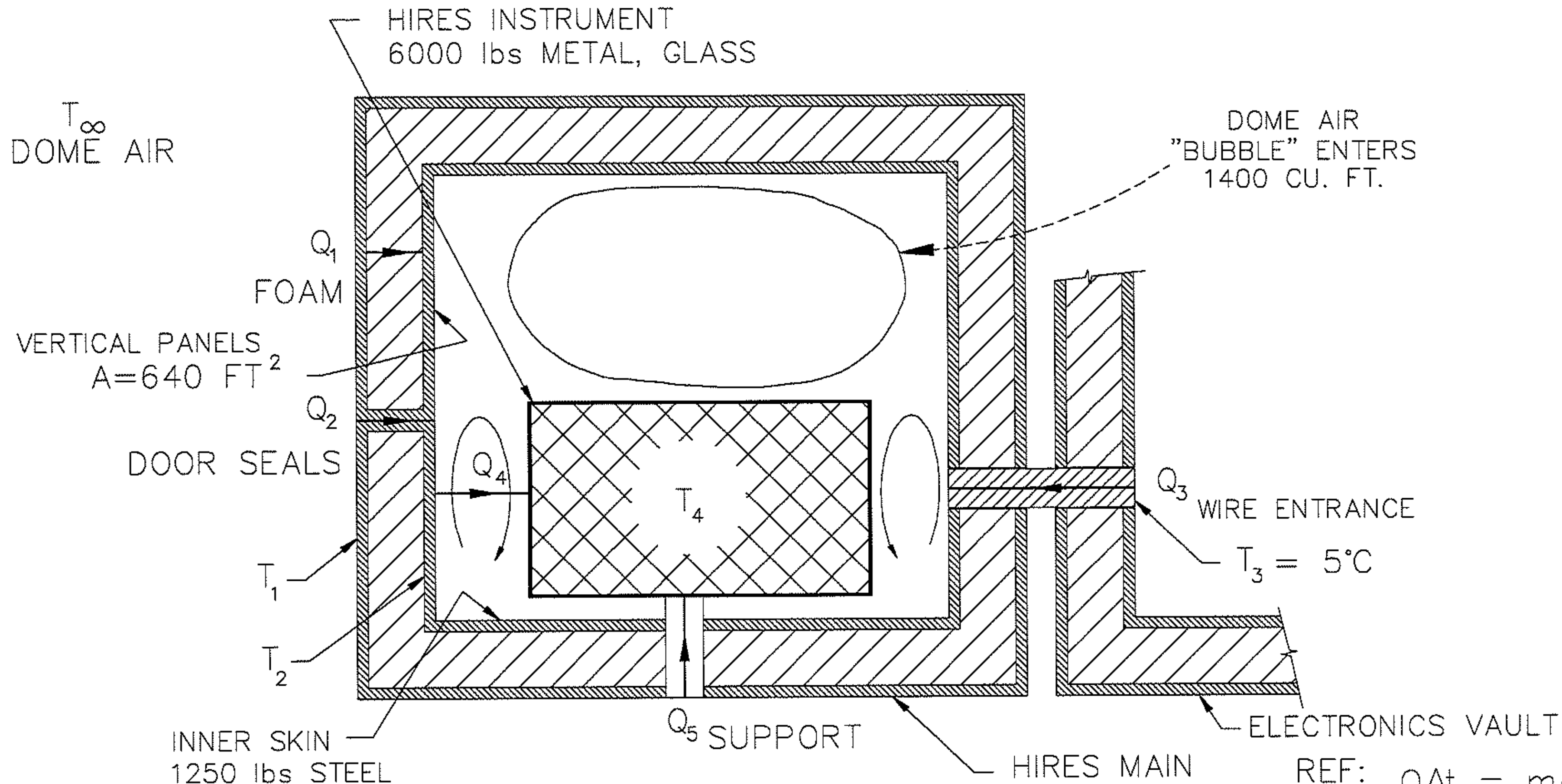
$m_4 = 6000$ lbs

$c_4 = 0.1$ Btu/lb–°F, STEEL

$\Delta T = 1.8°F$

$$\Delta t = \frac{(6000)(0.1)(1.8)}{(140)} = 7.7 \text{ HOURS}$$

$\Delta t_T = 1.7 + 7.7 = 9.5$ HOURS



REF: $Q\Delta t = mc\,\Delta T$

$$Q = \frac{kA\Delta T}{\Delta x}$$

and, $(mc\Delta T)_{air} = (mc\Delta T)_{inst}$

CASE II, AIR "BUBBLE"

1. $T_\infty = T_1 = T_2 = T_4 = T_5 = 0°C$
2. THEN T_∞ CHANGES TO 1°C, AND HALF THE VOLUME OF AIR INSIDE THE ENCLOSURE IS HEATED 1°C.

...T_4 CHANGES TO 0.04°C

KECK/HIRES
HOUSING THERMAL MODEL
CASE II: AIR "BUBBLE"
j.o. 10 12 88
H6403.B

CASE II:

1. $T_\infty = T_1 = T_2 = T_4 = T_5 = 0°C$
2. A "bubble" of dome air is admitted from the dome. The "bubble" is half the volume of the 'HIRES' enclosure. The "bubble" is 1°C warmer.

Assume that the bubble is mixed immediately and the 6000 pound instrument comes to equilibrium with the new air.

$$\Delta T = \frac{(m\ c)_{air}\,\Delta T_{air}}{(m\ \bar{c})_{instrument}}$$

$m_{air} = 1400$ cu.ft. x 0.07 lb/cu.ft

$m_{air} = 100$ lbs

$c_{air} = 0.24$ Btu/lb-°F

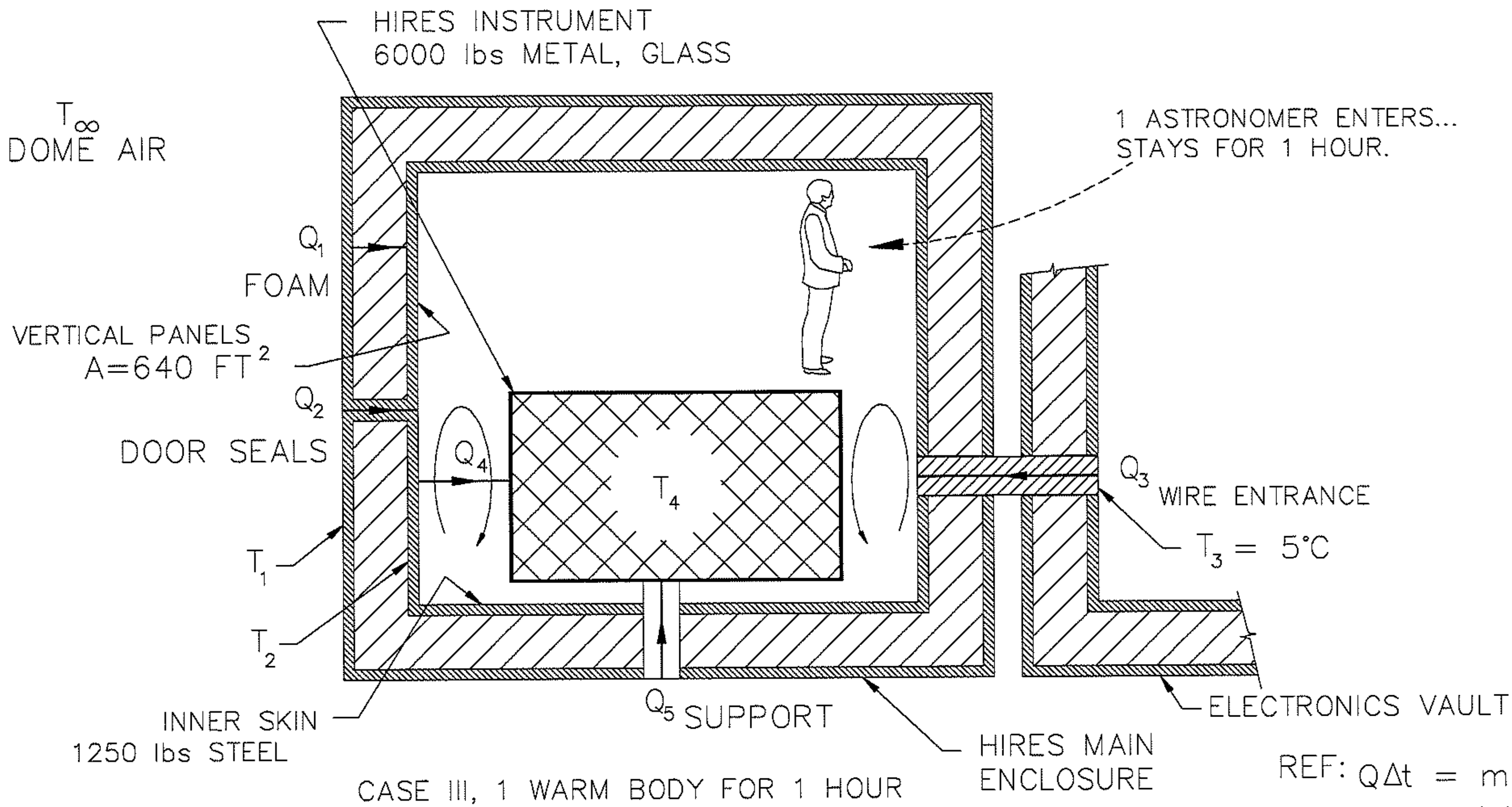
$\Delta T_{air} = 1.8°F$

$m_{instrument} = 6000$ lbs

$\bar{c}_{instrument} = 0.1$ Btu/lb-°F

$\Delta T = 0.07°F\ (0.04°C)$

This is tolerable for any time period.



CASE III, 1 WARM BODY FOR 1 HOUR

1. $T_\infty = T_1 = T_2 = T_4 = T_5 = 0°C$
2. THEN, A HEAT SOURCE IS APPLIED TO THE INSTRUMENT: 80 WATTS X 1 HOUR.

...T_4 CHANGES TO 0.25°C

REF: $Q\Delta t = mc\,\Delta T$

$$Q = \frac{kA\,\Delta T}{\Delta x}$$

NOTE–
A TYPICAL ASTRONOMER IS ASSUMED TO GENERATE 80 WATTS (273 Btu/hr)

KECK/HIRES
HOUSING THERMAL MODEL
CASE III

j.c. 10 13 88

H6405.B

CASE III

1. One human enters...

Similar to Case II, but instead of a bubble of warm dome air, a heat source is applied for a period of time. An astronomer is estimated to produce 80 watts*. For this case we assume a 1 hour time period. We assume this heat all goes into the instrument. (80 watt-hours)

$$\Delta T = \frac{Q \, \Delta t}{mc}$$

$$\Delta T = \frac{273 \text{ Btu/hr} \times 1 \text{ hr}}{6000 \text{ lbs} \times 0.1 \text{ Btu/lb-}^{\circ}\text{F}}$$

$$\Delta T = 0.46^{\circ}\text{F}$$

$$\text{or} \quad \Delta T = 0.25^{\circ}\text{C}$$

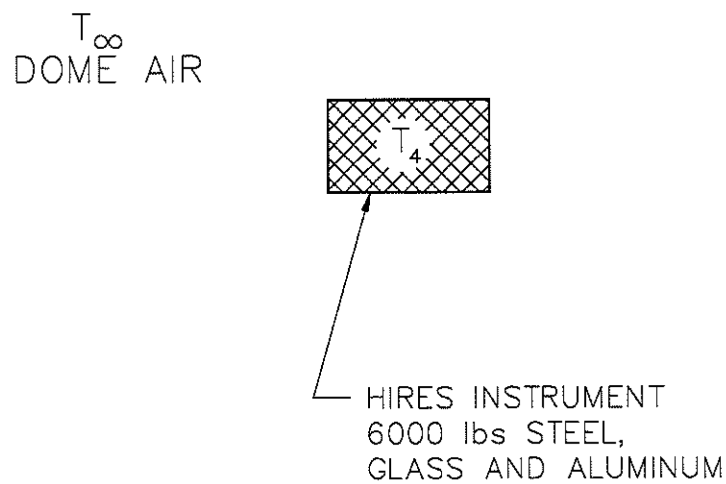
The conclusion here is that this is worse than a "bubble" and should be minimized. In fact, this effect will probably be accompanied by 2 "bubbles", once upon entering and once upon exiting the housing. The time constant will be on the order of 2 hours.

2. All motors run during setup...

A servo motor is 28 volts x 0.4 amps x 10 sec This is 0.03 watt-hours. All 13 servo motors running to initialize the instrument produce 0.4 watt-hours. This will raise the temperature 0.001°C, which is tolerable.

---

* Attributed to George Herbig as a rule of thumb. 80 watts is 273 Btu/hr.



CASE IV, NO ENCLOSURE

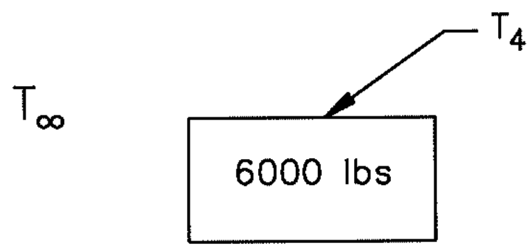
1. $T_\infty = T_4 = 0^\circ C$
2. THEN, $T_\infty = 1^\circ C$.

... LATER, T_4 IS ALSO $1^\circ C$.

THIS HAPPENS ON THE ORDER OF AN HOUR AND IS UNACCEPTABLE.

KECK/HIRES
HOUSING THERMAL MODEL
CASE IV
j.a.
H6404.A

CASE IV



We feel that this case has a short time constant; on the order of a few hours. If T_∞ changes 1°C, the instrument will de-focus badly.

THERMAL MODELLING–ASSUMPTIONS

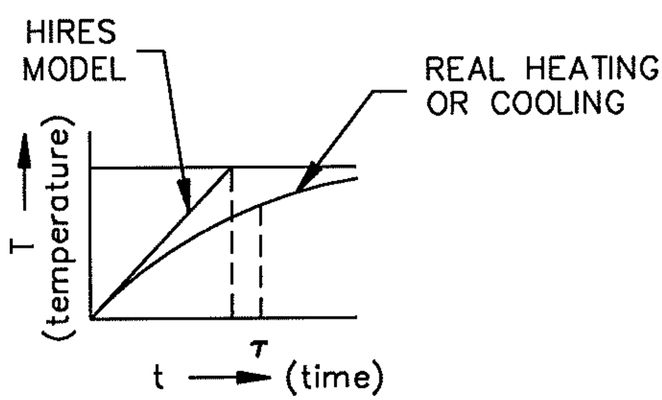
1. All calculations assume iso–thermal instrument conditions, that is, as the 6000 lb instrument is heating or cooling, it is all at the same temperature. This is not a bad assumption since the conductivity is large compared with the small temperature differences involved.

2. Convection in air is the dominant heat transfer mode. See Appendix 3 for details of the calculations. B

3. The real formula for T(t) during conductive heat transfer is:

$$\frac{T-T_{\infty}}{T_O-T_{\infty}}=e^{-\tau t}$$

We have assumed a constant ΔT and so have a linear value of T(t). This is easier to use in the models and gives shorter time periods for heating or cooling.



The time constant, τ, is the time it takes for ΔT to get to be about 1/3 of the original ΔT.

4. We have assumed that convection heat transfer is driven by 4 vertical walls and their height is 10 feet. The total area is 640 sq. ft.

5. The part of HIRES which is affected most by a temperature change is the Schmidt camera detector focus. The camera mirror has a focal length of 30" and this will change by the coefficient of expansion of Pyrex. The structure holding the mirror and detector is made of steel elements and will thus change a different amount since the coefficient is different. The relative change is then $\Delta L = L\Delta T\Delta\alpha$

where $\Delta\alpha = \alpha_{steel} - \alpha_{Pyrex}$

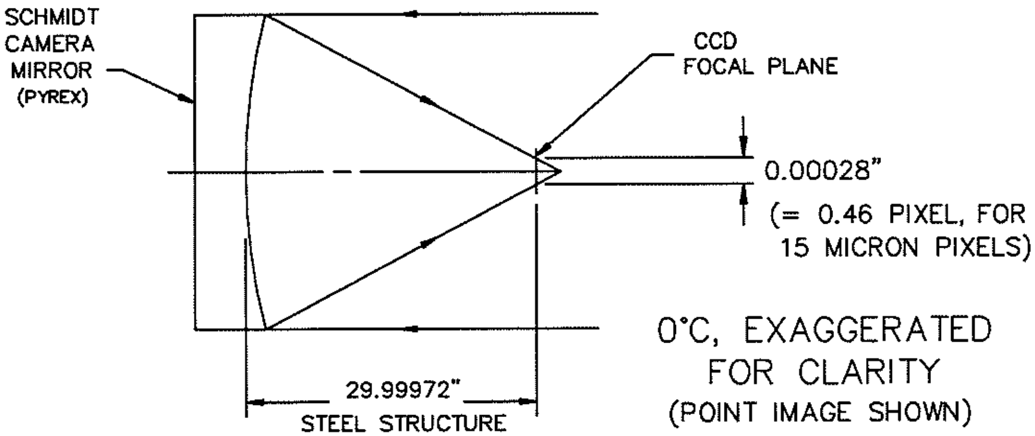
$$\alpha_{steel} = 6.5 \times 10^{-6} \text{ in/in-°F}$$

$$\alpha_{Pyrex} = 1.4 \times 10^{-6} \text{ in/in-°F}$$

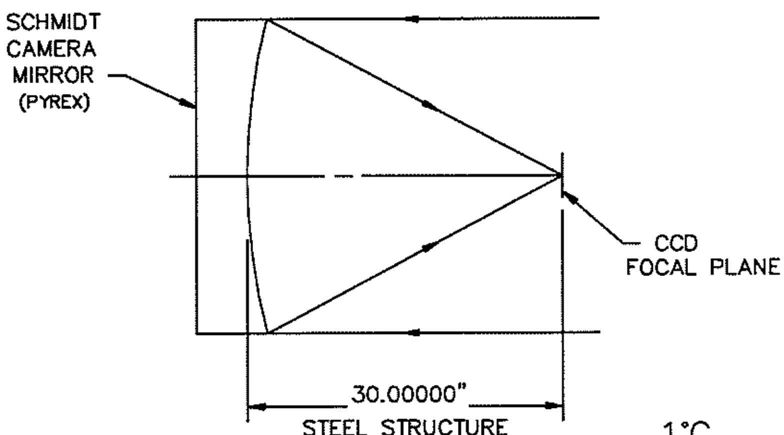
and $\Delta\alpha = 5.1 \times 10^{-6}$ in/in-°F

or $\Delta\alpha = 9.2 \times 10^{-6}$ in/in-°C

We have assumed that detectors with 15 micron pixels will be used. The Schmidt camera is approximately F/1.
A 1°C temperature change will produce a de-focus of 0.00028" (0.46 pixel). See accompanying drawings. Occurring instantaneously, this would be noticeable.
Our Case I model says that this de-focus amount will take 9.5 hours. The de-focus rate would be about 0.05 pixels/hour. This is tolerable.
We assumed that relative motion between slit and collimator would cause a smaller effect than the Schmidt camera focus change.



0°C, EXAGGERATED
FOR CLARITY
(POINT IMAGE SHOWN)



1°C
(POINT IMAGE SHOWN)

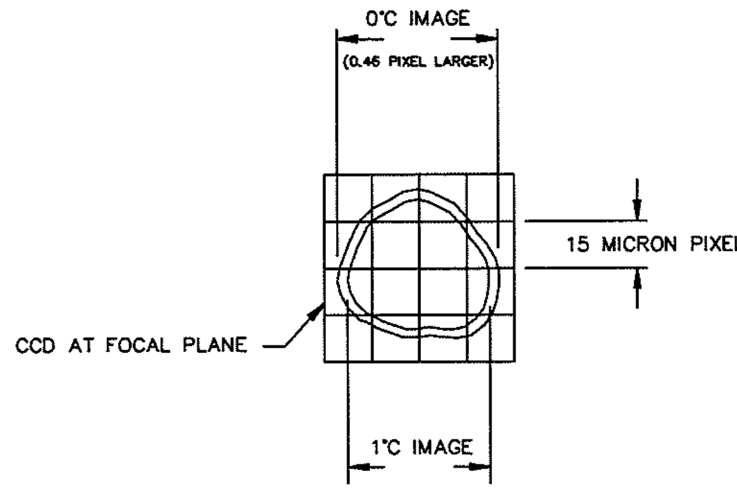


IMAGE DE-FOCUS
TYPICAL 3 PIXEL IMAGE
1°C CHANGE = 0.46 PIXEL CHANGE

REF:

$L = L_o\{1 - (\alpha_2-\alpha_1)(T-T_o)\}$

$\alpha_1 = 1.4 \times 10^{-6}$ in/in−°F

$\alpha_2 = 6.5 \times 10^{-6}$ in/in−°F

$L_o = 30"$

$L = 29.99972"$

THERMAL EFFECTS:

1. TEMPERATURE CHANGE RESULTS IN DE-FOCUS OF THE CAMERA.
2. NO "SEEING" CHANGE SINCE ΔT IS SLOW (< 1°C/5 HOURS)

KECK/HIRES
THERMAL ANALYSIS
DE-FOCUS

H6406.A

6. HIRES testing at UCSC:

We will be running HIRES in its enclosure for about 1 year prior to shipping to Hawaii. The temperature excursions in Santa Cruz will be much larger (e.g. 25°C in one day) and so we will see faster de-focus rates. T_∞ will be about 20°C.

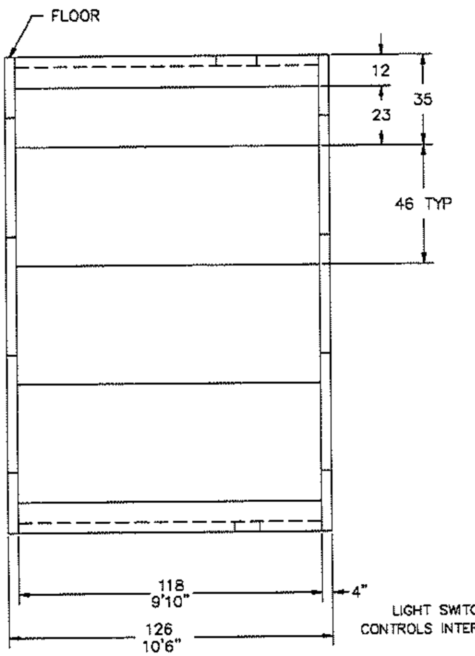
Q_1 could be 1625 Btu/hr (477 Watts)

Q_2 could be 1350 Btu/hr (396 Watts)

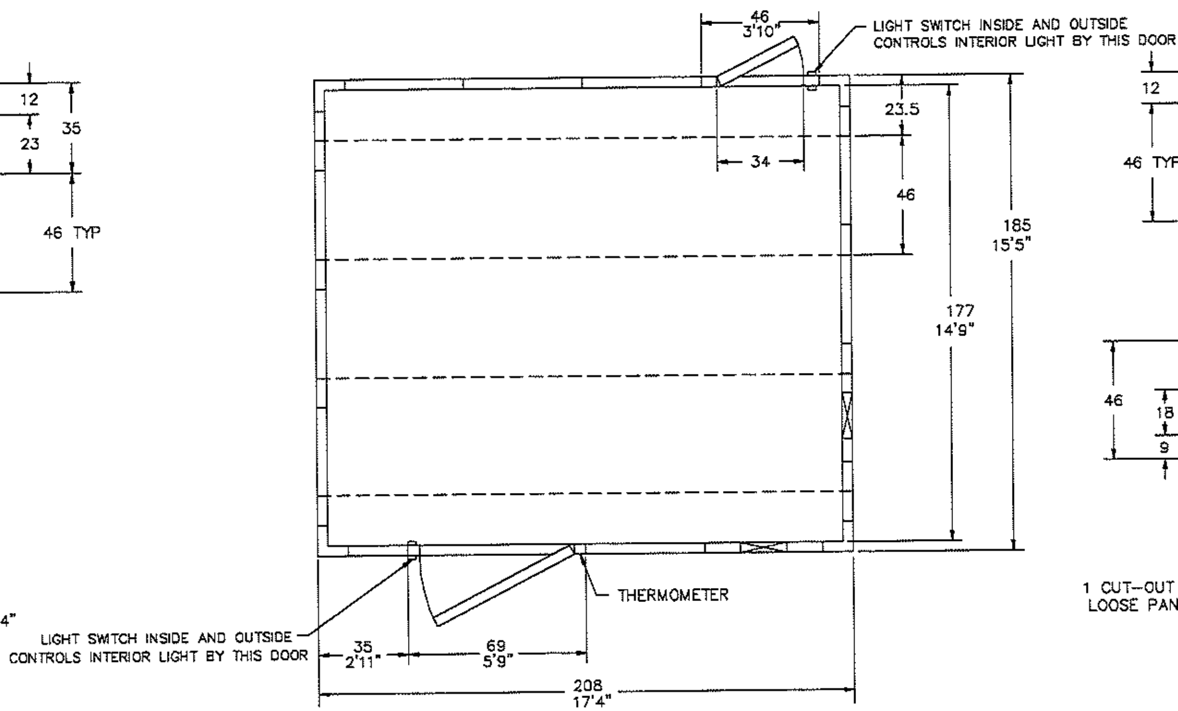
Q_3 will be 0

and Q_4 could be 3000 Btu/hr (880 Watts)

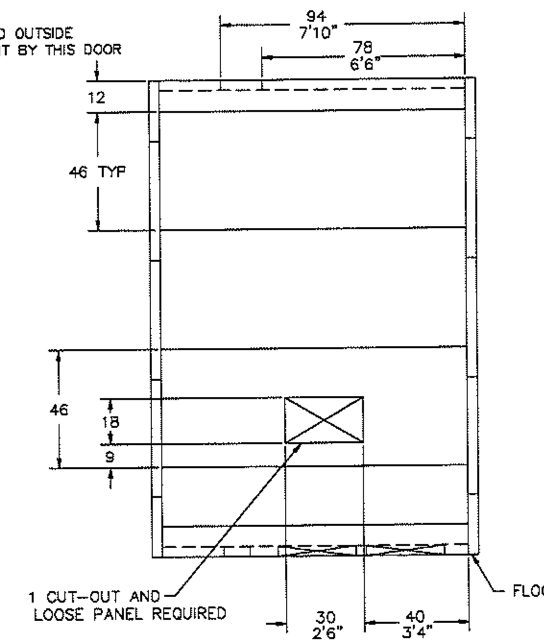
The time constant for the inner skin would be 1.64 hours (to increase 25°C) and the time constant for the instrument would be 9 hours (to increase 25°C) and the de-focus rate would become about 1 pixel/hour.
We will study the behavior of our thermal model while the entire spectrograph system is being tested at UCSC.



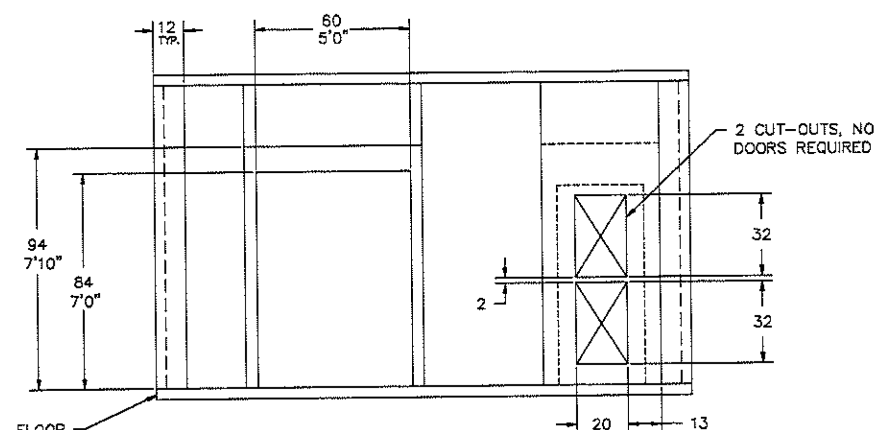
FRONT VIEW



PLAN VIEW



BACK VIEW



SIDE VIEW

(-1) BALLY ENCLOSURE

1 REQ'D
MODULAR CONSTRUCTION
WEIGHT: APPROX 6000 LBS

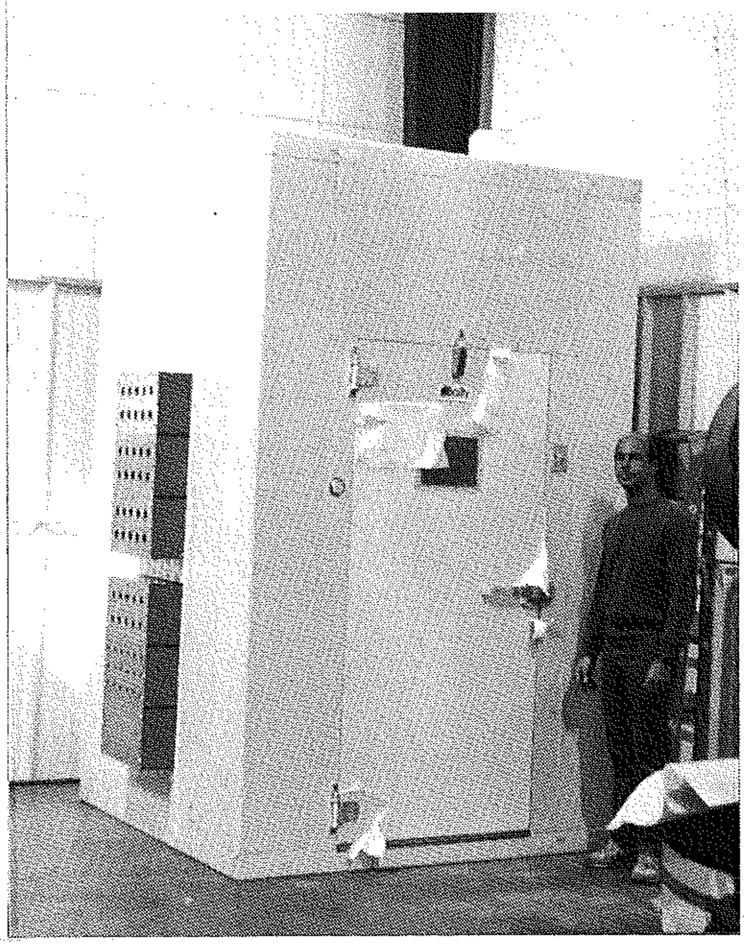
FLOOR PANELS – 455 AND 123 LBS
CEILING PANELS 298 AND 84 LBS
WALL PANELS, 46" = 175 LBS
CORNERS 79 LBS
WALL PANELS, 23" = 91 LBS
BIG DOOR 495 LBS
SMALL DOOR 303 LBS

NOTES:

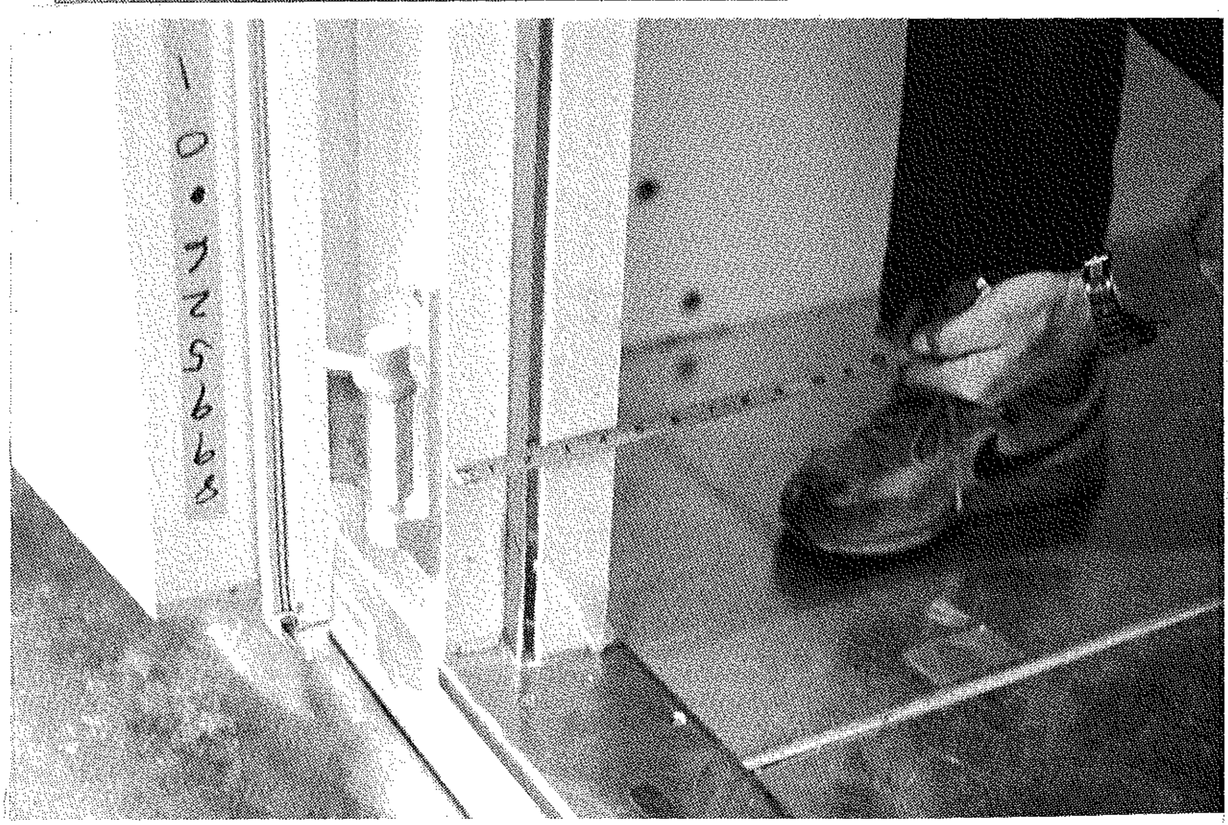
1. ALL PANELS ARE GALVANIZED WITH WHITE POLYURETHANE PAINT.
2. INSULATION IS POURED-IN-PLACE POLYURETHANE FOAM.
3. WALLS AND CEILING PANELS ARE 0.026" THK GALV STEEL
4. FLOOR PANELS ARE 0.073" THK GALV STEEL.
5. STANDARD DOORS WITHOUT HEATER STRIPS ARE REQUIRED, HINGED AS SHOWN.
6. NSF APPROVED FLOOR COVE (5/8" RAD)
7. 32 X 20 CUTOUTS TO REMAIN EMPTY; NO DOORS
8. 30 X 18 CUTOUT REQUIRES A FILLER PANEL THE PANEL WILL HAVE AN OVER-LAPPING FLANGE BUT NO HINGES OR LATCH.
9. CEILING AND FLOOR PANELS RUN 17'4" LONG.

1/20

KECK/HIRES
BALLY MODULAR ENCLOSURE
QUOTATION DRAWING
JA 3 1 90
H6430



ELECTRONICS
"VAULT"



DOORWAY DETAIL

Appendix 1: Convective Heat Transfer

Values	Reference
0.23 $Btu/hr-ft^2-{}^\circ F$	Holman
0.22	McAdams
0.13	Gebhart

The published data for heat transfer coefficients all were obtained with much larger temperature differences than we have here, and so we feel these values may be much larger than we will see in practice ($\Delta T = 1^\circ C$). Nevertheless, the biggest value of 0.23 was used in this report.

According to McAdams, we are in the turbulent boundary layer condition (transition from laminar happens 8 feet up the wall). In this region, $GrPr > 10^9$ and

$$h = 0.19\ (\Delta T)^{1/3}$$

$$\text{Where Gr (Grashoff Number)} = \frac{g\beta(T_w - T_\infty)L^3}{\nu^2}$$

and ν = kinematic viscosity = μ/ρ

μ = dynamic viscosity, 0.0427 lbm/hr–ft

ρ = density (20°F), 14,000 ft, 0.056 lbm/ft^3

g = 32.2 $ft/sec^2 \times (3600\ sec/hr)^2$

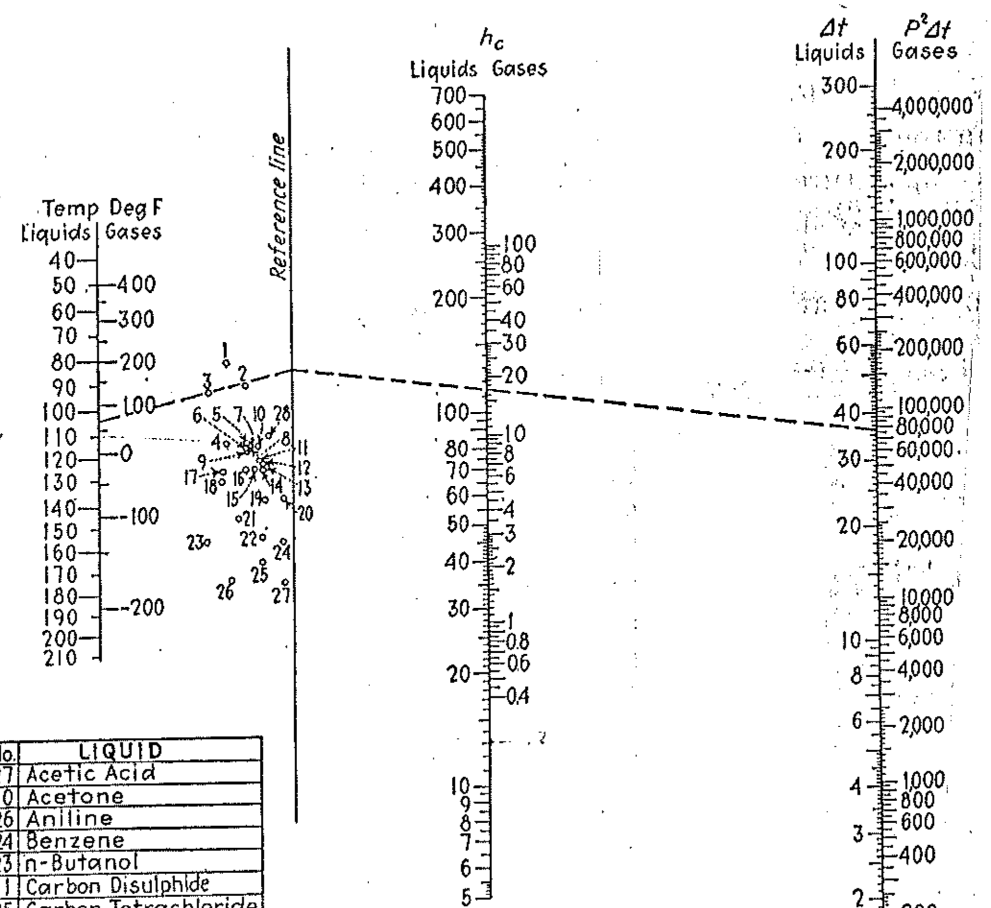
β = vol. coef. of expansion, $1/R^\circ$ = 0.002

L = 10 feet

and Pr (Prandtl Number) = 0.712 (50°F)

The alignment chart on the following page was used to obtain the McAdams value for h.

Touloukian *et al.*[36] reported data for a heated vertical cylinder inside an unheated cylinder containing liquid; the diameter ratio was 4.4. The heater had unheated cylinders at both ends.



No.	LIQUID
27	Acetic Acid
10	Acetone
26	Aniline
24	Benzene
23	n-Butanol
11	Carbon Disulphide
25	Carbon Tetrachloride
21	Ethanol
20	Ethyl Acetate
14	Ethyl Ether
1	Hydrochloric Acid 30%
15	Methanol
19	n-Pentane
18	Sulphuric Acid 110%
17	Sulphuric Acid 98%
16	Sulphuric Acid 60%
22	Toluene
3	Water

No.	GAS
7	Air
28	Ammonia
12	Carbon Dioxide
8	Carbon Monoxide
2	Hydrogen
4	Methane
9	Nitric Oxide
6	Nitrogen
5	Oxygen
13	Water

t_f	Mean Temperature of Film - Deg F
h	Film Coefficient Btu/(hr)(sq ft)(deg F)
Δt	Mean Temperature Difference - Deg F
p	Absolute Pressure of Gas - Atmospheres

FIG. 7-9. Alignment chart[4] for h_c with a *turbulent* boundary layer of air or liquid on *vertical* surfaces, based on Eq. (7-4*a*), for $X_L = N_{Gr,f} \cdot N_{Pr,f}$ from 10^8 to 10^{12}.
Note that Fig. 7-11 is for h_c with a *laminar* boundary layer on a *horizontal* cylinder, based on Eq. (7-6*b*), for X_D from 10^3 to 10^9. For vertical surfaces with X_L from 10^8 to 10^4, use Fig. 7-11, taking D' equal to $2L'/\pi$, expressed in *inches*.
For X_L below 10^4, use recommended curve on Fig. 7-7. With molten metals and fused salts, use Eq. (7-3*a*).

II. HORIZONTAL CYLINDERS

Theory. Temperature and velocity fields near a heated horizontal cylinder have been measured by Jodlbauer.[17] These results have been roughly predicted from theoretical considerations by Hermann;[13] for Grashof numbers exceeding 10^4, Hermann predicts $h_cD_o/k_s = 0.37N_{Gr,s}^{0.25}$ for natural convection from single horizontal cylinders to diatomic gases

Bibliography

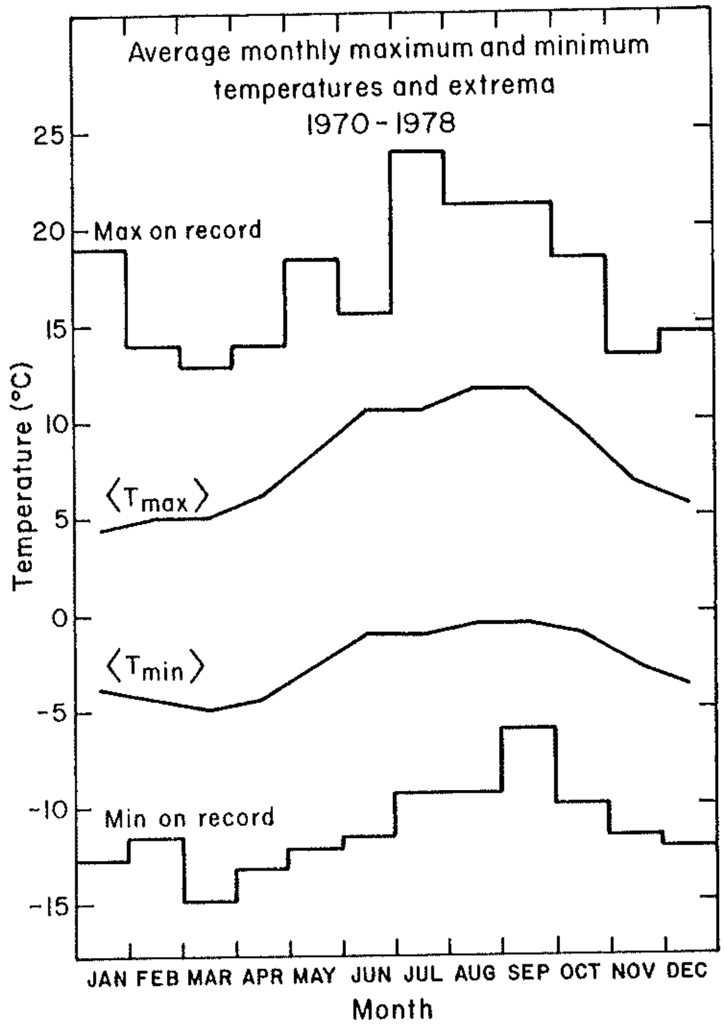
Holman, J.P.: "Heat Transfer", 2nd ed., McGraw-Hill, NY, 1968.

Gebhart, B.: "Heat Transfer", 2nd ed., McGraw-Hill, NY, 1971.

McAdams, W.H.: "Heat Transmission", 3rd ed., McGraw-Hill, NY, 1954.

Whitaker, S.: "Fundamentals of Heat Transfer", Pergamon Press, NY, 1977.

Nelson, J., et al: The Design of the Keck Observatory and Telescope, Report No. 90, UC Publ., Jan 1985.



XBL 8112-1672

Figure 10-4 Average monthly maximum and minimum temperatures and extrema. 1970 to 1978.

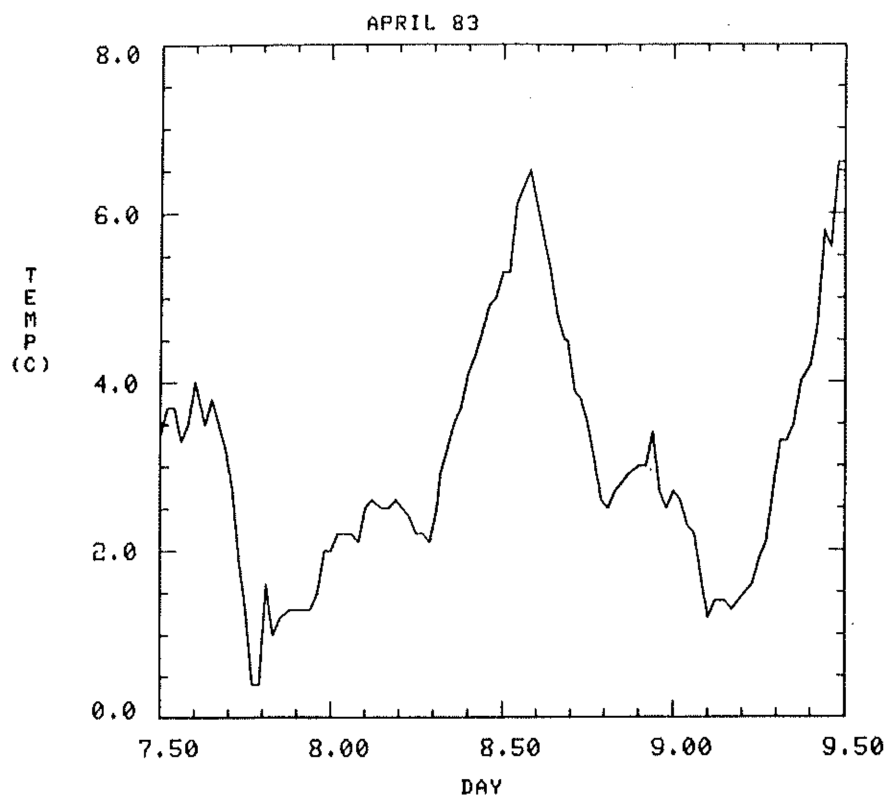


Figure 10-8 A typical 2 day temperature profile from the CFHT.

NOTES