HUMAN COMFORT AND THE HEAT PUMP

By

Abdallah Kamel Eid

Thesis submitted to the Faculty of the Graduate School
of the University of Maryland in partial fulfillment
of the requirements for the degree of
Doctor of Philosophy
1952

ACKNOWLEDGMENTS

The author wishes to express his sincere gratitude to Professor Charles Alfred Shreeve, Jr., of the Mechanical Engineering Department, whose guidance has made possible the research necessary to this thesis. He is deeply indebted to Dr. John E. Younger, Head of the Mechanical Engineering Department, under whose supervision and constant encouragement this study was made possible, and to Dr. J. R. Vanderslice for his directions in the mathematical solutions.

The sincere advice and assistance of the Egyptian Government Officials has also been greatly appreciated.

TABLE OF CONTENTS

Table of Contents, continued

LIST OF TABLES

LIST OF FIGURES

INTRODUCTION

It has long been the contention of the author that the Heat Pump, with proper adaption to the local resources and requirements, can be used to great benefit in his homeland in the Nile Valley, not only financially, but also through increased industrialization and a raising of the standard of living of the Egyptian people as a whole; thereby achieving, if only in a small way, his goal of bettering the lot of his nation.

It was with this purpose in mind, that the author carried out the necessary research, the main considerations of which were the relation of climate and soil to the Heat Pump and the relation of climate to the heat comfort zone. A final accumulation of these results and the conclusions reached are herein given.

Although there are several sources of heat for the Heat Pump, in this study the author chose to drive heat from the soil by using vertical pipes as evaporators. A portion of this soil heat comes from the air or outer layer of the earth's crust. Therefore, the amount of heat obtained will depend greatly upon the temperature and temperature changes at various seasons of the year. The ability of the medium to absorb, hold, and conduct heat must be studied carefully in relation to the conditions of the air, soil, and comfort heat temperatures. Experiments at the locality chosen for the Heat Pump should be made to determine the factors considered in the medium. Since weather bureau records and soil test results are quite generally available, accurate design for the Heat Pump can be accomplished.

Thus, the author looks forward to the day when the Heat Pump will be universally available to people in all income brackets for use in even the smallest home as well as for industrial and agricultural purposes.

TERMINOLOGY

Practical application of the Heat Pump requires a knowledge of physical terminology, definitions and units of measurement, physical characteristics of Freon 114, characteristics of the soil, heat transfer and available equipment for producing the required heat. Physical terminology includes the following common terms:

1. Comfort Zone: That zone within whose limits the variation of temperature leaves the human body in a state of well-being.

2. Degree Day: A unit, based upon temperature difference and time, used in estimating fuel consumption and specifying nominal heating load of a building in winter. For any one day, when the mean temperature is less than 65°F. (which has been found to be the average for most types of buildings) there exists as many degree-days as there are Fahrenheit degrees difference in the temperature between the mean temperature for the day and 65°F.

3. Density: The ratio of the mass of a specimen of a substance to the volume of the specimen: weight of a unit volume of a substance.

4. Dew Point Temperature: The temperature corresponding to saturation (100% relative humidity) for a given absolute humidity at constant pressure.

5. Dry Bulb Temperature: The temperature of a gas or mixture of gases indicated by an accurate thermometer after correction for radiation.

6. Effective Temperature: An arbitrary index which combines into a single value the effect of temperature, humidity, and air movement on the sensation of warmth or cold felt by the human body. The numerical value is that of the temperature of still, saturated air which would induce an identical sensation.

7. Energy: The capacity for performing work (kinetic or potential). In the form of heat; the unit of measure is the Btu which represents the quantity of heat required to raise the temperature of one pound of water one degree Fahrenheit.

8. Heat Transfer by Conduction: The flow of heat through a material medium in which kinetic energy is transmitted by the particles of the material from particle to particle without gross displacement of the particles.

9. Heat Transfer by Convection: The transmission of heat in a fluid by the differences in density of either natural or forced motion of the fluid (liquid or gas).

10. Heat Transfer by Radiation: The transmission of energy by means of electromagnetic waves of very long wave length.

11. Horsepower: The amount of power required to accomplish 33,000 ft. per pound of work per minute: $H.P. = \frac{\text{foot pound}}{\text{time} \times 33{,}000}$.

12. Isobaric: An adjective used to indicate a change taking place at constant pressure.

13. Isothermal: An adjective used to indicate a change taking place at constant temperature.

14. Latent Heat: A term used to express the energy involved in a change in state.

15. Length: The unit of measurement as inch, foot and yard.

16. Mean Radiant Temperature: The temperature of a uniform, black enclosure with which a human body would exchange the same amount of energy by radiation as in the actual environment.

17. Power: The rate of performing work; Btu per hour or watts.

18. Pressure: Force extended upon a unit area of a substance, usually expressed in pounds per square inch.

$$P \text{ (absolute)} = P \text{ (gauge)} + P \text{ (atmosphere)}.$$

19. Refrigeration Ton: Amount of heat necessary to melt one ton of ice at 32°F. in 24 hours. Since the latent heat of fusion is 144/Btu/# of water, the heat required to melt one ton of ice is:

$$2000\# \times 144 \text{ Btu} = 288{,}000 \frac{\text{Btu}}{\text{day}} = 12{,}000 \frac{\text{Btu}}{\text{hour}} = 200 \frac{\text{Btu}}{\text{minute}}.$$

20. Relative Humidity: The ratio (percentage) of the actual partial pressure of the water vapor in a space to the saturation pressure of pure water at the same temperature.

21. Sensible Heat: Used to indicate any portion of heat which changes only the temperatureof the substances involved.

22. Specific Gravity: For solids and liquids-the weight of the substance to the weight of an equal volume of water at 39.2°F. For gases-the ratio of the weight of an equal volume of dry air at the same temperature and pressure as the gas.

23. Specific Heat: The ratio of the amount of heat required to change a unit weight 1°F. to that required to change a unit weight of water by 1°F. (dimensionless).

24. Temperature: A measure of heat intensity. The unit of measure used throughout this thesis is the degree of Fahrenheit.

25. Time: A unit of measurement as the standard second, minute and hour.

26. Weight: The unit of measurement as ounce, pound and ton.

27. Wet Bulb Temperature: The temperature at which liquid or solid water, by evaporating into air, can bring the air to saturation adiabatically at the same temperature.

28. Work: That which is produced when a force acting upon a body over-

comes resistance and produces motion; a foot pound if a weight of one pound is lifted a distance of one foot.

Nomenclature

1. A--The area for heat transfer, perpendicular to the direction of heat transfer ft^2.
2. a--Thermal diffusivity $= \frac{K}{\rho C_p}$ ft^2/hour.
3. C_p--Heat capacity at constant pressure Btu/lb, °F.
4. h--Unit thermal conductance (subscripts; c-convection & r-radiation).
5. F_{AE}--Geometrical and emissivity influences on rates of heat transfer by radiation (dimensionless).
6. K--Thermal conductivity Btu/hr, ft, $(\frac{°F}{ft})$
7. L--Length (feet)
8. Nu--Nusselt modulus $= \frac{hL}{K}$ (K for fluid) (dimensionless).
9. p--Perimeter (feet). P-- Pressure (lb/ft^2)
10. q--Heat transfer rate (Btu/hr). Q--Heat transfer (Btu).
11. ρ--Weight density (lb/ft^3).
12. Δ --Finite increment.
13. γ--Mass density (slugs/ft^3).
14. μ--Absolute viscosity (lb/sec/ft^2)
15. ε --Emissivity of a surface (dimensionless).
16. σ--Stefan-Boltzman constant $= .173 \times 10^{-8}$ (Btu/hr, °F., $°R^4$).
17. Re--Reynolds modulus $= \frac{V l \rho}{\mu}$ (dimensionless).
18. Gr--Grashof modulus $= \frac{\beta \Delta t D^3 \rho^2 g}{\mu^2}$ (dimensionless).
19. Nu--Nusselt modulus $= \phi (Gr)^a (Pr)^b$. (dimensionless)

CHAPTER I—CLIMATE

A. THE NILE VALLEY.

The Nile Valley, extending from South to North for more than 32° latitude, is well equipped with meteorological stations, especially in the upper section. As the Valley passes through all the climatic belts from the Equator to the Mediterranean Sea, the records provide a particularly instructive picture of the climate of North Africa. The Valley is divided climatically into three sections:- The Egyptian Sudan, Egypt, and the Sahara.

Egypt has a Mediterranean climate; mild winters with but little rain and hot, arid summers. On the seashores, where the delta of the Nile touches six states and four important ports, the Mediterranean climate has three main characteristics:-

1-Most of the rain falls in winter and there is drought, more or less complete, in the four months of summer.

2-The winters have little rain, but very mild atmosphere. The mean temperature is about 54°F. Summer is hot and dry, the mean temperature in July being about 75°F.

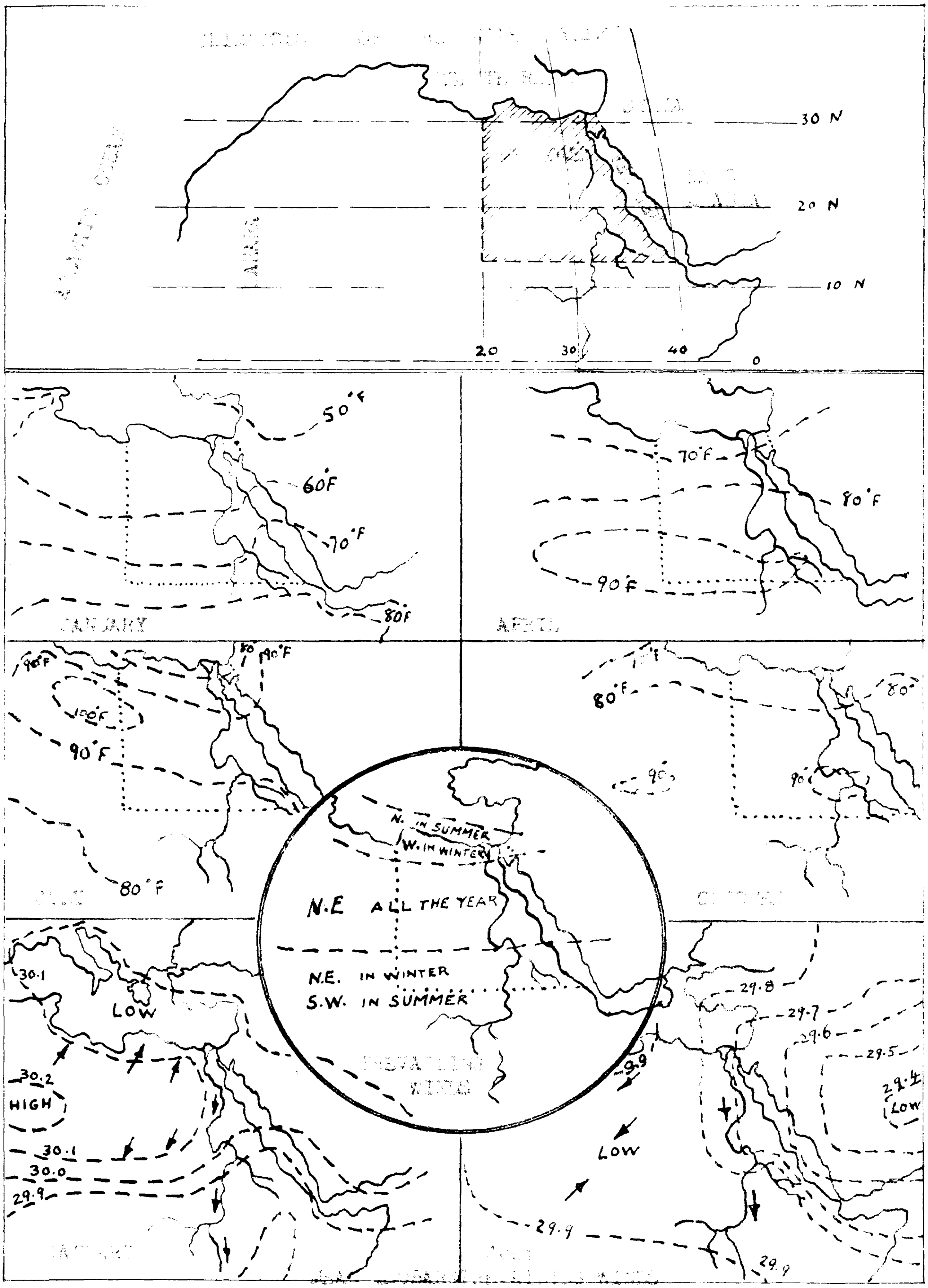
3-The bright sunny skies, clear in summer and practically cloudless in winter, are one of the far reaching influences on human development. The hot sunny weather is ideal for the ripening of the fruits for which the region is famous.

The almost constant north winds of the Egyptian Delta come from the North-east winds which blow fairly strong and steadily on the western half of the North African coast, under the control of Saharan low presures, reinforced by the effect of the fierce heat of the land adjacent to the cooler seas.

The climate of the western border of the Nile Valley closely resembles that of the Sahara. In the southern part of Egypt's latitude 21° where the sun's almost vertical rays are more powerful, the temperature in July is about 90°F., while in the Sahara it reaches more than 100°F. However, the dryness of the air and the resulting rapid evaporation, make the heat bearable, provided that there is protection from the direct rays of the sun. At night, there is rapid radiation, but nevertheless, it is always quite warm and dew is rare. In winter, North Africa is a region of high pressure while the Mediterranean Sea is a region of low pressure.

The mean annual rainfall at Alexandria is eight inches and at Port Said three inches. At the Delta it is only two inches. At Cairo, we reach the conditions of mean annual rainfall of 1 inch, all of it falling in winter. In spite of the lower latitude, the winters are cooler than on the shores of the Mediterranean, but the summers are far warmer and the air much drier. The mean temperature at Cairo in January is 53°F.

The humidity rises in summer and is at its lowest point in spring. In late summer, when the Nile is flooding, the atmosphere in the whole Nile Valley is somewhat hazy and the air may feel comparatively moist. At Helwan, according to official weather reports (15), the relative humidity is only 37% in May while it reaches 55% in September. On June 15, 1948, the temperature in Cairo reached 115°F. (46°C.) with a relative humidity of 32%. This has happened only twice in Cairo during the last century. The maximum temperature was 118.5°F. (48°C.) on June 13,



30 N
20 N
10 N
20
30
40
0
50°F
60°F
70°F
80°F
70°F
80°F
90°F
90°F
100°F
90°F
80°F
80°F
80°
90°
90°
N. IN SUMMER
W. IN WINTER
N.E ALL THE YEAR
N.E. IN WINTER
S.W. IN SUMMER
30.1
LOW
30.2
HIGH
30.1
30.0
29.9
29.8
29.7
29.6
29.5
29.4
LOW
29.9
LOW
29.9
29.9

Figure 1a

TABLE NO. 1a CLIMATOLOGY OF ALEXANDRIA AND PORT SAID

PERIOD		JAN.	FEBRUARY	MARCH	APRIL	MAY	JUNE	JULY	AUGUST	SEPT.	OCT.	NOV.	DEC.	YEAR
1901-1944	MEAN DAILY MAX. TEMP °C	18.5	19.2	21.3	23.6	26.2	28.2	29.6	30.4	29.1	28.4	25.0	20.6	25.1
1901-1944	MEAN DAILY MIN TEMP °C	10.6	11.0	12.9	15.0	18.0	20.8	23.0	23.6	22.5	20.2	16.9	12.6	17.3
1901-1944	MEAN OF DAY TEMP °C	13.7	14.2	15.9	18.1	21.0	23.6	25.4	26.2	25.3	23.3	19.9	15.7	20.2
1901-1948	HIGHEST READING °C	27.6	33	39.4	42.3	43.8	43.7	39.6	40.5	41.3	39.9	35.3	30.9	
	DATE	16-1941	17-1902	29-1901	23-1928	10-1941	16-1915	1-1940	27-1920	27-1929	3-1912	3-1919 5-1941	19-1937	
1901-1948	LOWEST READING °C	3.3	2.8	6.5	9.6	12.0	15.0	17.5	18.0	15.1	12.2	8.0	3.0	
	DATE	30-1921	16-1934	4-1913	7,8-1943	3-1905 6-1931	16-1902 2-1904	1,5-1903	31-1903	23-1910	20-1911	22-1908	10-1903	
1920-1929	RELATIVE HUMIDITY % at 8 AM	73	71	67	65	69	68	74	70	65	65	69	76	69
1925-1934	RELATIVE HUMIDITY % at 2 PM	61	59	57	58	64	68	72	68	63	61	61	61	63
19[illegible]-1944	RELATIVE HUMIDITY MEAN OF DAY	69	68	68	69	72	74	77	75	69	70	70	70	71
1901-1944	MEAN DAILY MAX TEMP °C	18.8	19.8	21.3	23.7	26.5	29.3	31.3	31.8	30.6	28.7	25.1	20.7	25.6
1901-1944	MEAN DAILY MIN TEMP °C	10.6	11.3	13.3	15.7	18.6	21.5	23.2	23.8	22.8	21.1	17.6	12.6	17.7
1901-1944	MEAN OF DAY TEMP °C	13.7	14.3	16.2	18.7	21.8	24.6	26.3	26.9	25.8	23.9	20.4	15.7	20.7
1901-1948	HIGHEST READING °C	29.0	33.1	38.0	40.4	45.0	43.9	38.1	37.2	40.1	37.6	36.5	29.8	
	DATE	30-1912	29-1924	29-1901	25-1906	10-1941	1-1914	3-1926	30-1924	27-1929	31-1918	7-1941	19-1937	
1901-1948	LOWEST READING °C	3.0	2.3	2.7	9.2	10.2	14.2	19.0	20.6	17.5	13.0	9.3	0.0	
	DATE	13-1902	11-1920	12-1917	5-1914	12-1917	1-1917	1-1942	30-1912 14-1916	26-1908	31-191[illegible] 30-1912	22-1904	15-1907	
1920-1929	RELATIVE HUMIDITY % at 8 AM	77	75	71	67	68	68	72	72	70	69	72	76	71
1925-1934	RELATIVE HUMIDITY % at 2 PM	59	55	58	59	60	60	60	60	57	58	60	59	59
1901-1944	RELATIVE HUMIDITY % MEAN OF DAY	76	75	73	73	[illegible]	75	76	76	73	73	73	76	74

TABLE NO. 1b CLIMATOLOGY OF CAIRO AND ASSIUT

PERIOD		JAN.	FEB.	MAR	APR.	MAY.	JUNE	JULY	AUG.	SEP.	OCT.	NOV.	DEC.	YEAR
1909-1947	MEAN DAILY MAX TEMP °C	19.8	21.4	24.[illegible]	28.7	32.8	35.3	36.0	35.2	32.6	30.6	26.4	21.4	28.7
1909-1947	MEAN DAILY MIN TEMP °C	7.7	8.4	10.7	13.6	17.2	20.0	21.7	22.0	20.0	17.6	14.1	9.6	15.2
1909-1944	MEAN OF DAY TEMP °C	12.3	13.5	16.3	20.2	24.1	26.8	27.7	27.6	25.2	22.7	18.7	14.0	20.8
1909-1948	HIGHEST READING TEMP °C	29.7	34.9	38	43.0	46.2	46.6	45.3	41.0	42.7	43.7	39.6	31.2	
	DATE	29-1912	22-1941	28-1917	30-1935	9-1941	13-1933	16-1947	27-1920	28-1939	3-1943	3-1941	18-1937	
1909-1948	LOWEST READING TEMP °C	-.8	2.6	2.8	7.0	9.9	14.2	16.8	18.1	14.5	12.0	7.3	2.0	
	DATE	13-1910	16-1934	1-1928	2-1946	1-1914	14-1912 2-1943	26-1913	1-1913	30-1916	31-1912	30-1920	13-1936	
1924-33	RELATIVE HUMIDITY % at 8 AM	78	72	71	62	59	64	72	76	79	80	82	78	73
1925-34	RELATIVE HUMIDITY % 2 PM	48	40	37	31	29	30	35	39	43	43	46	48	39
1909-47	RELATIVE HUMIDITY % MEAN OF DAY	74	67	64	58	52	55	61	65	69	71	74	75	65
1900-44	MEAN DAILY MAX. TEMP °C	20.1	22.5	26.7	31.6	35.5	37.2	37.2	36.6	33.7	30.7	26.6	21.6	30.0
1900-44	MEAN DAILY MIN TEMP °C	6.1	7.1	10.1	14.5	18.9	21.4	22.5	22.8	20.9	18.5	12.9	7.9	15.3
1900-45	MEAN OF DAY TEMP °C	11.7	13.3	17.1	22.2	26.6	28.8	29.4	29.1	26.5	23.8	18.6	13.6	21.7
1900-48	HIGHEST READING TEMP °C	33.0	35.8	42.5	45.0	48	49.5	45.0	44.6	44.0	40.0	39.0	32.6	
	DATE	19-1945	27-1936	30-1905	23,24 1905	14-1941 6-1942	13-1933	20-1915 23-1932 16-1947	11-1934	13-1940	8-1945	5-1936 3-1941	20-1933	
1900-48	LOWEST READING TEMP °C	0.0	0.0	2.0	5.6	10.5	14.6	15.5	18.2	16.5	10.5	4.5	0.0	
	DATE	13-1910 4-1942	14-1908	5-1913	2-1946	[illegible] 10-1910	3-1943	11-1900	30-1933	29-1905	31-1908	22-1904	27-1902	
1920-29	RELATIVE HUMIDITY % 8 AM	72	64	59	44	40	44	48	52	64	67	71	72	58
1925-34	RELATIVE HUMIDITY % 2 PM	42	33	25	19	17	18	21	24	31	41	43	44	30
1900-44	RELATIVE HUMIDITY % MEAN OF DAY	69	62	53	41	36	37	42	46	56	62	67	69	53

TABLE NO. 1c CLIMATOLOGY OF ASWAN

PERIOD		JANUARY	FEBRUARY	MARCH	APRIL	MAY	JUNE	JULY	AUGUST	SEPT.	OCT.	NOV.	DEC.	YEAR
1901-1944	MEAN DAILY MAX. TEMP °C	23.5	25.6	30.5	35.5	39.4	41.7	41.3	41.1	39.4	36.8	30.6	25.3	34.2
1901-1944	MEAN DAILY MIN TEMP °C	10.1	11.2	14.6	19.2	23.5	25.8	26.4	26.4	24.0	21.7	16.5	11.9	19.3
1901-1945	MEAN OF DAY TEMP °C	15.5	17.2	21.3	26.2	30.5	32.9	33.2	33.0	30.7	28.3	22.6	17.4	25.8
1901-1948	HIGHEST READING °C	37.8	38.9	43.4	46.6	48	50.6	51.0	49.0	47.0	45.2	41.5	37.0	—
	DATE	28-1935	1-1928	28-1936	27-1901	23-1929 15-1941	8-1932	4-1918	25-1924	several	4-194[illegible]	6-1903 3-1938	21-1933	—
1901-1948	LOWEST READING °C	3.0	1.7	6.0	9.3	11.0	19.0	20.2	19.6	17.0	12.2	3.0	2.5	—
	DATE	19-1904	9-1912	7-13 5-21 3-25	8-1926	7-1918	4-1904	17-1907	3-192[illegible]	27-1902	31-1912	22-1904	28-1902	—
1920-1929	RELATIVE HUMIDITY % 8 AM	56	48	38	31	31	30	30	34	41	43	48	53	40
1925-34	RELATIVE HUMIDITY % 2 PM	32	25	19	18	17	19	18	19	22	22	26	31	22
1901-44	RELATIVE HUMIDITY % MEAN OF DAY	45	40	32	28	27	25	26	29	33	35	40	45	34

$\times \frac{9}{5} + \quad =$

( )

( )

X =

X -4 =

6

1903. During the early afternoon hours of mid-June, all outdoor life comes to an abrupt standstill--man and beast drag themselves to a shady place for rest.

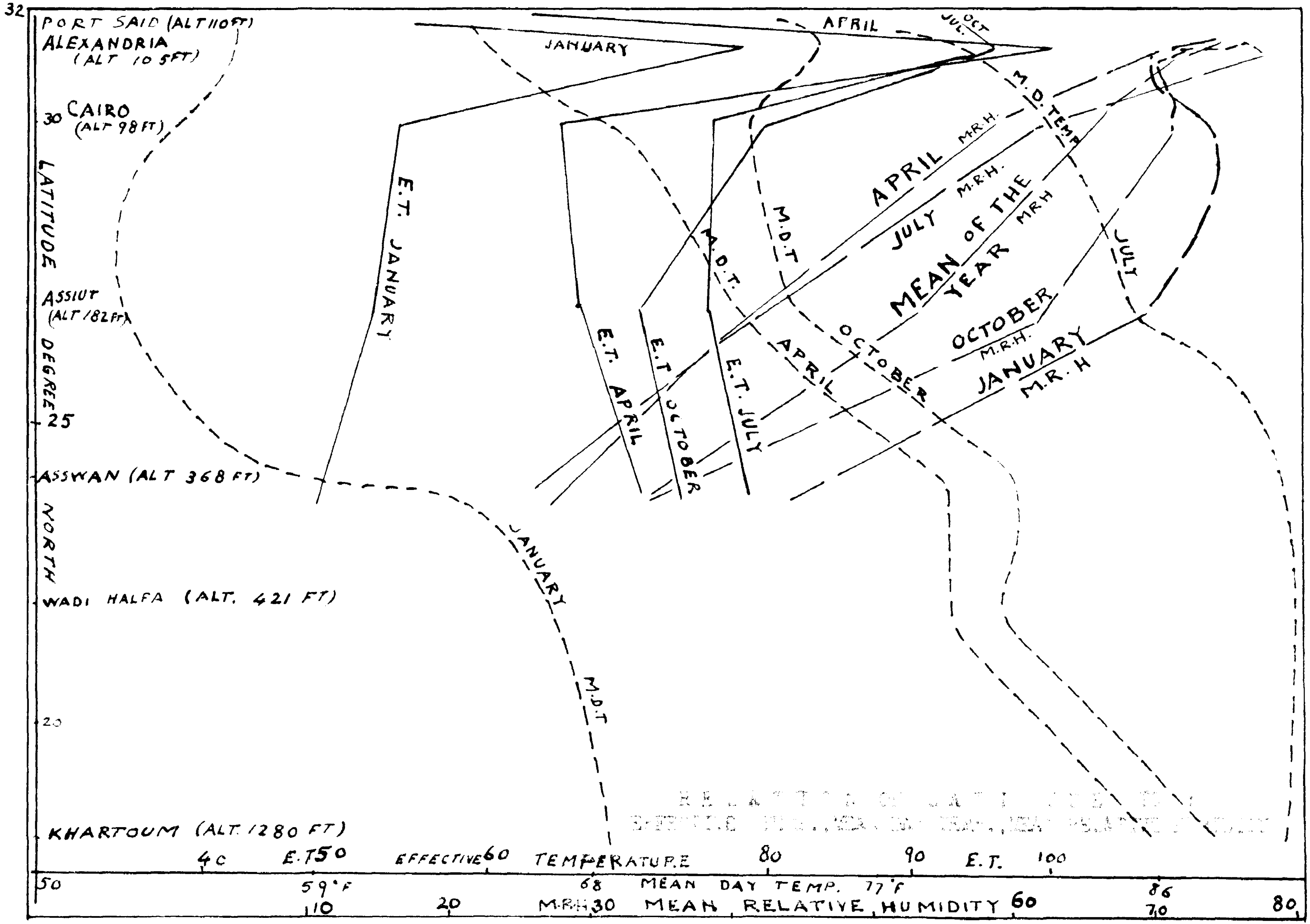
The "Khamsin" winds are caused by depressions moving North-east from the Sahara towards the Delta. Owing to the quarter from which they come, they blow hot and enervatingly with readings of 109°F. in Cairo. In winter, the south winds are cooler than normal and continue for two or three days. These are followed by waves of cold air lasting up to four months, necessitating the artificial heating of homes and business houses. The drop in temperature in Khartoum and most of the Sudan at this time is appreciable.

South of the Nile, in the Sahara, rain is almost unknown. Sutton described this case. "Siwa, Lower Egypt, on the 28th and 29th of December, 1930 received 1.5 inches, which did much damage to the houses of the oasis." (41) Usually, the mean rainfall at Siwa in December is almost never more than.1 inch!

At Cairo and in the Nile Valley south of it, the prevailing winds throughout the year are northerly and follow almost exactly the course of the river. These winds are so strong and constant that Nile boats easily make the journey upstream under sail and float down again with the current. South of Khartoum, at the junction of the Bahr-El-Ghazal and the Nile, the rains become heavier and last longer from April to October.

Mongalla (5°N. latitude) is in the region of summer rains and has the maximum rainfall from August till October. At Kadolai (3° N. latitude) the climate is equatorial; rain all of the year with the maximum in May and October.

Figure 2



PORT SAID (ALT 110 FT)
ALEXANDRIA (ALT 10.5 FT)
CAIRO (ALT 98 FT)
ASSIUT (ALT 182 FT)
ASSWAN (ALT 368 FT)
WADI HALFA (ALT. 421 FT)
KHARTOUM (ALT. 1280 FT)
LATITUDE DEGREE NORTH
32
30
25
20
JANUARY
APRIL
JULY
OCT
M.D. TEMP
E.T. JANUARY
E.T. APRIL
E.T. OCTOBER
E.T. JULY
M.D.T.
M.D.T
APRIL M.R.H.
JULY M.R.H.
MEAN OF THE YEAR M.R.H.
OCTOBER M.R.H.
JANUARY M.R.H.
APRIL
OCTOBER
JULY
JANUARY
M.D.T
40
E.T. 50
EFFECTIVE 60
TEMPERATURE
80
90
E.T.
100
50
59°F
68
MEAN DAY TEMP.
77°F
86
10
20
M.R.H. 30
MEAN RELATIVE HUMIDITY
60
70
80

The change from the Sahara to the Sudan is less obvious to the traveller up the Nile Valley, since the Nile forms a continuous oasis. To the North, the air is rather damp while the South is hot and arid.

The influence of the Mediterranean Sea appears at Alexandria, which is warmer in winter but cooler in summer than Cairo. The fall in temperature at Berber and Khartoum at midsummer is caused by the monsoon rains. Mongalla has the same conditions as the equatorial zone.

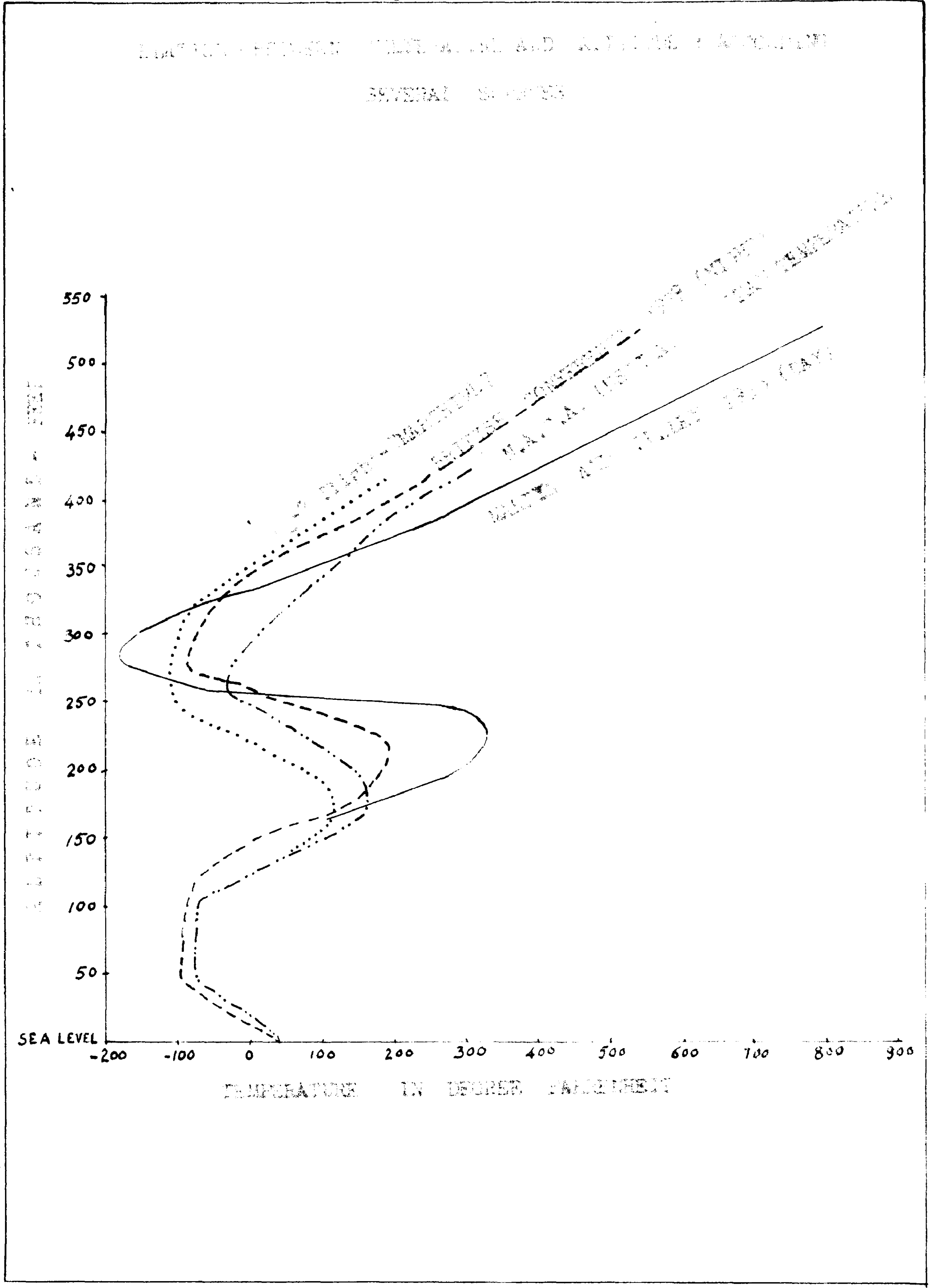
In Egypt, through the winter period, the buildings need heating from 50°F. Dry Bulb to 70°F. Dry Bulb and will require cooling from 100°F. to 80°F. Dry Bulb in summer with all the requirements for human comfort.

B. WASHINGTON, D. C.

The weather conditions at College Park, Maryland and Washington, D. C. are almost identical, the distance between the two being about fifteen miles with their latitude and longitude very similar. Washington, D. C. lies in the Middle-Atlantic Coastal Plain at a latitude of 38° 54' N. and a longitude of 77° 03' W. It is about 120 miles inland from the Atlantic Ocean. It has a temperate climate of fairly high relative humidity and moderate rainfall. The summers are rather warm, while the winters are mostly moderate. Spring and Autumn are generally pleasant.

The mean annual temperature for eighty years has been 55.6° F. The average temperature for individual months varies from 34.7° Dry Bulb in January to 77.1°F. Dry Bulb in July. The warmest year was 1949 with an average of 59.5°F. and the coldest was in 1875, and again in 1904, with an average of 52.2°F. The highest temperature on record is 105.6°F

Figure 3


550
500
450
400
350
300
250
200
150
100
50
SEA LEVEL
-200
-100
0
100
200
300
400
500
600
700
800
900

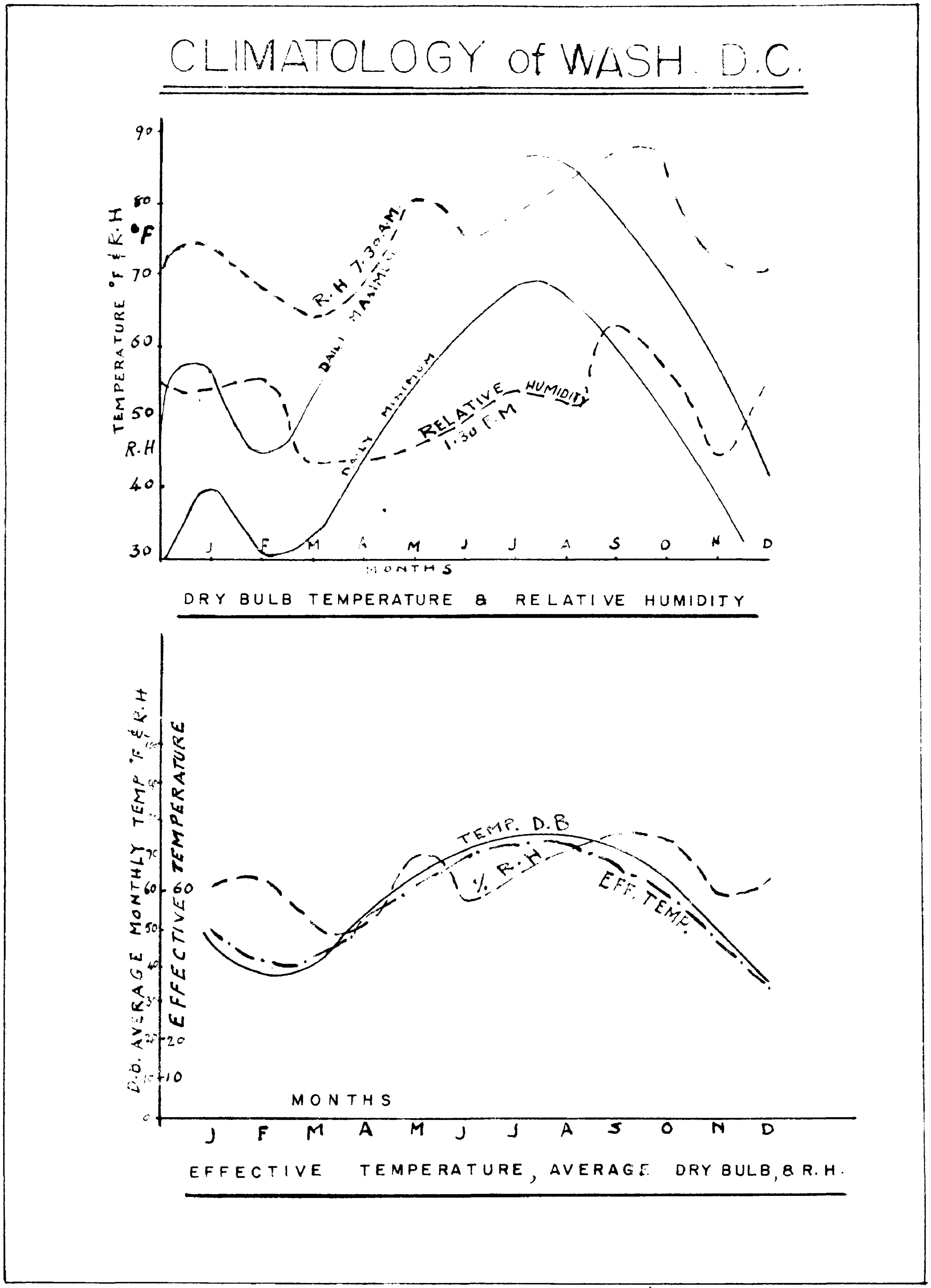
Dry Bulb on July 20, 1930; the lowest on record being -14.9°F. on February 11, 1899 during the worst blizzard in the climatic history of Washington. The extremes for the United States are 134°F. at Greenland Ranch, Death Valley, California, and -66° F. at the Riverside Ranger Station in Yellowstone Park, Wyoming, on July 10, 1913.

The relative humidity is high in August and September and lowest in April. The diurnal high normally occurs about 6:00 A.M. and the low about 3:00-4:00 P.M. The lowest ever recorded was 6% on March 11, 1929. The annual rainfall in Washington is 42.06 inches. July is the wettest month with 4.44 inches, while the driest is November with 2.63 inches. The maximum rainfall was 61.33 inches in 1889 and the minimum was 18.79 inches in 1826.

The average annual snowfall is 20.3 inches with snow occuring every year in January and February with average duration of eleven days per year. The greatest seasonal snowfall was 54.4" in 1899 while the lowest was 2.0" in 1950. The average sleet is expected in Washington and vicinity about eight days during the year. Washington seldom experiences severe damage from sleet storms. The maximum sleet was seven inches in 1927. The average annual number of hailstorms is one. In a period of 20 months, one may be expected. On April 29, 1938 a severe hailstorm caused one hundred thousand dollars worth of damage to greenhouses, windows, automobile tops etc.

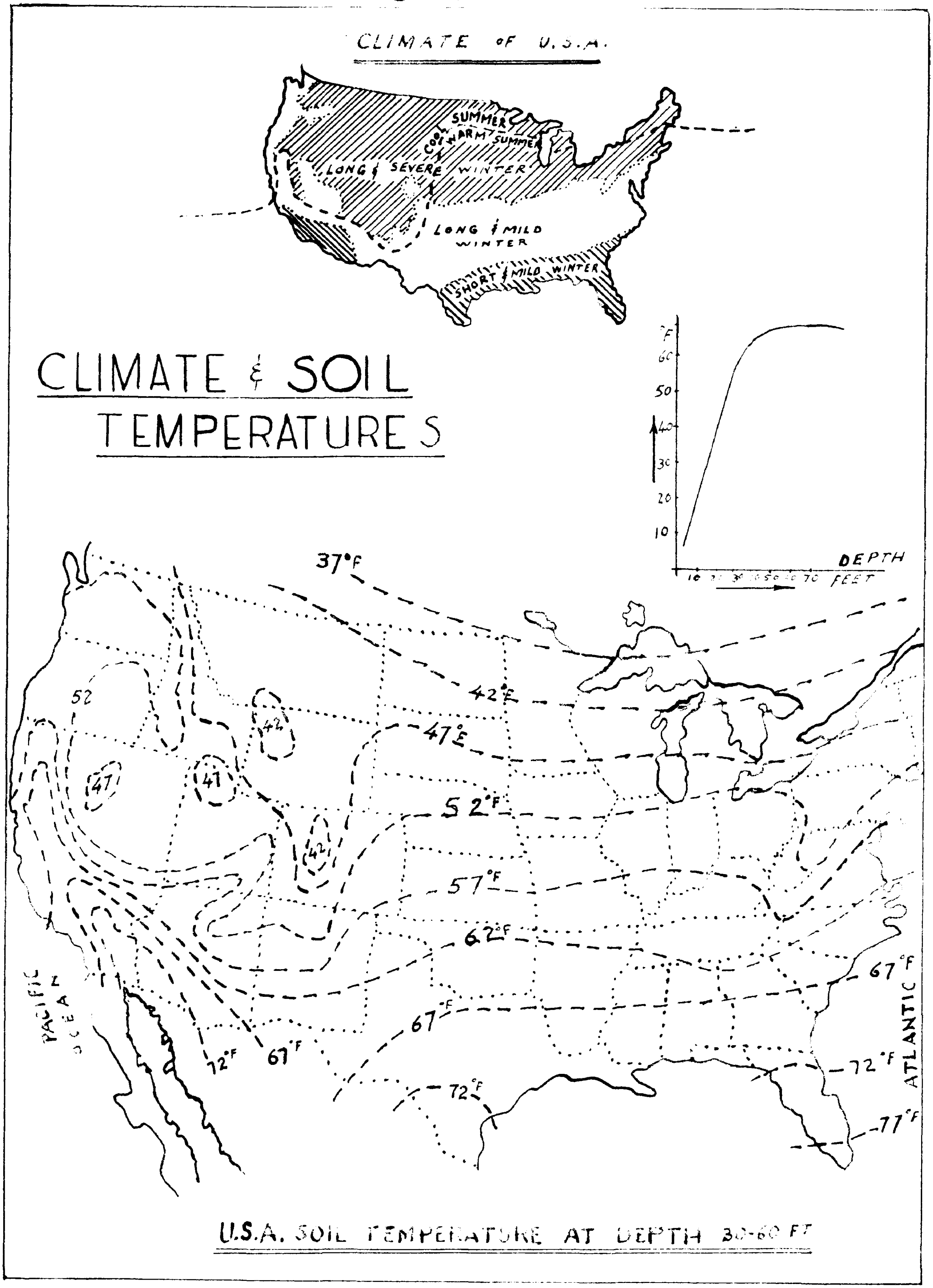
Floods occur in the Potomac River on an average of once every two years, mostly in March and April. The highest flood stage was 17.7' on the Wisconsin Avenue gage (flood stage-7 feet) in October 1942. The average annual wind speed is seven miles per hour, from the Northwest 23% of the time, 19% from the South. The extreme was 62 m.p.h. from the Southeast on September 29, 1896 during the passage of a tropical disturbance.

Figure 4


CLIMATOLOGY of WASH. D.C.
TEMPERATURE °F & R.H
°F
R.H
90
80
70
60
50
40
30
R.H 7.30 A.M
DAILY MAXIMUM
DAILY MINIMUM
RELATIVE HUMIDITY
1.30 P.M
J F M A M J J A S O N D
MONTHS
DRY BULB TEMPERATURE & RELATIVE HUMIDITY
D.B. AVERAGE MONTHLY TEMP °F & R.H
EFFECTIVE TEMPERATURE
60
20
10
TEMP. D.B
% R.H
EFF. TEMP.
MONTHS
J F M A M J J A S O N D
EFFECTIVE TEMPERATURE, AVERAGE DRY BULB, & R.H.

The average annual for thunderstorms is about thirty-one days. On June 9, 1928, a violent local thunderstorm swept over Bolling Field, destroying twelve airplanes. The maximum yearly degree-days was 5573 in the year from July 1903 till June 1904 while it was only 3804 in the year from July 1949 till June 1950.(54)

Figure 5


CLIMATE OF U.S.A.
COOL SUMMER
WARM SUMMER
LONG & SEVERE WINTER
LONG & MILD WINTER
SHORT & MILD WINTER
CLIMATE & SOIL TEMPERATURES
°F
60
50
40
30
20
10
DEPTH
FEET
37°F
42°F
47°F
52°F
57°F
62°F
67°F
72°F
77°F
52
42
47
41
42
PACIFIC OCEAN
ATLANTIC
U.S.A. SOIL TEMPERATURE AT DEPTH 30-60 FT

CHAPTER II--HUMAN COMFORT

Four of the main factors which have influenced life since the creation of the universe are geography, geology, climate and religion. Of these, climate may well be considered the most important in the evolution of civilization. It has had direct bearing on the type of dress, housing, work and recreation of each people and country. The gradual ability of man to utilize the resources of nature, has caused advancement in every nation.

Climatic conditions, together with an inherent fear of the great unknown beyond the actual experience of early man, caused him first to seek shelter in caves, to erect simple "homes" and gradually to cover the entrance, the floors, etc., either in protection from the cold or as insulation against the heat of local climatic conditions. In the same manner, climate caused early man to discover that by covering his body, he could attain "body heat comfort".

During the past quarter of a century, engineers in the field of heating, cooling and ventilating have made tremendous progress through research. This progress has been made, not only in the development and application of new and improved equipment for obtaining the desired environmental conditions, but also in a better understanding of the relationship of man to his physical environment.

While the physician works to improve the health of humanity, the engineer strives to improve the living conditions of this same humanity. Research has been carried out with normally healthy persons to determine the relation between heat loss and heat production under different environmental conditions and at different levels of activity.

It is important that heat transfer engineers and physiologists confer more often and learn to speak each other's language. The engineer speaks and writes in terms of Fahrenheit, square feet and British thermal units, while the physiologist lives in a realm of Centigrade, square meters and calories.

A. INTERCHANGE BETWEEN THE BODY AND ITS ENVIRONMENT.

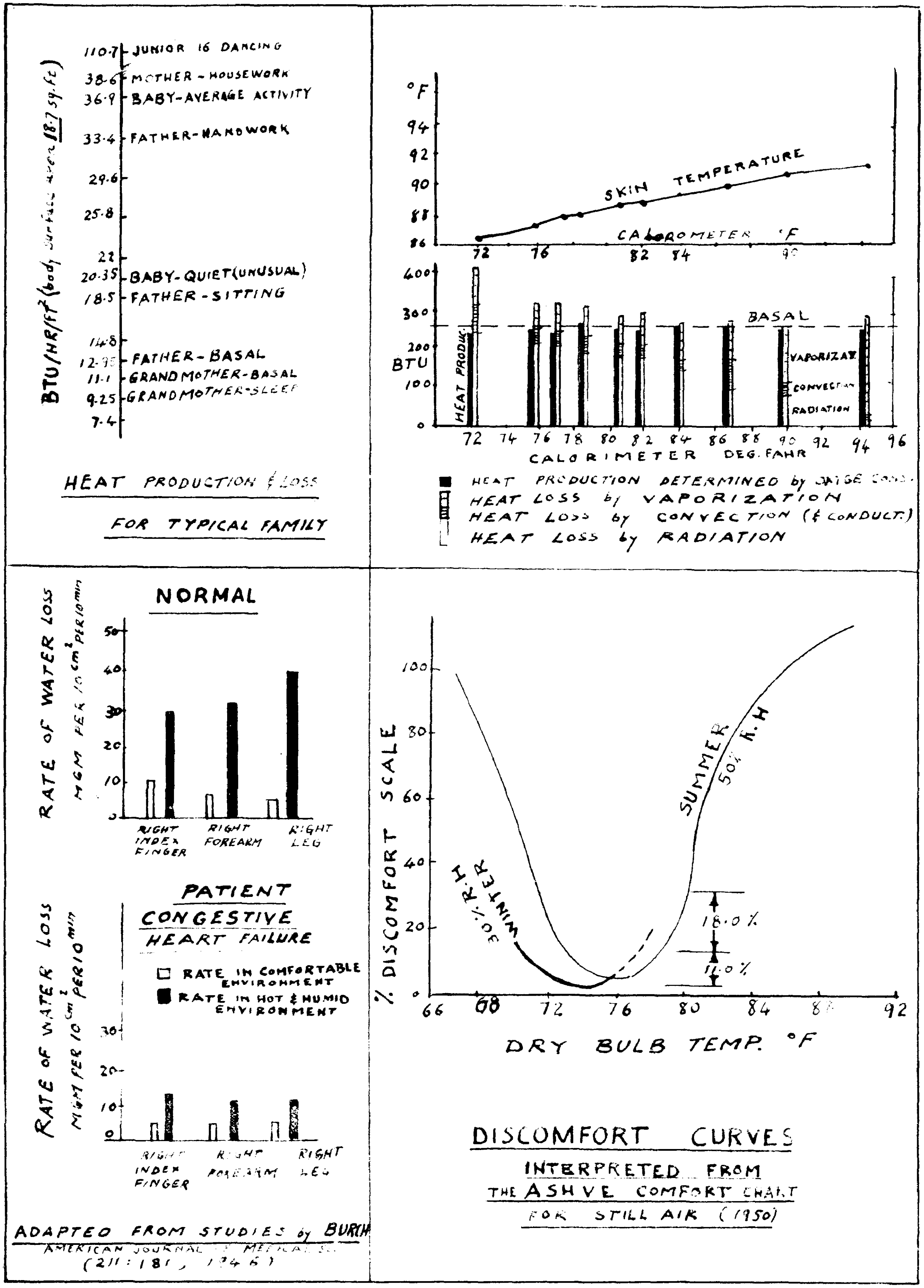
<u>The Human Body</u>: In itself, the human body is a great heat generator. Luckily, it also contains a very reliable heat regulatory system. The body produces heat from the chemical processes within it which are dependent upon the food intake. Muscle action and the glands, mainly the liver, produce great quantities of heat. If the body did not also give off heat, the temperature within would soon reach the boiling point. To keep the body temperature at a constant level, the body loses heat through the skin by radiation (150 Btu/hr), by conduction (100 Btu/hr), by evaporation (150 Btu/hr), from respiration, and from the dejecta.

The human body in relation to the surrounding atmosphere gives off or takes in heat and moisture. Thus, the physiological effect of climate upon the body depends mainly on the temperature and relative humidity of the surrounding air.

The mean average temperature in man is 98.4°F. by mouth, 98.5°F. in the axilla, and 99°F. by rectum. Several conditions modify the regulation of animal heat; day and night, age, muscular work, sleep, sex, race, idiosyncrasies, surrounding temperature, season of the year, and certain drugs. A variation in the temperature of man of 2°F. is normal. Temperatures below 80°F. and above 106°F., however, are most dangerous.

In cold weather, the loss of body heat is increased considerably and an extreme exposure may result in frost-bite/or death. Hot weather,

Figure 6-HUMAN THERMAL BALANCE



on the other hand, particularly when it is accompanied by a high percentage of humidity, produces a rise in body temperature owing to the interference with normal loss of body heat by evaporation. The pulse rate increases, the blood pressure decreases and a feeling of depression and general discomfort, or even total collapse, is experienced.

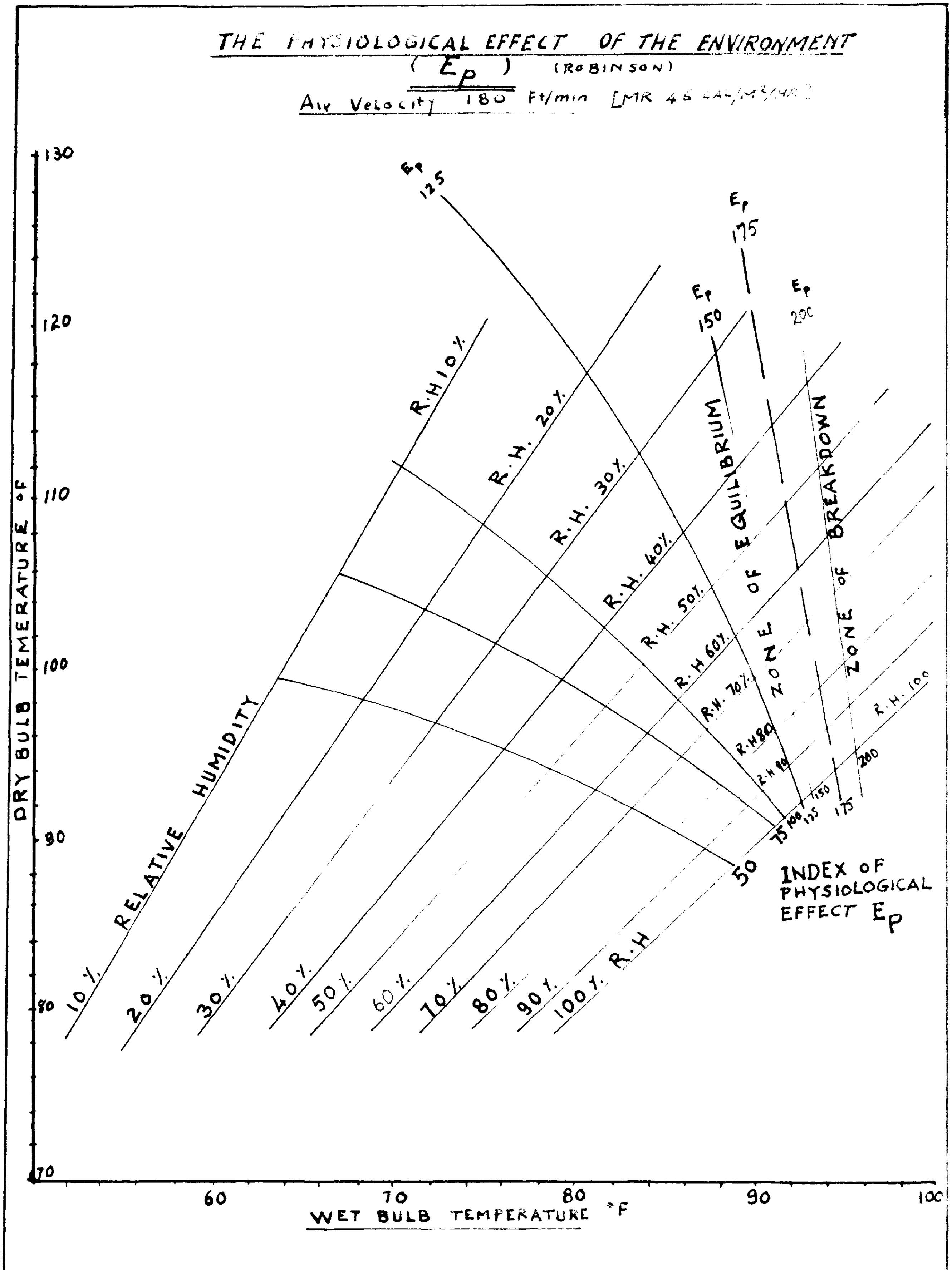
The result of exact balance of heat production and heat loss is the maintenance of the constant temperature of the body. The heat produced by the combustion of food usually maintains the body temperature well above the temperature of the surrounding air. By varying the circulation of blood in the vessels directly beneath the skin, the transfer of heat and a gradual cooling is effected. When sufficient cooling is not provided for in this way, the sweat glands take up the load, providing moisture for evaporation.

<u>Body Heat Gains</u>: The amount of heat produced in the body ranges from approximately 400 Btu/hr in a man seated at rest, to 1300 Btu/hr in a man doing hard manual labor. The human body will gain radiant heat if it is exposed to a surface at a higher temperature than that of the body. For instance, a body might gain as much as 1000 Btu/hr if it is exposed to direct sunlight. Inside walls, floors, and ceilings will radiate heat to the body if their temperatures are higher than that of the body.

Heat is also gained by the body through convection, that is, it is carried to the body by the air around it when the temperature of the air is higher than that of the body. This happens rarely indoors unless a person is standing directly over a heater or in the path of the warm air stream from a register.

<u>Body Heat Loss</u>: It has been found that the average normal person at rest has the greatest feeling of comfort when the body loses about

Figure 7


THE PHYSIOLOGICAL EFFECT OF THE ENVIRONMENT
(Ep) (ROBINSON)
Air Velocity 180 Ft/min
DRY BULB TEMERATURE °F
WET BULB TEMPERATURE °F
RELATIVE HUMIDITY
ZONE OF EQUILIBRIUM
ZONE OF BREAKDOWN
INDEX OF PHYSIOLOGICAL EFFECT Ep
Ep 125
Ep 150
Ep 175
Ep 200
R.H 10%
R.H. 20%
R.H. 30%
R.H. 40%
R.H. 50%
R.H 60%
R.H. 70%
R.H80
R.H 90
R.H. 100
10%
20%
30%
40%
50%
60%
70%
80%
90%
100% R.H
50
75
100
125
150
175
200
130
120
110
100
90
80
70
60
70
80
90
100

160 Btu/hr by radiation, 140 Btu/hr by convection to air, and 150 Btu/hr by evaporation of moisture from the skin, lungs and clothing. Dr. D. F. Dubois found through experiments that in standard conditions of light clothing, in air at 74°F. and 50% relative humidity, evaporative losses amount to 25% of the total heat loss, i.e., 100 Btu/hr. (11).

Insensible perspiration occurs from all parts of the body surface by convection and radiation. Raber and Hutchinson state that, "Under ordinary atmospheric conditions, radiation accounts for about 60% of the total heat loss from the body, about 240 Btu/hr from the normal person."(47). To distinguish between convection and radiation is difficult. The ratio between the two greatly depends on the conditions of the environment and the activity of the subject. Dr. Dubois suggests that 66% is lost by radiation and 15% by convection. "At the onset of shivering, the radiation fraction dropped to 49% (for the nude subject) while loss by convection increased to 29%." (11).

Example:

$$q_r = .172\, A_b \left\{ \left(\frac{T_b}{100}\right)^4 - \left(\frac{T_w}{100}\right)^4 \right\} = .172 \times 18.7 \times \left\{ \left(\frac{543}{100}\right)^4 - \left(\frac{534}{100}\right)^4 \right\}$$

$= 195$ Btu/hr at normal temperature by radiation

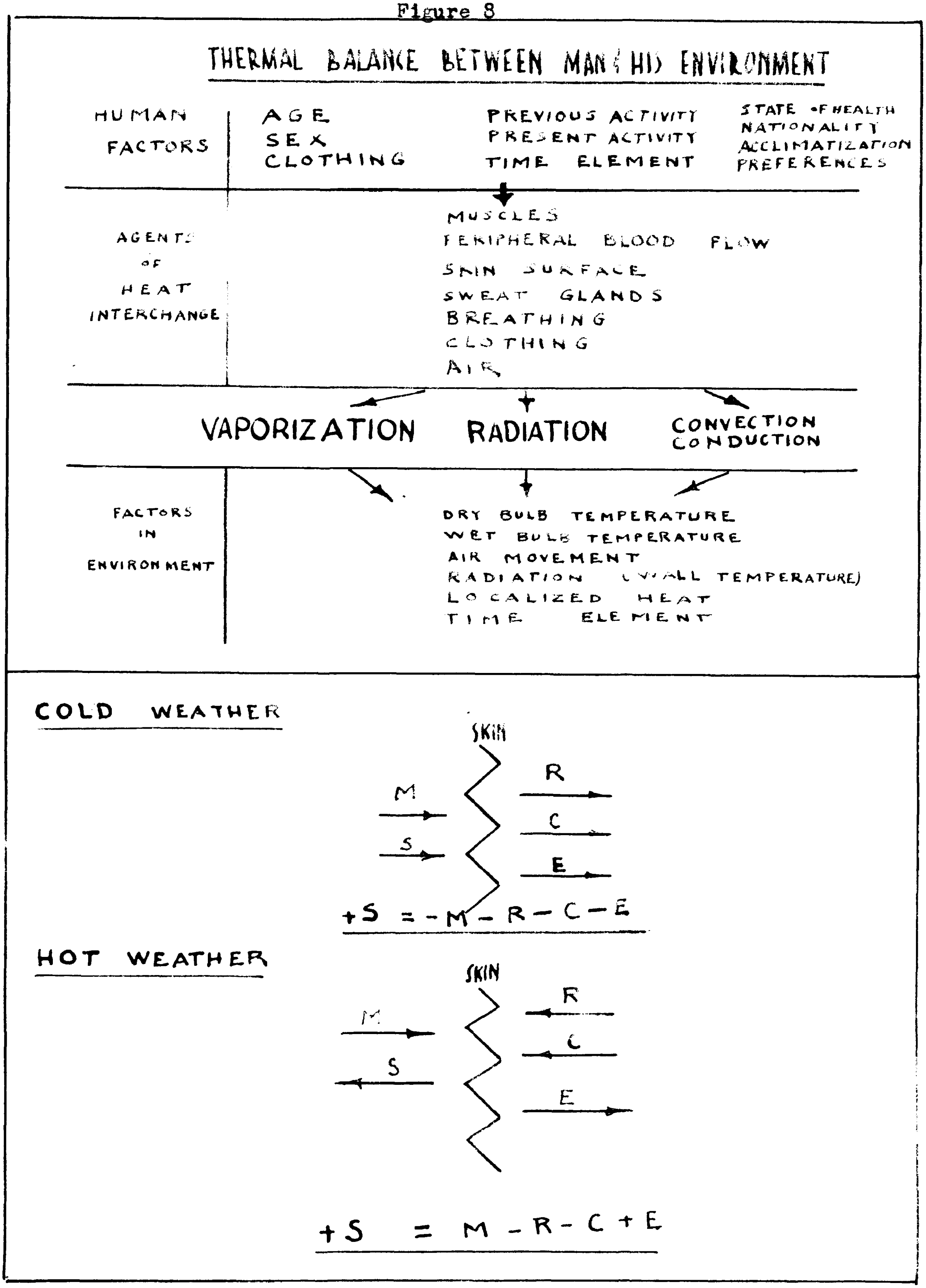
$$q_c = h_c A_b (t_b - t_a) = .67 \times 18.7 \times (83-74) = 116.60 \text{ Btu/hr}$$

by convection,

where we find the ratio between radiation and convection to be 50%-25%.

<u>Clothing as an Insulating Medium</u>: Clothing controls body heat loss by varying thermal resistance and film coefficient of heat transfer across the skin surface. Limitations as to practical types of clothing are necessary for maintaining a reasonable freedom of movement and the thermal capacity of the insulating layer sufficiently low in order to keep the body

Figure 8


THERMAL BALANCE BETWEEN MAN & HIS ENVIRONMENT
HUMAN FACTORS
AGE
SEX
CLOTHING
PREVIOUS ACTIVITY
PRESENT ACTIVITY
TIME ELEMENT
STATE OF HEALTH
NATIONALITY
ACCLIMATIZATION
PREFERENCES
AGENTS OF HEAT INTERCHANGE
MUSCLES
PERIPHERAL BLOOD FLOW
SKIN SURFACE
SWEAT GLANDS
BREATHING
CLOTHING
AIR
VAPORIZATION
RADIATION
CONVECTION
CONDUCTION
FACTORS IN ENVIRONMENT
DRY BULB TEMPERATURE
WET BULB TEMPERATURE
AIR MOVEMENT
RADIATION (WALL TEMPERATURE)
LOCALIZED HEAT
TIME ELEMENT
COLD WEATHER
SKIN
M
S
R
C
E
+S = -M - R - C - E
HOT WEATHER
SKIN
M
S
R
C
E
+S = M - R - C + E

temperature at a steady level. The psychological and physiological factors, including sensation, also play an important part in the design and material of the clothing used.

L. H. Newburgh found that sweating is a major hazard in overheating. (43). This has been emphasized in recent studies of arctic clothing. To explain, if a heavily clothed man in the Arctic exercises vigorously without removing or opening his clothes, he will sweat profusely. The water vapor will condense in the cold layer of clothing and give back some of the heat of condensation to the body. When the man stops exercising, the insulating properties of the clothes are harmful and heat loss is increased by the increased vaporization from the damp clothes. The same thing, in a lesser degree, can happen when going from a warm to a cold room.

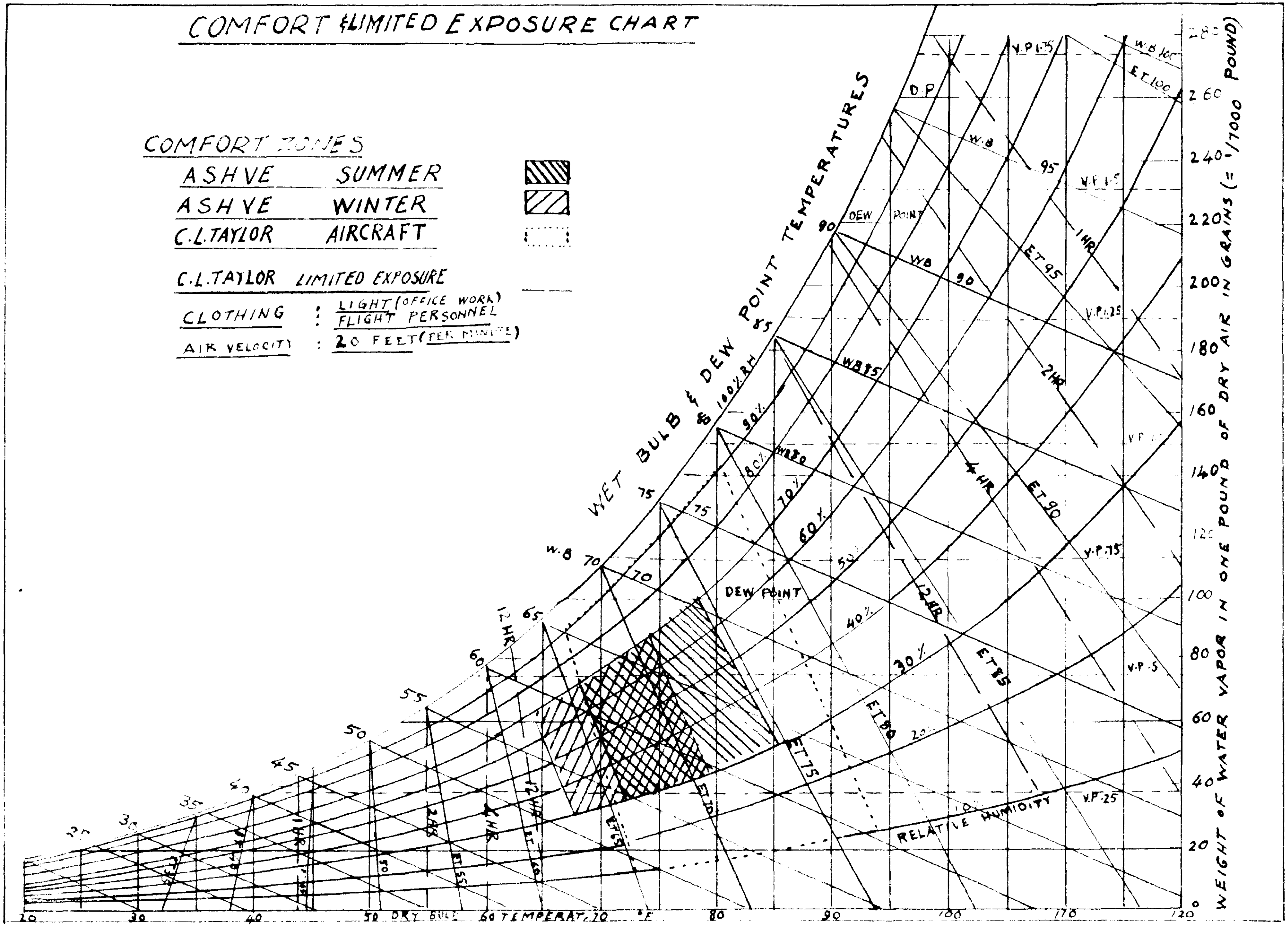
<u>Heat Interchanges</u>: The fundamental thermodynamic processes connected with heat exchange between the body and its environment may be described mathematically by the formula:

$$+ M = \pm S + E \pm R \pm C$$, where

M = rate of metabolism (heat produced within the body)
S = rate of storage (change in intrinsic body heat)
E = rate of evaporative heat loss
R = rate of radiative heat loss or gain
C = rate of convective heat loss or gain.

Metabolism heat is always positive, whereas storage heat may be either positive or negative, depending on whether heat is being stored or dissipated according to the rise or fall in the temperature of the body. Evaporative heat is always positive in the ordinary case when the dew point of the air is below the body surface temperature. This loss is balanced by the heat of metabolism. The heat of radiation and convection are positive when the surface temperature of the body is above that of the surrounding surfaces, and negative when it is below.

Figure 9


COMFORT & LIMITED EXPOSURE CHART
COMFORT ZONES
ASHVE SUMMER
ASHVE WINTER
C.L.TAYLOR AIRCRAFT
C.L.TAYLOR LIMITED EXPOSURE
CLOTHING : LIGHT (OFFICE WORK) FLIGHT PERSONNEL
AIR VELOCITY : 20 FEET (PER MINUTE)
WET BULB & DEW POINT TEMPERATURES
DRY BULB TEMPERATURE °F
WEIGHT OF WATER VAPOR IN ONE POUND OF DRY AIR IN GRAINS (= 1/7000 POUND)
RELATIVE HUMIDITY
DEW POINT

Heat Transfer by Convection: Convection is the transfer of heat from the flow of currents within a fluid body, whether the fluid is gas or liquid. When the lower portion of a fluid body is heated, it expands, becomes lighter and rises. The cooler and more dense portions of the fluid will replace the lighter ones. The constant flow, convection current, will convey heat from the hot surface to the cooler portions.

The rate of heat gain or loss from the human body by convection is the related function of four variables:

1-The average temperature of the skin and clothing.
2-The total area of the skin and clothing.
3-The Dry Bulb temperature of the air.
4-The air-motion in relation to the body.

The equation derived for an average person and for the air velocities from 9 feet per minute to 520 feet per minute is:-

$q_c = h_c A_c \sqrt{V} (t_b - t_a)$, where

q_c = rate of heat loss by convection
h_c = average film coefficient of heat transfer by convection from a clothed body and for an air velocity of 1 ft/minute in Btu/hr ft^2 °F.
A_c = area transferring heat by convection in ft^2
V = air velocity in feet per minute
t_b = average temperature of the skin and clothing in °F.
t_a = dry bulb temperature of the air in °F.

To determine the area of the body transferring heat by convection A_c, we use the Dubois formula (11) which is a function of area of the body A_b foot square, the height of the body Z in inches, and the weight of the body W in pounds:-

$A_b = 0.108 Z^{.725} W^{.425}$

Again using the Dubois formula, we find that the convection of the body in a standing position is 90% of the total area A_b.

Heat Transfer by Radiation: The effective radiating area of the skin and clothing is assumed to be between 70 and 75% of the total area of the body. The general equation of heat transfer by radiation, if we eliminate all convection transfer, is:-

$q_r = \sigma (T_1^4 - T_2^4) F_e F_a$, where

q_r = rate of heat transfer by radiation in Btu/hr sq. ft.
σ = (Stephan-Boltzmann constant) .174 x 10^{-8} Btu/hr ft^2 R^4
T_1 = absolute temperature of the body in ^oR.
T_2 = absolute temperature of the enclosure in ^oR.
F_e = factor allowing for emissivity of the two surfaces (body and surroundings) for complete blackness of body
F_a = configuration factor or angle factor for the position of the body relative to the surrounding surfaces.

Hutchinson treated the equation of heat radiation in a general way, considering several factors which other engineers tried to eliminate:-

Radiation Equation:-

$d_q = \sigma$ [illegible] dA , where

intensity $d_q = \sigma T_1^4 P_1 dA_1$

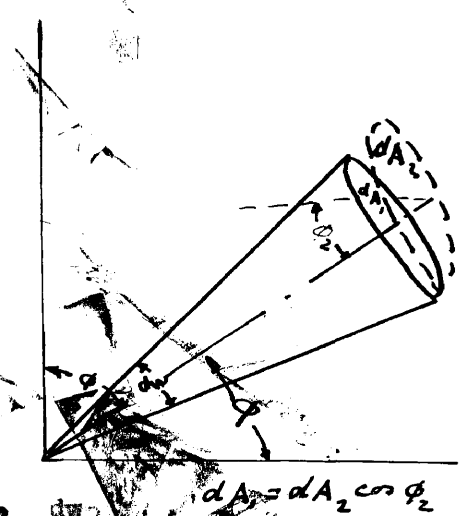
and $d^2q =$ [illegible] $dA_1 \; dw$

$= i dA_1 \; dw$

as $i = i_n \cos \phi$

$i_n = \dfrac{\sigma e_1 T_1^4}{\pi}$

thus $d^2q = \sigma e_1 T_1^4 dA_1 \cos \phi_2 \dfrac{dw_2}{\pi}$



The shape factor $dw_2 = \dfrac{\cos \phi_2 \, dA_2}{r^2}$

$$d^2q = \sigma e_1 T_1^4 dA_1 \times \frac{\cos \phi_1 \cos \phi_2 \, dA_2}{\pi r^2}$$

$$F \, dA_2 \, dA_1 = \frac{\cos \phi_1 \cos \phi_2 \, dA_2}{\pi r^2} = \frac{1}{dA_1} \; \frac{\cos \phi_1 \cos \phi_2 \, dA_1 \, dA_2}{\pi r^2}$$

and $$F \, dA_1 \, dA_2 = \frac{1}{dA_2} \; \frac{\cos \phi_1 \cos \phi_2 \, dA_1 \, dA_2}{\pi r^2}$$

$$F \, dA_2 \, dA_1 \, dA_1 = F \, dA_1 \, dA_2 \cdot dA_2$$

This relationship is known as the reciprocity theory and is used to evaluate shape factors of a system.

When dA is completely enclosed by other surfaces, the sum of the shape factors of enclosing surfaces with respect to the energy leaving dA, should equal 1.

$$d^2 q_1 = \sigma e_1 T_1^4 \, dA_2 \, F dA_2 \, dA_1$$

which is the fraction of heat energy leaving dA_1 and strikes $d A_2$ while

$dq = \sigma e_1 T_1^4 \, dA_1 \, (F_{A_2} \, dA_1)$ is the energy emitted by area dA_1 in the direction of finite area A_2. The energy emitted by one surface in the direction of another surface can be calculated by

$$q = E_1 \, A_1 \, F_{A_2 A_1} \quad \text{where}$$

$$F_{A_2 A_1} = \frac{1}{A_1} \int_{A_1} \frac{\cos \phi_1 \cos \phi_2 \, dA_1 \, dA_2}{\pi r^2}$$

If we consider reflective terms as having no importance (providing that room surface emissivity exceeds .93), the total energy emitted from A_2 to A_1 equals: $q = \sigma e_1 \, e_2 \, T_1^4 \, A_1 \, F_{A_2 A_1}$

and the net radiant transfer from A_1 to A_2 is:

$$q = \sigma e_1 \, e_2 \, A_1 \, F_{A_2 A_1} (T_1^4 - T_2^4)$$

Reducing this equation to simple form:-

$$q = e_1 \, e_2 \, h_r \, A_1 \, F_{A_2 A_1} (T_1 - T_2) \quad \text{where}$$

$$h_r = \left[\frac{\sigma (T_1^4 - T_2^4)}{T_1 - T_2} \right]$$

If the human body is placed within several completely enclosing black surfaces, which absorb all incident energy, the rate of heat loss by radiation from the body is:

$$q_r = 0.174 \times 10^{-8} \, e_b A_{br} ((F_{b1} (T_b^4 - T_1^4) + F_{b2} (T_b^4 - T_2^4) + F_{b3} (T_b^4 - T_3^4) \ldots\ldots))$$

where q_r = rate of heat transfer by radiation Btu/hr

$e_b = F_e$ = the emissivity of the human body

A_{br} = the effective radiation area of the skin and clothing in square feet (70--75% of A_b)

F_{b1}, F_{b2}, F_{b3}, etc. = the configuration factors or interchange factors depending on the geometry of the human body and enclosing surfaces

T_b = the absolute temperature of the human body °R.

T_1, T_2, T_3, etc. = the absolute temperature of the different enclosing surfaces °R.

Now, if we substitute for the actual black enclosure another fictitious black enclosure of constant temperature T_R, where the heat loss from the human body by radiation is the same as in the actual enclosure, the equation becomes:-

$$q_r = .174\ (10)^{-8}\ e_b\ A_b\ (T_b^4 - T_R^4),\ \text{then:-}$$

$$T_R^4 = (\ F_{b1}\ T_1^4 + F_{b2}\ T_2^4 + \ldots\ldots\ \text{etc.})\ \text{where}$$

T_R = the mean radiant temperature of an environment (black enclosing surfaces).

The mean radiant temperature, as defined before, is the temperature of a uniform, black enclosure with which a human body would exchange the same amount of energy by radiation as in the actual environment.

For small temperature difference and at common temperature, we find

$$\frac{.174 \times 10^{-8}\ (T_b^4 - T_R^4)}{T_b - T_R} \doteq 1.0 \doteq h_r$$

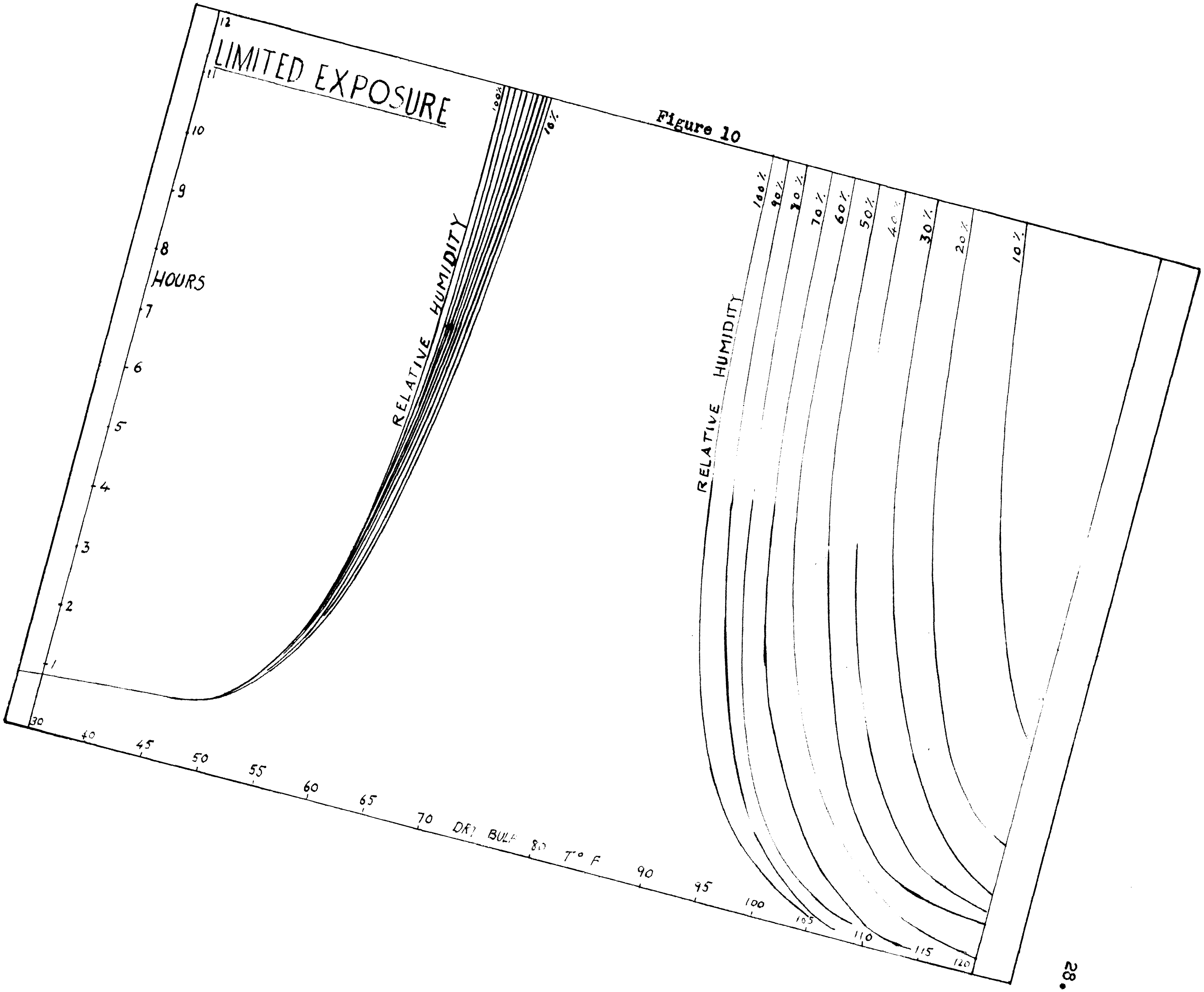
For engineering purposes, the rate of heat loss by radiation from the human body in an enclosure completely black is reduced to:-

$$q_r = e_b\ A_b\ (\ T_b - T_R\ )$$

Comparing this with Hutchinson's equation, we find that:-

$$e_b = e_1\ e_2\ h_r\ F_{A_2}\ A_1$$

Figure 10


LIMITED EXPOSURE
HOURS
12
11
10
9
8
7
6
5
4
3
2
1
RELATIVE HUMIDITY
100%
10%
RELATIVE HUMIDITY
100%
90%
80%
70%
60%
50%
40%
30%
20%
10%
30
40
45
50
55
60
65
70
DRY BULB 80 T° F
90
95
100
105
110
115
120

B. THE COMFORT HEAT EQUATION.

The comfort heat zone is the balance between heat loss and heat gain, maintaining the equilibrium of the human body within the comfort region. Messrs. J. D. Hardy and G. F. Soderstrom (Journal of Nutrition 16:493, 1938) demonstrated, by experiments with two normal men as subjects, the relationship of the factors involved in heat loss to the environmental temperature (20). As the heat production of the men was raised above the basal level, the comfort zone of easy balance was shifted to a cooler region.

In order to determine the comfort zone of thermal balance, the engineer must reduce many variables in order to study each factor separately. Dr. Eugene F. Du Bois of New York experimented for several days with a normal man in the Sage Respiration Calorimeter at environmental temperatures from 72°F. to 92°F. He eliminated the effect of clothing by using nude subjects, the effect of activity and digestion by having the subjects in basal condition, and radiation by having the air and wall temperatures identical. Humidity and air movement were low. The experiments, running for periods of two and three hours, followed a preliminary adjustment period in the box from one to two hours. The subject was comfortable, except in the cold zone, when he was on the verge of shivering, and in the hot zone, when he was sweating profusely.

Physiologists tell us that man responds to his environment as a whole rather than in a way which can be predicted by a knowledge of the separate variables which make up his environment. Therefore, the comfort zone may be considered as the area wherein everyone is neither too cold nor too hot.

In the region of the comfort zone, when the calorimeter temperature was over 82°F. and less than 90°F., the heat loss and heat production proved to be almost equal. In the comfort zone, physical adjustments, through slight changes in the peripheral blood flow and a slight increase in vaporization, are made so gradually that we are not conscious of them. Above this zone, the discomfort of profuse sweating warns us to slow down muscle action and to eat foods with less calories. The skin is kept at a fairly uniform temperature through the vaporization of sweat. Therefore it is not an indicator of stress. Below the comfort zone, however, the blood flow to the skin is almost shut off and the skin will not act as a heat transfer medium, but rather as an insulator, an excellent indicator warning us to exercise, don more clothing and eat "heavier" foods.

Under normal conditions, deaths from summer "heat waves" where the temperature exceeds 88°F., seldom exceed .02%. Elizabeth Schickele, (43) after studying the causes of death of 157 young men, put the equation of the heat-death line as:-

$$T = 119 - 2V$$

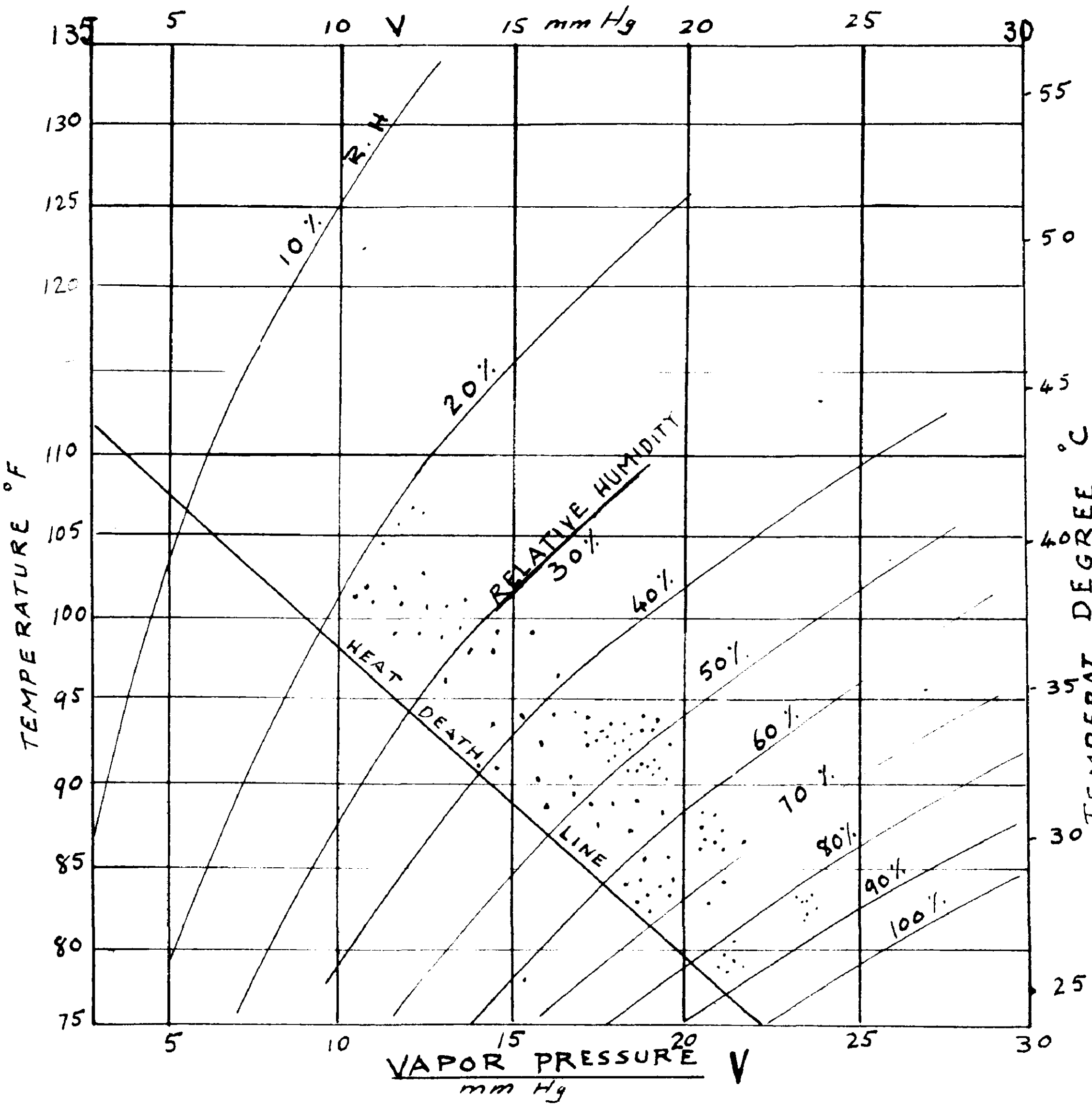
where: T = dry bulb temperature °F.

V = vapor pressure in millimeters of mercury

The APHA committee on atmospheric comfort, in a recent survey, found that an 85 degree effective temperature is the limit for continuing light work without immediate physiological strain and 80 degrees an effective temperature for heavy work. These two effective temperatures should serve as standards in various industries.

The limited and qualified conclusions of APHA reports interested the physiologists in the industrial heat problem. Additional thermal

Figure 11



HEAT DEATH LINE TEMP = 119 - 2V

SCHICKLE EXP.

standards are required in industry to cover the extensive region of environmental conditions lying between the upper limit of physiological tolerance and the lower limit of simple comfort. A definite solution cannot be reached without the complete cooperation between physiologists and psychologists.

Individual industries interested in the comfort of their workers must establish practical standards qualifying with the requirements of each industry. Cooperation of industrial, medical, engineering, personnel departments, physiologists and psychologists is needed for evaluating the suitability of thermal environments for work, or rest, and the comfort and health of the worker.

<u>Evolution of the Body Heat Comfort Equation</u>: In standard conditions where the state of comfort exits, the rate of body heat loss by radiation in Btu/hr (per °F. difference) between the clothed body surface temperature and the mean radiant temperature, can be considered practically constant. Similarly, the rate of body heat loss by convection in Btu/hr (per °F. difference) between the clothed body surface temperature and the ambient air temperature can be taken as constant.

The rates for the clothed subject in a room with air and walls at 74°F. are:-

By radiation: $q_r^1 = 200 \div (83 - 74) = 22.2$ Btu/hr (°F.)

By convection: $q_c^1 = 100 \div (83 - 74) = 11.1$ Btu/hr (°F.)

$$\left\{(T_{b1} - T_{a1}) - (T_{b2} - T_{a2})\right\} = 22.2\left\{(T_{b2} - mrt_2) - (T_{b1} - mrt_1)\right\}$$

$$11.1\Delta(T_b - T_a) + 22.2\Delta(T_b - mrt) = 0$$

$$-\frac{\Delta T_a}{\Delta mrt} = 2$$

$$\underline{T_b = AT_a + B\, mrt + K}, \text{ where } \frac{\Delta T_a}{-\Delta mrt} = K$$

Experimental work done by ASHVE indicates that a change in mrt of 1°F. requires an opposite change in effective air temperature of approximately $.5^\circ$F. The term "effective temperature" includes the influence of humidity but within the range of inside conditions usually found in comfortably heated or cooled rooms. A 0.5°F. effective temperature change would represent a variation in the dry bulb temperature of:-

$$1.75 \times .5 = 9^\circ \text{ F.}$$

Bedford Equation: Thomas Bedford found that 1°F. change in the dry bulb temperature required an opposite change in the mean radiant temperature of 1°F. The following equation is based on the theoretical solution of comfort relationship:-

t_b = body surface temperature = $.21\ t_a + .54$ mrt $+ 25.9$

The approximate relation can be considered as: t_a + mrt $= 140$

where mrt = mean radiant temperature $= F_{a1}\ t_1 + F_{a2}\ t_2 + F_{a3}\ t_3 + \ldots$etc.

F_{a1} is the shape factor of surface 1 of temperature t_1.

The shape factor relative to the surface area has already been calculated and plotted in diagrams by Raber & Hutchinson (47). The evaluation of the shape factor with the human body as the subject, with reference to ceiling, floors, and walls, was treated accurately.

Mackey Comfort Heat Equation: Professor C. C. Mackey, of the Cornell University Engineering Experimental Station, found that if we add the results of heat loss by convection:

$$q_c = h_c\ A_c \sqrt{V}\ (t_b - t_a)$$

to those of the equation of radiation heat transfer:

$$q_r = e_b\ A_b\ (\ t_b - t_R\ )$$

we can obtain the hourly rate of heat transfer (loss) from the human body by combined convection and radiation:

$$q_t = q_c + q_r = h_c A_c \sqrt{V}(t_b - t_a) + e_b A_b (t_b - t_R).$$

t_a and t_R with the relation of air velocity give us a new item to consider:

t_o = the operative temperature.

The operative temperature is based on three main factors: the air temperature t_a, the mean radiant temperature t_R, and the velocity of air at the time of measurement V. The still air velocity is V_o = 15 ft/minute. Assume firstly that t_o, the operative temperature = t_a = t_R and the air velocity V_o = V = 15 ft/minute, then the total rate of heat loss due to the combined convection and radiation from the human body is the same as in our fictitious environment.

In other assumptions, in either environment, the hourly loss of heat by combined convection and radiation is:-

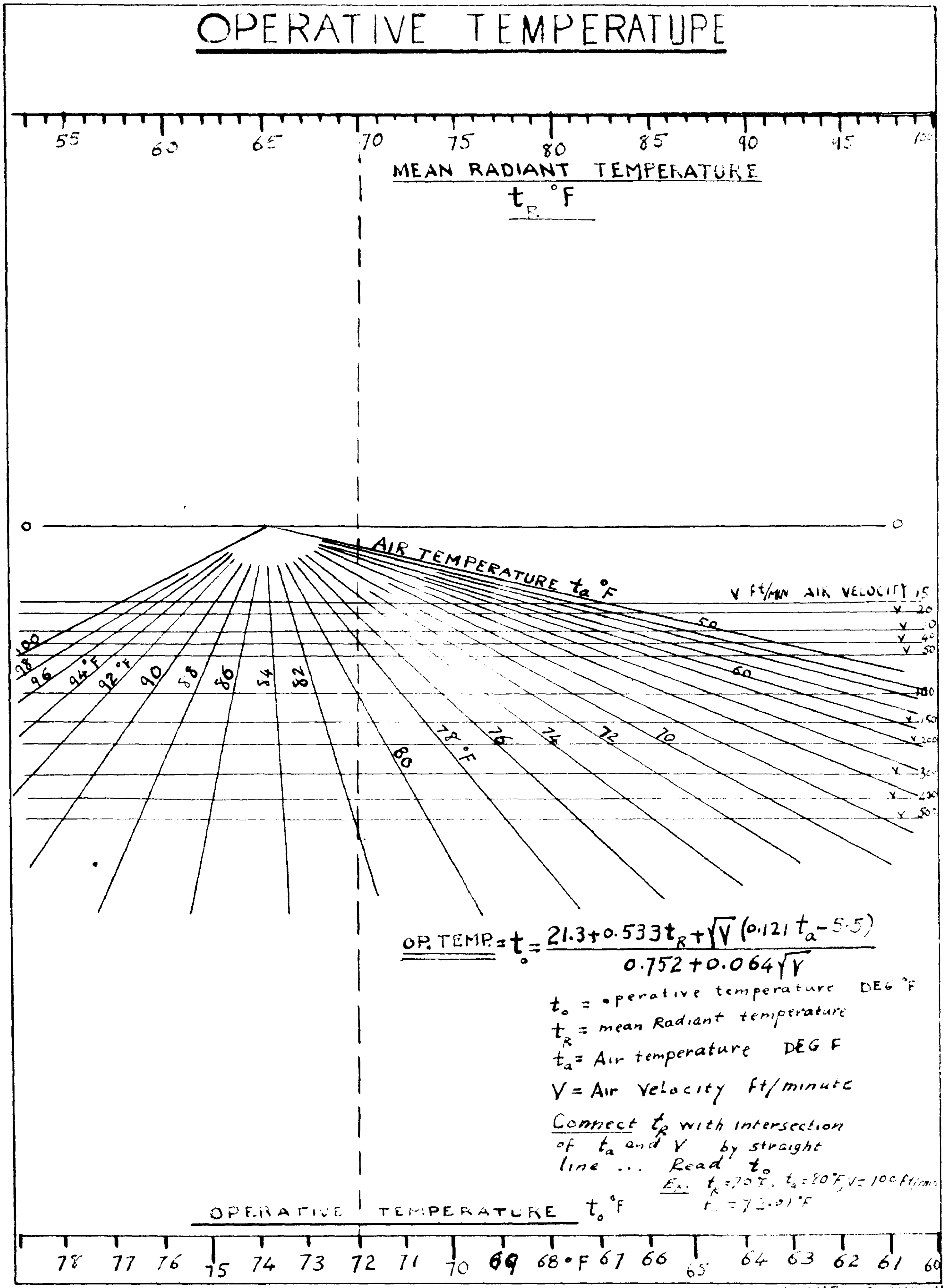
$$q_t = h_c A_c \sqrt{V} (t_b - t_a) + e_b A_b (t_b - t_R)$$
$$= h_c A_c \sqrt{V_o} (t_b - t_o) + e_b A_b (t_b - t_o), \text{ then}$$

$$t_o = \left\{ \frac{t_b (\sqrt{V_o} - \sqrt{V}) + \sqrt{V}\, t_a + \left(\frac{e_b A_b}{h_c A_c}\right) t_R}{\sqrt{V_o} + \frac{e_b A_b}{h_c A_c}} \right\}$$

= the operative temperature

In taking a particular case, the operative temperature of a standard environment (black enclosing surface) where the temperature of the air is equal to the mean radiant temperature, and in which the air velocity equals the standard velocity of 15 ft/minute, and where the total loss of heat from a given human body is at the same rate as in the actual environment, when the average temperature of the skin and

Figure 12



AFTER:- MACKEY·CORNELL U.

the clothing is the same in both, then the operative temperature can be defined as the air temperature t_a or the mean radiant temperature t_R (23).

Much experimental research has been done by the Pierce Laboratory in environments where $t_a = t_R = 70°F$.

and $V_o = 15$ ft/minute,

the result was that $q_c = 140$ Btu/hr

and $q_r = 160$ Btu/hr.

If we tried to solve the equation of operative temperature, on the basis of Pierce's results in these particular conditions to determine the ratio between the emissivity factor of the body (times the area of the body) and the convection coefficient (times the area), then

$$\frac{e_b A_b}{h_c A_c} = 4.43 \text{ , then}$$

$$t_o = t_b(.467 - .121\sqrt{V}) + .121\sqrt{V}\,t_a + .533\,t_R$$

It is found that as the operative temperature increases, the average temperature of the skin and clothing t_b increases until the zone of cooperative regulation is reached. There is a linear relation between t_o and t_b which leads to the equation, found experimentally:-

$t_b = a + b\,t_o$, where constants $a = 45.5$ and $b = .53$

Substituting this result in the operative temperature equation, we find that:

$$t_o = \frac{21.3 + .533\,t_R + \sqrt{V}(.121\,t_a + 5.5)}{.752 + .064\sqrt{V}}$$

(Professor Mackey used a graphical solution, known as the "operative temperature chart", qualifying with the previous equation.)

Ignoring the slight differences between t_o and t_b, Pierce Laboratory states, "When a given operative temperature is produced by low air temperature and high mean radiant temperature, then t_b (the average temperature of skin and clothing), will be slightly lower than when the same operative temperature is produced by nearly equal air and mean radiant temperature." For a limited equation, t_o (as given above) can be derived when assuming that we are going to maintain our environment at still air conditions (V = 15 ft/minute).

Then $t_o = .533\ t_R + .467\ t_a$

$$t_a + 1.14\ t_R = \frac{t_o}{.467}$$

This is the comfort equation as devised by Professor Mackey at Cornell University in 1944, where:

t_a = dry bulb temperature of air

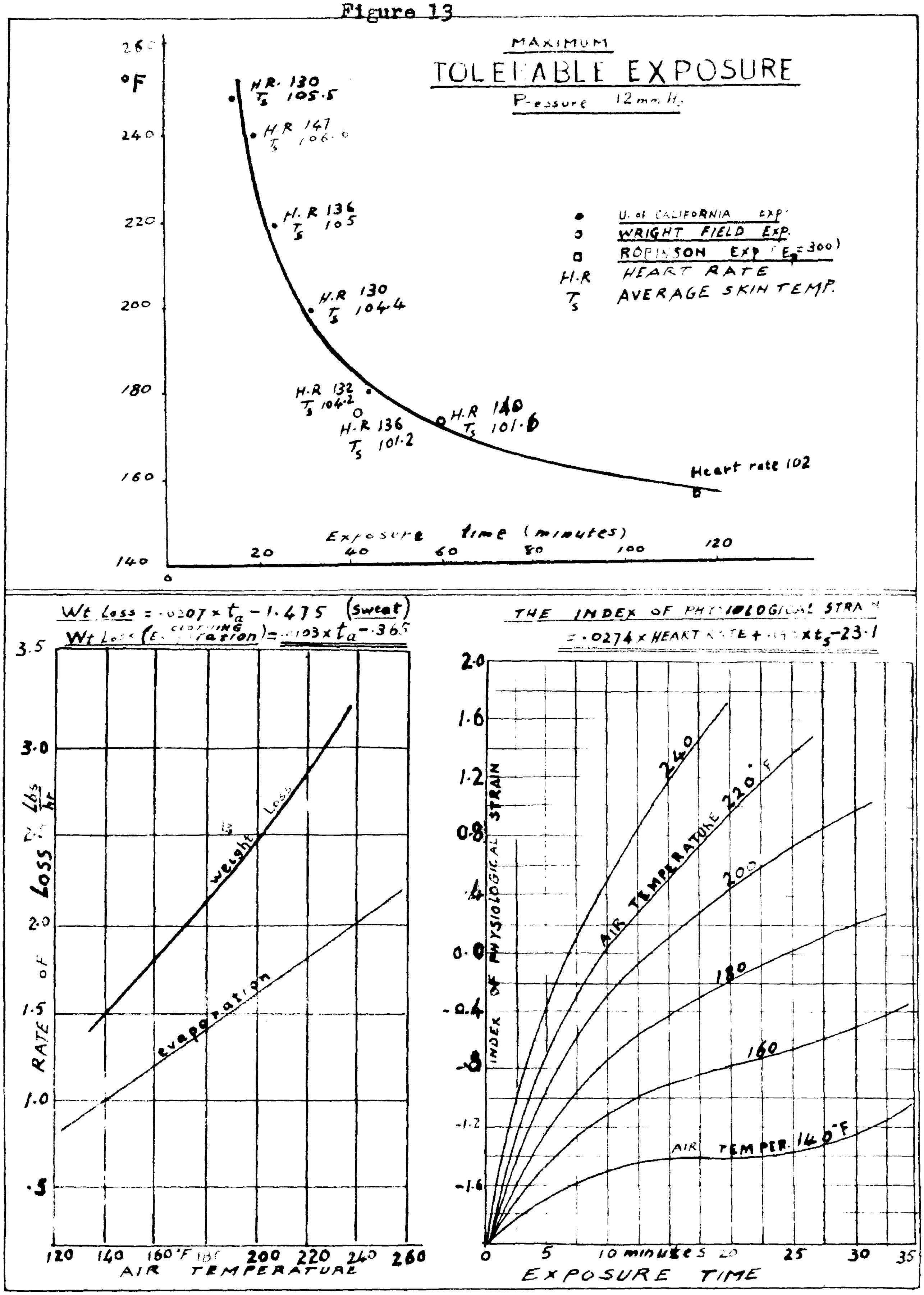
t_R = mean radiant temperature

t_o = the operative temperature

By using the operative temperature, the engineer may determine the relation and values of air temperature, mean radiant temperature and air movement which produce the same physical environment when that environment is created by heating systems.

<u>Conclusions</u>: The author contends that a practical definition of the comfort zone is that region wherein the most people find the least discomfort. Thereby, we can determine the factors of least discomfort to a group by statistical optimum. Experiments show that a 6.5 degree difference in temperature causes the normal person to begin to feel the difference between warm and cool. In other words, this temperature may be considered the boundary between comfort and discomfort to the majority. The author further contends that the question of inside temperature in relation to the outside temperature variations throughout the day need

Figure 13


MAXIMUM
TOLERABLE EXPOSURE
Pressure 12 mm Hg
°F
H.R. 130
T_s 105.5
H.R 147
T_s 106.0
H.R 136
T_s 105
H.R 130
T_s 104.4
H.R 132
T_s 104.2
H.R 136
T_s 101.2
H.R 140
T_s 101.6
Heart rate 102
U. of CALIFORNIA EXP.
WRIGHT FIELD EXP.
ROBINSON EXP (E_s = 300)
H.R HEART RATE
T_s AVERAGE SKIN TEMP.
Exposure time (minutes)
Wt Loss = .0207 × t_a − 1.475 (sweat)
Wt Loss (Evaporation) = .0103 × t_a − .365
RATE OF LOSS Lbs/hr
weight Loss
evaporation
AIR TEMPERATURE
THE INDEX OF PHYSIOLOGICAL STRAIN
= .0274 × HEART RATE + ... × t_s − 23.1
INDEX OF PHYSIOLOGICAL STRAIN
240
AIR TEMPERATURE 220°F
200
180
160
AIR TEMPER. 140°F
10 minutes 20
EXPOSURE TIME

not be taken too seriously when working with people spending eight or more hours indoors, as they would have no physiological consciousness of the outside temperature during this time. If this be accepted as fact, we can see that a 75°F. dry bulb temperature would be satisfactory for indoor comfort all year around with only slight differences in relative humidity: 50% in summer and 30% in winter.

For example: In a survey of two buildings, each thirteen years old, in Washington, D. C. with a total occupancy of over two thousand people ranging in ages from 18 to 80, ninety percent of the people were satisfied with a temperature of 75°F. all year around with 50% relative humidity in summer and 30% relative humidity in winter. The safe level of exposure depends upon the rate and duration of work and the age and physical condition of the worker. The relative humidity is the factor determining the comfort heat temperature by adjustment of the indoor dry bulb.

C. HUMAN ABILITY TO WITHSTAND EXTREME TEMPERATURES.

The associated fields of medical climatology (bioclimatology), heating and air conditioning engineering, physiology of body temperature regulation, and biophysics of heat exchange have cooperated to give humans their thermal heat requirements.

The bioclimatologist is concerned with climatic and geographical observation and classification in terms of biological adaptation. The engineer deals with specification and control of the artificial environment, which is the effect of microclimate. The biologist, whether he be morphologist, physiologist, or psychologist, looks at the characteristics of the organism which determine its adaptation and adjustment to the environment, either natural or artificial. The biophysicist acts to make internal relationship between these organisms and the environment properties on a general physical basis. Information from the abovementioned sources is necessary in order to complete formulas for scientific and technological equations.

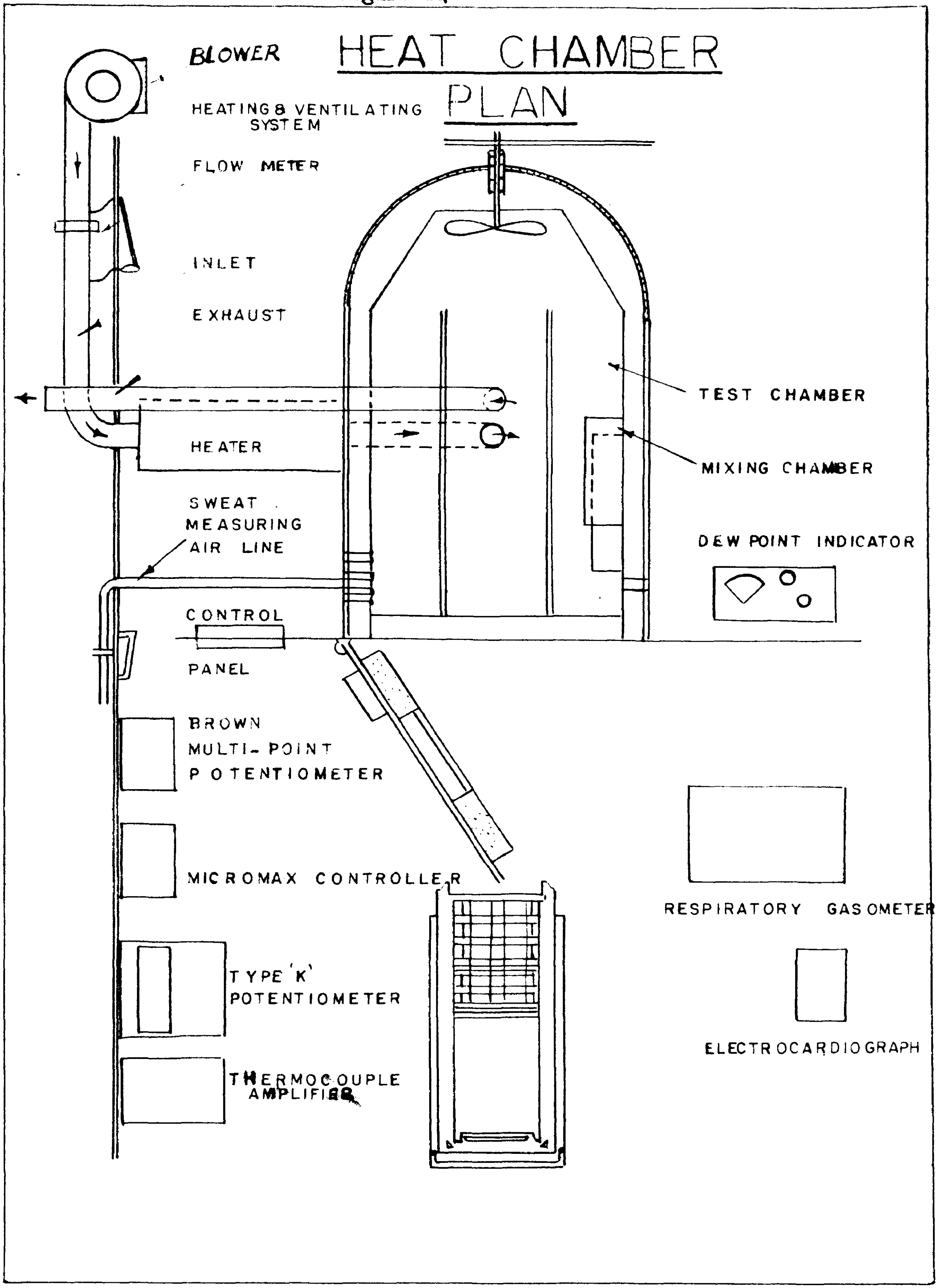
<u>Thermal Heat Requirements</u>: Mc Connell, Houghton, Yoglow, and coworkers, found the heart rate, rectal temperature, systolic pressure, and sweat loss to be proportional to relative humidity and dry bulb temperature (62). Their experiments were carried out with subjects exposed to 157°F. dry bulb, 15% relative humidity, and 112.5°F. dry bulb at 100 % relative humidity. The result, leading to effective temperature scale based upon the equivalent in thermal sensation, proved the effective temperature scale to be a correlate of the physiological response. The angle made by the effective temperature lines and dry bulb temperature lines on a standard psychometric chart gave the following equation:

$$\text{Angle (degrees)} = 42.0 + .474(T_{\text{dew point}} - 74).$$

In 1945, Robinson, Turrell, and Gerking, by using a more elaborate physiological index, compounded of heart rate, skin and rectal temperature and sweat loss, established tolerance contours which agree with those determined by thermal sensation and work performance done by C. L. Taylor in 1946, who also combined sweat loss, heat rate and skin and rectal temperatures into a physiological index. (52)

Heat transfer measurements and calculations give the heat load in terms of sensible heat gain (convection and radiation). The relation of this measurement, as well as the ambient temperature, to the thermal strain from the heart rate and the skin temperature can be deter-

Figure 14


HEAT CHAMBER
PLAN
BLOWER
HEATING & VENTILATING SYSTEM
FLOW METER
INLET
EXHAUST
TEST CHAMBER
HEATER
MIXING CHAMBER
SWEAT MEASURING AIR LINE
DEW POINT INDICATOR
CONTROL
PANEL
BROWN MULTI-POINT POTENTIOMETER
MICROMAX CONTROLLER
RESPIRATORY GASOMETER
TYPE 'K' POTENTIOMETER
ELECTROCARDIOGRAPH
THERMOCOUPLE AMPLIFIER

mined from a graphic plot of heat load (or ambient temperature) and exposure time. The actually observed tolerance times are compared to the indexes of strain, heat load and time, and a tentative tolerance curve established.

Self-Experiment: On April 15, 1949, the author experimented upon himself to ascertain the effects of high temperature upon the human body in relation to medical, physiological, psychological and other changes taking place. This experiment took place under the supervision of Dr. Craig L. Taylor, Professor of the Biotechnology of the Human Environment, in his research laboratory at the University of California.

1. The heat chamber was a cylindrical form, 70" long and 46" in diameter with a total volume of 70.4 cubic feet. It was insulated to a thickness of 6" inside the steel wall with an inner wall lining of galvanized iron. It contained a blower and duct system running from the outside of the building to an inlet at the bottom of the chamber. A meter in the duct employed a calibrated orifice with a differential gage to facilitate reading of the air flow. The meter calibrated over a range of 60 to 90 cubic feet per minute (air density .0736 pound/ft^3). This was in accordance with the method outlined in ASME FLUID METER REPORT FOR 1937. The air flow was adjusted to the gage reading by a valve in the air duct. There were twelve strip heaters of .75 KW capacity each, mounted in a heater box, heating the intake air. The heating unit was controlled by a micromax recording potentiometer of the on-off type, supplemented by a manual drop control, the primary element being a small iron-constantan thermocouple located in the chamber. It was thus possible to control the dry bulb temperature to within 1.5°F.

For rapid entry and exit from the chamber, a chair mounted on wheels ran in a track upon a carriage outside the chamber. In order to enter, the chamber door was opened, the carriage rolled into position, engaging the track within the chamber, and the chair rolled into the chamber. In order to induce thermal baseline conditions prior to entry into the chamber, a canvas tent was constructed to fit over the author in the chair carriage. Suspension ropes permitted it to be hoisted out of the way when not in use. A thermostated heater and air blower ventilated the tent with about one air change every two minutes, maintaining the temperature between 80 and 85°F.

2. Thermal environment: Radiation exchange was measured by a Dunkle and Gier (University of California, Berkeley) thermopile radiometer for measurement of the chamber wall emissivity. A thermal anemometer (Seban, Hillendahl, Gallagher and London 1943) based on the change in the cooling rate of a heated sphere as a function of air velocity, gave the equivalent air speed at a given point. The cooling rate was:–

$$\frac{\log \left(\frac{u\,V_o}{u\,V}\right)}{\theta} = \phi\, V \left(\frac{\rho}{\rho_o}\right)\left(\frac{T}{T_o}\right)^{0.71} \quad \text{feet/minute}$$

where: $u\,V_o$ = micro volts reading of temperature difference between "hot ball" and ambient air at time zero
$u\,V$ = micro volt reading at time ϕ minutes
ρ/ρ_o = dimensionless air densities lb/ft^3
T/T_o = dimensionless actual and standard temperatures in degree R.
θ = minutes

The temperatures of the air and walls were measured in °F. by means of numerous iron-constantan thermocouples connected to a Brown Electronic Self-balancing Potentiometer-48 station indicating model. Sufficient thermocouples were available to record the temperature at many different locations within a short interval of time. One thermocouple was suspended within a cylindrical double-walled radiation shield made of cardboard, while others were hung free in the air, and wall thermocouples were attached to the metal surface.

Humidity values were obtained from wet and dry bulb temperatures as read from thermocouples. Mercury thermometers were mounted at the intake to the blower and in the outlet duct leading from the chamber. A standard wick system was applied to one of each pair of thermocouples, and a small thermocouple was attached to the surface of each thermometer bulb. Vapor pressures were read from a monogram relating wet-bulb temperature, dry-wet differential, and vapor pressure, derived from the Carrier equation (Diedericks and Andrae 1930). These values were checked by means of a dew point indicator (General Electric Cat. #59938230) to which sampling streams of air from the inlet and exhaust ducts and from the chamber proper were drawn. The accuracy of the Dew Point meter is about 1°F. with careful manipulation.

3. Physiological Determination: Skin temperatures were measured at nine locations on the body by means of iron-constantan thermocouples mounted in Guillemin holders especially designed to insure a firm contact by pressing the junction into the skin. Temperatures were indicated by the Brown Automatic Potentiometer.

Hardy and Dubois (11 & 20) gave the fraction of surface area to every anatomical region:-

Heart	.07	Hand	.05	Arm	.14	Thigh	.19
Calf	.13	Foot	.07	Trunk	.35		

Heart rate data was obtained in four ways: the observer taking wrist pulse rates, the subject taking his own wrist pulse, measurement of electrocardiograph records, and reading of the heart rate from the electrocardiograph pulse-meter, (30 second wrist-pulse counts).

To record changes in the electrical behaviour of the heart, a new direct-writing electrocardiograph was designed and built. Potentials, picked up from the skin surface by electrodes of the conventional type, were electronically amplified to a level suitable for driving a Brush direct ink-writing oscillograph. The frequency response of the instrument as a whole is limited at the high frequency end by the characteristics of the Brush recorder (eg. pen friction) to a value of approximately 70 cycles per second and at the low end to a value of approximately ½ cycle/second by the coupling circuits utilized in the amplifier. (Tested by Council of Physical Medicine in Journal of American Medical Assoc., Vol.134,455-1947).

Blood pressure was taken in the pre- and post-exposure periods by using a standard Taylor Aneroid Sphygmomanometer and stethoscope. The operator controlled the pressure in the cuff from his position outside the chamber and listened to the amplified sounds by means of earphones.

Respiratory rates, volumes and oxygen consumptions were recorded upon a moving strip-chart by a pen attached to the gasometer bell of a closed circuit metabolism apparatus. A plastic Douglas valve set with a rubber mouth bit and a pair of hose to the gasometer completed the respiration circuit. A soda-lime absorber in the expired airway removed CO_2 so that excursions of the gasometer bell gave tracings from which we obtained:-

1-Respiration rate: total cycles in unit time (minute)
2-Expired air volume: total volume in unit time
3-Oxygen consumption: slope of volume change with time determined by the mean positions of expiratory peaks in the respiration record.

Sweat loss was measured by weighing the subject on a standard Taylor platform scale before and after heat exposure.

4. Procedure for Experiment No. F-8:

Date: April 15, 1949

Subject: Abdallah Kamel Eid

Station: Dr. C. L. Taylor Experimental Laboratory,
University of California, Los Angeles

Age of Subject: 27 years

Weight of Subject: 175 pounds

Height of Subject: 70 1/2 inches

Surface Area of Subject: 21.3 square feet

Mean Ambient Air Temperature in °F: 180

Range of Inlet Humidities at Vapor Pressure = .49 in. Hg.
= 12.45 mm Hg.

Duration of Exposure: 29 1/2 minutes

Clothing: Brief cotton swimming trunks only

Wall Temperature: at t_a = 180, the wall temperature t_w rapidly approached air temperature and $t_a = t_w = 180$°F.

Air Movement: for $t_a = 180$
V = 40.6 ft/minute
= average of position effect

Humidity: as a function in sweat loss, depends on the ambient temperature and the duration of exposure:-
mean $P_{wc} = .15 + 1.3\ P_{wi}$

P_{wc} = time-weighted average of chamber vapor pressure (inches Hg.)

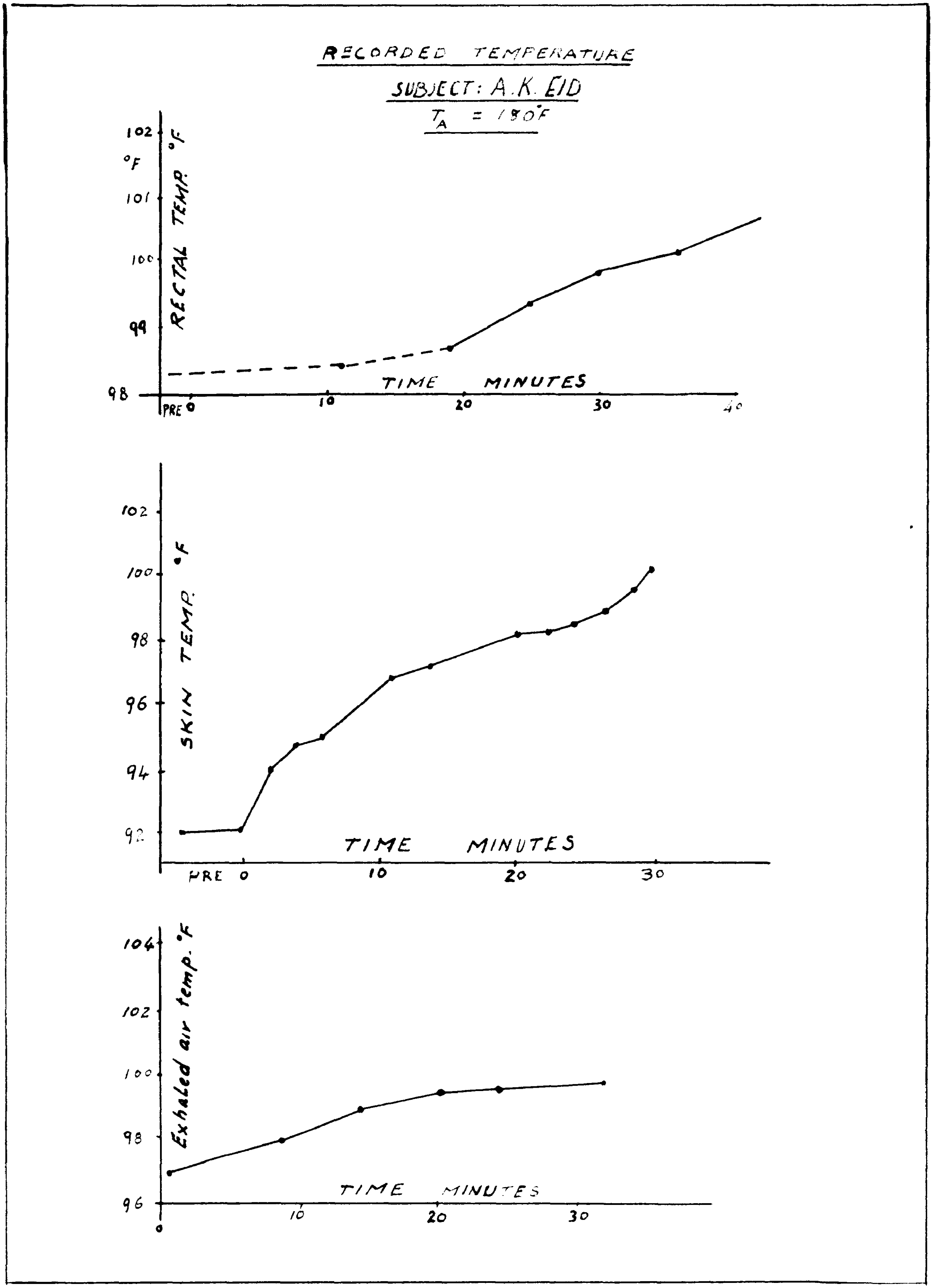
P_{wi} = average inlet vapor pressure (inches Hg.)

(There was no difficulty in measuring wet bulb temperatures up to 180°F. Dry Bulb)

The actual experiment proceedings were timed as follows:

1:00 P.M. Subject checked with student health office for approval, reported to laboratory, disrobed, inserted rectal thermocouple and was weighed in the nude.

Figure 15


RECORDED TEMPERATURE
SUBJECT: A.K. EID
T_A = 180°F
RECTAL TEMP. °F
°F
102
101
100
99
98
TIME MINUTES
PRE 0
10
20
30
40
SKIN TEMP. °F
102
100
98
96
94
92
TIME MINUTES
PRE 0
10
20
30
Exhaled air temp. °F
104
102
100
98
96
TIME MINUTES
0
10
20
30

1:30 - 2:00 p.m. Skin thermocouple harness was applied, subject was dressed and weighed with clothing. He was then seated in the chair mounted on the wheeled carriage and the canvas tent was lowered over the carriage.

2:00 - 2:30 p.m. Subject rested while pre-exposure measurements of skin temperature, respiratory rate and volume, etc. were taken.

2:30 - 2:59 pm. Tent was raised, thermocouple leads etc. were disconnected, and the subject rolled into the chamber which had been pre-conditioned to the desired temperature level. The time clock was started as the door closed on the subject. Leads were reconnected and the operators alerted to take readings. The first temperature reading was taken at the end of one minute exposure and all values obtained within this minute were recorded opposite exposure time 1. Skin temperatures were recorded every 2 minutes. During the intervals between the readings, the subject was directed to breathe through the valve set and periodically questioned regarding his subjective feelings and reactions.

2:59 - 3:30 p.m. At the end of the exposure time, connections to the subject were broken, the door was opened and he was rolled out onto the carriage. The elapsed time, as shown by the clock, was recorded at the instant of removal. The subject was weighed, undressed, toweled dry, and weighed again. He was then allowed to rest in a beach robe while further measurements of the pulse rate were made until his normal condition returned.

Figure 16 EFFECT OF EXTREME TEMPERATURE UPON RESPIRATION, HEART RATE, AND SWEAT RATE.

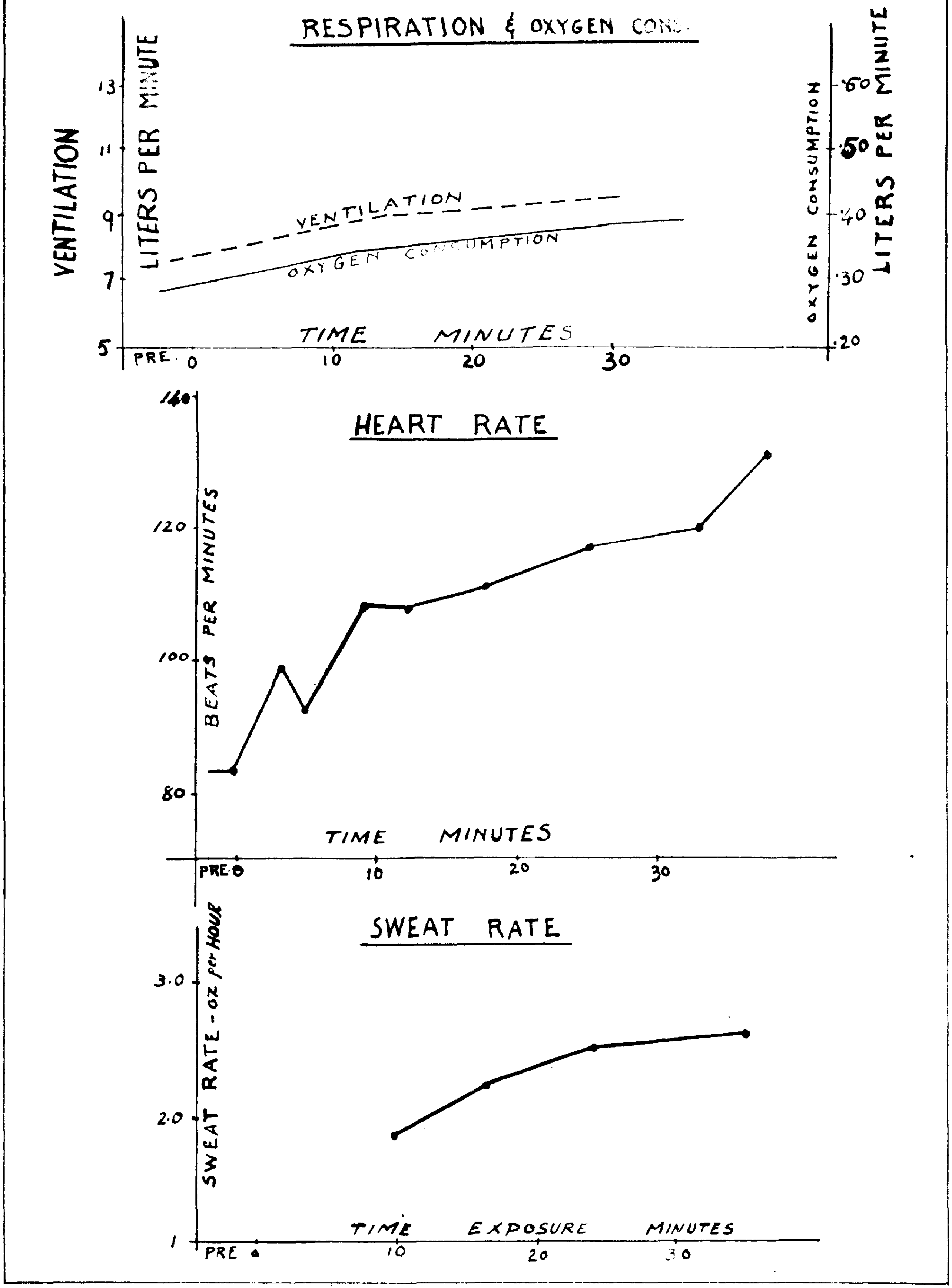
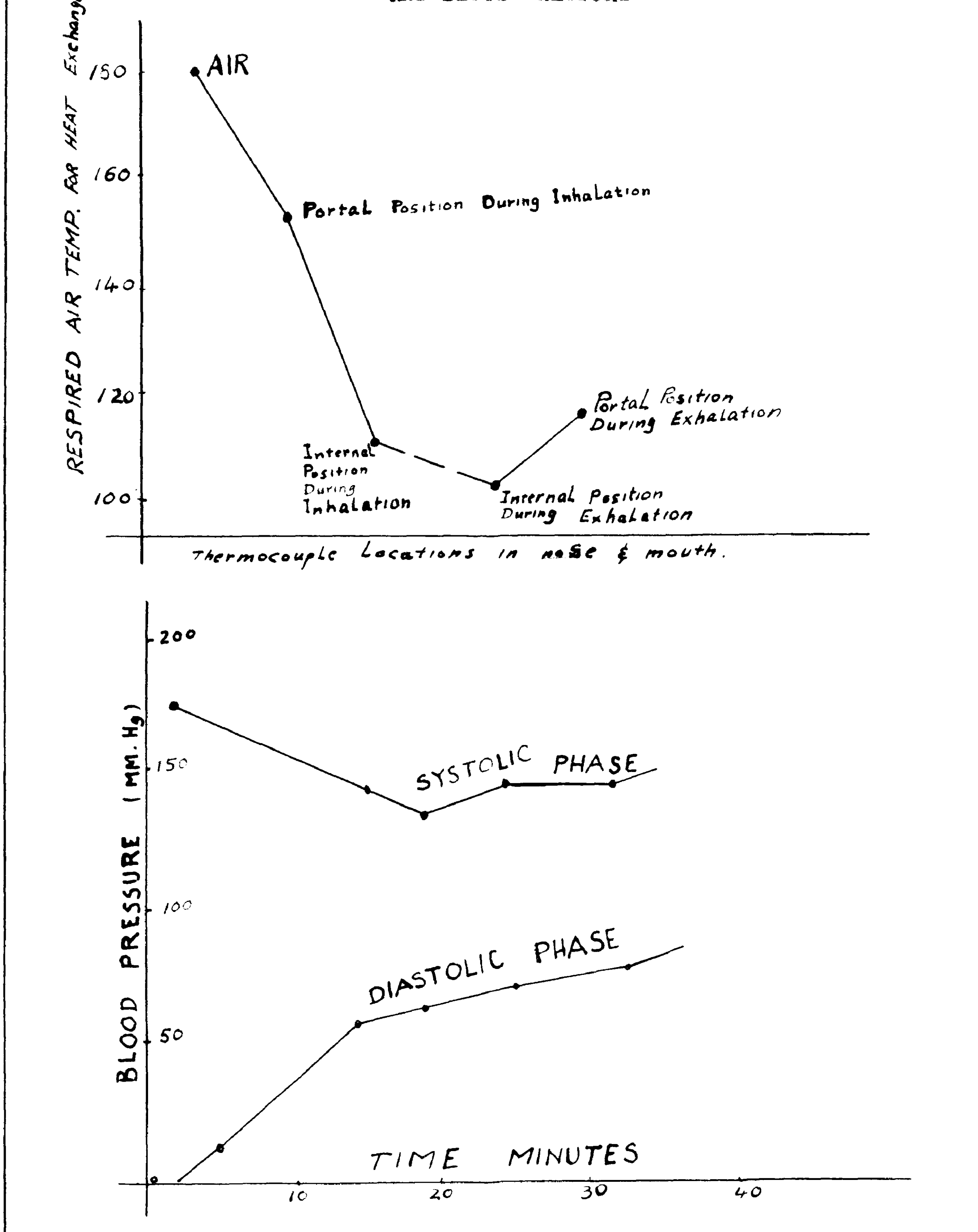


Figure 17 EFFECT OF EXTREME TEMPERATURE UPON RESPIRED AIR TEMPERATURE AND BLOOD PRESSURE



5-Results of Experiment:

Heart Rate: (beats/minute) = 54.7

Recovery Heart Rate: (beats/minute) = 140

Pulse pressure (Diastolic pressure) = .62

Breath holding (seconds) = 18.5

Average chamber air temperature °F = t_a: range 171.0-183.0 mean 180

Vapor Pressure of Inlet Air (in.Hg.) = P_{wi}: range .480.49 mean .48

Vapor Pressure of Outlet Air (in.Hg.)= P_{we}: range .60-.71 mean .67

Wet Bulb Temperature Outlet °F= WB_o: range 91.0-93.0 mean 92.6

Rectal Temperature °F. tr = range 98.2-99.7

Heart Rate: pre = 54, Post = 114

Nude Weight Loss: ΔW 1.5, ΔW/hr. ft^2 = .120

Skin Temperature °F. = t_a pre = 92.1

Experiment Duration: 29 1/2 minutes

θ minute	2	4	6	8	10	12	14	16	18	20
t_s °F. skin temp	93.8	94.8	95.1		96.9		97.3			98.2
θ minute	23	24	26	29	29½					
t_s °F. skin temp	98.3	98.4	98.7	100.1	100.4					

Free Respired Air Temperature:

Nose breathing: (after 7 minutes {internal inh. 109.1 / internal exh. 100.6
(after 8 minutes {portal inh. 153.1 / portal exh. 115.1

Mouth breathing: (after 13 minutes {internal inh. 115.3 / internal exh. 100.3
(after 15 minutes {portal inh. 125.7 / portal exh. 105.0

Respiratory Heat (after 13 minutes {internal exh. 100
(after 15 minutes {portal inh. 125.7(inhaled air). / portal exh. 105.0

Respired air temperature °F. {t_i = 153.0 / t_e = 115.0

Water vapor pressure (inch Hg.) $\begin{cases} P_{wi} = .67 \\ P_{we} = 1.87 \\ rh_e = .63 \end{cases}$

Volume of exhaled air: $V_e = 15.9$

$q_v = + 1.34$ respiratory heat exchange

6-<u>Conclusions from similar experiments</u>: Results of investigations of human response to high levels of environmental temperatures are represented in relation to the tolerable limits of heat exposure. Some individual experiments involving the author as subject (of which the foregoing is an example), consisted of multiple exposures to air temperature levels of 100, 140, 160, 180, 200, 220, 240, and 250 degree Fagrenheit. Humidity of the experimental environments was determined by the atmospheric humidity existing at the time of each exposure, as modified by the moisture production of the subject.

The average humidity during heat exposure ranged from .25 to .9 inches Hg. vapor pressure, but in most cases was of .8 inches of mercury. Wall temperatures approximated the air temperature to within 5 to 10°F. in all experiments.

The physiological reactions to the heat exposures were:-

a-The <u>Skin Temperature</u> rose continuously during the exposure, the rate being successively greater with increasing ambient temperature levels. Maximum mean skin temperature observed was 107°F., reached in an exposure to 240°F. with standard clothing (medium weight wool and cotton union suit).

b-The <u>Heart Rate</u> increased in the same manner as the skin temperature, usually reaching rates of 140-160 beats per minute. The maximum heart rate observed was 172 beats per minute.

c-The <u>Blood Pressure</u> changes showed greater individual differences than other physiological factors. Systolic pressure rose during exposure in all cases, while diastolic pressure declined continuously in one subject, but remained relatively stable, or slightly elevated, in another. Electrocardiograms taken during and after heat exposure showed minor variations from the normal resting records of the individuals, but no distinct heart damage.

d-<u>The Rectal Temperature</u> changes were relatively slight and slow compared to other physiological reactions.

e-<u>The Exhaled Air Temperature</u>, measured at the mouth during channeled breathing, responded rapidly to the heat load, and was a good indicator of thermal stress. Measurement of temperature at several locations in the respiratory passages, indicated that the mucous membranes exert a good regenerative influence.

f-<u>The Entire Respiratory System</u> continued to serve as an avenue of heat loss even under conditions close to the extremes in these tests, because of the evaporative loss.

g-<u>The Respiratory Ventilation</u> increased markedly under heat stress, often reaching values two or three times that of the resting levels, even though the respiration rate was not significantly affected.

h-<u>The Oxygen Consumption</u> apparently follows a line paralell to that of ventilation.

i-<u>The Rate of Sweat Loss</u> was highly variable, but tended to increase with successively higher levels of ambient temperature.

j-<u>The Rate of Actual Evaporation from the Skin</u> increased with the ambient temperature at a considerably slower rate than did the total loss.

k-<u>Absolute Values of the Weight Loss</u> observed ranged from .9 to 2.5 pounds, depending upon the characteristics of the individual, on the duration of exposure, and the ambient temperature level.

l-Subjective symptoms of <u>Thermal Strain</u> were numerous, but only a relatively few were observed consistently in each experiment. Some sensations, while definite, were difficult to describe in practical terms. One common feature of all exposures carried to tolerance was a feeling of air hunger, associated with deep and irregular respiration. This reaction was at times accompanied by restlessness and nervous irritability, while waves of dizziness intervened at terminal stages.

CHAPTER III--THE HEAT PUMP

Energy in nature is not easily harnessed or conserved, nor is it often usuable for human purposes in its natural state. It must go through changes in kind and quality, be transmitted through space as kinetic energy or stored in position as a potential supply available at the time of most need. The engineer, therefore, seeks to provide means by which the conversion of heat energy can be made by mechanical devices.

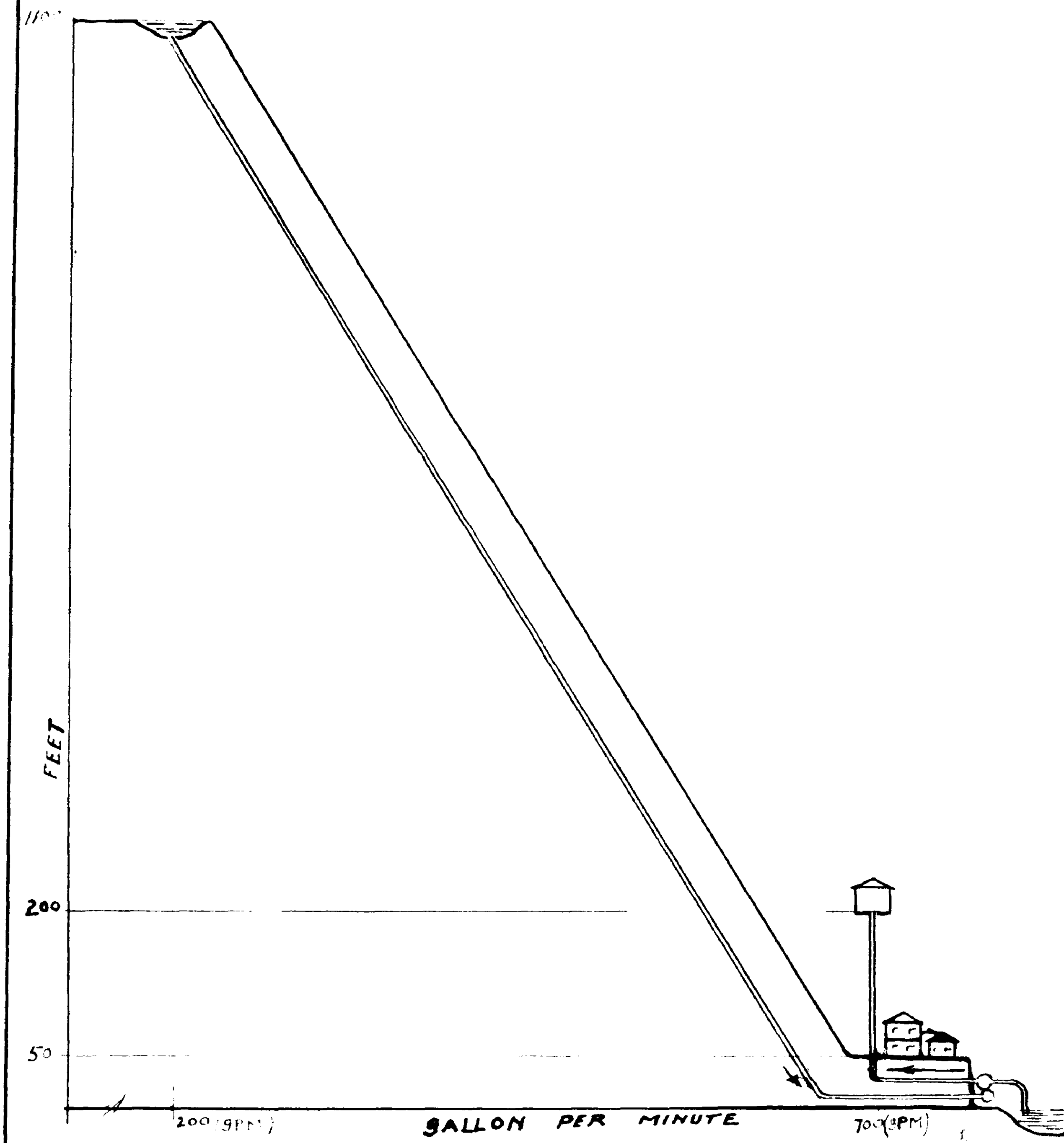
There are two main points to consider in controlling body heat: First, the human temperature regulating system (termed "natural autoregulator" in biotechnology) must coordinate with the control in order to qualify varying room and air conditions. Second, in order to assist the body mechanism in maintaining an energy balance, heating systems, then ventilating systems, and finally air conditioning systems were developed.

In perfecting comfort-giving installations, the engineer strives for the aim as described by the ASHVE code of minimum requirements for a comfort zone as:-

> "Air conditioning is the process by which simultaneously the temperature, moisture content, movement and quality of the air in enclosed spaces intended for human occupancy may be maintained within required limits."--ASHVE Transactions, Vol. 44, 1938 "Code of Minimum Requirements for Comfortable Air Conditioning".

In order to explain the operation of the heat pump, a hydraulic analogy of it should be considered. Imagine a community fifty feet above sea level where there exists an unlimited water supply. In order

Figure 18



HYDRAULIC ANALOGUE

THEORITICAL C.O.P = $\frac{1100}{200}$

C.O.P $\frac{700}{200}$

EFF. =.635

to furnish this community with a continuous supply, we shall fill a tank at a spot higher than any building in the area (200 feet altitude). We shall further postulate an adequate, but limited water supply at 1100 feet altitude. By drawing 200 gallons of water per minute from the water source at 1100 feet altitude, in order to drive a hydraulic turbine at sea level, it is possible to pump 700 gallons per minute to the tank at 200 feet altitude.

The term coefficient of performance (C. O. P.) is used instead of the term efficiency which in this system will equal $\frac{700}{200} = 3.5$. The theoretical coefficient of performance is $\frac{1100}{200} = 5.5$. The efficiency of the system is the ratio between the actual C. O. P. and the theoretical coefficient of performance $\frac{3.5}{5.5} = .635$. The losses encountered in this system are due to an inadequate entrance and exit. There would also be frictional losses in the pipes and in the hydraulic turbine and pumping unit.

A. IN RELATION TO CLIMATE AND SOIL

1. Heat Sources and Sinks: Reliable design data for heat sources and sinks is necessary. Four of the principal potential heat sources are air, water, earth, and solar energy. Heat from air, water and earth may be used separately while solar energy will be developed as an auxiliary to the other three.(27).

a. Air: A study of air as a heat source would involve research on typical climatic data for winter and summer (for regions with annual degree days of 1000, 2500, 4000, 5500) including such factors as the mean monthly temperature (including dry bulb, wet bulb, and dew point), extreme monthly temperatures, length of growing season and occurrence of fog and sleet. A knowledge of the effect of the foregoing upon the design of the pump is necessary in order that it may be practical for winter and summer use, etc.

Data of the daily cyclic temperature variations, a report of the daily spread between maximum and minimum temperatures and a typical 24 hour chart should be available.

It has been found that the advantages of using air as a heat source lie in its universal availability at favorable temperatures during 75-95% of the heating season in most parts of the United States. Satisfactory design knowledge for its use is already available and the first cost of the equipment is lower than for earth and water sources. The use of air heat permits the design of self contained, factory assembled and tested units. When the system is used for space cooling, it forms an inexhaustible heat source and a satisfactory heat sink.

This system also has its drawbacks. The air temperatures are lowest when heating demands are highest. Therefore, its performance efficiency is reduced. It calls for a bulky heat-exchanger, unless it is cut down at the expense of large circulating power requirements. We also find that extensive ducting of air may be necessary.

b. Water: A study of water as a heat source would involve research of City (river, lake and underground), Surface (rivers, streams, and lakes), and Waste water with regard to its maximum temperatures, its quality (hardness, other solids, organic materials), reliability of a constant supply, etc.

City water is readily available and requires no pumping. The temperature variation is less than for air and it has the lowest first cost for source equipment and forms a good heat sink. Well water is another very satisfactory heat source if the water is chemically suitable. The cost is reasonably low and the heat transfer surfaces may be comparatively small and inexpensive. Surface water has the advantage of being tax-free when available. The operation costs are relatively inexpensive and it forms a good heat source and sink. Its use is limited, however, due to very probable corrosion, scaling and algae foulings. Waste water is of limited application, usually of insufficient supply and also likely to be corrosive and to contain solids. Sewage and waste outflow are not reliable sources, as the quantities are small and the flow inadequate.

e. <u>Steam generating plants</u>: There are many advantages of the steam-generating plant as a source and sink for the heat pump (as applied to the heating and cooling of all-electric houses) over the use of natural heat sources. Provision is made for a heat source controlled within close temperature limits, and unaffected by climate, location or legal restrictions. Provision is made for a suitable heat sink. A higher over-all annual coefficient of performance is obtainable. A smaller heat pump is required in places where the maximum heating load is greater than the maximum cooling load. The annual load factor on the power system is below that supplied from a large modern power station, necessitating decentralization of generating capacity; and, in order to fill the heat source requirements of the heat pumps in the condenser water alone, the total load on the power plant must be at least 1.6 times the power consumed by the heat pumps. (Even though the load imposed by the all-electric houses meets this condition during most of the heating season, additional load is necessary for the coldest weeks of the year).

The suggestion of Professor Charles A. Shreeve, Jr., of the Mechanical Engineering Department of the University of Maryland, that steam power generating stations would be good heat sources and sinks for handling the heating and cooling requirements of nearby housing developments was a part of the author's study.

Looking to existing examples, we find that by using the reservoir water from Austin Dam in Texas, year around air-conditioning is furnished to the Lower Colorado River Authority's three-story office building. So in steam electric generating plants, the heat pump unit takes heat out of the water and supplies it to the space to be heated.

The coefficient of performance will be equal to the (electric energy supplied plus the energy drawn from the heat source) over (the electrical energy supplied). The coefficient of performance of the heat pump will be increased by decreasing the temperature difference between the condenser and evaporator. The temperature of the heat source should be fairly uniform over the year, as any variation in the temperature will change the C. O. P. materially. For that reason, and because of large quantities of water available at high temperatures at practically no cost, a favorable and suitable source of heat pump system can be maintained.

In using a steam generating station as a source and sink for the heat pump, extended decentralization of the stations and a considerable decrease in the installed capacity of these stations as compared with present large stations will be necessary. If a new market for electric energy can be established by this process, decentralization would appear justified, as it has been proved to be for various business activities. In addition, any enterprise, whose objective coincides with increased standards, is bound to prosper in the long run. It is hoped that such a future awaits the type of development proposed.

d. Solar: In a consideration of the availbility of solar energy we shall be concerned with four factors. The first of these is the element of variation in the incident total radiation received daily, weekly, seasonally, and yearly. The second concerns recommended design values for incident solar energy at representative localities, with their geographical and seasonal variations. Thirdly, we must note the degree to which incident radiation is below the recommended design values, a factor involving the maximum number of consecutive days of operation, and the amount of thermal deficit. Lastly, we must consider certain geographically correct areas having sufficient solar energy for heat pump operation. Under this topic, we must study solar energy both as a primary and an auxiliary heat source.

As previously stated, solar energy, unless used with an adequate storage system, is an auxiliary, and not a primary, heat source, as it does not form a heat sink. The use of solar collectors would require many architectural revisions. The utilization of a storage system, operating upon a heat, or fusion cycle, supplied by supplementary heat from air or solar sources might be feasible. Heat storage devices are limited by the cost and space requirements, but combinations of heat sources result in improvement of the coefficient of performance over an entire heating season. The drawback would be a greater initial cost of the system.

e. <u>Earth</u>: Utilization of ground heat and the technical aspects involved therein in connection with the heat pump were considered to be the main objectives of this study and of particular interest to the author. Earth is the most readily available source of heat for the heat pump system, taking into consideration the resistance of the soil to heat flow. The moisture of the soil has great influence upon the ground temperature. One of the most striking features of the rates of underground temperature increases, as observed in borings, is their variations from place to place related to local causes of which the nature is unknown. Undoubtedly, the variation in conductivity of various rocks, underground tension, mineralization, and movement of underground water, influence the rate of temperature increases.

1-<u>Soil temperature as affected by its own properties</u>: The temperature of the soil is determined to a considerable extent by its own properties, depending upon factors responsible for the difference in the intensity of absorption of heat such as color, variations in the specific heat of the soil, such as composition and water content, and the differences in heat conductivity, as compaction and moisture content.

Soil identification according to the Army Airfield (or Casagrande) classification system requires a knowledge of such types of soil as gravel and gravelly soils; sands and sandy soils; fine-grained soils with low to medium compressibility; fine-grained soils with high compressibility; fibrous organic soils with high compressibility and further classification into the fifteen groups established under the above-listed devisions of this system.

Soil-testing methods include such types as determining the size of samples needed, sampling methods for undisturbed soil, field tests for identification, laboratory test procedures, moisture determination in place. Classification systems are concerned with a comparison of the recognized systems and a system preferred for heat-pump work.

In studying groundwater appearance, we must note several factors, such as the height of the water table (noting here the normal state and seasonal fluctuations); the horizontal or vertical circulation of natural water; the saturation percentage of the earth.

In order to study the natural vertical temperature gradient in the earth at selected geographical locations, we shall need winter-month data (including earth data from the Casagrande system and the effect of water content on the depth-temperature relation) and similar summer-month data.

Factors which enter into a study of natural cyclic temperature variations of the earth are depths; daily cycles; periodic (meaning between successive cold or hot spells); seasonal (cumulative effect); yearly (influence of the seasonal reversal).

The points to be considered as affecting the earth temperature and cyclic variation are ambient air temperature (which will determine conduction and convection); solar heat absorption; radiation (including such states as clear and cloudy skies and day and night times); precipitation, such as rain, dew and snow; evaporation; wind; and conduction from and to the substrata.

The effect of the top surface must be considered both on solar and other heat absorption and on radiation and other heat dissipation. Concerned with the relative effect of the heat input and withdrawal on the earth temperature and gradients are heat input tests for heat withdrawal versus separate tests for heat "reservoir" capacity and temperature gradients; the influence on vegetation (including lawns, gardens, etc.) on trees and deep-rooted plants; retarding (winter killing); stimulating (summer parching); the effect on insects, etc. and on vegetation.

Knowledge of the factors to be considered in a study of the temperature gradient in the earth with heat pump operation includes data on location and size of the heat pick-up device; rate of heat withdrawal or input by the heat pump; the total cumulative heat withdrawal and input for heating and cooling seasons; type of soil; water table elevation; water content of the stratum; and top-surface conditions. We must also note here the depth and extent of frost penetration, as it affects forces on, and movement of, underground structures.

Material for backfill in trenches includes soil removed from excavations; soil selected for its conductivity; metallic admixtures; concrete and bitumastic compounds; etc. Methods for obtaining high thermal conductivity include puddling with water, tamping by hand tools, compacting by vibrators, and impacting devices applied on pipes.

Schübler did much experimenting with the relation of the effect of different colors upon the intensity of the absorption of heat by various types of soils. From his table, which follows, we can see that black surfaced soils were more heat absorbing than natural color soils and that white surfaced soils were least heat absorbing of the three. These findings proved to be true in experiments made by the author, of soil in California.

TABLE NO. 2-Effect of Color on Heat-Absorption of Soils (Degree Centigrade)

SOIL TYPE	NATURAL COLOR	WHITE	BLACK
Yellow-gray quartz sand	44.7°C.	43.2°C.	50.9°C.
White-gray quartz sand	44.5	43.2	51.1
Yellow clay	44.1	42.4	49.7
Loam	44.5	42.2	49.5
Blackish-grey humus	47.4	42.6	49.4
Blackish-gray garden soil	45.2	42.4	50.9

(A difference of about 8°C. exists between the white and black surfaced soils.)

TABLE NO. 3 THERMAL DIFFUSIVITY AS AFFECTED BY VARIOUS COMPOSITIONS OF SOIL AND ITS WATER CONTENT.

MATERIAL	SPECIFICATION			THERMAL CONDUCTIVITY		DENSITY	SP. HT. C.		$a = \frac{K}{\rho C_p}$
		°F.	Moisture	K	Dims	Sp. wt. lb/ft³ $\rho = \frac{w}{v}$	t°F.	C_p	
SAND:									
Moist-	quartz	68		.65	$\frac{\text{Btu}}{\text{ft/hr/°F}}$	102.4		.5	.0127
Dry-	medium	68		.19	"	94.9	68)	.191	.0105
	fine	32		.15	"	103	208)	.19	.008
Moist-		32	8.3%	.34		109		.24	.013
SANDSTONE:									
Moist-		68		.97		141.1	32)		
Dry-		68		.75		140.5	210)	.179	
CALCAREOUS EARTH:		32	43%	.41		104		.53	.007
Very dry soil		32	0%	(.10 (.20					.008) .012)
Damp soil				(.8 (2.0				.02) .04)	
Wet mud				0.50		94		.6	.009
SANDY CLAY:			15%	.53		111		.33	.015

During the period from July 7th to 13th, 1949, the author made soil and temperature tests on the campus of the University of California in Los Angeles. Three holes in all were drilled: one to a depth of 71½ feet, at which point water seepage forced removal to another site; one to a depth of but 20 feet before large rocks broke the drilling bit and forced removal again; and the final one to a depth of 83 feet. Each of these holes was ten inches in diameter.

The writer lowered a glass thermometer, fastened to the end of a steel tape, at intervals of two feet during the drilling, to determine the temperature of the soil at the various underground levels. The readings given in the following tables are the result of several re-checked and compared tests taken at this time.

Soil characteristics tests were also taken at two foot intervals. From these findings, we conclude that there can be great variance in the characteristics of the soil at so frequent intervals as every twelve inches during the drilling. During the tests, a drop of 10°F. was noted within a vertical distance of two feet where the soil changed from soggy states to the water level.

The water level on the U. C. L. A. campus varies from 70 to 100 feet deep depending on the site. The moisture content differs from month to month as the soil reaches the saturation point in winter and approches the dry state in summer. When the freezing point is reached, the flow of heat from the soil to the pipes will be affected by the increased resistance to the heat flow.

Figure 19 SUB-SOIL TEMPERATURES AT THREE POINTS ON U.C.L.A. CAMPUS

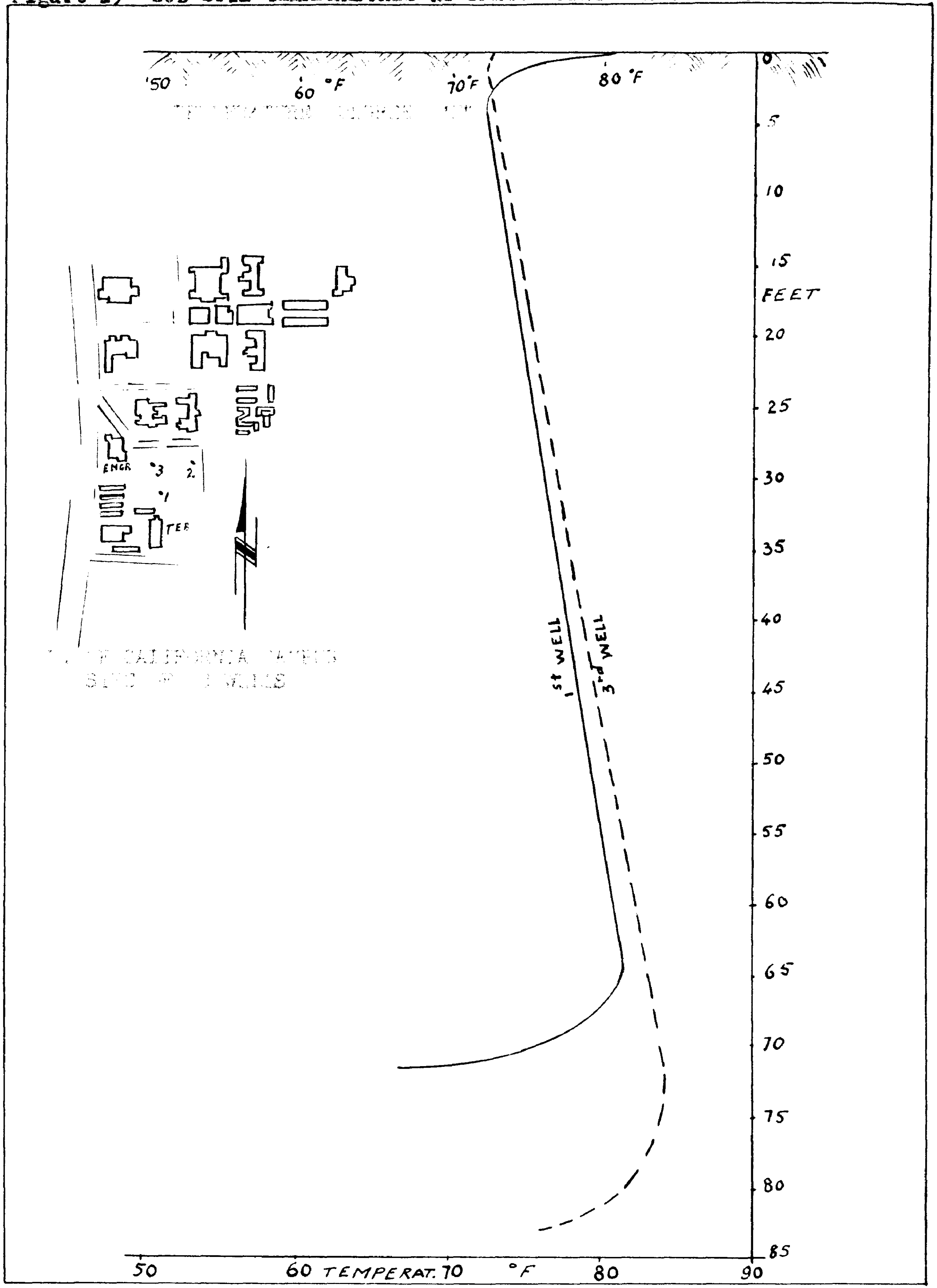


TABLE NO. 4 TYPES OF SOIL ENCOUNTERED AT VARIOUS DEPTHS ON U.C.L.A. CAMPUS

DEPTH (feet)	SOIL DESCRIPTION HOLE #1	HOLE #3
Surface	Dry	Br., silty-fine sand
5	"	Br., clayey, sand
6	Br., silty-fine sand	"
10	Br., clayey sand	Sandy clay
12	Sandy, clay & silt	"
14	Silty sand	Clayey coarse sand & gravel
16	Medium sand & gravel	" "
18	Clayey coarse sand & grav.	"
20	"	Br. med. sand & gravel
25	Br. med. sand & gravel	Med. sand & big gravel
32	Br. sand & 2" size gravel	Lt. br. silty clay
33	Br. sand & 1½" size gravel	"
37	Lt.br. silty clay	"
41	Silty fine sand	"
43	Br. silty clay	"
44	Br. silty clay, fine gravel	"
45	Br. clayey fine sand	Br. sandy clay, fine gravel
47	Br. fine sand, clayey silt	Br. sandy clay
50	Br. sandy clay	Fine sand, clayey silt
55	Fine med. sand, some gravel	Fine med. sand, some gravel
58	Clayey fine sand, fine grav.	Clayey fine sand with grav.
59	Medium sand & gravel	Medium sand & gravel
61	Silty fine sand	"
63	Dark br. sandy clay	"
65	Light br. silty clay	Silty fine sand
66	Lt. br. silty med. sand, some gravel	Dark brown sandy clay
68	Br. sand with gravel (wet)	Br. silty med. sand, some gravel
70	Coarse sand & gravel(water)	"
73		Coarse sand & gravel
78		Fine sand
80		Wet clay
83		Very wet clay

Lt.= light Med.=medium Br.= brown Grav.= gravel

TABLE NO. 5 - UNDERGROUND DEPTH AND TEMPERATURE AT U.C.L.A. CAMPUS

DEPTH (feet)	TEMPERATURE °F. FIRST READING		TEMPERATURE °F. SECOND READING	
	HOLE #1	HOLE #3	HOLE #1	HOLE #3
Surface(0)	80	72.8		
2	73			
4	72.5			
9	73.5			
10	73.5	73.6		
15	74	74.5		
20	74.8	75.5	74.5-74.2	75
25	75.6	75.2	75.8	
30	76.6	77.2	77	77
35	77	78		
40	77.8	78.8		
45	78.5	79.8		
50	79.6	80.5		
55		81.5		81.7
57	80.1			
58	80.2			
60	81.0	82		
61½	81.2			
64	81.5			
65	81.7	83		
67	80			
70	74	83.8		
71½	67			
73		84		
75		83.5		
80		82.0		81.8
83		76		

Note: the temperature readings of Hole #2 were identical to those found in Hole #3 down to a depth of 20' at which point drilling ceased.

Correction for errors in the temperature readings was found to be .0045 °F. as calculated in Appendix III.

2. Soil Temperature as Affected by Climatic Conditions: Climatic conditions on the surface of the earth change periodically. These changes affect the temperature of the soil below the surface and are of great importance to heat transfer study.

Assume a one-dimentional heat flow in a homogeneous soil with constant coefficients:

$$\frac{dT}{d\theta} = a\,\nabla^2 t = a\frac{\partial^2 T}{\partial y^2}$$

with a boundary condition relation:

$$T = Ae^{b\theta}\,Be^{cy} = De^{b\theta + cy}$$

$$A,\ B,\ b,\ c,\ \&\ D \equiv \text{Constants}$$

From:

$$\frac{dT}{d\theta} = a\,\nabla^2 t$$

$$b = ac^2$$

$$c = \pm\sqrt{\tfrac{b}{a}}$$

$$T = De^{b\theta \pm \sqrt{\frac{b}{a}}\,y}$$

Assume:

$$b = \pm i\gamma$$

$$T = De^{[\pm i\gamma\theta \pm y\sqrt{\frac{\gamma}{a}}\sqrt{\pm i}\,]}$$

As

$$(1+i)^2 = 1 + 2i - 1 = 2i$$

$$\therefore\ \sqrt{i} = \pm\frac{1+i}{\sqrt{2}}$$

$$\sqrt{-i} = \pm\frac{1-i}{\sqrt{2}}$$

$$T = De^{[\pm i\gamma\theta \pm y\sqrt{\frac{\gamma}{2a}}(1\pm i)]}$$

$$= De^{\pm i\gamma\theta}\,e^{\pm y\sqrt{\frac{\gamma}{2a}}(1\pm i)}$$

which will satisfy all boundary conditions.

One of the solutions is

$$T = \left(D e^{-y\sqrt{\frac{\gamma}{2a}}}\right)\left[e^{i\left(\gamma\theta - y\sqrt{\frac{\gamma}{2a}}\right)} - e^{-i\left(\gamma\theta - y\sqrt{\frac{\gamma}{2a}}\right)}\right]$$

then: $$T = E e^{-y\sqrt{\frac{\gamma}{2a}}} \sin\left(\gamma\theta - y\sqrt{\frac{\gamma}{2a}}\right) \quad 1$$

The same is true for the other three solutions:

$$T = F e^{y\sqrt{\frac{\gamma}{2a}}} \sin\left(\gamma\theta + y\sqrt{\frac{\gamma}{2a}}\right) \quad 2$$

$$T = G e^{y\sqrt{\frac{\gamma}{2a}}} \cos\left(\gamma\theta + y\sqrt{\frac{\gamma}{2a}}\right) \quad 3$$

$$T = H e^{-y\sqrt{\frac{\gamma}{2a}}} \cos\left(\gamma\theta - y\sqrt{\frac{\gamma}{2a}}\right) \quad 4$$

Using the initial condition that when $y = 0$, $T = T_M + T_o \sin \gamma\theta$, where T_M = mean annual temperature, and T_o is the amplitude of temperature variation, then T at any depth y is given by:

$$T = T_M + T_o e^{-y\sqrt{\frac{\gamma}{2a}}} \sin\left(\gamma\theta - y\sqrt{\frac{\gamma}{2a}}\right)$$

which gives the temperature at depth y when the time is θ.

This equation is of an undulating motion of amplitude =

$$= T_o e^{-y\sqrt{\frac{\gamma}{2a}}}$$

The maximum range of temperature will be twice the amplitude

$$= 2 T_o e^{-y\sqrt{\gamma/2a}}$$

$$\tau = \frac{2\pi}{\gamma}$$

Therefore, the maximum variation of temperature at fixed y is:

$$= 2 T_o e^{-y\sqrt{\frac{\pi}{a\tau}}}$$

Hence, the maximum temperature at fixed y occurs when θ satisfies

$$\sin\left(\gamma\theta - y\sqrt{\frac{\gamma}{2a}}\right) = 1 \qquad \therefore \theta = \frac{\sin^{-1} 1}{\gamma} + \frac{y}{\sqrt{2a\gamma}}$$

<u>For fixed θ time</u>: the maximum temperature when $\frac{dT}{dy} = 0$

$$-\sqrt{\frac{\gamma}{2a}}\, e^{-y\sqrt{\frac{\gamma}{2a}}} \sin\left(\gamma\theta - y\sqrt{\frac{\gamma}{2a}}\right) - \sqrt{\frac{\gamma}{2a}}\, e^{-y\sqrt{\gamma/2a}} \cos\left(\gamma\theta - y\sqrt{\frac{\gamma}{2a}}\right) = 0$$

$$\tan\left(\gamma\theta - y\sqrt{\frac{\gamma}{2a}}\right) = \frac{+\sqrt{\frac{\gamma}{2a}}}{-\sqrt{\frac{\gamma}{2a}}} = -1$$

$$y = -\sqrt{\frac{2a}{\gamma}} \tan^{-1}(-1) + \sqrt{2a\gamma\theta^2}$$

$$\left(\gamma\theta - y\sqrt{\frac{\gamma}{2a}}\right) = (2n+1)\frac{\pi}{2} \qquad y \text{ const}$$

$$n = 0,\ 1,\ 2,\ 3,\ 4\ \ldots$$

$$\theta_{t\,max} = \frac{1}{\gamma}\left[(2n+1)\frac{\pi}{2} + y\sqrt{\frac{\gamma}{2a}}\right]$$

time θ and y depth both increase till θ maximum reaches the surface, when n = 1.

Therefore:

$$\theta = \frac{1}{\gamma}\left[\left(\frac{3\pi}{2}\right) + y\sqrt{\frac{\gamma}{2a}}\right]$$

if:

$$\gamma\theta - y\sqrt{\frac{\gamma}{2a}} = 0$$

$$\theta = \frac{y}{\gamma}\sqrt{\frac{\gamma}{2a}} = \frac{y}{2}\sqrt{\frac{\tau}{a\pi}}$$

then the apparent velocity of such a wave $= \dfrac{y}{\theta} = 2\sqrt{\dfrac{a\pi}{\tau}}$

The flow of heat:

$$q = KA\frac{\partial T}{\partial y}$$

$$T = T_0 e^{-y\sqrt{\frac{\gamma}{2a}}}\left[\sin\left(\gamma\theta - y\sqrt{\frac{\gamma}{2a}}\right)\right]$$

$$\frac{\partial T}{\partial y} = T_0\left(-\sqrt{\frac{\gamma}{2a}}\right)e^{-y\sqrt{\frac{\gamma}{2a}}}\left[\sin\left(\gamma\theta - y\sqrt{\frac{\gamma}{2a}}\right) + \cos\left(\gamma\theta - y\sqrt{\frac{\gamma}{2a}}\right)\right]$$

$$\frac{Q}{A} = -K\int_{\theta_1}^{\theta_2}\left(\frac{dT}{dy}\right)_{y=0} d\theta$$

where: $\theta_1 = \dfrac{\tau_1}{8}$, $\theta_2 = \dfrac{3\tau_2}{8}$ & $\dfrac{Q}{A} = KT_0\sqrt{\dfrac{2\tau}{\pi a}}$ $\dfrac{BTU}{ft^2}$

[illegible] Tamura (Monthly Weather [illegible] Rev. [illegible], 396 (190[illegible])), using the equation $T = 2T_0 e^{-y\sqrt{\frac{\gamma}{2a}}} = 2T_0 e^{-y\sqrt{\frac{\pi}{a\tau}}}$ and checking by experiment, using diffusivity a = .0027 C.g.s., tabulated the following results:

TABLE NO. 6 TAMURA HEAT FLOW TIME LAG TABLE

DEPTH (feet)	OBSERVED ANNUAL RANGE °F.	CALCULATED ANNUAL RANGE °F.	TIME LAG (DAYS) $\theta = \frac{x}{2}\sqrt{\frac{T}{\pi a}}$	
			OBSERVED	CALCULATED
Surface	50.8	50.8	0	0
.98	41	42.2	2.5	10.6
1.96	33.7	35.0	9	21.6
3.92	25.5	24.4	35	42.3
9.8	9.38	8.3	93.5	106
16.3	2.35	2.35	117.5	176.5
22.7	.722	.722	267.0	247.0

Thus we see that at a depth of 22.7 feet, the temperature range is very small, only .722°F., and could be negligible after 25 feet deep.

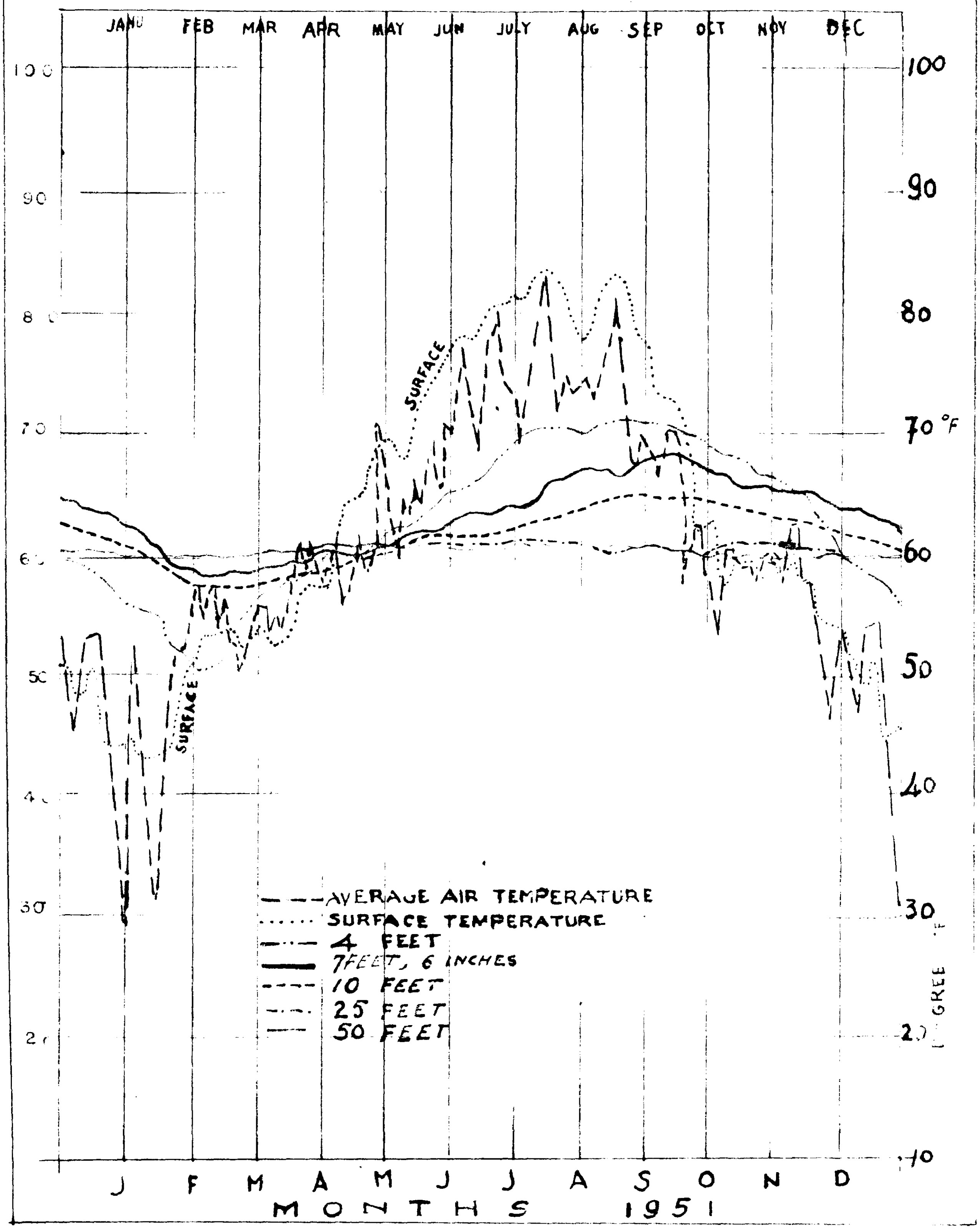
Weather Bureau records from Seattle, Washington, Portland, Oregon, and Boise, Idaho are closely related. The average air temperatures and soil temperature at depths of 4 feet, 7½ feet, 10 feet, 24 feet, and 50 feet were recorded at intervals of 12 minutes and averaged for each day over a period of two years. The graph shows the relation of these records with the data of surface soil temperatures and air temperature. At the three sites chosen, there is a complete similarity in the temperature of the soil at certain depths. The temperature at the soil surface followed closely the average air temperature while sometimes it was several degrees higher due to heat absorbed by radiation or other means.

The unusual minimum or maximum weather records made a temperature wave, which was conducted rather slowly into the ground, appearing later at a greater depth with little change. At a depth of 40-50 feet below the surface of the ground, the soil remained very nearly at a constant temperature.

At a meeting of the ASHVE in Portland, Oregon during July 1951, Professor George S. Smith, of the Electrical Engineering Department of

Figure 20

AVERAGE AIR TEMPERATURE & SUBSURFACE TEMPERATURE



the University of Washington, presented a very interesting method for plotting the temperature recorded at Seattle, Washington, where the lag in the change of temperature at the various depths is clearly shown for isothermal temperature lines. There appears to be, in effect, a lag of at least a year, even at the 50 ft. level, as indicated by the rise of 50°F. temperature line nearly a year after an unusually cold winter.

With continually recorded temperatures at various depths below the soil surface, the transient state of the flow of heat can be determined by knowing the thermal conductivity and diffusivity of the soil.

Using the Fourier differential equation:

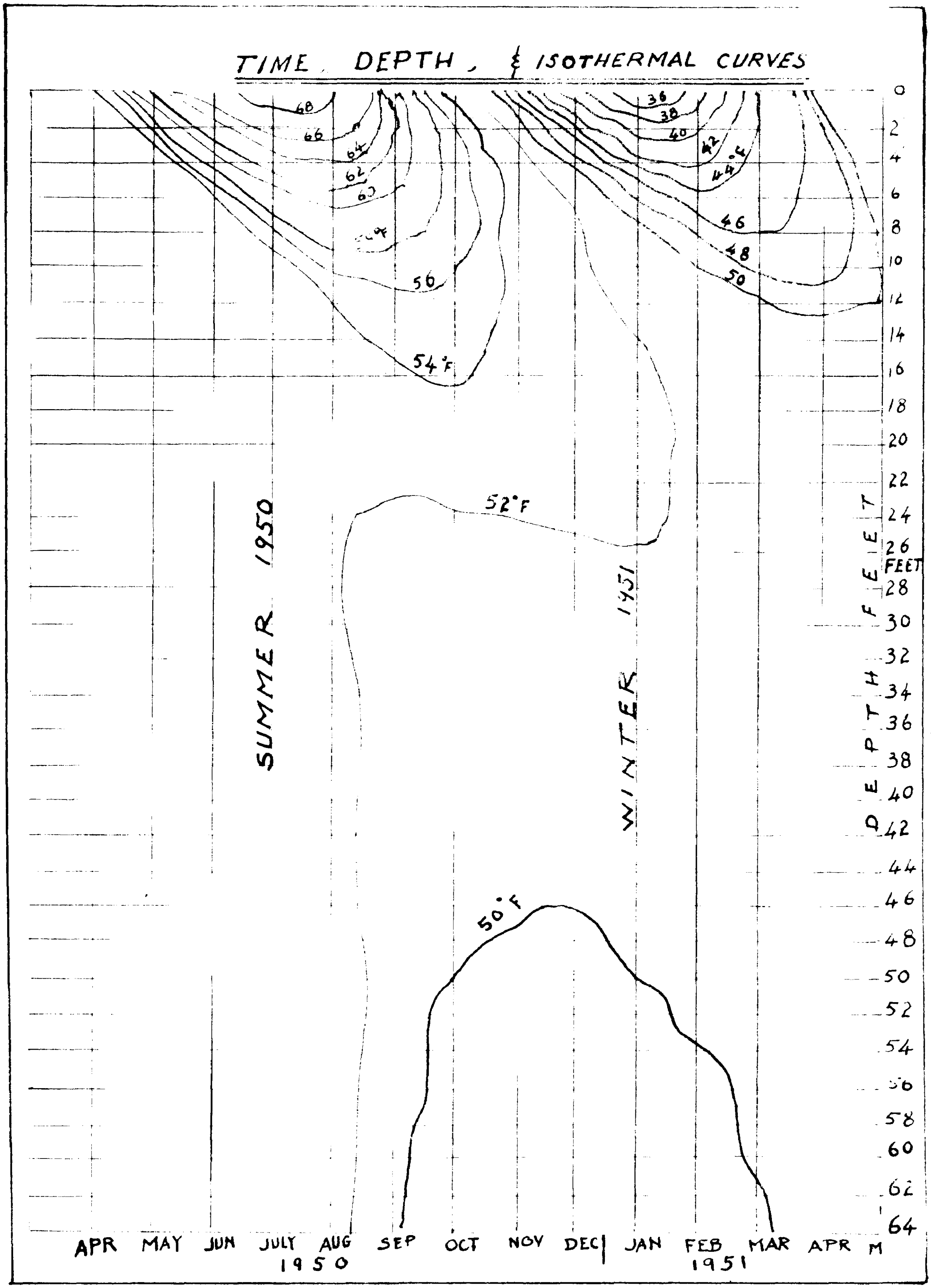
$$a = \text{thermal diffusivity } (\text{ft.}^2/\text{hr}) = \frac{K}{\rho C_p} = \frac{X^2 P}{4\pi t^2}$$

where: K = conductivity
P = time for one cycle of sinusoidal temperature change
t = time required for a given temperature
x = distance either upward or downward
C_p = specific heat
ρ = density

The graph of soil thermal diffusivity relative to depth can show clearly the relation, assuming that soil characteristics do not change.

In many localities, the rainfall carries a considerable amount of heat into the soil, and the ground moisture, due to the rain, will increase the soil's capacity to maintain this heat energy in storage. The amount of heat added by rainfall can readily be computed by using as data the weight of rainwater falling on each square foot and the difference between the average air temperatures and the temperature of the soil (assuming that rainwater temperature is approximately the same as the air temperature).

Figure 21


TIME, DEPTH, & ISOTHERMAL CURVES
68
66
64
62
60
58°F
56
54°F
52°F
50°F
36
38
40
42
44°F
46
48
50
SUMMER 1950
WINTER 1951
DEPTH FEET
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30
32
34
36
38
40
42
44
46
48
50
52
54
56
58
60
62
64
APR MAY JUN JULY AUG SEP OCT NOV DEC JAN FEB MAR APR M
1950
1951

The total heat added to the soil will, therefore, be equal to the heat gained by radiation plus the heat gained by rainwater which, to a certain extent, avoids the temperature drop caused by operation of the pump.

B. THE MECHANICAL AND TECHNICAL ASPECTS.

Let us first consider the mechanical and technical problems involved in using the natural heat sources and sinks just discussed:-

1. Air: Procedure for coordination of winter-summer coil design conditions involves such points as relative capacities for evaporator and condenser service; air quantities to be handled; refrigerant quantities required; and the effect of wet bulb temperature upon coil capacity.

Several factors which effect frost formation and accumulation include maintainence of the coil temperature; temperature of the air upon entering and leaving; content of the water vapor in the entering air; fog and sleet in the entering air; air velocity; dust and vapor prevention in the air; the arrangement of screens and inlet louvers.

Types and properties of frost include granular, hoar, ice film, density and porosity, and thermal conductivity. The effect of frost accumulation must be considered as to its effect on air velocity and quantity, on heat transfer coefficient for the coil, and on heat pump capacity and the coefficient of performance.

Types of coils and other evaporator-condenser units to be noted are: extended surface coils (finned), plain tubes (100% primary surface), shell and tube with air flow through tubes, cylindrical shells and flat surfaces (100% primary surface) and other types. The terms "Single section", "multiple sections"(in parallel), and "multiple sections" (in series) represent types of arrangement of evaporator condenser units.

Various features of the finned coil design include: tube rows in depth, tube arrangement and spacing, fin size and metal thickness, fin spacing and shape, fin stagger and angular position, refrigerant manifolding for uniform temperature, and refrigerant manifolding for seasonal reversal of coil functions.

The means by which the detection of frosting and automatic operation of defrost is accomplished include: temperature changes of air, changes in super-heat of the refrigerant, air resistance changes and others.

Frost prevention methods include:air velocity (gives a scouring effect), air filters for dust removal, and temperature differentials between air and coil surface. Methods of defrosting are: air flow cut-off or reversal, hot gas defrost cycle, refrigerant circuit changeover, water spray, antifreeze solution film, electric resistance heating units, mechanical devices, and other means.

Points to be noted in the application of evaporative condensers and cooling towers for heat pick-up include: analysis of commercial designs, variations in standard models, choice of antifreeze solutions, capacity obtained from the heat pick-up, the effect of antifreeze on summer heat dissipation, cost as compared with dry coil type condensers.

2. Water: As mentioned before, well water is an excellent source of heat because its temperature remains close to the annual mean and is not subject to sudden changes. In residential heating, disposal be means of an extra well is not economical. Extensive use of well

water without restoration (pump back) tends to deplete the supply. Problems of corrosion and upkeep of the pumping equipment make the use of directly pumped well water as a source of heat for residential pumps impractical in the long run.

In using underground water which is not pumped, as a heat source, the heat exchanger can be employed, if the underground water is available and circulating. Heat exchange fluid may consist of water or a special solution. In order to avoid the danger of freezing, a pre-installation knowledge of any stationary underground water is desirable.

A water analysis should be carefully made for possible chemical treatment required to protect the metals of the pump system. Advice of purveyors and proprietary compounds have been found to be misleading and prejudiced. As to well water, its use will result in the corrosion of metals. Steel pipe can be used, except in the panel piping (which utilizes copper tubing). The system should be regularly treated with modest amounts of suitable chemicals. Well water development should precede design, since, in many localities, information based on the performance of wells in the vicinity may not completely apply.

Sand traps (such as settling tanks) in the well water, are justified, even for wells apparently entirely free of sand. The shafts of pumps which handle well water should be made of hardened metals, such as stainless steel, or of the molybdenum group of alloy steels.

In the use of waste water, a knowledge of industrial plants, power plants, distribution economics and sewage, the use of sanitary and storm sewers, the design of shallow dry wells, slush disposal possibilities, the effects of disposal wells on supply wells, and the restrictive ordinances affecting disposal is necessary.

Water source auxiliaries include: well and other type pumps, strainers and sand traps, water treatment, antifreeze devices, alternate source switchover control, and finally, a submerged heat exchanger design.

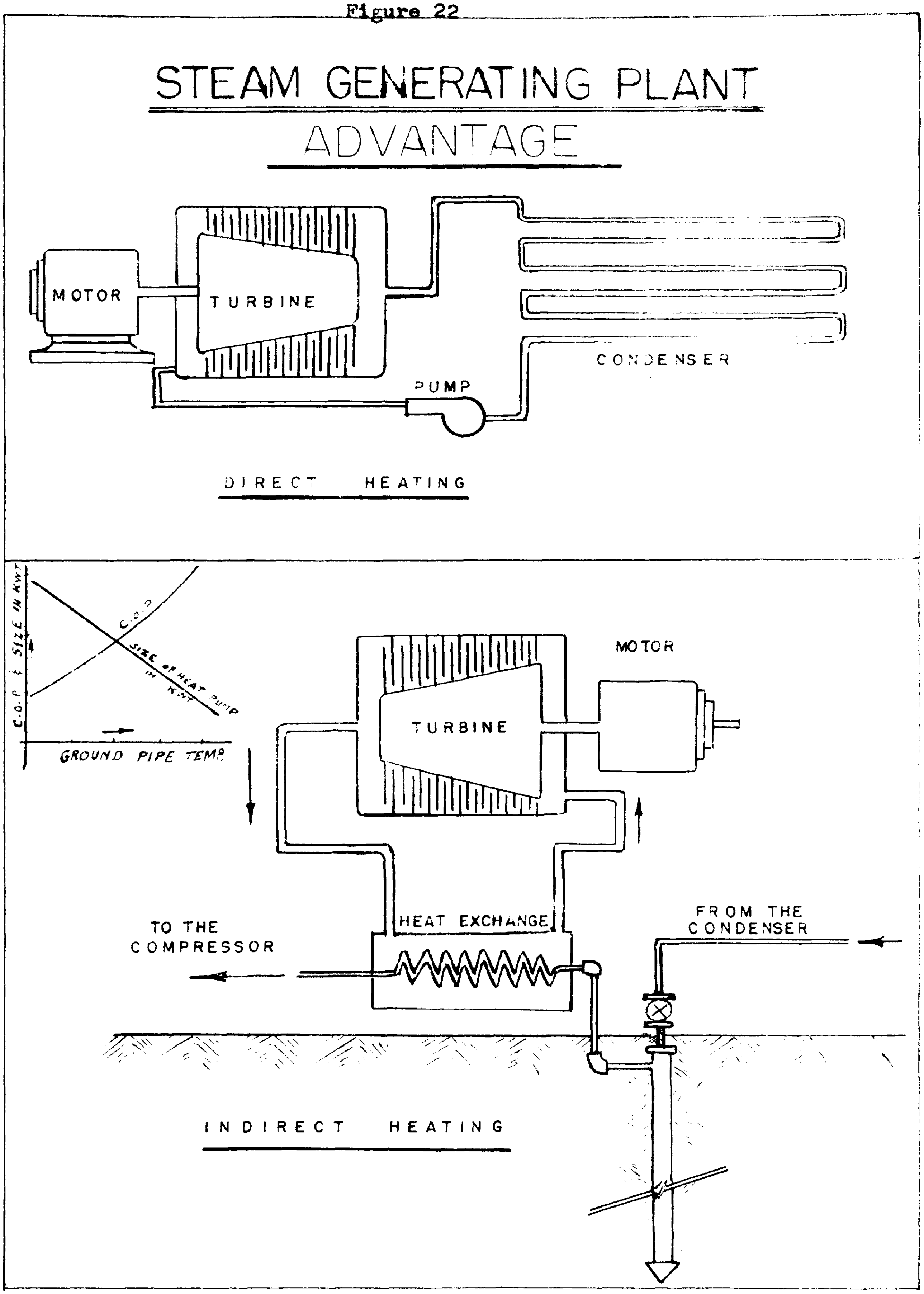
A workable system with the economic disadvantage of combining the complexity of the air-to-air system warrants the additional cost of a water storage system. This reduces the size of the air-to-air heat pump due to its ability to handle difficult peaks of short duration.

3. <u>Steam-electric plants</u>: The climate in the United States makes summer residential cooling desirable. Such cooling has not been widely applied because of the high initial cost involved, partly due to the fact that cooling has not been associated with heating as closely as is necessary.

One possibility already investigated, is that of using the waste heat in the condensing cooling water from steam-power generation as a heat source for heat pumps in dwellings. For cooling, a centrally located spray pond could serve as a heat sink. Of necessity, in each dwelling, not only the heating and cooling functions, but cooking service and water heating, household refrigeration, lighting and the operation of other appliances would be performed by electricity.(12)

The application would involve installation of an underground system of piping (the loop) through which water is circulated between power plant and heat pumps located in the dwellings. During the heating

Figure 22


STEAM GENERATING PLANT
ADVANTAGE
MOTOR
TURBINE
CONDENSER
PUMP
DIRECT HEATING
C.O.P & SIZE IN KWT
C.O.P
SIZE OF HEAT PUMP IN KWT
GROUND PIPE TEMP.
MOTOR
TURBINE
HEAT EXCHANGE
TO THE COMPRESSOR
FROM THE CONDENSER
INDIRECT HEATING

season, loop water, by passing through the steam condensers of the turbogenerators, would serve as a cooling water condenser and also as heat sources for the heat pump.

With the heat pump at the center of the loop, the temperature of the loop water is easily controlled by regulating the rate of flow and by mixing the loop water with cold water from a cooling pond. During the summer months, the loop water becomes a heat sink for the heat pumps and is pumped directly to a spray pond (cooling tower), thus by-passing the steam condensers.

The heat, in condensing cooling water, most nearly approaches the initial heat source for the pump. Also, it is just as "free" as the natural heat sources. The loop water is not affected by climatic conditions or local legal restrictions, and since its temperature can be controlled, a high coefficient of performance can be obtained by its use, a factor which would minimize both the size of the unit and its power consumption.

Heat transfer between water and the refrigerant is especially suitable to small and efficient heat exchangers. Scaling and corrosion of heat transfer surfaces are kept to a minimum. The pressure drop of the refrigerant is small in relation to that in ground coils. In addition, much less refrigerant is required. However, the application must be confined to an area immediately surrounding the plant.

Questions arising in connection with a power plant sized to meet the demands of a housing project include the following:-

a. During the coldest weeks of the year, additional heat would be required. Ways in which the deficiency could be met include auxiliary heating with low pressure steam from the turbine, tapping-up with electric strip heaters, or taking an additional load such as from a nearby shopping-center. The last would seem preferable.

b. The cost of generating power in the small plant is to be considered. Although the resulting savings might be large, it is not yet possible that a small plant can generate and distribute power as cheaply as a large central station.

If we considered, as an example, the steam-electric generating plants in Baltimore, Maryland, the aggregate rated capacity of 462,000 kilowatts are as follows:

Westport	125,000	Kilowatts at	25	cycles
"	125,000	"	60	"
Riverside	120,000	"	60	"
Gould Street	72,000	"	60	"
Pratt Street	20,000	"	25	"
Total	462,000	Kilowatts		

The 60 and 25 cycles are tied together by 30,000 KVA frequency changers located at the Westport plant.

If the generators require eight pounds of steam for every kilowatt hour (average for most generators): then,

$$462{,}000 \text{ kw} \times 8 \text{ lb. (for every kW/hr)} = 3{,}696{,}000 \ \frac{\#\text{steam}}{\text{hour}}.$$

Normally, the steam is discharged at an atmospheric temperature of 79°F. and of approximate enthalphy 950 $\frac{\text{Btu}}{\text{hr}}$.

Sixty thousand Btu/hr are required for heating a well-insulated six room residence of average size, when the outside temperature is -10°F. If this is true, then the plants in Baltimore could supply heat to 50,000 homes and under the same principle, the three turbines of 200,000 kw each in Washington, D. C. could supply 76,000 homes.

$$600,000 \text{ kw} \times \frac{8}{\text{kw}} = 4,800,000 \frac{\# \text{ steam}}{\text{hour}}$$

$$4,800,000 \times 950 = 4,560,000,000 \frac{\text{Btu}}{\text{hour}}$$

$$\frac{4,560,000,000}{60,000} = 76,000 \text{ homes}$$

Could this energy heat be stored or transferred to other forms of energy, we would have an endless supply to draw upon.

4. Solar: In the matter of solar radiation absorbers, we shall again note four factors, including firstly, the definition and characteristics of an ideal absorber; secondly, the description and performance characteristics of successfully used solar absorbers; thirdly, the treatment of absorbing media; and lastly, the placement of the solar absorber. Under the second item, such equipment as plates, tubing, irregular surfaces, filled translucent blocks, mirror-type devices (plane, parabolic and other), other solar absorbers and the factor of the earth, both treated and untreated, must be studied. Under the third item, treating of absorbing media, we shall note both surface treatment, and treatment of mass or aggregate. With the fourth point, placement of the solar absorber, we shall study location (as relative to the building), orientation and the angle of elevation.

When we come to the heat pump pickup device, we note two points; firstly, application, as regards use of the device in combination with the solar radiation absorber, with a collecting device or medium and with storage; secondly, the kind and design of the device.

In a consideration of solar energy storage, we are obliged to consider such factors as an ideal storage media, with its chemical and physical properties; existing storage media and properties, such as liquids (water and other); chemical compounds with phase change, including those with a change between 30° and 60°F., and those with change at temperatures beyond 60°F; solids, including crushed aggregates, earth and other. Two remaining points to note in connection with solar energy storage include its location in basement, attic, or elsewhere and its purpose, including factors such as heat storage <u>after</u> delivery from the heat pump, and solar energy heat storage <u>before</u> delivery to the heat pump.

5- <u>Earth</u>: Earth is the most widely available heat source. It has a smaller temperature change than does air and its installation costs less than any other system. It is a practical source of heat, providing radiation and warm convection currents to the surface. It is this heat from underground water which is used to replenish the heat drawn from the earth by the heat pump. The effect of soil and the factors affecting earth temperature and its cyclic variation should be taken into consideration.

Provision for maintaining high thermal conductivity in earth includes water application on the surface, water injection pipes buried in the earth, and other proposed methods. Professor Chas. A. Shreeve, Department of Mechanical Engineering at the University of Maryland, has suggested a reversal in the cycle and restoration of the heat in the ground for more efficiency during the heating season and at the same time, use of the heat pump for cooling purposes. This problem is being considered, as the control of the heat pump lies in the climate and soil factors. (18 & 51)

In a study of heat pick-up and disposal pipes and devices installed in earth, we shall need to note the types of application (for indirect heat transfer fluid circulation, and for direct refrigerant use); the kinds of device, such as plain pipe or tube, serpentine pattern tube, concentric tubes and flat plates; the position of the device (vertical, horizontal, inclined, looped, etc).

Methods for improving heat transfer between earth and metal necessitates the study of fins, webs, studs, etc. on the exterior of tubes; other methods supplementary to the sections on backfill materials and provisions for mainteinance of high thermal conductivity in earth.

The possibility of using earth for thermal storage with heat pump units must be studied in order to utilize both surplus heat from the heat pump and the off-peak energy in direct resistance heating elements.

Data required on heat transfer fluids for the indirect method includes information on the physical chemical properties required and the fluids available, such as antifreeze solutions for automotive vehicles (including ethylene glycol (prestone), glycerine, and alcohol); brine for ice and cold-storage plants; other thermal process fluids.

As to the matter of performance and comparative economics of methods and devices, we shall need to note the coefficient of performance for heating, and the power kilowatt per ton of refrigeration for cooling. In addition to these points, we should study the annual energy consumption for heating and cooling and the annual cost including operating expense and fixed charges.

6-<u>Miscellaneous</u> general problems encompass a study of possibilities of space heating units, an analysis in detail of hot water heating features, methods of determining heating requirements, radiant heating possibilities using heat pumps, humidity control problems, home designs to take advantage of heat pumps, the possibilities of combination units, an analysis of the effect of storage on performance and a thermodynamic analysis of heat pump cycles.

Investigation of gas, oil and coal as primary energy sources should be undertaken and the cost with relation to heat pump installation compared.

Other technical problems to be met are:-

a-Automatic defrosting
b-Automatic capacity modulation
(The response must reverse from summer to winter conditions.)
c-Automatic switching
d-Compressor unloading and sequence starting of auxiliaries
e-Automatic reversals of refrigerant flow
f-Dissipation of losses from auxiliary motors
g-Noise and vibration
h-Serviceability (a vital requirement)

C. APPLICATION AND COST.

Contrary to popular opinion, the advent of the Heat Pump depends much more upon the economic factors involved than upon technological advances yet to be made.

1-<u>Industrial</u>: The industrial application of the heat pump should be considered first. One important point to note is the fact that the heat pump has not been used in industry in this country to the extent possible, in contrast to Switzerland where there is a scarcity of ordinary fuels and an abundance of water power. Possibly, in western and northwestern parts of the United States, the heat pump will find increasing uses in industrial processes.

The concentration of solutions, such as milk, fruit juices, etc., is an industrial possibility for the heat pump. This process can be performed economically by three methods:

a. <u>Multi-stage evaporation</u>: One of the most economical methods, providing the fuel is inexpensive, and initial investment costs reasonable.

b. <u>Direct compressions of the vapor</u>: This methods requires an increase of vapor pressure so as to obtain the necessary temperature differential between the distillate and the solution. It has the advantage of eliminating temperature drops between the fluids and the heat transfer surfaces, which are required with an indirect compression system. This method is used successfully in Switzerland.

c. <u>Indirect compression system</u>: This method makes use of separate heat transfer surfaces and separate thermodynamic fluid in a closed cycle. The disadvantage of this method lies in its requirement of temperature differences between the distillate and the heat transfer surfaces in order to keep the size of the latter within economical limits. The closed cycle, however, insures freedom from corrosion and eliminates the precipitation problem. The density of the thermodynamic fluid is such that a physically smaller machine can be used instead of the larger one required by a direct compression system.

The heat pump can be used to advantage in factories. A heat pump and dehumidifying unit was used at the Chevrolet plant of General Motors Corporation in Southern California. Cable replacements began to be a noticeable maintenance item in the underground electrical power distribution system of the plant as a result of disintegration of the sheathing of the electrical cables due to the corrosive action of water distilled from relatively high dew point air coming into contact with the walls of the cableway, which are at the same low temperature as the ground. When this item reached thousands of dollars per month, the company installed a 5HP heat pump to provide 1300 cubic feet per minute of dry warm air at a pressure of seven inches water gage forced into the cableway system. Since this heat pump has been in operation, no conduits have been replaced and all manholes have been dry and easy to work in. (22)

For the Oregonian Building, completed in June, 1948, in Portland, Oregon, there was designed a device known as the water-to-water heat pump. This building was especially constructed for a modern newspaper with a daily circulation of 219,000, (which reaches 276,000 copies on Sunday) but also contains offices for editorial and business groups associated with the production plant. In addition to these groups, it houses a radio station and studios connected with the publishing operation. It was necessary, in the design of this building, to use complete air conditioning, because ventilating courts had been eliminated in order to facilitate the uninterrupted flow of material in the mechanical processes of publication.

Analysis of the heat pump load of 6,000,000 Btu/hr for cooling and 4,8000,000 Btu/hr for heating was made for the building. Three water wells were drilled, two to a depth of 200 feet, with the well-water temperarure at 55°F. in cold months, and the third was drilled to 930 feet with the well-water temperature at 58°F. in the hot months.

During the heating season, well-water from the 200 ft. depth supplied the settling tanks. Well-water at 55°F. was pumped to the suction of heating pump to a condenser. The water travels from the condenser to the heating coils and then to the deep well through a relief valve.

During the cooling season, well-water from the 930 ft. well at 58°F. is pumped firstly into the settling tank, then to the suction line of a cooling pump. The pump then supplies the water to the evaporators, after which the water is conveyed to the cooling coils of the air conditioning systems and then discharged to the sewer through the pressure relief valve.

2-Commercial: An example of the use of the heat pump in an office building is the Equitable Building in Portland, Oregon, the first non-utility office building in this country to use the heat pump. Since air conditioning, which was a basic concept in its design, required cooling capacity greater than design heating capacity, and since comparatively low electrical costs indicated economic feasibility, the design included the heat pump for both cooling and heating(33)

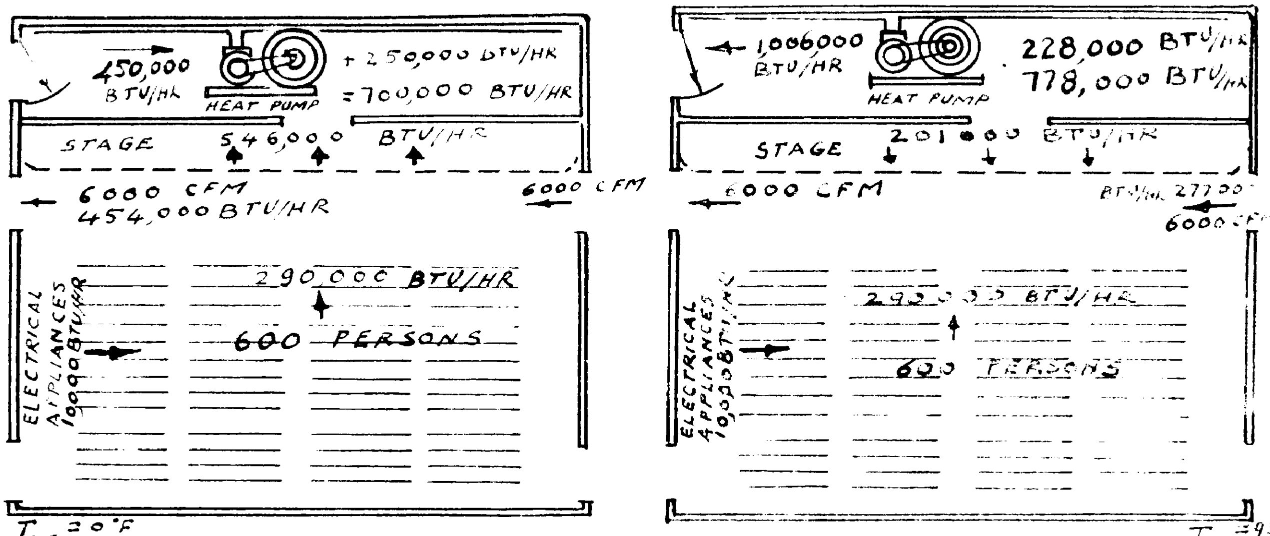
Dehumidification, which is required frequently in Portland at temperatures between 60° and 80°F., was necessary in this case. Heating and cooling by air entirely was the system adopted.

As to the air conditioning system, warm and cool plenum systems were provided and air from them was mixed as required by 11 zone thermostats for the typical floors. One humidistat per floor (subject to reset from the basement mechanical equipment room) controlled dehumidification.

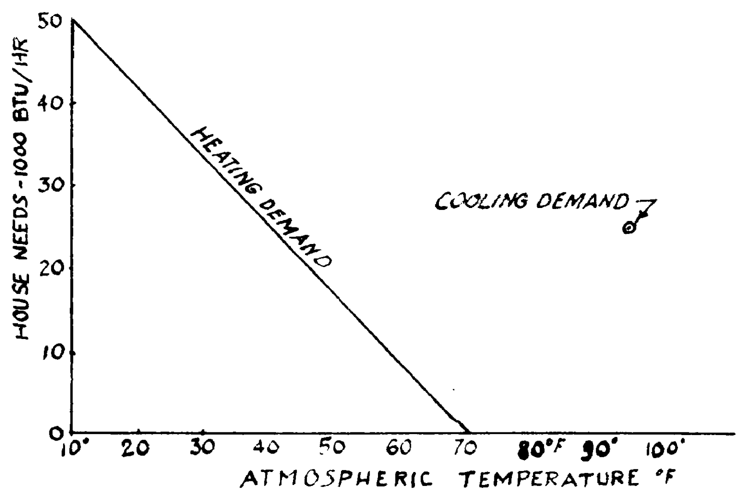
The heating (warm water circuit) was a closed system when heating was greater, during which time a thermostat in the return determined the number and selection of condensing units to be operated. Use of a recovery coil was made; preconditioning air with waste water resulted in reduction by one-third of the heating load at design conditions.

Figure 23

HEAT BALANCE



WINTER HEAT BALANCE SUMMER HEAT BALANCE



EFFECT OF CLIMATE ON HEATING & COOLING DEMAND FOR SMALL HOUSE

YEARLY ENERGY CONSUMPTION OF ELECTRICAL APPLIANCES

ELECTRICAL APPLIANCES	(KWHR) ENERGY	ELECTRICAL APPLIANCES	YEARLY ENERGY (KWHR) WITHOUT HEAT PUMP	YEARLY ENERGY (KWHR) WITH HEAT PUMP
LIGHTING MISC. APPLIANCES	346			
REFRIGERATOR	31[illegible]			
RADIO	80	IRONING MACHINE	100	100
FLAT IRON	64	PERCOLATOR	72	72
WASHING MACHINE	24	WATER HEATING	2625	1308
VACUUM CLEANER	20	RANGE	975	975
TOASTER	15			
ELECTRIC CLOCK	14	HOUSE HEATING	0	8900
ROASTER	225	HOUSE COOLING	0	2000
SPACE HEATER	45			

As the outside temperature rises, or as the cooling load became greater, an immersion thermostat in the return from the cooling coils took over the condensing unit operation through the step controller and selector. Certain valves were repositioned and the chilled water circuit became a closed system.

One feature of the heat pump control was a provision for closing the valves in the hot and chilled lines to the individual machines when a machine was not in operation. This is a factor which prevents the mixing of return and discharge water and the resulting extreme temperatures required from the machines in operation to maintain the temperatures at which there were two thermostatic controls.

The changeovers were completely automatic and no auxiliary heat was provided. Heating was largely by-product, as a supply of by-product heat was available from the cooling, made possible by the heat pump.

3-<u>Residential</u>: A primary application study of the heat pump should be in relation to its ability to provide satisfactory results for the private home at less cost than present-day heating and cooling methods. Here application problems involve a study of comfort requirements in terms of heat pump operation, an analysis of air distribution systems, the determination of noise and vibration requirements, the collection of performance data, the study of service problems, and the possibilities of applying heat pumps to existing homes.

4-<u>Cost</u>: The heat pump is, of course, simply a refrigeration unit with an electric motor, compressor, condenser and evaporator operated to make use of heat. It may be used to cool space needing conditioning in the summer, as it heats in winter by changing the functions of the condenser and the evaporator. The heat load assumed for the cooling load is 50% of that assumed for the heating load. The expense of the initial installation of the heat pump is still high for practical over-all use for industrial and private purposes. These conditions will improve as building costs decrease generally.

Weather bureau records offer a rather simple means of calculating the approximate percentage of the total monthly time in which the heat pump will operate, by supplying heat on the basis of the average coldest day that may be expected, calculated from records of several years previous.

Since the heating system must run almost continuously on this day, the degree days for this average coldest day multiplied by the number of days in this month will give the equivalent monthly degree days. By dividing the weather bureau's average degree days for the month by the equivalent monthly degree days and multiplying by 100, the percentage of time the heat pump should operate in that month may be ascertained.

Table No. 7

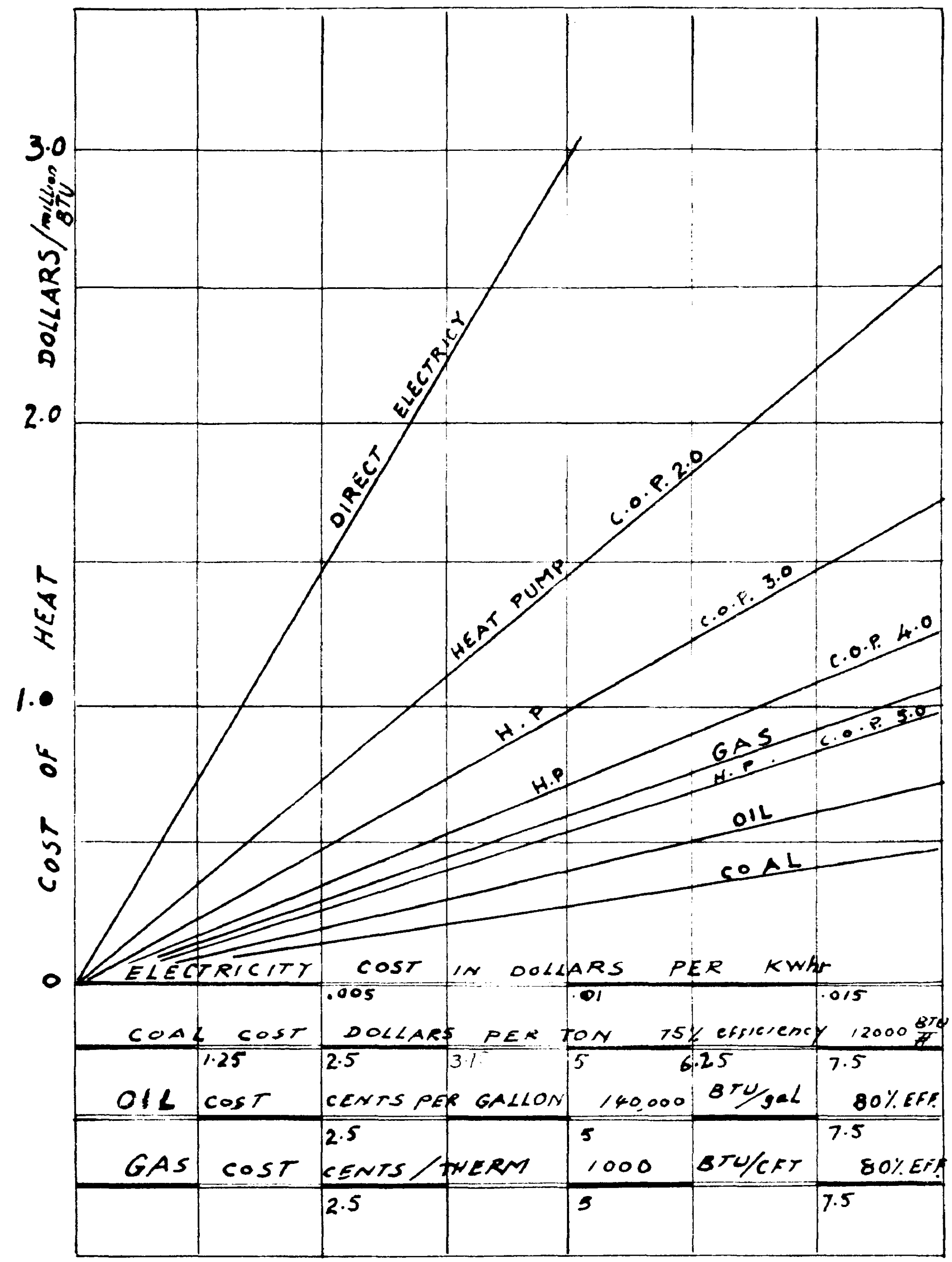
AVERAGE YEARLY DEGREE DAYS IN THE UNITED STATES FOR A PERIOD OF 80 YEARS

Atlanta	3002	Cleveland	6171	New York	5306
Boston	5943	Denver	5863	New Orleans	1208
Chicago	6287	Detroit	7989	Philadelphia	4749
Portland	4379	San Francisco	3143	St. Louis	4610
		District of Columbia	4085		

Table 8 STEAM CONSUMPTION FOR VARIOUS TYPES OF BUILDINGS
(pound per degree-day per thousand cubic ft. of heated space)

Apartment buildings	1.78	Banks	.88
Residences	1.32	Auto Sales & Service	.83
Hotels	1.46	Religious Assemblies	.58
Printing Establishments	1.25	Department Stores	.57
Clubs & Lodges	.96	Garages (Storage)	.42
Retail Stores, Theaters	.90	Office Buildings	.975
		Loft and Manufacturing Buildings	.89

Figure 24 COMPARATIVE COSTS OF HEAT CONDITIONING INSTALLATIONS



As an example, let us assume an office building having 1,200,000 cubic feet of heated space; then the seasonal steam consumption in Washington, D. C., on the basis of the last 1950 record of degree-day, would equal:

$$4,046 \times \frac{1,200,000}{1,000} = .975$$

or 4,730,000 pounds.

Using the coefficient of performance (C.O.P.) for comparing the cost of the heat pump with other heat conditioning installations, and assuming that each 1 Kw hour energy delivered is equivalent to 1 theoretical C.O.P. for direct heaters; having the assumptions of:

3413 Btu/hr = 1 Kw hr. = .75 cents cost (SPA rate).

Experience indicates that the average C.O.P. for many domestic and commercial heat pump installations is approximately $3.30. The graph considers this assumption in showing:-

Natural gas of efficiency 80% and 1,000 Btu/cubic ft.

Oil of efficiency 30% and 140,000 Btu/gallon

Coal of efficiency 75% and 12,000 Btu/lb.

Generally, the heat pump can become competitive with other fuels on the basis of operating costs at a power rate of 1%, or less, per kilowatt hour.

Since there is apparently no serious engineering problem involved, the over-all cost is the principal factor affecting the feasibility of its application. However, the costs should decrease considerably through mass production and research.

To summarize, the heat pump offers a clean, odor-free heat, eliminating the necessity for a chimney and the use of fixed double-glazed windows, gives the advantage of storm windows and eliminates sliding sash costs. It offers low-grade heat of greater comfort, with air-conditioning both winter and summer in a single unit of one price. Heat can be delivered quickly, as there is no time lag involved, and there are no waste products to be disposed of.

C. UTILIZATION OF VERTICAL GROUND PIPE.

The aspect of the heat pump in which the author is most interested is the utilization of ground heat for its installation. As much research is being done with horizontal grid tubes buried in the ground, the author studied vertical pipe installation and its economies in comparison with other methods of utilizing the various heat sources for the heat pump. Its aims were to determine the quantity of heat the pump can supply and the coefficient of performance for several variables.

The reversal in the cycle and restoration of the heat in the ground for more efficiency during the heating season and at the same time, use of the heat pump for cooling purposes, are considered in this research in their relationship to the controling factors of climate and soil.

Heat absorbed by each square foot of soil can be calculated from:

$$\frac{Q}{A} = KT_o \sqrt{\frac{2Pa}{\pi a}}$$

where: Q= heat, Btu/lb, K= conductivity, T_o= maximum temperature
A= area (square ft.), a = diffusivity
P= duration of sinusoidal temperature variation.

Figure 25

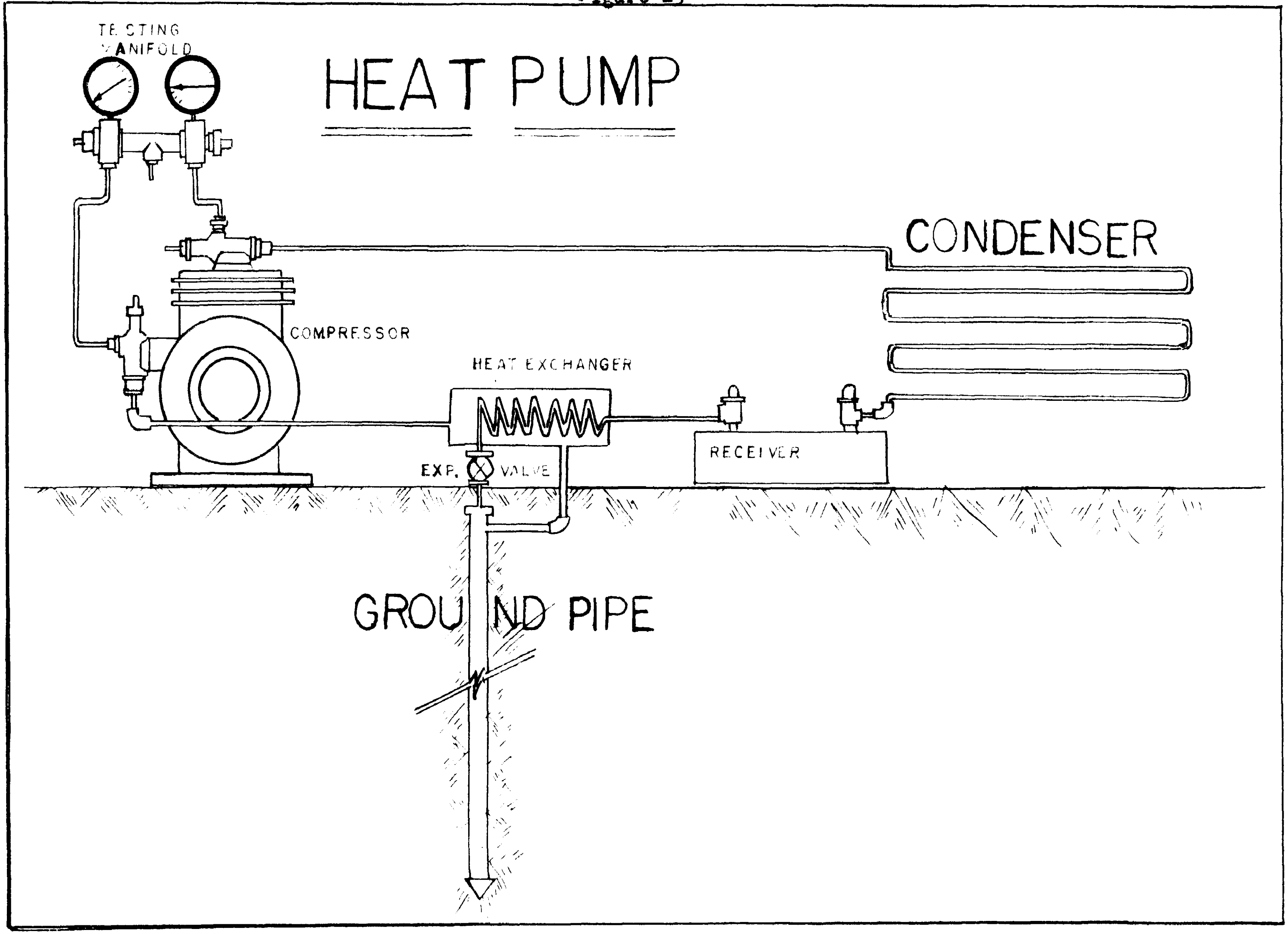
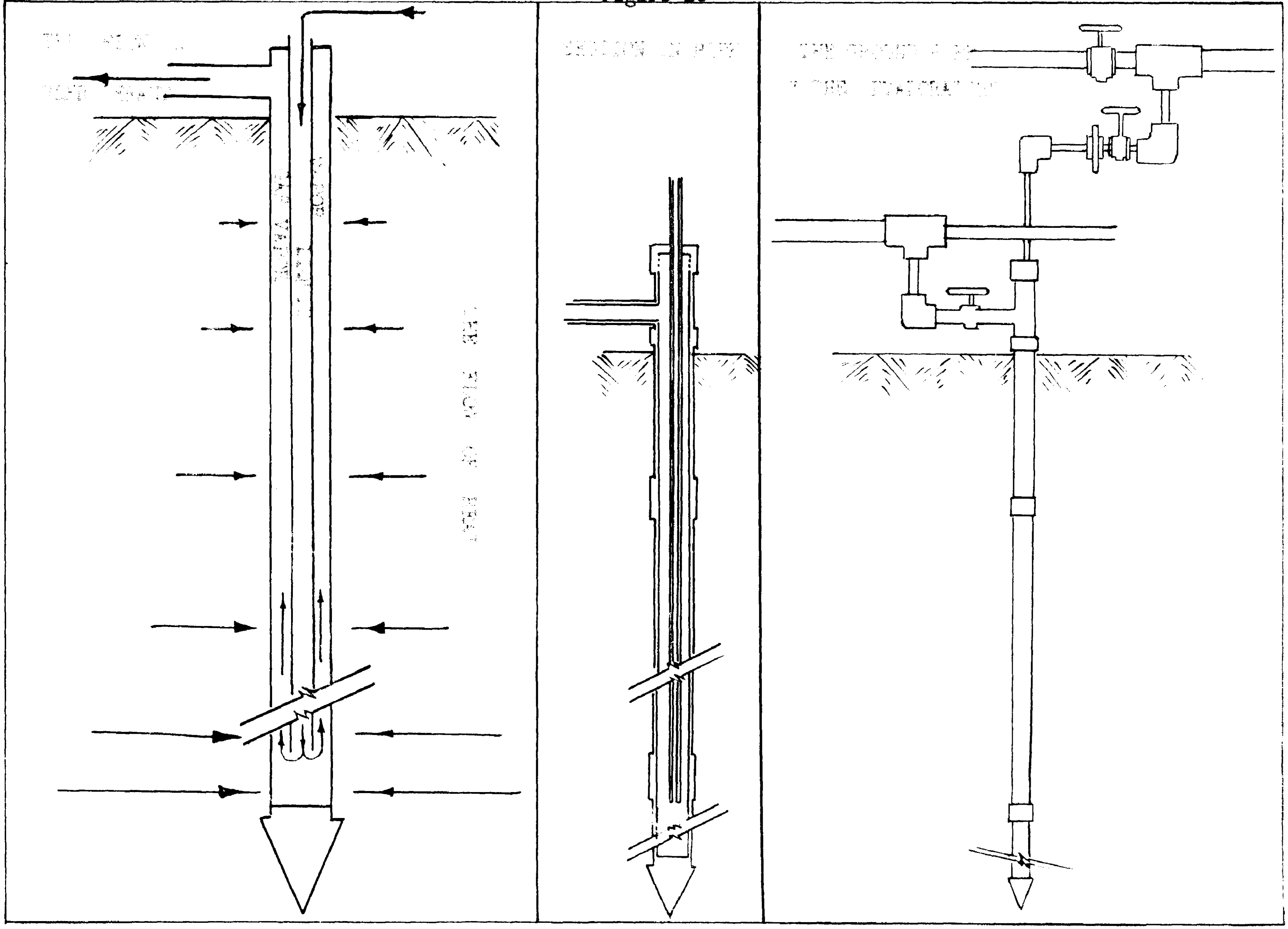

TESTING
MANIFOLD
HEAT PUMP
CONDENSER
COMPRESSOR
HEAT EXCHANGER
RECEIVER
EXP. VALVE
GROUND PIPE

Figure 26



In most temperate climates, rain during the heating season adds a good deal of heat. In winter months the sun may radiate considerable heat to the soil even when the air temperatures are rather low. Such additions might be approximated by a study of weather bureau records giving daily rainfall and sunshine data. Efforts were made to determine moisture migration, which generally aids in heat absorption for space heating, and reverses its effect for space cooling.

In colder climates, the vertical ground pipes of the heat pump will have the advantage of obtaining the heat from a greater depth, thus by-passing the temperature lag which is found near the surface.

In warm climates, as in Washington, D. C., and in the Nile Valley, where the sun shines most of the winter, the shallow pipe will utilize the solar radiated heat to avoid the effects of this temperature lag.

In winter, the average temperature at any depth can be calculated from records prior to operation of the heat pump. The average temperature will be low after the heat pump starts. Measurements might well be made in advance to cover a long period of time. Assuming that the soil has regained normal temperature conditions during the summer, the same design as was used for winter may continue on without change. If, after the summer season, the heat of the soil has not attained normal temperature conditions, the ground pipe may no longer be adequate, and adjustments in the design to compensate for the change can be made.

Proper design of ground pipe requires knowledge of the following factors:

1-The maximum thermal units per hour necessary for heating or cooling.

2- Soil characteristics; types of soil, natural density, etc.

3-Reliable information on heat transfer coefficients for vertical pipe at various depths.

4-The average number of hours of actual operation during the heating or cooling season.

5-The annual heat absorption per square foot of soil area.

Success of a ground pipe depends as much on heat availability in the soil as it does on the pipe size, spacing, length, etc. The soil heat is supplied from sun radiation, from heat carried into the soil by rain, from underground moisture migration, and from heat originating in the earth's core.

The discrepancies found in the use of vertical pipe to absorb and deliver heat from and to the soil, have been due to a lack of knowledge of all the factors involved, especially the thermal constants of the soil. The aim of this research is to determine some of these factors.

The surface area is an important item in business sections, and, the heat needed is important in order to determine the minimum length of pipe necessary. Indications are that the average city lot is too small to supply a home with sufficient heat, so that the horizontal ground pipe is practical only where unlimited depths are available, or where special soil, moisture and subsurface conditions are found.

The aim in the study of earth as a storage medium (as a more promising medium than any utilizing air, well water, etc.) was to examine some of the factors determining the practibality of a system involving a heat-exchanger, (the vertical pipe inserted in the soil).

Solutions are presented in general terms to permit application to various sets of circumstances.

Let us assume the surfaces as located in a conductive soil, having the following specifications:

$K = 0.896$; $c_p = 0.45$; $\rho = 103$; $a = .0193$

The heat withdrawal is at a constant rate at the pipe surface and the contact with earth of infinite extent and uniform initial temperature. The temperatures at various points are calculated from the equation:

$$t = \frac{A}{5.4575\ K} \int_{-\infty}^{\log_{10} \frac{a\theta}{r^2}} e^{-\left(\frac{r^2}{4a\theta}\right)} d \log_{10}\left(\frac{a\theta}{r^2}\right)$$

where t = temperature change in degree Fahrenheit

θ = time in hours

a = thermal diffusivity = $\frac{K}{\rho c_p}$

r = radius in feet.

The temperature change t, at the surface of the pipe, is of interest, since, in the use of the pipe as a heat source, the design must be such that the heat receiving surface will not fall below a certain minimum temperature after an extended period. These temperatures may be found by inserting values for r in the above equation which correspond to the radii of the cylinders of the dimensions of r to be considered.

The problem was solved with the aid of certain tables giving the values of a related integral. An empirical expression is used which gives the value of the integral with negligible error for values $\log_{10}\left(\frac{a\theta}{r^2}\right)$ greater than one. Therefore, the equation will be:-

$$t = \frac{A}{5.4575\ K}\left[\log_{10} \frac{a\theta}{r^2} + .106 \frac{r^2}{a\theta} + .351\right]$$

From this equation, we notice that the temperature changes quickly at the pipe surface at the beginning of the withdrawal of heat from the earth, but much less rapidly as time proceeds. Increasing the pipe diameter results in a small decrease in the temperature difference between the initial ground temperature and the surface temperature.

For best results, the heat transfer surface should be located deep in the earth in a form of vertical pipe. If the heat exchanger is in the form of a planer coil, the surface required is only about one fourth as great for balanced heat extraction and return, as for heat withdrawal only. It is only when heat interchange with the atmosphere is minimized, that the relatively small surface for the cyclic case can be realized. This required the heat exchanger to be driven deep into the soil, with the coil consisting of a row of vertical pipes.

The Engineering Building of the University of Maryland at College Park, is underlain by about 150 feet of Coastal Plain sediments, beneath which are hard crystalline rocks. The upper part of the Coastal Plain sediments are of Pleistocene Age and the lower part of Cretaceous Patuxent Age. They consist of alternating lenticular beds of clay, sand and gravel. Ground water occurs in the porous sands and gravels, and since the beds are lenticular, the exact depth of water at a specific locality cannot be predicted.

Mr. J. T. Singewald, Jr., Director of the Department of Geology, Mines and Water Resources, of Baltimore, Maryland, gives the water table level as about 25 feet deep. The temperature of the water most of the year is 57.5°F. Weather Bureau records give the soil surface temperature for spring and fall (almost eight months yearly) as 65.4°F., while the dry bulb temperature of the air is 60°F.

The temperature gradient, under these conditions, and for the type of soil at the University of Maryland is 1°F. increase per 60 feet in depth. However, at a depth of 100 feet, the normal gradient is likely to be affected by atmospheric temperatures and may vary several degrees during the cycle of the year.

Soil Characteristics at the University of Maryland

Well No. 1 (2 miles east of Mechanical Engineering Building)

Bedrock reported 159 feet below land surface.

Well No. 2 (4 miles west, south-west of Mechanical Engineering Building)

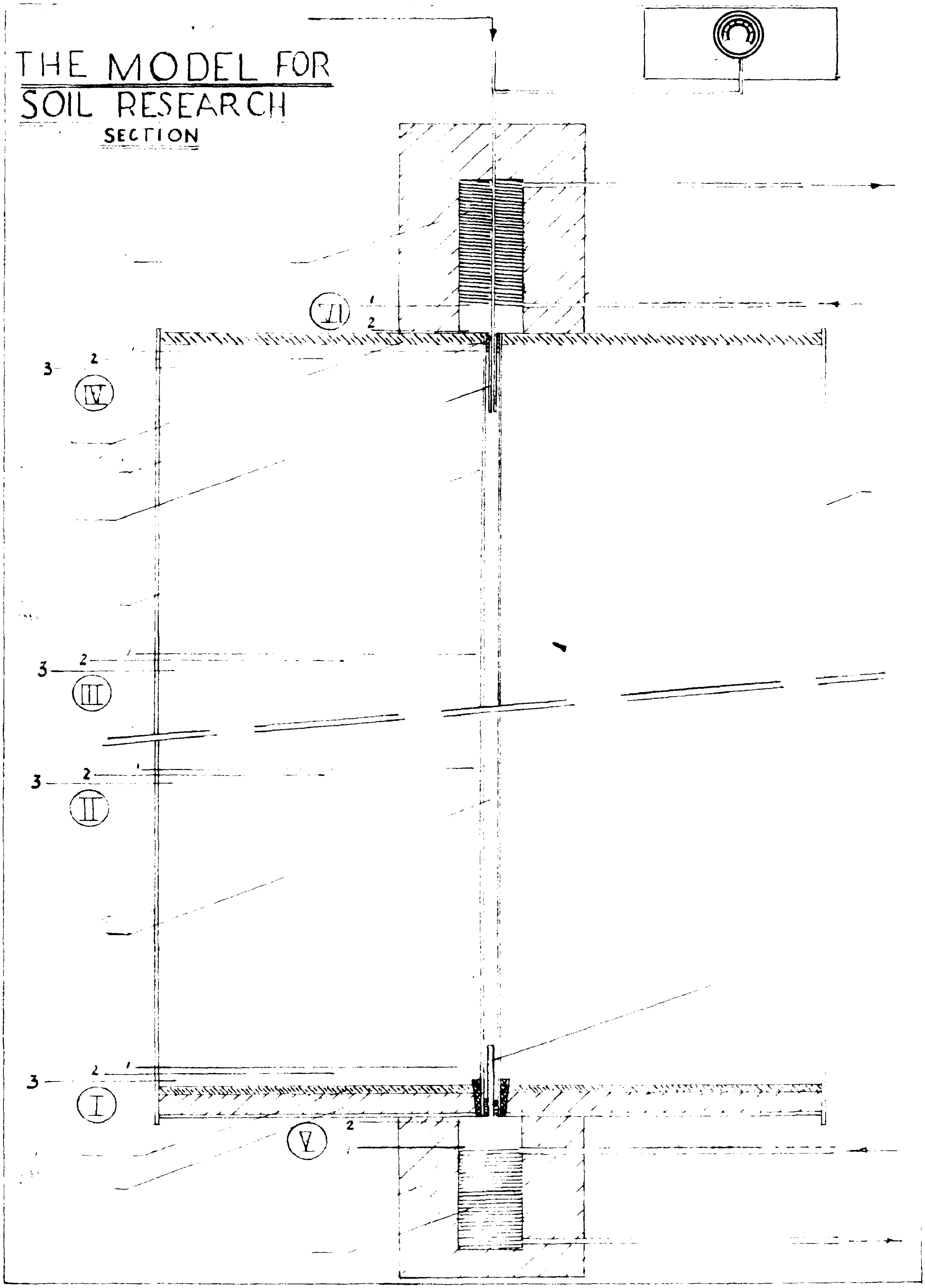
Log:	0-11	feet	sandy, yellow clay with some gravel
	11-18	"	dry, yellow sand
	18-32	"	red clay
	32-49	"	blue clay
	49-120	"	red clay
	120-126	"	blue clay
	126-145	"	blue clay with streaks of sand bearing considerable water (much iron content)
	145-191	"	soft rock of soapstone nature
	191-193	"	quartz
	193-209	"	soft gneiss rock
	209-284	"	hard gneiss rock

Well No. 3 (1.4 miles south by east of Mechanical Engineering Building)

Log:	0-3	feet	yellow clay
	3-33	"	gravel
	33-40	"	sandy clay
	40-60	"	red clay
	60-80	"	blue clay
	80-110	"	red clay
	110-118	"	brown clay
	118-136	"	blue clay
	136-151	"	red and brown clay
	151-171	"	water-sand

Well No. 4 (1.4 miles south by east of Mechanical Engineering Building)

Figure 27



Log:	0-8	feet	top soil and clay
	8-21	"	coarse
	21-28	"	sand and gravel
	28-43	"	yellow sandy clay
	43-55	"	red clay
	55-83	"	yellow clay
	83-104	"	Gray clay
	104-148	"	red clay
	148-152	"	blue clay and wood
	152-165	"	medium blue water sand
	165-168	"	medium coarse gravel
	168-174	"	medium coarse blue sand

1. EXPERIMENT ON HEAT TRANSFER IN THE SOIL.

a. <u>Description of experiment</u>: The experiment model consisted of a cylindrical steel tank, 35 inches high, 22 inches in outside diameter, 21.5 inches inside diameter, and having conductivity equal to 26.2, density equal to 493 pounds per cubic ft. and specific heat equal to 0.11 Btu per pound. This was filled with sand for which certain tests were carried out in order to obtain detailed specifications:

The specific gravity of dry sand	= 2.57
The specific gravity of saturated sand	= 2.60
The apparent specific gravity	= 2.66
Absorption	= 1.2 %

The bulk specific gravity as defined in the Standard Definitions of Terms relating to specific gravity (ASTM DESIGNATION: E12) of the American Society for testing materials, calculated from the formula:-

$$G_{BD} = \frac{A}{V-W}$$

A is weight in grams of ovendried sample in air
V is volume in millimeters of flask
W is weight in grams or volume in millemeters of water added to flask

The same equation for $G_{BS} = \frac{500}{V-W}$

The apparent specific gravity is calculated from the formula:-

$$G_{app} = \frac{A}{(V-W) - (500-A)}$$

The percentage of absorption is calculated from the formula:-

$$A_{BS}\% = \frac{500 - A}{A} \times 100$$

Duplicate determinations should check to within .02 in the case of specific gravity and .05% in the case of percentage of absorption.

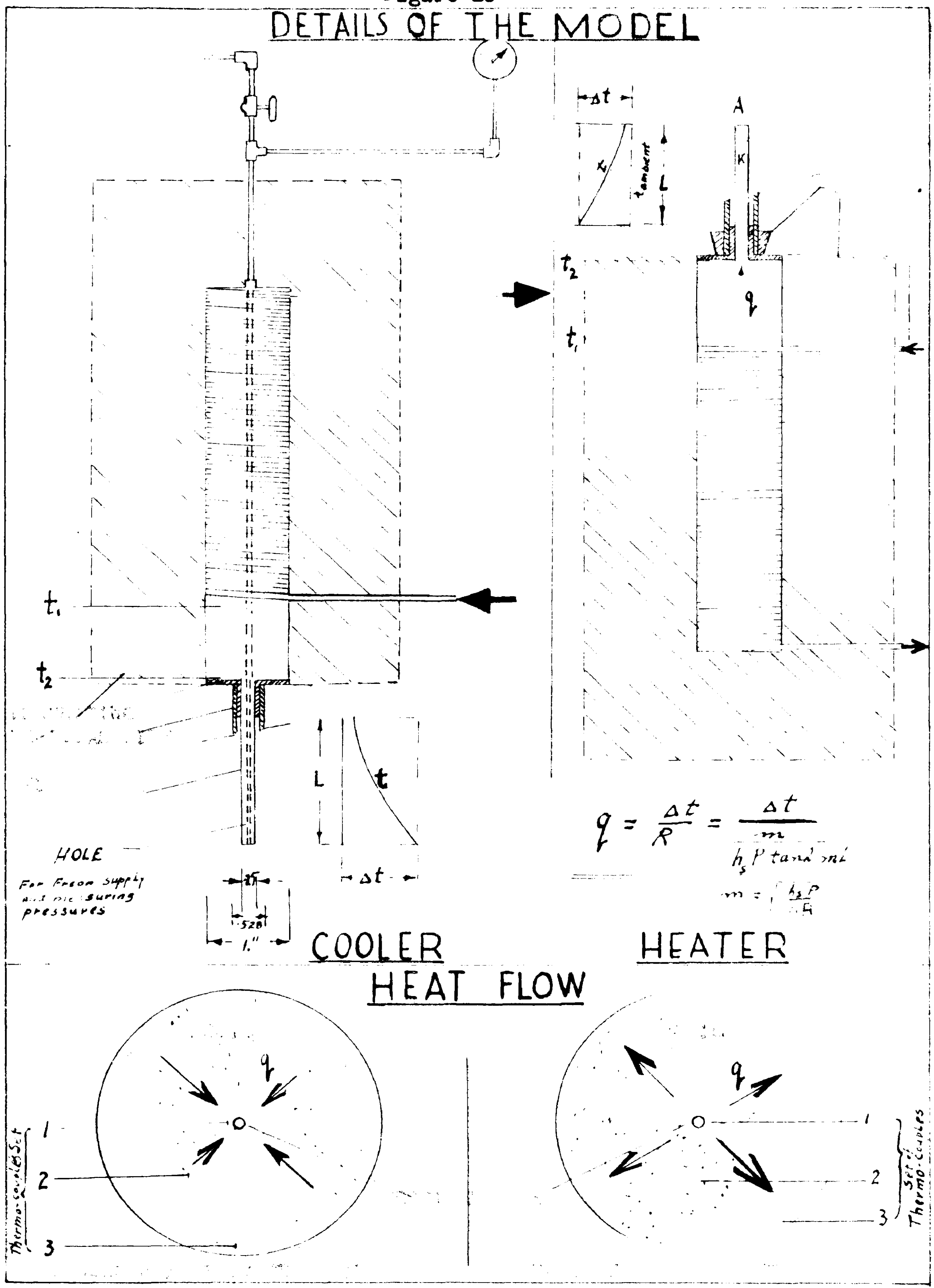
(Sieve Analysis) This method of testing covers a procedure for the determination of the particle size distribution of fine and coarse aggregates, using sieves with square or round perforations.

Samples were dried to substantially constant weight at a temperature not exceeding 110°C. (230° F.). They were then separated into a series of sizes using sieves of #4 (4760-micron), #8, #16, #30, #50, #100, F. M., and #200. F. M. is the finest modulus which has an accumulative percentage of sand retained upon the abovementioned sieves. This classification is necessary in order to comply with the specifications for the sand used in the heat pump experiments.

The sieving operation, conducted by lateral and vertical motion, was accompanied by jarring action in order to keep the sample moving continuously over the surface of the sieve. This continued until no more than one per cent by weight of the residue passed through any sieve during one minute. The weight of each size was then determined on a scale. The result of the test is seen below:

Sieve #	4	8	16	30	50	100	F.M.	200
Test	3.0	12.4	21.8	40.5	82.9	94.6	2.55	2.44

Figure 28


DETAILS OF THE MODEL
Δt
t
tambient
L
A
K
t₂
t₁
q
t₁
t₂
L
t
Δt
HOLE
For Freon supply and measuring pressures
.528
1."
q = Δt/R = Δt / (m / (h_s P tanh mL))
m = √(h_s P / kA)
COOLER
HEATER
HEAT FLOW
q
q
1
2
3
Thermo-couples Set
1
2
3
Set of Thermo-couples

Dry sand: density (ρ) = 94.8
specific heat (c_p) = 0.19
conductivity (K) = 0.19
diffusivity (a) = 0.01056

Wet sand: density (ρ) = 102
specific heat (c_p) = 0.5
conductivity (K) = 0.65
diffusivity (a) = 0.01275

In the center of the tank was placed a vertical pipe of brass with conductivity equal to 49.3, specific heat equal to 0.09, density equal to 500 and having an outside diameter of 0.528 inches and an inside diameter of 0.3622 inches.

The cooler, of brass also, was tightly attached to the top of the vertical pipe. It was composed of three parts: The top part was covered with copper tubing containing running cold water. The second part, below this, was attached at both ends with two thermocouples, one inch apart, to measure the heat flow in the cooler. The bottom part (the fin), was inside the top part of the vertical pipe to cool the refrigerant. All parts were thoroughly insulated. A narrow opening ran through the fin, into the cooler, to the refrigerant supply tank with an appendage to the pressure gage for measurement of the specific values of the refrigerant. The top part of the cooler was one inch in diameter and 4.5 inches in height, surrounded by 0.25 inches of copper tubing. The second part of the cooler was also one inch in diameter and one inch in height. The fin section was 0.25 inches in diameter and 0.236 inches long.

The heater, of brass, was tightly attached to the lower end of the vertical pipe. Like the cooler, it was composed of three parts: the bottom section was 1 inch in diameter and 3.25 inches in height and covered with copper tubing (outside diameter equal to 0.25 inches) which conveyed running hot water. The second part was 1 inch in diameter and 1 inch in height, with one thermocouple at each end to measure the heat flow to the refrigerant. The top part of the heater was a fin 0.25 inches in diameter, which supplied heat to the refrigerant. The fin was 0.225 inches long. All parts of the heater werecompletely insulated.

Dichlorotetrafluoroethane ($C\ Cl\ F_2 - C\ Cl\ F_2$), which is known as Freon 114, was chosen as the refrigerant for the experiment because of the atmospheric conditions and the state of the model. Its physical properties are:-

Molecular weight: 170.9

Boiling point (at 760 - 1 atm) = 38.4°F. (3.55°C.)

Melting point: -137°F (-94°C.)

Critical temperature: 294.3°F. (145.7°C.)

Critical pressure: 474 lb/sq. inch abs.

Color: clear and water white

Odor: faint and like ether

Moisture content: not more than .0025% by weight

Non-condensible gases: not more than 5.0% by volume in vapor phase

High boiling impurities: not more than 0.05% by volume

Chlorides: none

Flammability: noncombustible and noninflammable

Specific heat of vapor (1 atm):

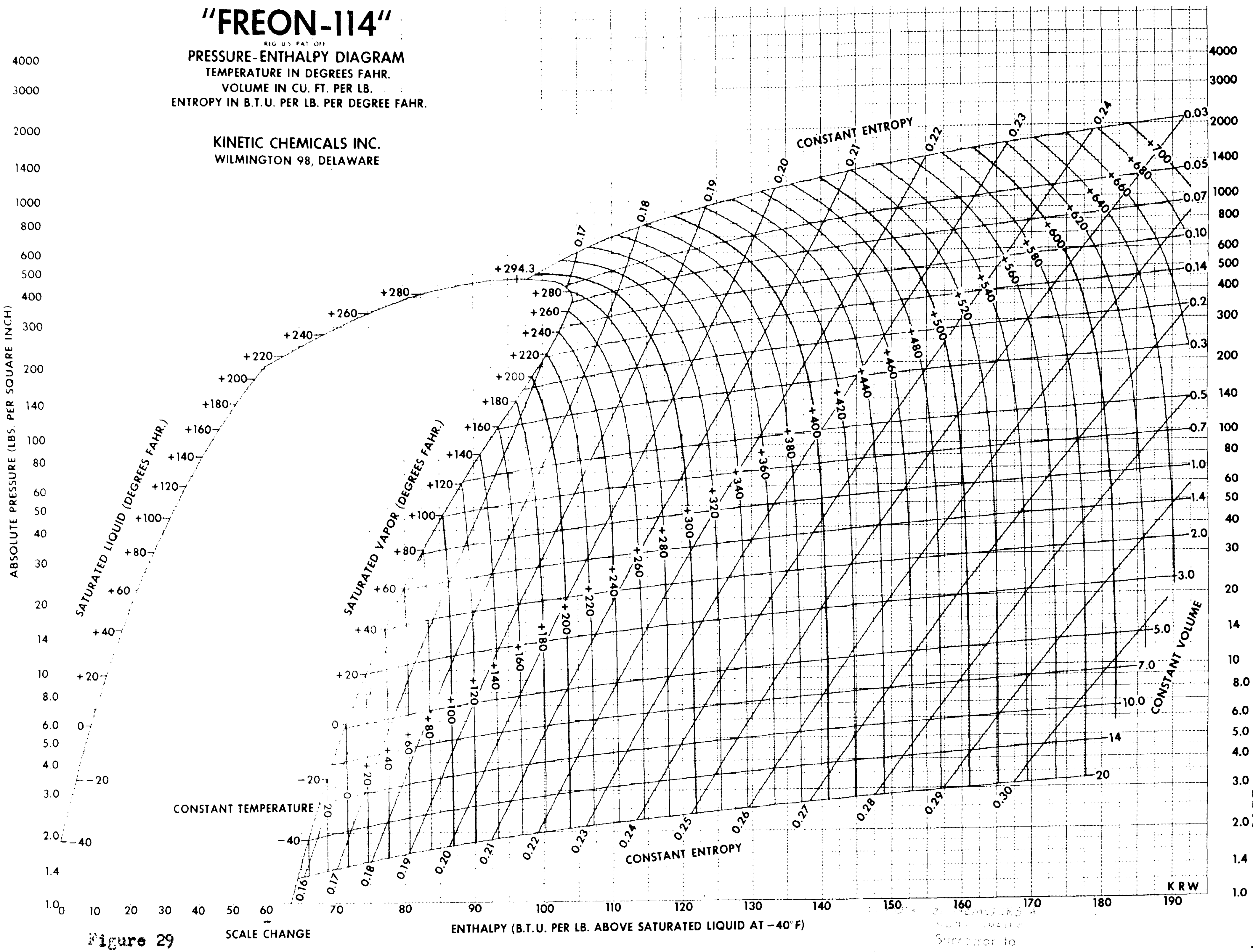

"FREON-114"
REG. U.S. PAT. OFF.
PRESSURE-ENTHALPY DIAGRAM
TEMPERATURE IN DEGREES FAHR.
VOLUME IN CU. FT. PER LB.
ENTROPY IN B.T.U. PER LB. PER DEGREE FAHR.
KINETIC CHEMICALS INC.
WILMINGTON 98, DELAWARE
ABSOLUTE PRESSURE (LBS. PER SQUARE INCH)
SATURATED LIQUID (DEGREES FAHR.)
SATURATED VAPOR (DEGREES FAHR.)
CONSTANT ENTROPY
CONSTANT TEMPERATURE
CONSTANT VOLUME
+294.3
SCALE CHANGE
ENTHALPY (B.T.U. PER LB. ABOVE SATURATED LIQUID AT −40°F)
KRW

Figure 29

Temperature °F.	C_p	C_v	$\frac{C_p}{C_v}$
110	.1629	.1502	1.085
160	.1696	.1571	1.08
210	.1763	.1641	1.074

Viscosity:

Temperature °F.	Vapor (1 atm.) (centipoises)	Liquid (sat'n pressure) (centipoises)
-40	.0095	.879
-20	.0098	.711
0	.0102	.598
20	.0106	.516
40	.0109	.454
60	.0112	.405
80	.0116	.366
100	.0119	.334
120	.0122	.307
140	.0125	.284

Thermal conductivity: (a) Freon 114 (liquid)

Temperature °F.	K
32	.0515
104	.0424
167	.0344

(b) Freon 114 (vapor)

Temperature °F.	K
86	.00646
194	.00811

Heat transmission coefficient:(a) Liquid:-

$$\frac{h_{av}}{K} D = 1.65\left(\frac{wC}{KN}\right)^{\frac{1}{3}}\left(\frac{\mu}{\mu_f}\right)^{\frac{1}{3}}\left(1 + .015\sqrt{\frac{d^3\rho^2 \Delta t \beta g}{\mu_f^2}}\right)$$

simplified to:

$$h_{av} = \left(\frac{V}{Nd_i}\right)^{\frac{1}{3}}(5.58 - .0095t)(1 - .001\,\Delta t)\left[1 + d_i(\Delta t)^{\frac{1}{3}}(1.54 + .007t_f)\right]$$

where: V = velocity of flow ft/min
N = pipe length in feet

(b) Condensing vapor:

$$h_{av} = \frac{.725}{n^{.25}}\left(\frac{\lambda\rho^2K^3g}{\mu D_o(\Delta t)}\right)^{.25}$$

simplified to:

$$h_{av} = \left[n\,d_o(\Delta t)\right]^{-.25}\left[5630 - .33t - .0043t^2\right]$$

where: t = temperature of condenser
n = height of pipe in ft.
d_o = outside diameter in inches

(c) Vapor

$$h = 36\, C_p\, G_1^{.8}\left(\frac{Z}{d_i}\right)^{.2}$$

simplified to:

$$h d_i = (V d_i)^{.8}(.611 + .011t + .00016t^2)$$

where V = velocity of flow ft/sec.
d_i = inside diameter in inches

The Freon 114 pressure enthalpy diagram shows clearly the relation of volume, entropy, temperature, pressure and enthalpy.

EXPERIMENTAL THERMAL CONDUCTIVITIES AT ONE ATMOSPHERIC PRESSURE

Freon Vapor

Temperature	Conductivity	
86°F.	.00646	Btu ft./ft.^{2} hr. °F.
194°F.	.00811	Btu ft./ft.^{2} hr. °F.

Freon Liquids

Temperature	Heat input Btu/hr °F.	K conductivity (Btu ft./ft.^{2} hr. °F.)
32°F.	1.232	.0515
104°F.	1.117	.0424
167°F.	1.017	.0344

Copper and constantan proved to be the best materials for measurement of heat temperatures and were therefore used in the thermocouples. There were two for the cooler, two for the heater and four sets of three each placed at four levels within the tank, each placed at a certain distance from the central axis. These sixteen thermocouples were firstly calibrated with a reference junction at 32°F. They were connected to a portable precision potentiometer.

A very sensitive pressure gage was chosen to accurately measure the pressure of the refrigerant between 5 and 100 pounds per square inch.

b. <u>Experiment Procedure</u>: Wet and dry sand temperatures were recorded every five minutes for the transient state. A lapse of twenty-six hours for dry sand and forty-seven for wet sand was necessary in order to assure a steady heat flow. For both states, the temperature-time-distance history was recorded. (Tests on dry sand were made first.)

Following this, a certain percentage of moisture was added to a certain amount of sand in the upper half of the tank, after which were conducted rapid tests in order to obtain accurate data.

The experiments were repeated to determine the heat flow function at various depths and at certain distances from the center of the pipe. Isothermal lines and heat flow curves were figured and drawn to show the resistivity of the sand at different states.

Heat flows from higher to lower temperatures. The outside temperature was 70°F., while the refrigerant temperature was 40°F. when absorbing heat from the soil and 100°F. when returning heat to the soil.

Heat changed the state of the refrigerant (FREON 114). When the heat flowed from the sand to the refrigerant, the Freon was evaporated until it reached the cooler fin. The cooler at 32°F. condensed the Freon to a liquid. This process of evaporation and condensation of the refrigerant was continued for the transient state, the steady state, and for dry and moist sand. Results were recorded.

When heat flowed from the heater at 100°F. to the refrigerant, the Freon was evaporated. When the sand, at a lower temperature than the refrigerant, absorbed heat from the Freon, the refrigerant was condensed. Having reached the heater fin, the Freon was again evaporated. This process of evaporation and condensation of the refrigerant was repeated and the results recorded for the transient and steady states, dry and moist sand.

c. Experiment Results:

1-The air-side film coefficient = h_o.

$$h_o = 5.05\left(\frac{P}{P_o}\right)^{.5}\frac{(\Delta t)^{.25}}{T^{.5}}$$

$$= .22 \text{ Btu/hr. ft.}^2\ {}^\circ\text{F.}$$

2-The Freon and film coefficient = 2.82 Btu/hr. ft.^{2} °F.

conductivity = .0515 Btu ft./hr.ft.^{2} °F.

3-The heat flow from atmospheric room temperature to the refrigerant inside the pipe:

$$q = \frac{2\pi\ (\Delta t)\times L}{\frac{1}{.89\times .22}+\frac{\ln\frac{11}{10.9}}{26.2}+\frac{\ln\frac{10.9}{.528}}{.19}+\frac{\ln\frac{.528}{.3622}}{49.3}+.355}$$

$$= \frac{2\pi\times 30\times L}{5.1+\frac{.01}{26.2}+\frac{3.03}{.19}+\frac{.379}{49.3}+.355}$$

$$= \frac{188.2\ L}{5.1+.000381+15.95+.00769+.355}$$

$$= \frac{188.2\,L}{21.4138071}$$

$$= 8.80 \qquad \text{Btu/hr/(ft. height of the pipe)}$$

$$= 8.80\times\frac{35}{12} = 26.0 \text{ Btu/hr.}$$

4-The unit thermal conductance (for $2\pi L\ (\Delta t) = 1$)

$$U = \frac{1}{21.4138071} = .0473$$

5-For measurement of the length of the cooler fin (L):

The steady heat flow equation is:

$$\nabla^2 t = 0$$

$$q = -K\,A\,\frac{dt}{dy}$$

$$T'' - \frac{hc}{K} \times \left(\frac{\pi d}{\frac{\pi d^2}{4}}\right) T = 0$$

h_c of Freon 114 at 40°F.
K of brass fin

h_c of refrigerant at condensing state

$$h_{av} = \frac{.725}{n^{.25}} \left(\frac{\lambda \rho^2 K^3 g}{\mu D_o (\Delta t)}\right)^{.25}$$

$$= (n\,d_o\,\Delta t)^{-.25} (563.0 - .33\,t - .0043\,t^2)$$

$$= 303.2 \quad \text{Btu/ft.}^2\ \text{hr.}\ ^{\circ}\text{F.}$$

$$t_w - t_\alpha = t_w - 40 = T$$

$$\frac{d}{dx}\left(KA\frac{dt}{dx}\right) = h_c\,P\,(t - t_\alpha)$$

$$\frac{d^2t}{dx^2} = \frac{h_c P}{KA}(t - t_\alpha)$$

$$A = \frac{\pi}{4} \times \left(\frac{.25}{12}\right)^2 = .00034 \text{ square feet}$$

$$P = \pi d = 3.1416 \times \frac{.25}{12} = .0652 \text{ feet}$$

$$T'' - \frac{303.2}{49.3} \times \left(\frac{\pi d}{\frac{\pi d^2}{4}}\right) T = 0$$

$$T'' - 24.6 \times \frac{12}{.25}\,T = 0$$

$$T'' - 1180\,T = T'' - F^2\,T = 0$$

$$F = \sqrt{1180} = \pm 34.4$$

The solution for temperature T:

$$T = A\, e^{-34.4y} + B\, e^{+34.4y}$$

where A and B are constants.

The initial and boundary conditions are:

$y = 0 \quad t = t_w$ = temperature of the fin wall $^{\circ}F$.

$y = L \quad t = t_\alpha$ = temperature of the vapor Freon

$$t_w < t_\alpha$$

therefore: $t_w - t_\alpha = A + B$

$$t_\alpha - t_\alpha = 0 = A\, e^{-34.4L} + B\, e^{+34.4L}$$

$$A = -\frac{e^{+34.4L}}{e^{-34.4L}}\, B = -B\, e^{68.8L}$$

$$t_w - t_\alpha = -B\, e^{68.8L} + B$$

$$= B\,(1 - e^{68.8L})$$

if $t_w - t_\alpha = 10^{\circ}F$

$$B = \frac{10}{1 - e^{68.8L}}$$

$$A = -\frac{10\, e^{68.8L}}{1 - e^{68.8L}}$$

$$T = \frac{-10\, e^{68.8L}\, e^{-34.4y}}{1 - e^{68.8L}} + \frac{10}{1 - e^{68.8L}}\, e^{34.4y}$$

$$= \frac{10}{1 - e^{68.8L}} \left(-\frac{e^{68.8L}}{e^{+34.4y}} + e^{34.4y} \right)$$

$$\left.\frac{dT}{dy}\right|_{y=L} = \frac{10}{1 - e^{68.8L}} \left(34.4\, e^{34.4L} + 34.4\, e^{68.8L - 34.4L} \right)$$

$$= \frac{344}{1 - e^{68.8L}} \left(2 e^{34.4L} \right)$$

$$= \frac{688\, e^{34.4L}}{1 - e^{68.8L}}$$

$$q = -KA \frac{dT}{dy}\Big]_{y=L} = \frac{-49.3 \times .00034 \times 688 e^{34.4L}}{1 - e^{68.8L}}$$

$$= -11.5 \quad \frac{e^{34.4L}}{1 - e^{68.8L}}$$

$$= -26.0 \quad \text{Btu/hr.}$$

$L = .23$ inches

= the length of the cooler fin

check: $q = \frac{\Delta t}{R}$ Btu/hr

$$R = \frac{m}{h_s \, P \tanh mL}$$

$mL > 2.3$ then $\tanh mL = 1$

$$m = \sqrt{\frac{h_s P}{KA}}$$

A = conduction cross section area ft.^{2}

P = perimeter of cross section ft.

h_s = unit conductance to the surroundings from the fin surface

K = thermal conductivity fin material

$$m = \sqrt{\frac{303.2 \times .0652}{49.3 \times .00034}} = \pm 34.4$$

$$R = \frac{+34.4}{303.2 \times .0652 \tanh mL}$$

$$q = -26.0$$

$$= \frac{30 \times 303.2 \times .0652 \tanh mL}{34.4}$$

$$\tanh mL = 1 - \frac{2}{e^{2mL}} + \frac{2}{e^{4mL}} - \frac{2}{e^{6mL}} + \cdots = -1.52$$

$$-.52 = -\frac{2}{e^{2mL}}$$

$$e^{2mL} = 3.86$$

$$2mL = 1.355$$

$$L = \frac{1.355}{2 \times 34.4} = .0197 \text{ feet}$$

$$= .236 \text{ inch fin length}$$

6. The length of the heater fin:-

$$\nabla^2 t = 0$$

$$q = -KA \frac{dt}{dy}$$

$$\therefore \frac{d^2t}{dy^2} = \frac{h_c P}{KA} (t - t_\alpha)$$

A = area of cross section = square feet

$$= \frac{\pi}{4} \times \left(\frac{.25}{12}\right)^2 = .00034$$

P = perimeter $= \pi d = \pi \times \frac{.25}{12}$

$$= .0652 \text{ feet}$$

take $q = 8.8$ (Btu/hr) per ft.

therefore use $h_c = .0424$ Btu ft/hr. ft.^{2} °F.

$$\frac{d^2t}{dy^2} - \frac{h_c P}{KA} T = 0$$

$$T'' - \frac{.0424 \times .0652}{49.3 \times .00034} T = 0$$

$$T'' - .1655\, T = T'' - P^2 T = 0$$

$$P = \sqrt{.1655}$$

$$= \pm .4065$$

$$T = A e^{-.4065y} + B e^{+.4065y}$$

where A and B are constants

from boundary conditions

when $y = 0$ $\quad t = t_w$ °F. temperature of the wall of the fin

$y = L$ $\quad t = t_\alpha$ °F. temperature of hot Freon liquid

$$t_w > t_\alpha$$

$$t_w - t_\alpha = A + B$$

$$t_\alpha - t_\alpha = 0 = A\, e^{-.4065L} + B\, e^{+.4065L}$$

$$A = -B \frac{e^{.4065L}}{e^{-.4065L}} = -B e^{.813L}$$

$$t_w - t_\alpha = -B\, e^{.813L} + B$$

$$= B\,(1 - e^{.813L})$$

if $t_w - t_\alpha$ = the difference between the temperature of the fin wall and the average temperature of the Freon liquid

$$= 10°F.$$

therefore $B = \dfrac{10}{1 - e^{.813L}}$

$$A = -\frac{10}{1 - e^{.813L}}\, e^{.813L}$$

$$T = \frac{-10\, e^{.813L}}{1 - e^{.813L}}\, e^{.4065y} + \frac{10}{1 - e^{.813L}}\, e^{.4065y}$$

$$= \frac{10}{1 - e^{.813L}} \left[-\frac{e^{.813L}}{e^{.4065y}} + e^{.4065y} \right]$$

$$\left.\frac{dT}{dy}\right]_{y=L} = \frac{10}{1 - e^{.813L}} \left[.4065 e^{.4065L} + .4065 e^{.813L - .4065L} \right]$$

$$= \frac{8.13}{1 - e^{.813L}}\, e^{.4065L}$$

$$-K\,A\,\frac{dT}{dy}\bigg]_{y=L} = 8.8 \text{ (Btu/hr) per ft.}$$

$$-8.8 = -\frac{49.3 \times .00034 \times 8.13\, e^{.4065L}}{1 - e^{.813L}}$$

$$= .1375\ \frac{e^{.4065L}}{1-e^{.813L}}$$

$$1 - e^{.813L} = .0156\, e^{.4065L}$$

$$1 - e^{2x} = .0156\, e^{x}$$

X = .4065L	LEFT SIDE	RIGHT SIDE
.005	.0101	.0154
.0100	.0202	.0154
.015	.0305	.0154
.0075	.015	.015

$X = .0075$ feet

$.4065\,L = X = .0075 \times 12$ inches

$$L = \frac{.0075 \times 12}{.4065} = .225 \text{ inches}$$

7-Rate of heat flow:

The most useful information for the design of a ground pipe is that concerning the rate of heat flow from the soil to the vertical pipe or vice versa.

$$q = A\,U\,(\Delta t)$$

i.e. $q = AU\phi(\Delta t)$ Keeping A and U constants.

The line curve indicates the relation between the rate of heat flow (q) and the difference in temperature variations. Assuming that the pipe is kept at a constant temperature, the unbroken line curve gives a similar relation when heat withdrawal per unit length of tube is assumed to be constant. The coordinates q and (Δ t) may be used for any case in which conductivity and diffusivity of the soil is known. Curves, such as q and (Δ t), can be made only for a given type of soil and size of pipe of certain material.

Most check tests were carried on for twenty-four hours or more. The value of K for these check points was in each case determined by the change in the temperature per unit distance away from the pipe surface by means of the equation:

$$K = \frac{Q \ln \frac{r_2}{r_1}}{2\pi\,(t_2 - t_1)}$$

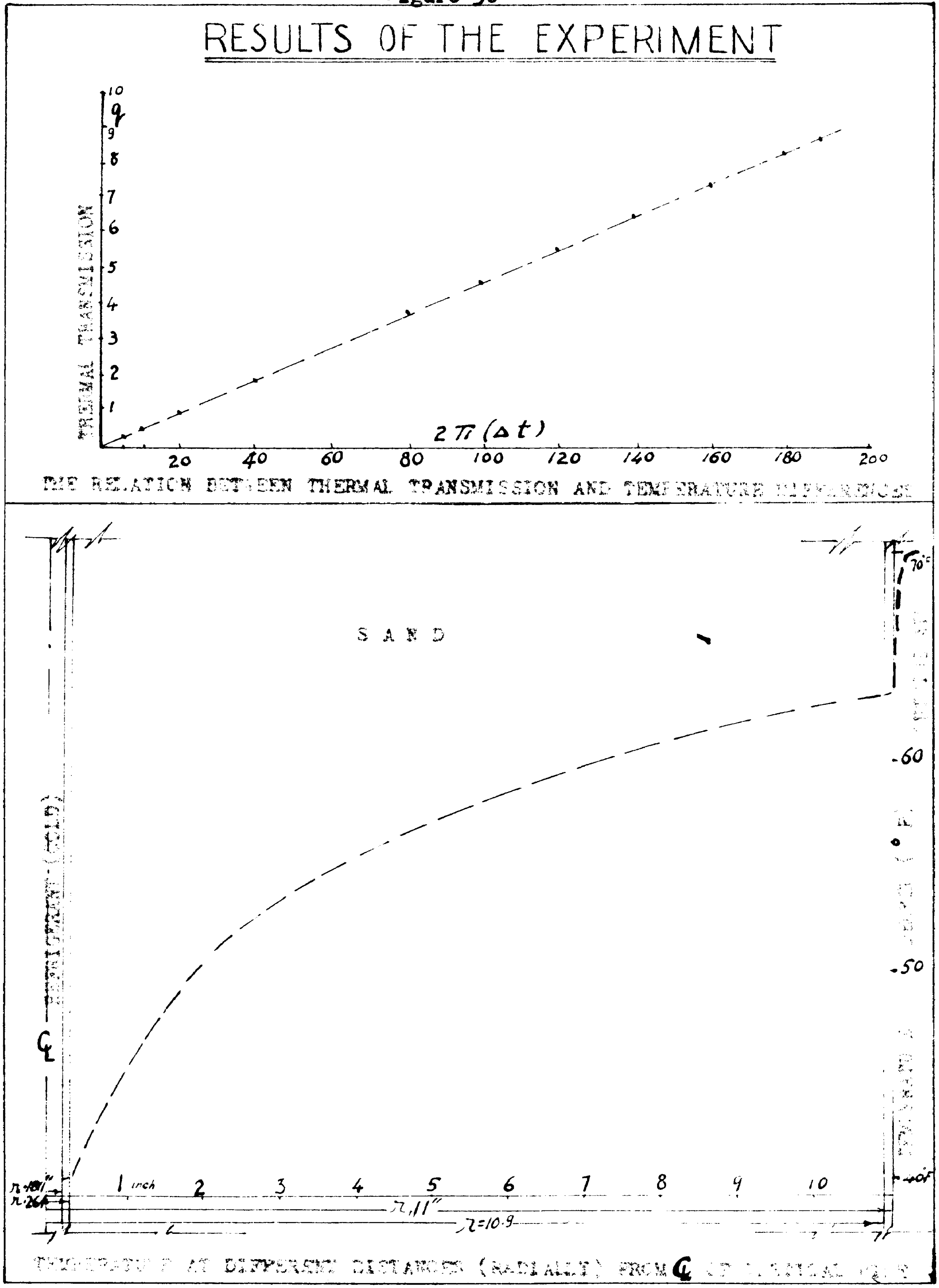
$$q = \phi \; 2\pi \; (\Delta t)$$

$2\pi(\Delta t)$	q	$2\pi(\Delta t)$	q
0	0	30	1.419
1	.0473	40	1.892
2	.0946	50	2.36
3	.1419	60	2.83
4	.1892	70	3.3
5	.236	80	3.78
6	.283	90	4.25
7	.33	100	4.73
8	.378	150	7.59
9	.425	188.2	8.8
10	.473	200	9.46
20	.946		

3-The change in thermal resistance (degree per unit of heat transferred per unit of time), was due to the change in temperature difference.

$$R = \frac{t_1 - t_2}{q} = \frac{\Delta t}{q} = \frac{L}{KA}$$

Figure 30


RESULTS OF THE EXPERIMENT
THERMAL TRANSMISSION
q
10
9
8
7
6
5
4
3
2
1
2π(Δt)
20
40
60
80
100
120
140
160
180
200
THE RELATION BETWEEN THERMAL TRANSMISSION AND TEMPERATURE
SAND
70°
-60
-50
40°F
1 inch
2
3
4
5
6
7
8
9
10
r,11"
r=10.9

Δ t	R	Δ t	R
0	0	9	1.0
1	.1115	10	1.115
2	.223	12	1.34
3	.3345	14	1.56
4	.446	16	1.782
5	.558	18	2.01
6	.67	20	2.23
7	.78	30	3.345
8	.89		

9-The temperatures were taken at different distances (radial) from the center line of the vertical pipe. The following equation gives the temperature of the soil when the Freon is 40°F. and the outside atmospheric temperature is 70°F.:

$$q = \frac{2\pi L\,(T_o - T_i)}{\frac{1}{r f}} = 26.0 \text{ Btu/hr.}$$

$$T_{11''} = -\frac{q}{2\pi r f L} + T_0 \quad \text{(for air side convection drop)}$$

$$= 70 - \frac{26.0}{2\pi \times .89 \times .22 \times \frac{35}{12}} = 62.75^\circ\text{F.}$$

$$T_{10.9''} = 70 - \frac{8.95 \times (5.1 + .000381)}{6.285}$$

$$T_{10''} = 70 - \frac{8.95}{6.285} \times \left(5.100381 + \frac{\ln\frac{10.9}{10}}{.19}\right)$$

$$= 70 - 1.43\left(5.100381 + \frac{.0862}{.19}\right) = 70 - 7.92 = 62.08^\circ\text{F.}$$

$$T_{9''} = 70 - 1.43\left(5.100381 + \frac{\ln\frac{10.9}{9}}{.19}\right) = 70 - 1.43 \times 6.100381$$

$$= 70 - 8.78 = 61.22^\circ\text{F.}$$

$$T_{8''} = 70 - 1.43\left(5.1 + \frac{\ln\frac{10.9}{8}}{.19}\right) = 70 - 1.43 \times 6.72 = 60.35^\circ\text{F.}$$

$$T_{7"} = 70 - 1.43\left(5.1 + \frac{\ln\frac{10.9}{7}}{.19}\right) = 70 - 10.62 = 59.38^{\circ}F.$$

$$T_{6"} = 70 - 1.43\left(5.1 + \frac{\ln\frac{10.9}{6}}{.19}\right) = 70 - 11.8 = 58.2^{\circ}F.$$

$$T_{5"} = 70 - 1.43\left(5.1 + \frac{\ln\frac{10.9}{5}}{.19}\right) = 70 - 13.2 = 56.8^{\circ}F.$$

$$T_{4"} = 70 - 1.43\left(5.1 + \frac{\ln\frac{10.9}{4}}{.19}\right) = 70 - 14.8 = 55.2^{\circ}F.$$

$$T_{3"} = 70 - 1.43\left(5.1 + \frac{\ln\frac{10.9}{3}}{.19}\right) = 70 - 17 = 53^{\circ}F.$$

$$T_{2"} = 70 - 1.43\left(5.1 + \frac{\ln\frac{10.9}{2}}{.19}\right) = 70 - 20 = 50^{\circ}F.$$

$$T_{1"} = 70 - 1.43\left(5.1 + \frac{\ln\frac{10.9}{1}}{.19}\right) = 70 - 25.3 = 44.7^{\circ}F.$$

$$T_{.264} = 70 - 1.43\left(5.1 + \frac{\ln\frac{10.9}{.264}}{.19}\right) = 70 - 1.43 \times 20.85 = 40.1^{\circ}F.$$

$$T_{.1311} = 70 - 30 = 40^{\circ}F.$$

10-The soil and tank temperatures at radial distances from the central axis of the tank (when the Freon Temperature is $100^{\circ}F.$ and the outside air temperature is $70^{\circ}F.$) are as follows:

$$q = \frac{2\pi L\,(T_i - T_o)}{\frac{1}{r_f}} = 26.0 \quad \text{Btu/hr.}$$

$$T_{11} = T_o + \frac{q}{2\pi r_o L f} = 70 + \frac{26.0}{2\pi \times .89 \times \frac{35}{12} \times .22} = 77.25^{\circ}F.$$

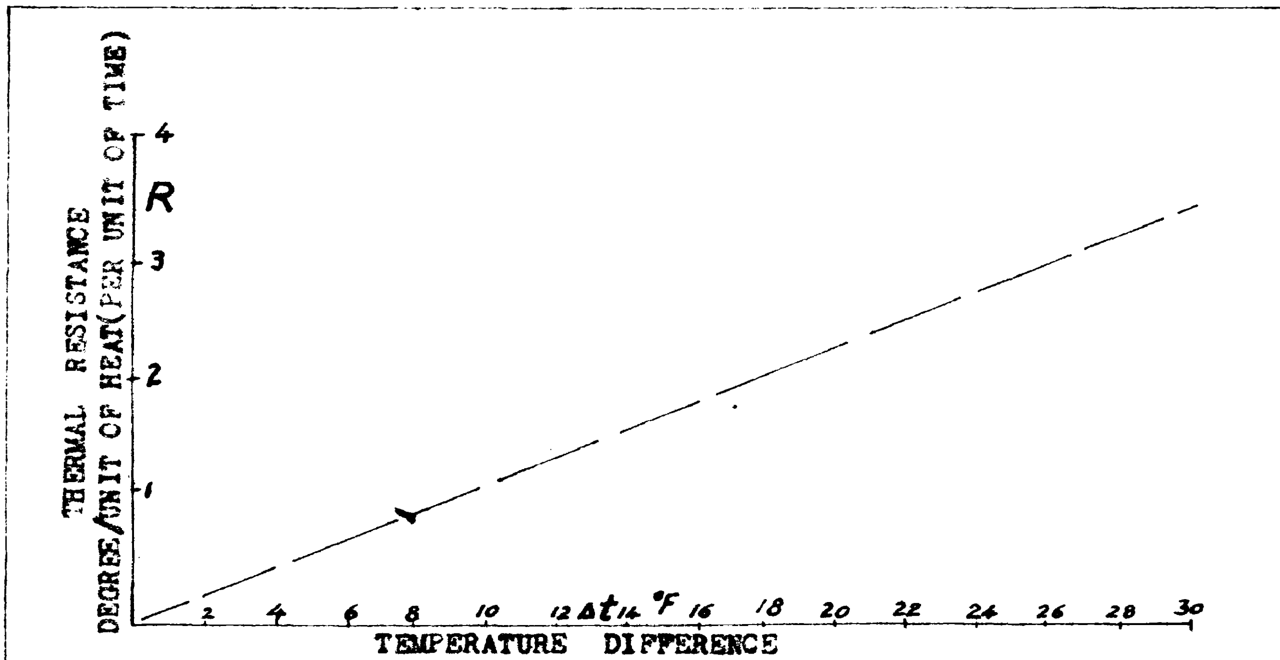
$$T_{10.9} = 70 + \frac{q}{2\pi L U}\ln\frac{r_2}{r_1} = 70 + \frac{8.95 \times (5.1 + .000381)}{6.285} = 77.257^{\circ}F.$$

$$T_{10} = 70 + \frac{8.95}{6.285} \times \left(5.1 + \frac{\ln\frac{10.9}{10}}{.19}\right) = 70 + 7.92 = 77.92^{\circ}F.$$

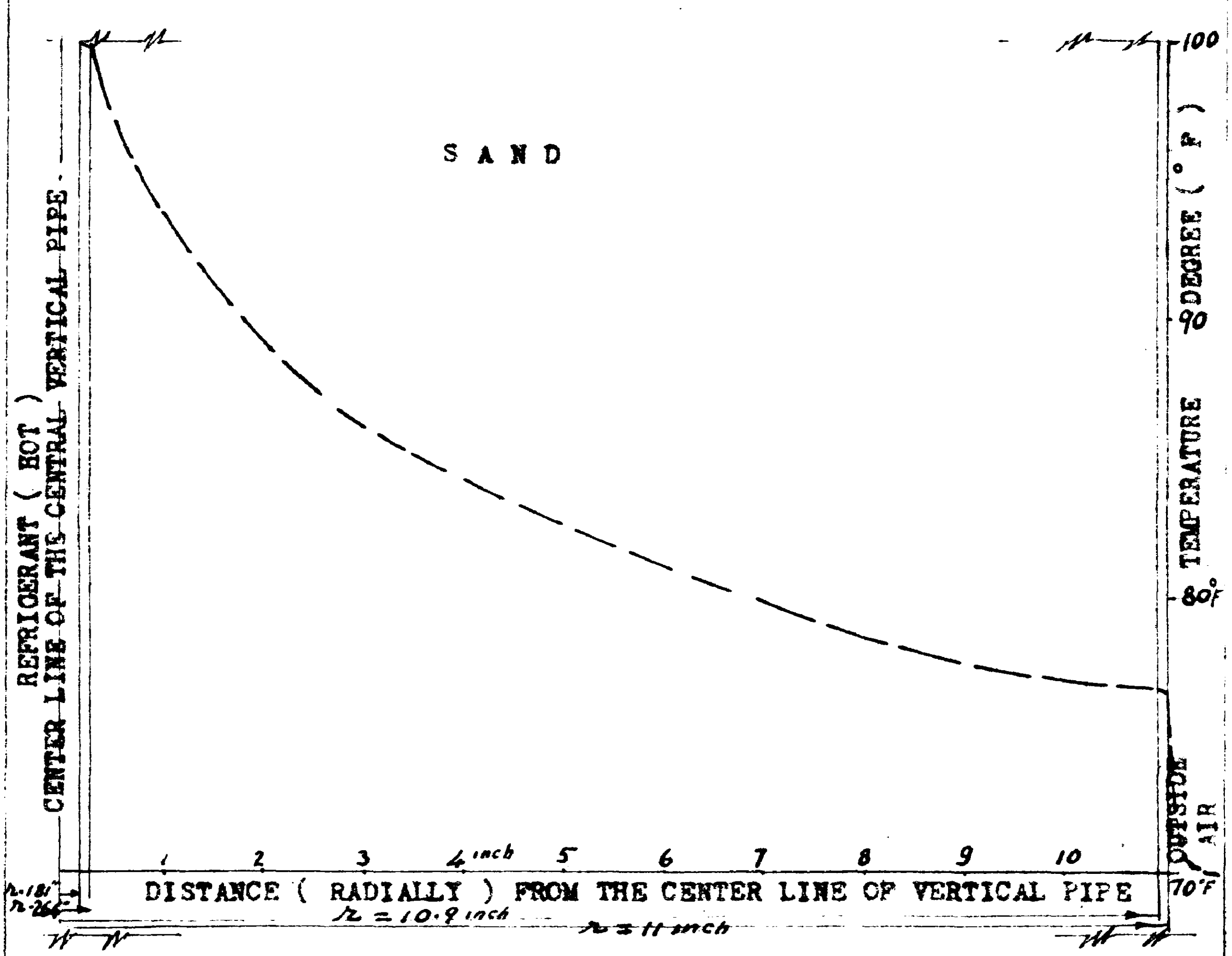
$$T_{9} = 70 + 1.43\left(5.1 + \frac{\ln\frac{10.9}{9}}{.19}\right) = 78.78^{\circ}F.$$

$$T_{8} = 70 + 1.43 \times 6.72 = 79.65^{\circ}F.$$

Figure 30b Results of Experiment



THE RELATION BETWEEN THERMAL RESISTANCE AND TEMPERATURE DIFFERENCES



TEMPERATURE AT DIFFERENT DISTANCES (RADIAL) FROM ℄ OF VERTICAL PIPE

$T_7 = 70 + 10.62 = 80.62°F.$

$T_6 = 70 + 11.8 = 81.8°F.$

$T_5 = 70 + 13.2 = 83.2°F.$

$T_4 = 70 + 14.8 = 84.8°F.$

$T_3 = 70 + 17 = 87°F.$

$T_2 = 70 + 20 = 90°F.$

$T_1 = 70 + 25.3 = 95.3°F.$

$T_{.264} = 70 + 29.8 = 99.9°F.$

$T_{.1311} = 70 + 30 = 100°F.$

Heat transfer in the soil varied at the beginning of the experiment with the amount of time consumed before a steady flow of heat was obtained. Heat flow in the soil varies, due to the type of soil and its moisture content.

10-The period of time for which the dry sand is in the transient state (assuming that the tank is a cylinder of sand with an outside radius (r_o) of 10.9 inches, inside radius (r_i) of 264 inches) is explained in the following calculation:

$h_{air} = 22$, $K = .19$, $\rho = 94.8$, $C_p = .19$, a (diffusivity) $= .01056$

$$\frac{dt}{d\theta} = a \frac{\partial^2 t}{\partial x^2}$$

$$\frac{t - t_a}{t_i - t_a} = \varphi \left[\frac{hR}{K} , \frac{a\theta}{R^2} , \frac{r}{R} \right]$$

$$\text{Biot} = \frac{hR}{K} = \frac{.22}{.19} \times \frac{10.9}{12} = 1.06$$

$$\left(\frac{1}{\text{Biot}} \right) = .945$$

$$\text{Fourier} = \frac{a\theta}{R^2} = \frac{.01056}{.84}\theta = .0125\ \theta$$

$$\left(\frac{r}{R}\right)_{.264''} = \frac{.264}{12} \times \frac{12}{10.9} = .0241$$

$$\frac{t - t_a}{t_i - t_a} = \frac{22.54}{30} = .74$$

from heat transfer notes graph of page V-43

$$\frac{a\theta}{R^2} = .32$$

$$.0125\,\theta = .32$$

$$\theta = \frac{.32}{.0125} = 25.6 \text{ hours}$$

$$= 25 \text{ hours } 36 \text{ minutes},$$

which approximates the result in Appendix III.

11-The period of time for which the wet sand is in the transient state (assuming that the tank is a cylinder of sand with an outside radius (R) of 10.9 inches, inside radius (r_i) of .264 inches) is explained in the following calculation:

$h_{air} = .22$, $K_{sand} = .65$, $\rho = 102$, $C_p = .5$

a (diffusivity) $= .01275$

$$\frac{dt}{d\theta} = a\frac{d^2t}{dx^2}$$

$$\frac{t - t_a}{t_i - t_a} = \varphi\left[\frac{hR}{K}, \frac{a\theta}{R^2}, \frac{r}{R}\right]$$

$$\text{Biot} = \frac{hR}{K} = \frac{.22}{.65} \times \frac{10.9}{12} = .31$$

$$\frac{1}{\text{Biot}} = 3.22$$

$$\text{Fourier} = \frac{a\theta}{R^2} = \frac{.01275}{.84}\,\theta = .0151\,\theta$$

$$\left(\frac{r}{R}\right)_{.264''} = \frac{.528}{2 \times 12} \times \frac{2 \times 12}{21.8} = .0241$$

$$\frac{t - t_a}{t_i - t_a} = \frac{22.54}{30} = .74$$

from the Boelter ... "heat transfer notes" graph (V-43)

$$\frac{a\theta}{R^2} = .7$$

$$.0151\,\theta = .7$$

$$\theta = \frac{.7}{.0151} = 46.1 \text{ hours} = 46 \text{ hours } 6 \text{ minutes.}$$

d. <u>Conclusions and Recommendations:</u> From the theoretical studies and experiments completed, the following general statements and conclusions are found to be of great importance:

1-The temperature drop between the ground and vertical pipe is in proportion to the rate of heat absorbed or emitted.

2-For any given pipe surface area, a small, long pipe is more effective than a large shorter one.

3-Soil conductivity is a more important factor than soil diffusivity.

4-The isothermal surfaces will be cylinders when close to the pipe surface; further away, they will be affected by the temperature lag; and for still further depths, will be almost horizontal planes.

5-The heat flow, for one vertical pipe, will be radially perpendicular to the isothermal surfaces and pipe surface when close to it. For wider radii, the heat flow curves will take the vertical directions, in accordance with the higher temperatures located deeper in the ground.

6-From the above conclusions, the vertical ground pipes, besides their own advantages, will have the additional advantage of horizontal grids.

7-For short period, high capacity operation, a horizontal grid has approximately the same heat capacity as two vertical isolated pipes.

8-For a community, or in a busy city, where housing space is limited, the vertical pipes will be sufficient to supply each home with its needs. Also, the vertical pipes will transfer heat from the same area as the horizontal grids and the depth will result in greater heat and efficiency.

9-It is thought that projections on the pipe surface fail to give as much return (for the cost) as an equal amount expended for more pipe.

10-A horizontal coil is less efficient, for any given length of pipe, than the same length of pipe driven straight into the soil.

11-Even though deep vertical wells have certain advantages, if they are operated for long periods of time with more heat extracted in winter than is returned in summer, it is necessary to consider the effect of a local cooling of the ground near the pipe.

12-From the foregoing, we may conclude that the small, long and deeply buried vertical pipe is best in the use of earth as a heat source, all other factors being equal.

SUMMARY

In recent years, the heat pump has been recognized as a practical device for residential year-round air-conditioning. It offers a means of obtaining both heating and cooling in addition to all the benefits which a year-round air-conditioning system can give. From the thermo-dynamic point of view, the heat pump is a new application of old refrigeration cycles. However, the device, as it is proposed at the present time, gives an automatic type of heating and cooling system not heretofore possible. Research work upon the device has been accomplished, involving the auxiliary equipment, by manufacturers and technical societies.

Heat sources, as in the past, form the most important heat pump problem, as far as residential applications are concerned. Two of these sources, which should be developed for universal application, are air and earth. Air, as a heat source, is not as feasible, due to mechanical difficulties involving frost formation, complex controls, etc. The earth, however, offers many advantages, due to the rather uniform temperature at which it is available, even in northern areas. Its disadvantages involve the difficulty of installing sufficient heat transfer surface to obtain good performance. Some of the problems needing solution are theoretical investigation of heat flow under various conditions, information on the types of heat sources already proposed, and an investigation of special heat sources.

The source of the heat for the pump is naturally of interest to the user, as the cost involved forms a large portion of the pump installation regardless of the type of unit, because the heat source is an application problem. Utility companies are one possible source of support for heat source research, as they have a particularly large market in the heat pump in the form of electric energy sales. Even though the expense of this research will fall upon the user, as he pays for it indirectly through his electrical rates, nevertheless, the improvement in performance which would result from research can in the end result in lower total electrical cost to the user.

Accessory problems include the development of special refrigerants and production of low-cost accessories such as electrostatic dust precipitators and germicidal lamps.

Application problems of the heat pump are rather numerous. One of these is the study of comfort requirements in terms of heat pump operation. Some of the heat pump features are similar to warm air (radiant heating) systems. The minimum requirements for satisfactory home cooling have not been too throughly investigated. Probably, all the various requirements in connection with heat pump operation will become more important as the heat pump becomes a necessity, rather than a luxury item. Other application problems include analysis of air distribution systems for heating and cooling, study of noise and vibration limitations, insulation requirements, design procedures development, and study of heat pump application to present-day housing. General research on temperature, air velocity distribution, and human comfort requirements, in residential sections, can be partly adapted to heat pump installations.

REFERENCES

1. Abbott, C. G. 1950 Periodic influences on Washington and New York weather of 1949-1950, Smithsonian Institute, Washington, D. C.

2. Allen, J. R. 1920 Theory of heat losses from pipes buried in the ground, ASHVE transactions, 26, 335.

3. American Automobile Association, The climatic Handbook for Washington, D. C., Weather Bureau Technical Paper No. 8

4. Avery, L. R., E. F. DuBois, C. E. Leopold, T. P. Hatch, R. A. Keeton, and C. Schneider, 1951 Man and his relationship to air, Symposium of 57th annual meeting of ASHVE, Philadelphia, Pa.

5. Bedford, Thomas: Equivalent Temperature, What it is, How it's measured. August 1951 ASHVE Heating, Piping and Air Conditioning 87.

6. Boelter, L. M. K., V. H. Cherry, H. A. Johnson, & R. C. Martinelli 1946 Heat transfer notes, U. of California Press, Berkeley, Calif.

7. Blockley, W. V. and John Lyman Oct. 1950, Studies of human tolerance to extreme heat and mental performance under heat stress and checking tests, U. S. Air Force-Wright Patterson Airforce Base, Dayton, Ohio, Technical Report 6022.

8. Blockley, W. V. and C. L. Taylor Feb 1950, Studies in tolerance for extreme heat, U. S. Airforce, Wright Patterson Airforce Base, Dayton, Ohio, Technical Report 5831.

9. ______________________________, June 1949, Human tolerance limits for extreme heat, ASHVE meeting, Minneapolis, Minnesota.

10. Downing, F. T., June 1951, Application of the heat pump to hydro-electric power house, Heating, Piping and Air Conditioning.

11. DuBois, E. F., 1936 Mechanism of heat loss in health and disease. Transactions of Associciition of American Physicians 51:252

12. Eibling, J. A. and B. A. Landry, 1951, The steam-generating station as a source and sink for the heat pump, ASHE spring meeting, Atlanta, Georgia.

13. Eichna, L. W., W. F. Ashe, W. B. Bean, and W. B. Shelley, 1945, The upper limits of environmental heat and humidity tolerated by acclimatized men working in hot environments, Journal of Industrial Hygiene and Toxicology 27 and 59.

14. Eid, Abdallah Kamel, January 1951, The economic and technical aspects of vertical pipe for the heat pump, ASHVE 57th annual meeting, Philadelphia, Pennsylvania.

15. Egyptian Government, Department of Meteorology, Ministry of War & Marine, 1949, 1950 & 1951, personal correspondence with the Director General.

16. Carslaw and Jaeger, 1947, Conduction of heat in solids, Oxford Clarendon Press, pp. 222, 272, 273 and 284.

17. Geological Society of America, Special papers-QEI G 298 Nos. 36 & 37.

18. Guernsey, E., W. P. Blotz, P. L. Skan, June 1949, Earth as a heat source or storage medium for the heat pump, ASHVE meeting.

19. Hadley, W. D., December 1949, Operating characteristics of heat pump ground coils, Edison Electric Institute, Bulletin, p. 457.

20. Hardy, J. D. and C. F. Sanderstrom, 1938, Heat loss from the nude body and peripheral blood flow at temperature 22°C.-35°C., Journal of Nutrition 16:493.

21. Hofmann, July-August 1940, Uber die Gesetzmassigkaten der Wärme- und der Stoffübertragung auf Grund der Strömungsvorgangen im Rohr, Forschung auf dem Gebiete des Ingenieurwesens, Ausgabe A, Band 11 No.4, 159-169.

22. Hess, A. J., February 1951, Heat Pump protects underground electric cables against corrosion, Heating, Piping, and Air-conditioning.

23. Ingersoll, L. R. and H. J. Plass, 1948, Theory of the ground pipe heat source for the heat pump, ASHVE transactions 54 (presented in Bretton Woods, N. H. meeting).

24. ________________, and Zobel, 1913, Mathematical theory of heat conduction, Ginn and Co. pp. 34, 43 and 162.

25. Jahnke, E. and F. Emde, 1943, Tables of functions with formulae and curves, Dover Publications, pp. 6-9.

26. Johnson, W. E. and N. J. Bloomfield, February 1948, Economic and technical aspects of the heat pump, ASHVE 54th annual meeting, New York City.

27. Jordan, R. C., November 1950, The heat pump, heat sources and sinks, ASHVE committee composit work, Heating, Piping and Air-conditioning, page 87.

28. Kelvin, Sir W. Thomson, Mathematical and physical papers, II, 41ff.

29. Kemler, E. N., December 1946, Heat for the heat pump, (Heating, Piping and Air-conditioning).

30. ______________, and H. Kuklik, 1948, Research aspects of the heat pump, ASHVE 54th annual meeting in New York City.

31. King, W. F. and M. B. Martus, July 1947, Heat transfer from ground to water in buried coils, Heating and Ventilating p. 71.

32. Kingston, V. M., 1946, Technical memorandum No. 630, U. S. Bureau of Reclamation, Denver, Colorado.

33. Krooker, J. D. and R. C. Chewning, February 1948, A heat pump in an office building, ASHVE 54th annual meeting, New York City.

34. Lane Medical Lectures, The mechanism of heat loss and temperature regulation, Stanford University press, p. 348. 1937.

35. Los Alamos Scientific Laboratory, The effects of atomic weapons, Federal Civilian Defence Administration.

36. Macintire, J. J. and F. W. Hutchinson, July 1950, Refrigeration Engineering, John Wiley & sons, New York, 2nd Edition.

37. Mackworth, N. H., 1948, Definition of the upper limit of environmental warmth by physiological tests of human performance, Royal Society Empire Scientific Conference June, July 1946 I: 423.

38. McAdams, 1942, Heat transmission, McGraw-Hill Book Company, New Jersey.

39. McLachlan, N. W., 1934, Bessel function for engineers, Oxford University Press, p. 157.

40. ______________, and Meyers, October 1934, The polar form of the Ker and Kei functions with applications to eddy current heating, Philosophical magazine, 120:610.

41. Meteorological Magazine, 1930

42. Moon, P., 1940, Proposed standard solar radiation curves for engineering use, Journal of Franklin Institute, 230, No. 5.

43. Newburgh, L. H., 1949, Physiology of heat regulation and the science of clothing, Saunder's Company, London.

44. Newman, A. B., 1931, Temperature distribution in internally heated cylinders, Industrial and Engineering Chemistry, 23: 19.

45. Parkerson, W., 1949, Seasonal performance factor of the heat pump, Heating and Ventilating, January, p. 61.

46. Raber, B. F. and F. W. Hutchinson, Refrigeration and air conditioning engineering, Wiley and Sons, New York City.

47. ______________________________, Panel heating and cooling analysis, Wiley and Sons, New York City.

48. Schickele, Elizabeth, 1947, Environment and fatal heat stroke: An analysis of 157 in the United States during World War II.

49. Smith, G. S., July 1951, Climatology as an aid in heat pump design, ASHVE meeting in Portland, Oregon.

50. __________, February 1951, Factors useful in ground grid design for the heat pump, ASHVE 57th annual meeting in Philadelphia, Pa.

51. __________, and T. Yamairchi, July 1950, Thermal conductivity of soils for design of heat pump installations, ASHVE Journal, Heating, Piping and Air-conditioning section p. 129.

52. Taylor, C. L., June 28, 1951, Human responses to extreme heat exposure (speech given in Army Medical Service Graduate School, Washington, D. C.)

53. ten Bosch, M., Die Wärmeübertragung, 3d. ed.; Berlin: Julius Springer 1936, pp. 171, 125 and 126.

54. U. S. Department of Commerce, Climatological and Hydrologic Services Division, Data re climatological summary for Washington, D. C., personal information given author by R. E. Spencer, Chief, and W. A. Wood, Director, in July 1951.

55. U. S. Department of Geology, Mines and Water Resources, Board of Natural Resources, personal information from Joseph T. Singewald, Jr. Director, given author 1950-1951.

56. U. S. Geological Survey, District of Columbia: its rocks and their geological history, Bulletin No. 967.

57. U. S. Office of Civilian Defense, Survival under atomic attack, U. S. Government Printing Office, Washington, D. C.

58. Utilities Research Commission, February 1, 1949, Heat sources in the Chicago area for the heat pump, Report No. 94, Chicago, Illinois.

59. Walfanger, L. A., The major soil divisions of the United States, Ph.D. thesis, Columbia University, May 1929.

60. Williams, R. L., Sales-refrigerants, Kinetic Chemicals Division of E. I. DuPont De Nemours and Co. Publication, Wilmington, Delaware and personal contact with author July-August 1951.

61. Winslow, C. E. A. and L. P. Herrington, 1949, Temperature and human life, Princeton University Press.

62. Yaglow, C. P., Thermal standards in industry, by committee on atmospheric comfort, American Public Health Association Yearbook 1949.

63. Zeigler, E. C., Civil Engineering Department, University of Maryland, personal conferences with author, July-August 1951.

64. Zobel, O. J. and A. C. Ingersoll, 1948, Heat conduction with engineering and geological applications, McGraw Hill Book Co., Inc., New York City, p. 47, 152 and 154.

STATION	ALT. (feet)	JAN.	FEB.	MAR.	APR.	MAY	JUNE	JULY	AUG.	SEP.	OCT.	NOV.	DEC.	YEAR	RANGE
Port Said	11	56.1	57.2	60.4	65.1	70.3	75.4	78.4	79.5	77.5	73.9	67.5	59.4	68.4	23.4
Alexandria	105	56.1	57.2	60.1	63.7	68.5	73.4	77	78.1	76.3	73	66.4	59.4	67.5	22
Cairo(Abbasia)	98	52.7	55.4	60.8	67.6	74.1	79.0	81.0	80.6	76.1	71.9	64.0	55.9	68.2	28.3
Asyut	182	52.9	55.9	63.0	71.8	78.9	83.3	84.9	84.4	79.3	74.3	64.8	56.3	70.9	32.0
Wadi Halfa	421	57.9	61.0	68.5	78.1	84.6	88.2	88.5	87.4	84.6	79.5	70.0	60.1	75.7	30.6
Atbara	1163	63.7	70.9	77.5	84.4	90.9	92.3	90.7	89.6	90.3	86.9	79.3	71.7	82.8	24.1
Khartoum(Sudan)	1290	70.3	73.4	79.2	86.0	90.7	91.4	88.5	86.5	88.2	87.4	[illegible]	72.1	82.8	21.1
Hillet Doleib	1293	80.9	82.9	87.1	88.3	85.1	81.3	79.9	79.5	81.3	82.0	81.7	79.2	82.4	9.1
Mongalla	1440	80.4	81.7	82.6	81.0	79.0	77.4	75.9	75.7	77.2	78.1	79.0	79.2	79.0	6.9
El-Obeid	1866	67.3	70.5	75.9	83.1	85.8	84.7	80.5	79.0	80.1	81.3	76.1	68.7	77.7	18.5
Kassala	1666	74.3	76.3	81.7	87.3	90.3	87.9	82.8	81.3	83.3	85.8	83.3	76.3	82.6	16.0
Port Sudan	15	73.2	73.0	74.7	79.2	84.0	88.3	92.1	92.3	88.3	83.7	80.4	75.9	82.0	19.3
Berbera	31	76.1	76.1	78.1	81.7	87.3	95.7	97.4	96.6	91.2	83.3	79.1	77.1	85.0	21.3

APPENDIX I--TABLE No. 9 MEAN TEMPERATURES (Degree Fahrenheit) AS RECORDED BY [illegible] WEATHER STATIONS

	AL[illegible]			[illegible]UT			AS[illegible]			CAIRO			PORT SAID		
	D.B.	R.H.	E.T.	D.B.	R.H.	E.T.	D.B.	R.H.	E.T.	D.B.	R.H.	E.T.	D.B.	R.H.	E.T.
JAN.	82	69.4	77.8	53	69	52.2	50	45	48.9	54.2	74	54	56.7	76	55
APR.	108.2	69	100	72	41	66.8	79.4	28	71.4	68.5	58	65.2	65.6	73	63.5
JULY	103	77	96	85	42	76.3	92	26	78	82	61	76.4	100.5	76	94
OCT.	103.9	70	96	75	62	71.2	83.2	35	74.3	72.6	71	69.6	99.5	73	94

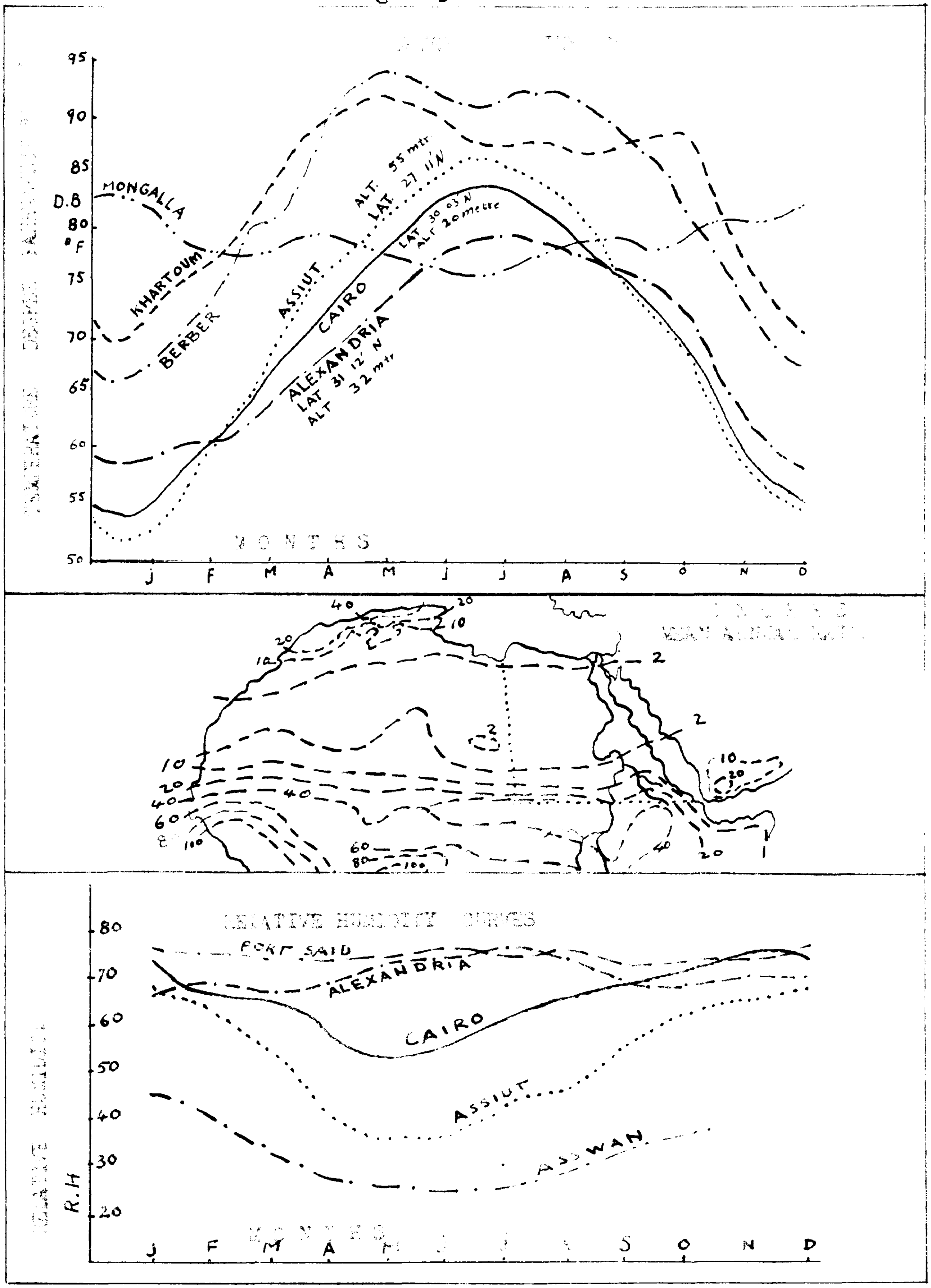
D.B.--Dry Bulb Temperature °F　　R.H.--Relative Humidity　　E.T.--Effective Temperature

APPENDIX I--TABLE NO.10 CLIMATIC DATA FOR LOWER EGYPT

	Wadi Halfe	Saleh	Borku	Barbar	MEAN DAILY RANGE OF DAKHLA OASIS
Highest recorded(April) (mid-day noon)	126	127			J F M A M J 30 33 34 36 36 33
Maximum Temperature(June)	106	118	112	112	J A S O N D 31 31 29 29 29 32
Lowest recorded(midnight)	28				SAHARAN MEAN TEMP.
Minimum (January)	46	70	45	49	Winter(Jan) 50°F. Summer(July)100°F.

APPENDIX I—TABLE NO. 11 SAHARA TEMPERATURE RECORDS

Figure 31



95
90
85
D.B
80
°F
75
70
65
60
55
50
MONGALLA
KHARTOUM
BERBER
ASSIUT
CAIRO
ALEXANDRIA
LAT 31 12' N
ALT 32 mtr
ALT. 55 mtr
LAT. 27 11'N
LAT 30 03 N
ALT 20 metre
MONTHS
J F M A M J J A S O N D
40
20
10
2
10
20
40
60
80
100
RELATIVE HUMIDITY CURVES
80
70
60
50
40
30
20
R.H
PORT SAID
ALEXANDRIA
CAIRO
ASSIUT
ASSWAN
J F M A M J J A S O N D

APPENDIX I--TABLE NO. 12

MEAN WIND VELOCITIES IN MID-WINTER FOR EGYPT

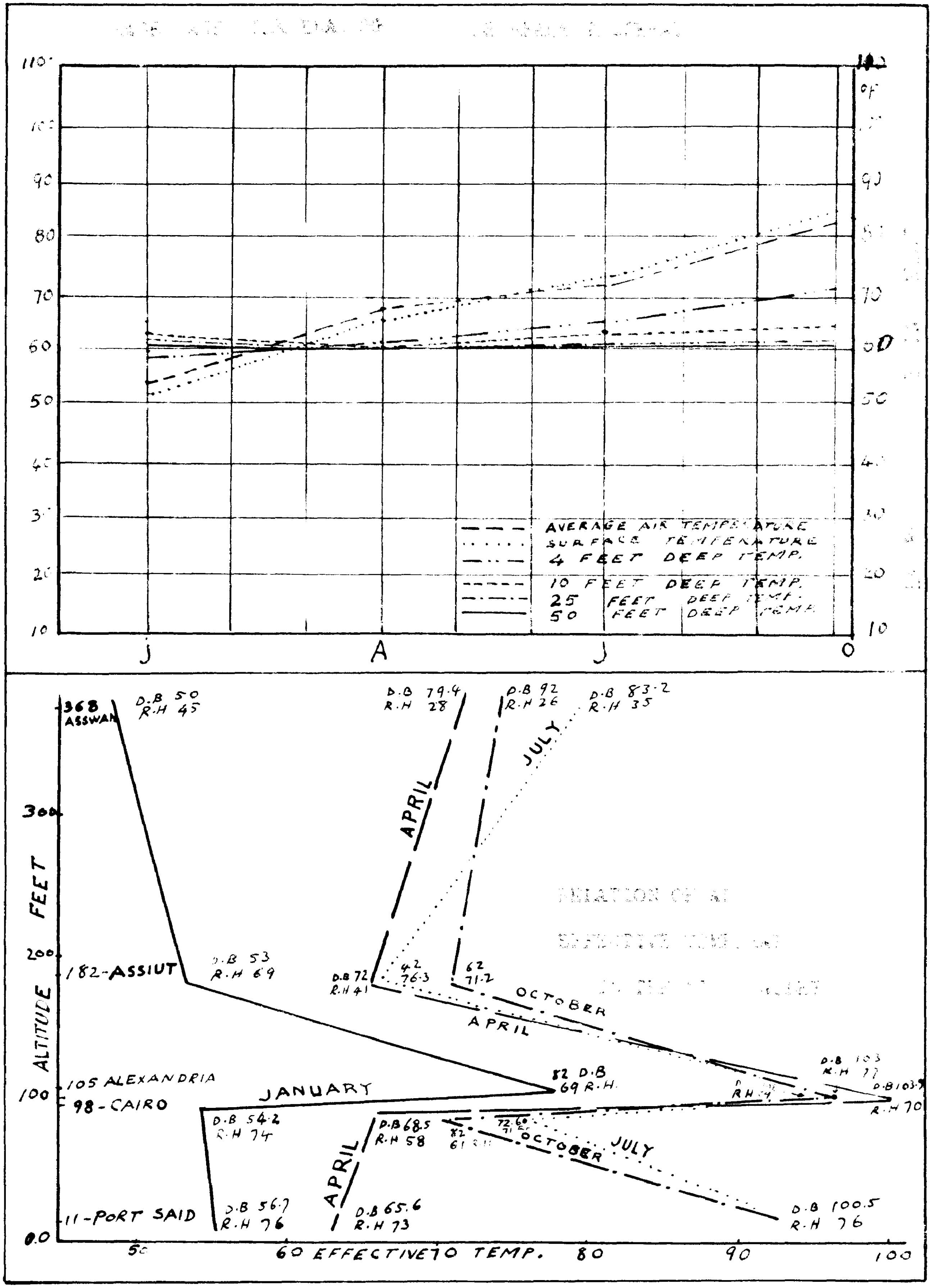
	N.	N.E.	E.	S.E.	S.	S.W.	W.	N.W.	CALM
CAIRO	10	2	1	3	28	6	3	3	44
SALAH	12	44	13	3	3	6	3	3	13
ASSIUT	80% blow from N.W., N., or N. E.								

APPENDIX I--TABLE NO. 13

PERCENTAGE WIND VELOCITY IN MID-SUMMER FOR EGYPT

	N.	N.E.	E.	S.E.	S.	S.W.	W.	N.W.	CALM
CAIRO	43	8	3	0	1	1	12	15	17
ASSWAN	83	3	0	0	1	2	1	5	5
SALAH	6	41	30	7	2	6	3	1	4
ASSIUT	80% northerly or easterly								

Figure 32.



AVERAGE AIR TEMPERATURE
SURFACE TEMPERATURE
4 FEET DEEP TEMP.
10 FEET DEEP TEMP.
25 FEET DEEP TEMP.
50 FEET DEEP TEMP.
J
A
J
O
368 ASSWAN
D.B 50 R.H 45
182-ASSIUT
D.B 53 R.H 69
105 ALEXANDRIA
98-CAIRO
11-PORT SAID
ALTITUDE FEET
JANUARY
APRIL
JULY
OCTOBER
D.B 79.4 R.H 28
D.B 92 R.H 26
D.B 83.2 R.H 35
D.B 72 R.H 41
82 D.B 69 R.H.
D.B 54.2 R.H 74
D.B 68.5 R.H 58
D.B 56.7 R.H 76
D.B 65.6 R.H 73
D.B 100.5 R.H 76
60 EFFECTIVE 70 TEMP. 80
90
100

MONTH	MEAN TEMPERATURES			DEGREE DAY	TOTAL PRECIPITATION	RELATIVE HUMIDITY		PERCENT POSSIBLE SUNSHINE
	DAILY MAX	DAILY MINIMUM	MONTHLY			1:30 A.M.	1:30 P.M.	
JANUARY	57.0	39.5	[illegible]	733	1.74	70	54	44
FEBRUARY	[illegible]	31.5	38.7	732	2.68	69	55	45
MARCH	51.0	32.0	41.8	717	4.05	61	44	58
APRIL	63.0	42.5	52.8	377	1.87	66	44	57
MAY	73.9	56.1	65.0	74	5.96	82	58	46
JUNE	83.9	63.9	73.9	0	3.44	75	50	73
JULY	84.9	67.8	76.4	0	4.20	85	54	60
AUGUST	85.0	67.0	76.0	0	5.30	81	54	67
SEPTEMBER	74.4	59.5	67.0	57	7.27	85	69	44
OCTOBER	70.6	51.1	60.9	149	3.33	85	58	55
NOVEMBER	57.8	33.1	48.0	520	2.66	69	45	59
DECEMBER	[illegible]	25.5	35.6	907	3.69	69	55	46
YEARLY	65.9	48.2	57.0	4046	46.39	75	53	55

TABLE NO. 14 CLIMATIC CHART FOR [illegible], N. C.

APPENDIX II

Equivalent Temperature: The scale of warmth known as effective temperature allows for air temperature, humidity and air movement, but makes no specific allowance for radiant heat. Considering the fact that the mean radiant temperature differs considerably from the air temperature, the effective temperature gives an inaccurate impression of the overall level of warmth.

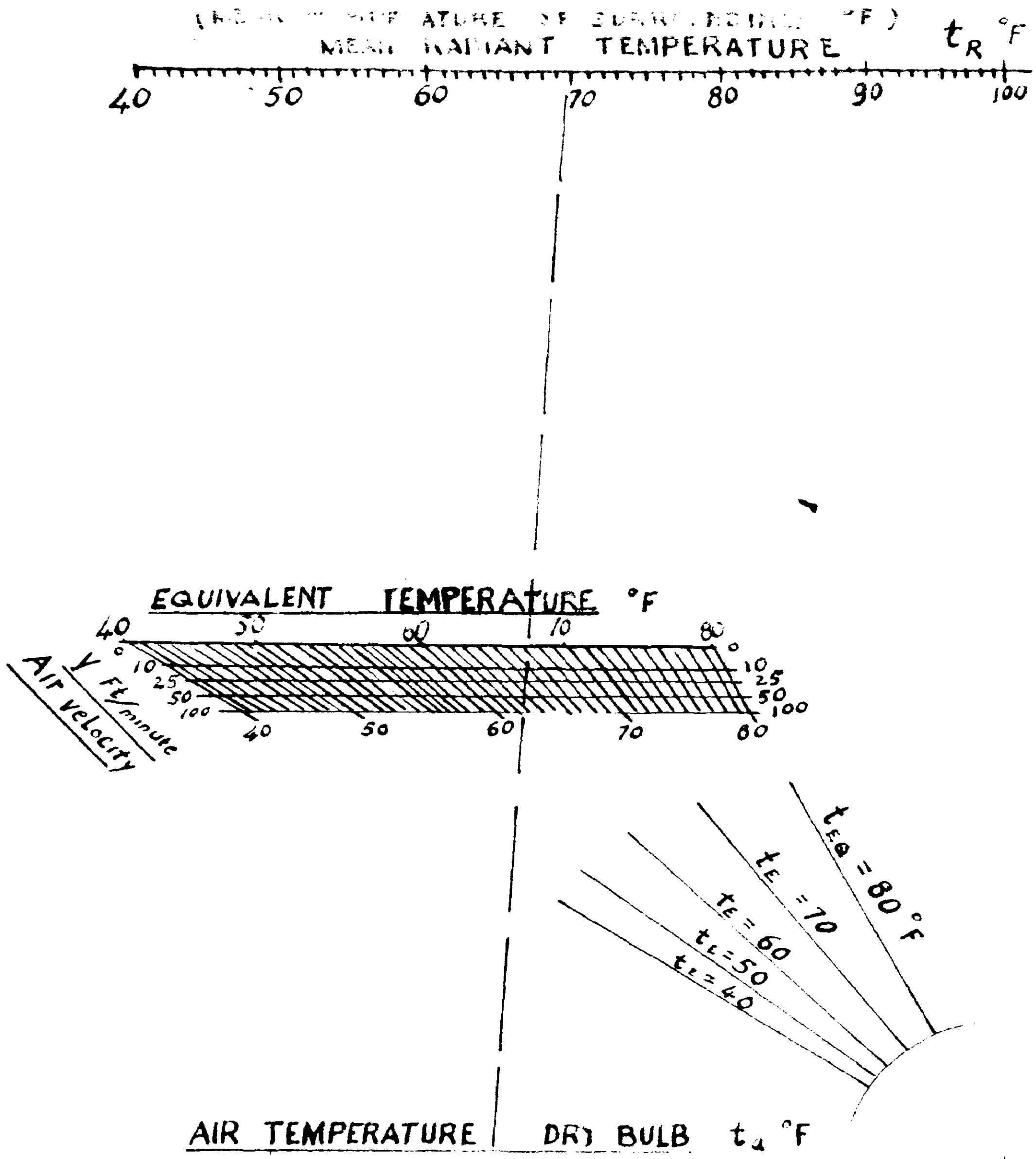
In contrast, the scale of equivalent temperature, as used in Great Britain for the past twenty years, does take into account radiant heat. In 1929, there was introduced by Dufton, a thermostatic device called the eupatheostat, which was sensitive to radiation from surrounding objects, to sun-shine and drafts, and which kept itself "comfortable" by closely regulating the heating of the room.

The comfort indicator (eupatheoscope) was later introduced in 1930-32. In the direct reading form of the instrument, a coil carrying a proportion of the heating current was wound round the bulb of a thermometer, the stem of which was fixed to the outside of the cylinder. In this way, the reading of the thermometer depended on the combined effects of convection and radiation. The thermometer read in terms of equivalent temperature, meaning "that temperature of a uniform enclosure, with still air, in which a sizable black body would lose heat at the same rate as that observed." (5)

As the Eupatheoscope would be unsuitable for warmer conditions, Willard, Kratz and Fahnestock in 1933 constructed a similar instrument with a surface temperature of 83°F.

Figure 33

HUMAN COMFORT EQUIVALENT TEMPERATURE



$$t_{\text{EQUIVALENT T.}} = 0.522\, t_a + 0.478\, t_R - 0.01474\, V^{\frac{1}{2}} (100 - t_a)$$

t_a : Dry bulb air temperature °F
t_R : mean radiant temperature °F
V : air velocity in Ft per minute

Courtesy of His Majesty's Stationery Office ; THOMAS BEDFORD LONDON 1936

A modified eupatheoscope was constructed by Dufton in 1932 and later adopted by the British Inter-Departmental Committee on Heating and Ventilation Research. The definition of equivalent temperature was changed to: "That temperature of a uniform enclosure in which, in still air, a black body of sufficient size would lose heat at the same rate as in the environment, the surface temperature of the body being one-third of the way between the temperature of the enclosure and 100°F."

Equivalent Temperature

$$= 0.522\ ta + 0.478\ t_r - 0.01474(v)^{\frac{1}{2}}(100-ta)\ldots\ldots$$

where: ta = air temperature
t_r = mean radiant temperature in degree Fahrenheit
v = air speed, fpm.

The most convenient method for calculating the direct measurement of the mean radiant temperature is with a radiation thermopile. The mean radiant temperature can also be obtained from readings of the black globe thermometer (Bedford and Warner, 1934) if the air temperature and velocity are also known.

After observations have been made with the six inch globe thermometer, the equivalent temperature can be estimated (without calculating the mean radiant temperature, by using the following relationship:-

Equivalent Temperature

$$= 0.522\ ta + 0.478\ tg + (v)^{\frac{1}{2}}(0.0808\ tg - 0.0661\ ta - 1.474)\ldots$$

where: tg is the temperature indicated by the 6 inch globe thermometer.

For normal indoor conditions, equivalent temperature is an excellent scale of warmth, valid up to those temperatures at which evaporative losses from the body become important.

<u>Effect of Atomic Blast Radiation on Human Comfort</u>: Since atomic energy radiation has become another hazard to human comfort, the author made the following study.

There are three types of explosions by atomic bomb; high in the air, on the surface, and in water. Considering the bomb exploded high in the air, such as the ones used at Hiroshima and Nagasaki, we find that an unprotected city would be affected in several ways.(21) In an area of .197 square miles below the burst, there would be approximately 70,000 fatal injuries and complete ruin. Outward from this area to a distance of two miles from the center, the damage would range from heavy to light and casualty effects would vary from severe to mild. Beyond the two mile radius, the damage would consist mainly of broken windows and falling plaster. No effect would be discernable beyond the eight mile radius. After the atomic explosion, in severe and moderate areas, flesh burn cases in all degrees of severity will occur from exposure to the light and heat produced by the explosion. Of the almost 70,000 fatal injuries, from 15 to 20% would be due to radio-activity.

The atomic explosion will give rise to various types of penetrating radiations, namely Gama Rays (which are electromagnetic rays similar to X-rays); neutrons (electrically neutral particles thrown from atomic nuclei); Alpha and Beta Rays (nuclear particles of negative and positive charges) respectively.

Gama Rays are the source of trouble as they produce ionization within the blood-forming cells, the intestinal tract, and tissues of the body, causing chemical disruption of the cellular contents.

Radiation injuries affect the body's ability to form white blood cells, while injury to the intestinal tract prevent clotting, resulting in severe intestinal bleeding. The severity of the illness depends upon the amount of radiation received. If the amount was not initially fatal, the injured tissue will recover if the body is aided in its recovery process by proper medical treatment, rest, adequate intake of nutrients and fluids, including blood transfusions, use of aureomycin and similar drugs. The procedure for self-protection, outlined by Civil Defense Officials, include the admonition to wear loose clothing of a light color covering most of the body, to erect shelters according to specifications and to keep a supply of preserved foods and liquids on hand.

	COMFORT	HOT-DRY	HOT-WET
CONDITION OF ENVIRONMENT (TEMPERATURE °F.)			
Dry Bulb	84.0	99.5	99.5
Wet Bulb	64.6	71.6	90.0
HEAT PRODUCTION ($Btu/ft^2/hr$)	13.4	13.6	15.81
HEAT DISPOSAL-($Btu/ft^2/hr$) Storage:	0	.59	2.85
Conduction:	9.1	3.85	2.67
Convection:	4.36	9.15	10.04
BLOOD FLOW-($Liters/hr/meter^2$)	4.2	20.2	33.81
CARDIAC OUTPUT (Liters/hr)	144	174	243
PERCENTAGE CARDIAC OUTPUT FOR HEAT LOSS (SM^2)	2.9	11.7	13.9
RECTAL TEMPERATURE °F.	97.5	98.4	99.5
PULSE RATE (Beats per minute)	63	68	87

TABLE NO. 15

PHYSIOLOGICAL EFFECTS OF VARIOUS TEMPERATURES UPON THE HUMAN BODY.

APPENDIX III-The Temperature Gradient.

In special papers No. 36 & 37 CEI G 29 S of the Geological Society of America and in the A.A.P.G. Journal No. 8, 1944, the equation of a straight line T = a + by

where: t = temperature in degree Fahrenheit
y = depth in feet
a.b = constants; varies for certain depths and constants between two depth limits. They depend on the geological, geographical, and natural effect for different countries.

i.e. The temperatures at various depths in a certain site, vary for different limits due to the variation of the constants. The temperature depth flow curve is a curve of straight connected lines.

This equation was tested with values for constants a and b for Long Beach, California to a depth of 2743 feet and at Santa Fe Spring, Carnarvon, Johannesberg, and Kiwatersrand, Africa. The following temperatures were calculated and recorded for Santa Fe Spring:

$$a = 21.9,\ b = 1.379,\ y = 34.67',\ 40.17' \text{ and } 41.12'$$

$$T_{34.67} = 21.9 + 1.379 \times 34.67 = 69.5^{\circ}F.$$

$$T_{40.17} = 21.9 + 1.379 \times 40.17 = 77.1^{\circ}F.$$

$$T_{41.42} = 21.9 + 1.379 \times 41.12 = 78.7^{\circ}F.$$

(As the temperatures of the soil at various depths have boundary conditions which are dependent upon the time due to the climate and its changes throughout the year and considering soil specifications are constants for certain depths.)

Evaluating the work done by Gröber (Temperaturverlauf und Wärmestroemungen in Periodisch-erwärmten Korpern, Forshungsarbeiten-Heft 300) for the case we have in this research, and considering its boundary conditions

at: $y = 0$ $\qquad t_o = \phi(\theta)$
$y = \infty$ $\qquad t_{\infty} \neq \infty$

and taking the initial temperature for a cold climate $t = 0$ at time $\theta = 0$.

$$\frac{dt}{d\theta} = a\frac{\partial^2 t}{\partial y^2}$$

$$T = \Theta \cdot \Upsilon$$

Θ Υ

θ

θ

$$\frac{\Theta'}{a\Theta} = \frac{\Upsilon''}{\Upsilon} = \pm i\lambda^2$$

$\pm i\lambda^2$ const

$i = \sqrt{-1}$

$$\Theta' - (\pm i)\lambda^2 a\Theta = 0$$

$$\Upsilon'' - (\pm i)\lambda^2 \Upsilon = 0$$

$$T = C e^{\pm i\lambda^2 a\theta} e^{\pm\lambda\sqrt{\pm i}\, y}$$

$$T_1 = C_1 e^{+i\lambda^2 a\theta \pm \lambda\sqrt{+i}\, y}$$

$$T_2 = C_2 e^{-i\lambda^2 a\theta \pm \lambda\sqrt{-i}\, y}$$

$$(1+i)^2 = 2i \qquad i = \frac{(1+i)^2}{2}$$

$$(1-i)^2 = -2i \qquad -i = \frac{(1-i)^2}{2}$$

$$t_1 = C_1 e^{-\sqrt{\frac{1}{2}}\lambda y} e^{+i(\lambda^2 a\theta - \sqrt{\frac{1}{2}}\lambda y)}$$

$$t_2 = C_2 e^{-\sqrt{\frac{1}{2}}\lambda y} e^{-i(\lambda^2 a\theta - \sqrt{\frac{1}{2}}\lambda y)}$$

$$t_3 = C_3 e^{+\sqrt{\frac{1}{2}}\lambda y} e^{+i(\lambda^2 a\theta + \sqrt{\frac{1}{2}}\lambda y)}$$

$$t_4 = C_4 e^{+\sqrt{\frac{1}{2}}\lambda y} e^{-i(\lambda^2 a\theta + \sqrt{\frac{1}{2}}\lambda y)}$$

For $t_\infty \neq \infty$
t_3, t_4
ELIMINATED

$$t = e^{-\sqrt{\frac{1}{2}}\lambda y}\left[C_1 e^{+i(\lambda^2 a\theta - \sqrt{\frac{1}{2}}\lambda y)} + C_2 e^{-i(\lambda^2 a\theta - \sqrt{\frac{1}{2}}\lambda y)}\right]$$

$$e^{\pm i(\lambda^2 a\theta - \sqrt{\frac{1}{2}}\lambda y)} = \cos(\lambda^2 a\theta - \sqrt{\tfrac{1}{2}}\lambda y) \pm i\sin(\lambda^2 a\theta - \sqrt{\tfrac{1}{2}}\lambda y)$$

$$t = e^{-\sqrt{\frac{1}{2}}\lambda y}\left[(C_1 + C_2)\cos(\lambda^2 a\theta - \sqrt{\tfrac{1}{2}}\lambda y) + (C_1 - C_2)\, i\sin(\lambda^2 a\theta - \sqrt{\tfrac{1}{2}}\lambda y)\right]$$

$$C_1 = A - i\frac{B}{2}, \qquad C_2 = A + i\frac{B}{2}$$

$$t = e^{-\sqrt{\frac{1}{2}}\lambda y}\left[A\cos(\lambda^2 a\theta - \sqrt{\tfrac{1}{2}}\lambda y) + B\sin(\lambda^2 a\theta - \sqrt{\tfrac{1}{2}}\lambda y)\right]$$

[illegible] is [illegible] by:

$$t = C e^{-\sqrt{\frac{1}{2}}\lambda y} \cos\left[\lambda^2 a\theta - \sqrt{\tfrac{1}{2}}\lambda y - \alpha\right]$$

[illegible] $C = \sqrt{A^2 - B^2}$ and $\alpha = \tan^{-1}\frac{B}{A}$

and C, α, λ, A and B are determined [illegible] [illegible] conditions.

$$t = \phi(\theta) \qquad \text{at ground surface } y = 0$$

[illegible] series $t_0 = \dfrac{a_0}{2} + \displaystyle\sum_{n=1}^{n=\alpha}\left[a_n \cos\frac{2\pi n}{\theta_0}\theta + b_n \sin\frac{2\pi n}{\theta_0}\theta\right]$

$y = 0$ $\qquad t_0 = A\cos\lambda^2 a\theta + B\sin\lambda^2 a\theta$

when $\lambda = \sqrt{\dfrac{2\pi n}{\theta_0 a}}$, $A = a_n$, $B = b_n$

the [illegible] equation which satisfies the boundary conditions is:

$$t = \frac{a_0}{2}\left[1 - \Theta\left(\frac{y}{2\sqrt{a\theta_0}}\right) + \sum_{n=1}^{n=\alpha} e^{-\sqrt{\frac{n\pi}{a\theta_0}}y}\left[a_n\cos\left(\frac{2\pi n}{\theta_0}\theta - \sqrt{\frac{n\pi}{a\theta_0}}y\right) + b_n\sin\left(\frac{2\pi n}{\theta_0}\theta - \sqrt{\frac{n\pi}{a\theta_0}}y\right)\right]\right]$$

[illegible] initial condition of $t_{\theta=0} = 0$

[illegible]

$$t_1 = \frac{1}{2\sqrt{a\pi\theta}}\int_0^{\alpha} f(y')\left[e^{-\frac{(y-y')^2}{4a\theta}} - e^{-\frac{(y+y')^2}{4a\theta}}\right]dy'$$

[illegible] general equation to [illegible] general form:

$$t = \frac{a_0}{2}\left[1 - \Theta\left(\frac{y}{2\sqrt{a\theta_0}}\right)\right] + \sum_{n=1}^{n=\alpha} e^{-\sqrt{\frac{n\pi}{a\theta_0}}y}\left[a_n\cos\left(\frac{2\pi n}{\theta_0}\theta - \sqrt{\frac{n\pi}{a\theta_0}}y\right) + b_n\sin\left(\frac{2\pi n}{\theta_0}\theta - \sqrt{\frac{n\pi}{a\theta_0}}y\right)\right]$$
$$- \frac{1}{2\sqrt{a\pi\theta}}\int_0^{\alpha}\sum_{n=1}^{n=\alpha} e^{-\sqrt{\frac{n\pi}{a\theta_0}}y'}\left[a_n\cos\sqrt{\frac{n\pi}{a\theta_0}}y' - b_n\sin\sqrt{\frac{n\pi}{a\theta_0}}y'\right]\left[e^{-\frac{(y-y')^2}{4a\theta}} - e^{-\frac{(y+y')^2}{4a\theta}}\right]dy'$$

From tests made in Cairo (Egypt), College [illegible] ([illegible]), and [illegible] [illegible], where [illegible] [illegible] [illegible] ([illegible]).

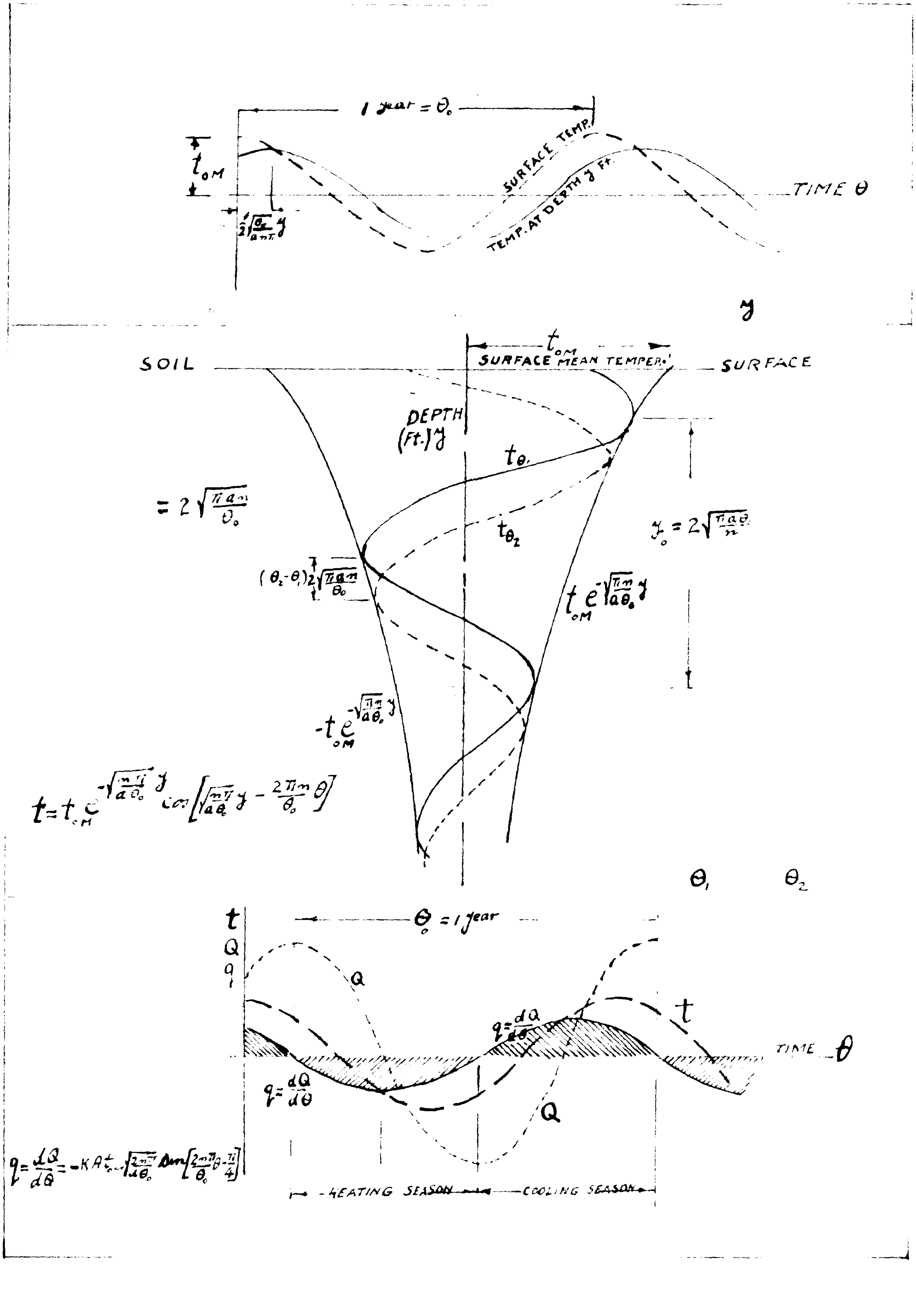
Below this depth, the temperature will [illegible] [illegible]:

$$t = \frac{a_0}{2} + \sum_{n=1}^{n=\alpha} e^{-\sqrt{\frac{n\pi}{a\theta_0}}y}\, C_n\cos\left[\frac{2\pi n}{\theta_0}\theta - \sqrt{\frac{n\pi}{a\theta_0}}y - \alpha_n\right]$$

where $C_n = \sqrt{a_n^2 + b_n^2}$, $\alpha_n = \tan^{-1}\dfrac{b_n}{a_n}$

[illegible] temperature oscillation [illegible] values for [illegible] [illegible] over a period of six months every year, [illegible] [illegible] $= \frac{a_0}{2}$.

Figure 34


1 year = θ_0
t_{oM}
SURFACE TEMP.
TEMP. AT DEPTH y FT.
TIME θ
y
SOIL
t_{oM}
SURFACE MEAN TEMPER.
SURFACE
DEPTH (Ft.) y
t_{θ_1}
t_{θ_2}
$= 2\sqrt{\frac{\pi a n}{\theta_0}}$
$y_0 = 2\sqrt{\frac{\pi a \theta_0}{n}}$
$(\theta_2-\theta_1)2\sqrt{\frac{\pi a n}{\theta_0}}$
$t_{oM} e^{-\sqrt{\frac{\pi n}{a\theta_0}}y}$
$-t_{oM} e^{-\sqrt{\frac{\pi n}{a\theta_0}}y}$
$t = t_{oM} e^{-\sqrt{\frac{n\pi}{a\theta_0}}y} \cos\left[\sqrt{\frac{n\pi}{a\theta_0}}y - \frac{2\pi n}{\theta_0}\theta\right]$
θ_1
θ_2
t
Q
q
$\theta_0 = 1$ year
Q
t
$q = \frac{dQ}{d\theta}$
TIME θ
$q = \frac{dQ}{d\theta}$
Q
$q = \frac{dQ}{d\theta} = -KAt_{oM}\sqrt{\frac{2n\pi}{a\theta_0}} \sin\left[\frac{2n\pi}{\theta_0}\theta - \frac{\pi}{4}\right]$
HEATING SEASON
COOLING SEASON

Considering (t_o) is a single harmonic

$$t = t_{oM} \cos \frac{2\pi n}{\theta_o} \theta$$

and

$$t = t_{oM} e^{-\sqrt{\frac{n\pi}{a\theta_o}} y} \cos \left[\frac{2\pi n}{\theta_o} \theta - \sqrt{\frac{n\pi}{a\theta_o}} y \right]$$

which is the same as

$$t = t_{oM} e^{-\sqrt{\frac{n\pi}{a\theta_o}} y} \cos \left[\sqrt{\frac{n\pi}{a\theta_o}} y - \frac{2\pi n}{\theta_o} \theta \right]$$

The maximum temperature, at any given y is at time θ satisfying.

$$\cos \left[\frac{2\pi n}{\theta_o} \theta - \sqrt{\frac{n\pi}{a\theta_o}} y \right] = 1 \qquad \text{(y fixed)}$$

$$\cos 2 m \pi = 1 \qquad m = 0, 1, 2, 3 \ldots$$

where $$\frac{2\pi n}{\theta_o} \theta - \sqrt{\frac{n\pi}{a\theta_o}} y = 2 m \pi$$

Therefore $$\theta_{max} = \frac{m}{n} \theta_o + \frac{1}{2} \sqrt{\frac{\theta_o}{a n \pi}} y$$

which gives the maximum time for a certain temperature to reach a certain depth.

The maximum time for the surface at certain temperature is when $$\frac{2\pi n}{\theta_o} \theta - 0 = 2 m \pi \qquad y = 0$$

$$\theta_{t\,max} = \frac{m}{n} \theta_o \qquad y = 0$$

The time lag for depth y (feet)

$$\theta = \frac{1}{2} \sqrt{\frac{\theta_o}{a n \pi}} y$$

$$t = t_{oM} e^{-\sqrt{\frac{n\pi}{a\theta_o}} y} \cos \left[\sqrt{\frac{n\pi}{a\theta_o}} y - \frac{2\pi n}{\theta_o} \theta \right]$$

Dividing this equation into two partial functions:

1st function $$f_1(y) = t_{oM} \cos \sqrt{\frac{n\pi}{a\theta_o}} y$$

where the wave length = $$y_o = 2 \sqrt{\frac{\pi a \theta_o}{n}}$$

$$\therefore f_2(y) = t_{oM} \cos\left[y\sqrt{\frac{n\pi}{a\theta_o}} - \frac{2\pi n}{\theta_o}\theta\right]$$

is the same as $f_1(y)$ except the wave moved a distance:

$\frac{2\pi n}{\theta_o}\theta$ as θ increased.

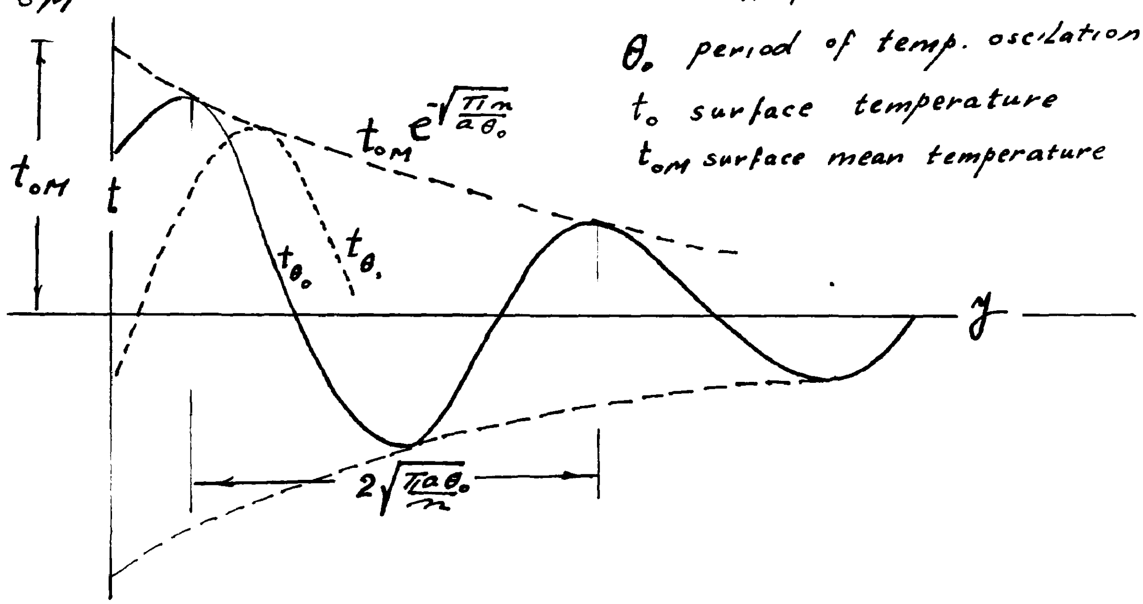
The period is $= \frac{\theta_o}{n}$

The velocity of propagation $= \frac{\text{the wave length}}{\frac{\theta_o}{n}} = 2\sqrt{\frac{\pi a\theta_o}{n}} \Big/ \frac{\theta_o}{n}$

$$= 2\sqrt{\frac{\pi a n}{\theta_o}}$$

General temperature equation: $t = t_{oM}\, e^{-\sqrt{\frac{n\pi}{a\theta_o}}\, y} \cos\left[\sqrt{\frac{n\pi}{a\theta_o}}\, y - \frac{2\pi n}{\theta_o}\theta\right]$

represents the same wave as $f_2(y)$ in which the amplitude t_{oM} decreases to $t_{oM}\, e^{-\sqrt{\frac{n\pi}{a\theta_o}}\, y}$ for increasing values of (y)



Heat flow: The heat absorbed or rejected, through the area A of the flat plate during the half period may be determined from the equation of conduction:

$$dQ = -KA\left(\frac{\partial t}{\partial y}\right)_{y=0} d\theta$$

from equation: $t = t_{oM}\, e^{-\sqrt{\frac{n\pi}{a\theta_o}}\, y} \cos\left[\frac{2\pi n}{\theta_o}\theta - \sqrt{\frac{n\pi}{a\theta_o}}\, y\right]$

$$\frac{\partial t}{\partial y} = -t_{oM}\sqrt{\frac{n\pi}{a\theta_o}}\, e^{-\sqrt{\frac{n\pi}{a\theta_o}}\, y}\left[-\sin\left(\frac{2\pi n}{\theta_o}\theta - \sqrt{\frac{n\pi}{a\theta_o}}\, y\right) + t_{oM}\left(-\sqrt{\frac{n\pi}{a\theta_o}}\right) e^{-\sqrt{\frac{n\pi}{a\theta_o}}\, y} \cos\left[\frac{2\pi n}{\theta_o}\theta - \sqrt{\frac{n\pi}{a\theta_o}}\, y\right]\right]$$

$$\left(\frac{\partial t}{\partial y}\right)_{y=0} = t_{oM}\sqrt{\frac{n\pi}{a\theta_o}}\left[\sin\frac{2\pi n}{\theta_o}\theta - \cos\frac{2\pi n}{\theta_o}\theta\right]$$

$$q = \frac{dQ}{d\theta} = -KA\,t_{oM}\sqrt{\frac{n\pi}{a\theta_o}}\left[\sin\frac{2\pi n}{\theta_o}\theta - \cos\frac{2\pi n}{\theta_o}\theta\right]$$

[illegible] $\lambda = \frac{\pi}{2}$ [illegible] $\cos\lambda = \frac{1}{\sqrt{2}}$, $\sin\lambda = \frac{1}{\sqrt{2}}$

$$q = -KA\,t_{oM}\sqrt{\frac{n\pi}{a\theta_o}}\sqrt{2}\,\sin\left[\frac{2\pi n}{\theta_o}\theta - \frac{\pi}{4}\right]$$

[illegible] flow [illegible]

$$\int dQ = Q\Big]_a^b = KA\,t_{oM}\sqrt{\frac{\theta_o}{2\pi n a}}\cos\left[\frac{2\pi n}{\theta_o}\theta - \frac{\pi}{4}\right]_a^b$$

$$= KA\,t_{oM}\sqrt{\frac{\theta_o}{2\pi n a}}\left[\cos 0 - \cos\pi\right]$$

$$= KA\,t_{oM}\sqrt{\frac{2\theta_o}{\pi n a}}$$

[illegible]

[illegible] Fourier solution is:

$$t = A_o + A_1\cos\frac{2\pi}{\theta_o}\theta + B_1\sin\frac{2\pi}{\theta_o}\theta + A_2\cos\frac{4\pi}{\theta_o}\theta + B_2\sin\frac{4\pi}{\theta_o}\theta + \cdots$$

$$= A_o + C_1\cos\left(\frac{2\pi}{\theta_o}\theta - \delta\right) + C_2\cos\left(\frac{4\pi}{\theta_o}\theta - \delta_2\right)$$

[illegible]

$$C_n = \sqrt{A_n^2 + B_n^2}$$

$$\delta_n = \tan^{-1}\frac{B_n}{A_n}$$

[illegible] δ_n

[illegible] the

( )	[illegible]	[illegible]	1	δ degrees	δ [illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]°	[illegible]'
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]°	[illegible]'
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]°	[illegible]'
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]°	[illegible]'

[illegible] between two depths for a certain temperature:

$$\theta = \frac{y' - y}{2}\sqrt{\frac{\theta_o}{a\pi}}$$

APPENDIX III--CORRECTION FOR ERRORS IN TEMPERATURE READINGS

The measurement of soil temperatures in the experiments at the U. C. L. A. and the University of Maryland, was corrected by following the equation of Gröber (Die Wärmeübertragung) because of heat conducted through the film surrounding the well, along the well, and to the wall of the well.

Disregarding the possibility of radiation between the soil and the thermometer, and following heat by convection and conduction, the temperature distribution along the well is an exponential function of an oscillatory wave:

$$t = t_{oM}\, e^{-\sqrt{\frac{n\pi}{a\theta_0}}\, y} \cos\left[\frac{2\pi n}{\theta_0}\theta - \sqrt{\frac{n\pi}{a\theta_0}}\, y \right]$$

which can be expressed as a function of the hyperbolic cosine of the distance:

$$t = t_{oM} \cdot \frac{1}{\cosh my}$$

where: t = the difference in temperature between the air at the bottom of the well and the wall of the well near the point at which the thermometer is immersed.
It is the approximate error due to heat conduction from the well.

$t_{o}M$ = the surface mean temperature

$m = \sqrt{\frac{h\, 2\pi r}{KA}}$ per ft.

$h = \frac{Btu}{ft^2\, hr\, {}^\circ F}$ film conduction between the wall and air

r = inside radius of the well in feet

K = thermal conduction of the soil Btu/ft^2 hr. $\frac{{}^\circ F.}{ft.}$

A = cross-sectional area of walls of well in ft.^{2}

y = depth of the well in feet.

$m = \sqrt{\frac{h\, 2\pi r}{K\, 2\pi r\, dr}} = \sqrt{\frac{h}{Kr}}$, dr = (difference in radii between the thermometer and the inside radius well) = r

$t = t_{oM} \frac{1}{\cosh y\sqrt{\frac{h}{Kr}}}$

As an example for each 4.3 feet deep and using:

$$t = t_{oM} \cdot \frac{1}{\cosh y\sqrt{\frac{h}{Kr}}}$$

h = 1.022 (for still air inside the well of one ft. radius—ASHVE Guide)

K = .19 sand

dr = 1 feet

y = 4.3 feet

$$t = t_{oM} \cdot \frac{1}{11013}$$

$= 50 \times .0905 \times 10^{-3}$

$\doteq .0045°F.$

APPENDIX III---SOIL TEMPERATURE AT VARIOUS DISTANCES FROM CENTER OF VERTICAL GROUND PIPE IN A HORIZONTAL PLANE

The theoretical and analytical solution of heat flow to the pipe from the ground at initial temperature T_o, was studied by Lord Kelvin and presented in his mathematic and physical papers (Sir W. Thomson), Vo. II, p. 41ff. Lord Kelvin's solution is used in determining soil temperature at any distance from the center of the pipe after a certain period of time. Following is a rather simple treatment of the Heat Source Theory.

Since heat is obtained from a steady, permanent source of heat or sink, in an infinite medium at the initially uniform temperature T_o, then the subsequent temperature at any point in the medium is given as:-

$$T - T_o = \frac{Q}{2\pi K} \int_x^\infty \frac{e^{-\beta^2}}{\beta} d\beta$$

where: T = temperature in soil at any distance from the vertical pipe in horizontal direction, °F.

T_o = initial temperature of soil before the operation, °F.

$+Q$ = heat emission of pipe) } Btu per linear foot

$-Q$ = heat absorption of pipe) } vertical height

$X = \frac{r}{2\sqrt{a\theta}}$

r = distance from center line of pipe in horizontal plane, ft.

K = thermal conductivity of the soil

a = thermal diffusivity of the soil $= \frac{K}{C_p \rho}$

ρ = density of soil lb/cubic ft.

C_p = specific heat

θ = time since start of operations, hours

β = variable of integration

For various values of $\frac{r}{2\sqrt{a\theta}}$ we find the values of the integration $\int_x^\infty \frac{e^{-\beta^2}}{\beta} d\beta$ in table 1.

For the values of $\frac{r}{2\sqrt{a\theta}}$ less than .2

$$\int_x^\infty \frac{e^{-\beta^2}}{\beta} d\beta = 2.303 \log_{10} \frac{1}{x} + \frac{x^2}{2} - \frac{x^4}{8} - .2886$$

while in table 2, we get the values of temperature difference for distance from the center line of the vertical pipe in a horizontal plane.

TABLE I

[illegible]

$\frac{r}{2\sqrt{a\theta}}$	$\int_x^{\alpha} \frac{e^{-\beta^2}}{\beta} d\beta$	$\frac{r}{2\sqrt{a\theta}}$	$\int_x^{\alpha} \frac{e^{-\beta^2}}{\beta} d\beta$	$\frac{r}{2\sqrt{a\theta}}$	$\int_x^{\alpha} \frac{e^{-\beta^2}}{\beta} d\beta$
.0001	8.9217	.25	1.1235	1.2	.0547
.0005	7.3123	.3	.9594	1.3	.0379
.001	6.6191	.35	.8206	1.4	.0259
.0048	5.06	.4	.7046	1.5	.0174
.005	5.0097	.5	.5221	1.6	.0115
.01	4.3166	.6	.3872	1.7	.0075
.05	2.7084	.7	.2860	1.8	.0048
.1	2.019	.8	.2090	1.9	.003
.15	1.6197	.9	.1525	2.0	.0019
.2	1.3406	1.0	.1097	2.2	.0007
		1.1	.0780		

TABLE II

[illegible] $(T - T_0)$ [illegible]
[illegible]

[illegible] (r)	T - T_0									
	1 wk.		2 wks.		1 mth.		2 mths.		6 mths.	
	Soil 1	2	1	2	1	2	1	2	1	2
.0208	.66	1.86	.71	2.0	.76	2.16	.81	2.3	.88	2.53
.0417	.57	1.61	.62	1.72	.67	1.88	.72	2.02	.79	2.24
.083	.48	1.31	.54	1.45	.58	1.62	.62	1.74	.70	1.96
.33	.3	.76	.34	.90	.40	1.05	.44	1.19	.52	1.41
.67	.21	.49	.25	.63	.30	.73	.35	.92	.42	1.13
1	.16	.34	.2	.47	.25	.62	.3	.75	.37	.97
1.5	.11	.21	.15	.33	.2	.46	.24	.60	.31	.81
2.5	.05	.07	.09	.16	.13	.28	.18	.4	.25	.61
5.0			.02	.02	.06	.08	.09	.17	.15	.35

Soil 1-wet clay-(k = 1.2, c_p = .45, ρ = 102, a = .0261)
Soil 2-moist soil (k = .4, c_p = .3, ρ = 100, a = .0133)

It is found that the Kelvin equation can give good values of pipes less than 2 inches in diameter, which are more efficient than larger ones, as the small and longer pipe have more earth from which to draw heat and so the temperature will be kept high for a longer period at the same rate of heat absorption per unit area. This equation is built on radial heat flow. That means that the vertical pipe should be long enough to allow all heat flow to be radial.

Ingersoll and Plass illustrated Kelvin's equation with the following example: (23)

A long pipe, one inch in outside diameter, vertically buried in soil of conductivity $K = 1.2$, specific heat $C_p = .45$, density $\rho = 102$, diffusivity $a = .0261$, at an initial temperature $T_o = 50^\circ F.$, and absorbing heat at a rate of $q = 20$ Btu/hr.

Find the temperature of the pipe surface in contact with the soil after one month (731 hours of operation).

Solution:

$$\frac{r}{2\sqrt{a\theta}} = \frac{.0417}{2\sqrt{.0261 \times 371}} = .0048$$

$$\int_x^\infty \frac{e^{-\beta^2}}{\beta}\, d\beta \qquad \text{from table 1 is } 5.06$$

Using the Kelvin equation:

$$T - T_o = -2.66 \times 5.06 = 13.4^\circ F.$$

$$\text{therefore: } T = T_o - 13.4 = 50 - 13.4 = 36.6^\circ F.$$

from table 2 for one Btu per hour per linear foot

$$T - T_o = -.67$$

therefore for 20 Btu per hour per linear foot $= -13.4^\circ F.$

therefore $T = 50^\circ F. - 13.4^\circ F. = 36.6^\circ F.$ (the same result).

For two vertical pipes of 1" outside diameter with intervals of 30 inches, the same conditions as before hold true:

$$T - T_o = -20 \times (.67 + \frac{.67}{4}) = -16.8$$

T = temperature at the pipe surface = 50-16.8 = 33.2°F.

which shows that each pipe draws heat, to some extent, away from the soil surrounding the neighboring pipe.

In the case of buried horizontal pipe, the Kelvin equation can be used with the assumption that there is a negative image of the pipe, i.e. a source of heat at the same distance above the surface.

The temperature drop depends upon the period operation as the amount of heat absorbed varies for each month's needs.

Example: Assume an isolated vertical pipe having an outside diameter of 1 inch and operating at an average absorption of 10 Btu/hr/ft during the month of November and 20 Btu/hr/ft during December. What temperature might be expected in the pipe on January 1st under the same conditions? From Table 2, we find that:-

$$T-T_o = (-10)\ ((T-T_o)\text{ two months} -(T-T_o)\text{one month})+(-20)((T-T_o)\text{1 month})$$

$$= (-10)(.72-.67) + (-20)(.67) = -13.9°F.$$

$$T = 50 - 13.9 = 36.1°F.$$

Ingersoll and Plass made a study of the temperature in ½ inch pipe buried vertically in soil #1 and soil #2 for a period of 100 years. It was interesting to note that after 10 years, the decline in temperature became very gradual. They also found that, in certain types of deep dry wells, the summer return of heat may be much less than the winter heat absorption.

Deep vertical wells, even if dry, have certain obvious advantages. A study of the possible effect of a progressive local cooling of the ground near the vertical pipes should be carefully made even though it may not be a serious problem after several years of operation.

APPENDIX III--Experiment Problems

Sand was chosen as the soil in the experiment for the heat pump. The model and the heat pump parts were arranged to draw heat at 18.8 Btu/hr per unit area of the pipe. The temperature of the sand was 70°F. with the following specifications:

For dry sand: density $\rho = 94.8$ lb/cubic ft.

Specific heat $C_p = .19$ Btu/lb.

Conductivity $K = .19$ Btu/hr ft^2 $\left(\frac{°F}{Ft}\right)$.

Diffusivity $a = \frac{K}{\rho C_p} = .01056$

For wet sand: Density $\rho = 102.0$

Specific heat $C_p = .5$

Conductivity $K = .65$

Diffusivity $a = .01275$

The Freon 114 (refrigerant for heat transfer) evaporated and condensed under normal pressure at temperature $T = 40°F$.

The brass pipe had an outside diameter = .528 inches = .022 feet radius.

The initial temperature of all the machine prior to operation was $T_o = 70°F$.

Using the Kelvin equation:

$$T - T_o = \frac{Q}{2\pi K} \int_x^{\alpha} \frac{e^{-\beta^2}}{\beta} d\beta$$

$$= \frac{Q}{2\pi K} I(x)$$

where

$$X = \frac{r}{2\sqrt{a\theta}}$$

$$I(x) = \int_x^{\alpha} \frac{e^{-\beta^2}}{\beta} d\beta$$

With this data and calculating for the time required for the [illegible] of the outside surface of the brine pipe to reach 40°F

$a = (.?)^2$ hrs 15 minutes. we find that:-

[illegible]

1 at 8.80 Btu/hr...

$$\int_x^\alpha \frac{e^{-\beta^2}}{\beta}\,d\beta = \frac{-30}{-8.8} \times 2\pi \times .19 = 4.05$$

for $I(x) = 4.05$

$$X = .019$$

$$\theta = \left(\frac{.022}{2 \times .019 \times .1025}\right)^2 = \text{[illegible] hours.}$$

(in Chapter III this was found to be 25.6 hours)

2 at 10 Btu/hr driven by the pump

$$\int_x^\alpha \frac{e^{-\beta^2}}{\beta}\,d\beta = \frac{T - T_0}{\frac{-10}{2\pi \times .19}} = \frac{40-70}{-10} \times 2\pi \times .19$$

$$= 3.56$$

From table I (appendix

for $I(\ ) = 3.56$

$$X = \frac{r}{2\sqrt{a\theta}} = .021$$

$$.021 = \frac{.022}{2\sqrt{.01056\theta}}$$

$$\theta = \left(\frac{.022}{2 \times .021 \times \sqrt{.01056}}\right)^2$$

$$= 26.2 \text{ hours}$$

3 at 13.4 Btu/hr extracted by the heat pump.

$$\int_x^\alpha \frac{e^{-\beta^2}}{\beta}\,d\beta = \frac{T - T_0}{\frac{-13.4}{2\pi \times .19}} = \frac{-30}{-13.4} \times 2\pi \times .19$$

$$= 2.67$$

for $I(x) = 2.67$ from Table I (Appendix III)

$$x = .05167$$

$$.05167 = \frac{.022}{2 \times \sqrt{\theta} \times \sqrt{.01056}}$$

$$\theta = \left(\frac{.022}{2 \times .05167 \times \sqrt{.01056}}\right)^2$$

$$= 4.30 \text{ hours.}$$

For wet soil:

at 8.95 Btu/hr

$$I(x) = \int_x^\alpha \frac{e^{-\beta^2}}{\beta}\,d\beta = \frac{-30}{-8.95} \times 2\pi \times .65 = 13.75$$

(Larger than the values given in the tables.)

[illegible]

$$I(x) = \int_x^\alpha \frac{e^{-\beta^2}}{\beta}\,d\beta = \frac{40-70}{-10} \times 2\pi \times .65$$

$= $ [illegible] (Larger than values in table 1)

[illegible]

$$\int_x^\alpha \frac{e^{-\beta^2}}{\beta}\,d\beta = \frac{48-70}{-10} \times 2\pi \times .65$$

$= $ [illegible]17

$$\frac{r}{2\sqrt{a\theta}} = .00[illegible]1$$

$$\theta = \left(\frac{.022}{2\times .113\times .0001}\right)^2 = $$ [illegible] x 10^4 hours.

[illegible]

$$\int_x^\alpha \frac{e^{-\beta^2}}{\beta}\,d\beta = \frac{\frac{T-T_0}{-13.4}}{2\pi\times .65} = \frac{48-70}{-13.4} \times 2\pi \times .65$$

$= $ 6.7[illegible]

$$\frac{r}{2\sqrt{a\theta}} = .[illegible]$$

$$\theta = \left(\frac{.022}{2\times .113\times .0009}\right)^2 = $$ 11750 hours, or ([illegible] weeks).

at [illegible] Btu/hr:

$$\int_x^\alpha \frac{e^{-\beta^2}}{\beta}\,d\beta = 4.700$$

$$\frac{r}{2\sqrt{a\theta}} = .0[illegible]7$$

$$\theta = \left(\frac{.022}{2\times .113\times .007}\right)^2 = $$ 192 hours.

APPENDIX III- Using water instead of Freon 114 in the heat pump system.

The experiment explained in Chapter III was repeated using water in place of Freon 114. Some changes in the set was necessary to qualify with the new experiment. The grid pipe was designed in Figure 26 (page 98). Water was driven at 32°F. to absort heat from the surrounding sand (dry and wet) and the heat gained by the water through the cycle was equal to the heat lost by the sand.

Repeating the experiment with hot water at 128°F and equalizing the heat gained by the sand (wet or dry) with the heat lost from the hot water and with the water flow at 98.425 lb. per hour, the results of the experiment had four noticeable functions:-

1-The temperature variation along the radius changed by small degrees near the tank wall while a larger drop or raise in the temperature was noticeable nearer the vertical pipe.

2-The amount of heat flow in various sections of the vertical pipe depended greatly upon the amount of exposed area.

3-The coefficient of conductance varied in the case of hot water and showed the largest temperature drop between the pipe and the nearby sand.

4-The moisture effect decreased the film conductance coefficient between the pipe and the nearby sand resulting in an increase of the temperature difference.

APPENDIX III--Preliminary experiments to determine factors included in the research.

A set was fixed to measure thermal conductivity of sand for different characteristics and to calculate from the results the thermal heat conductance between the sand and the surface.

Comparing characteristics of sand with well known material (cork) and equalizing the heat flow from a heater to cork, sand, and to cooling water, we find that:-

$$q = KA \frac{\Delta t}{\Delta x}$$

A-fine (medium) dry sand

1-1" sand thickness

$$\frac{.0232 \times 1 \times 165}{1/12} = \frac{66.8}{2/h + 1/12K}$$

$$\frac{2}{h} + \frac{1}{12K} = 1.453$$

2- ½" sand thick plate

$$\frac{.0232 \times 1 \times 163.6}{1/12} = \frac{57}{2/h + 1/24K}$$

$$\frac{2}{h} + \frac{1}{24K} = 1.256$$

from 1 and 2:

$$\frac{1}{24K} = .197$$

$$K = \frac{1}{24 \times .197} = .212 \quad \frac{Btu}{hr.\,{}^{\circ}F.\,ft^2/ft.}$$

and

$$\frac{2}{h} + \frac{1}{2.55} = 1.453$$

$$h = \frac{2}{1.453 - .392} = 1.88$$

B-Coarse dry sand:

$$\frac{2}{h} + \frac{1}{12K} = \frac{59.8}{.0232 \times 188 \times 12} = 1.15$$

$$\frac{2}{h} + \frac{1}{24K} = \frac{50}{153 \times .0232 \times 12} = .98$$

$$\frac{1}{24K} = .17$$

$$K = \frac{1}{24 \times .17} = .246$$

$$h = 2.46$$

C- Wet fine sand:

1- 1" sand thickness

$$q = KA \frac{\Delta t}{\Delta x}$$

$$.0232 \times 1 \times \frac{174}{1/12} = \frac{72.8}{2/h + 1/12K}$$

$$2/h + 1/12K = 1.51$$

2- $$.0232 \times 1 \times \frac{175}{1/12} = \frac{67}{2/h + 1/24K}$$

$$2/h + 1/24K = 1.38$$

$$1/24K = .13$$

$$K = .321$$

$$h = \frac{2}{1.51 - .26} = 1.6$$

<u>Results of the experiments:</u>

1-The thermal conductivity of the sand varies as to its characteristics, being highter for coarse sand than for fine sand.

2-The presence of large size quartz increases the conductivity coefficient while minerals such as plagioclase, felspar and pyrexene decrease the conductivity.

3-The specific heat values of the sand decreases with the decrease in temperature, the average value at zero °F. being .16 Btu/hr. and .19 Btu/hr at 140°F.

4-The conductivity of dry sand is less than that for wet sand.

5-Surface thermal conductance coefficients for dry sand were higher than for wet sand (assuming the other factors to be equal), as it appeared in the preliminary experiments.

6-The possibility of using coarse sand to increase pore spaces for good thermal conductivity has been considered and proved good results.

ASHVE Guide, Vol. 29-1951, pp. 181-183
Verschoor, J. D. & Paul Greebler, Heat transfer by gas conduction and radiation in fibrous insulations.(Manville Research Central)
ASME Paper No. 51-A-54

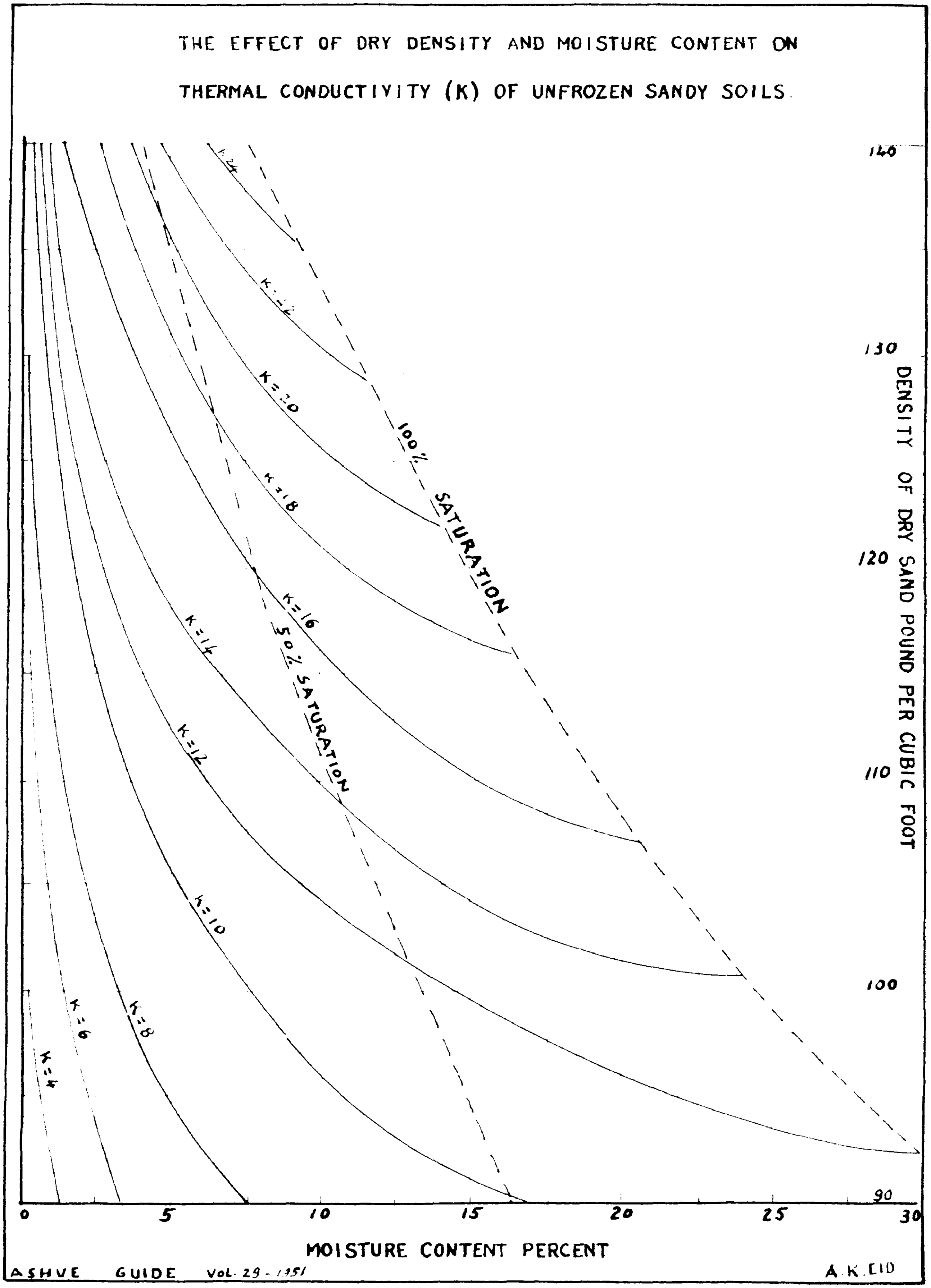

THE EFFECT OF DRY DENSITY AND MOISTURE CONTENT ON
THERMAL CONDUCTIVITY (K) OF UNFROZEN SANDY SOILS.
K=24
K=22
K=20
K=18
K=16
K=14
K=12
K=10
K=8
K=6
K=4
100% SATURATION
50% SATURATION
140
130
120
110
100
90
DENSITY OF DRY SAND POUND PER CUBIC FOOT
0
5
10
15
20
25
30
MOISTURE CONTENT PERCENT
ASHVE GUIDE VOL. 29 - 1951
A.K.EID

Figure No. 35

Conclusions: 1-It takes longer for the temperature to drop from 70°F. to 40°F. in wet sand than it does in dry sand, because of the higher heat capacity of the wet sand as compared to that of the dry sand.

2-Themore heat that is absorbed by the ground pipe, the less time is needed for the temperature to drop, provided all other factors remain constant throughout operation of the heat pump.