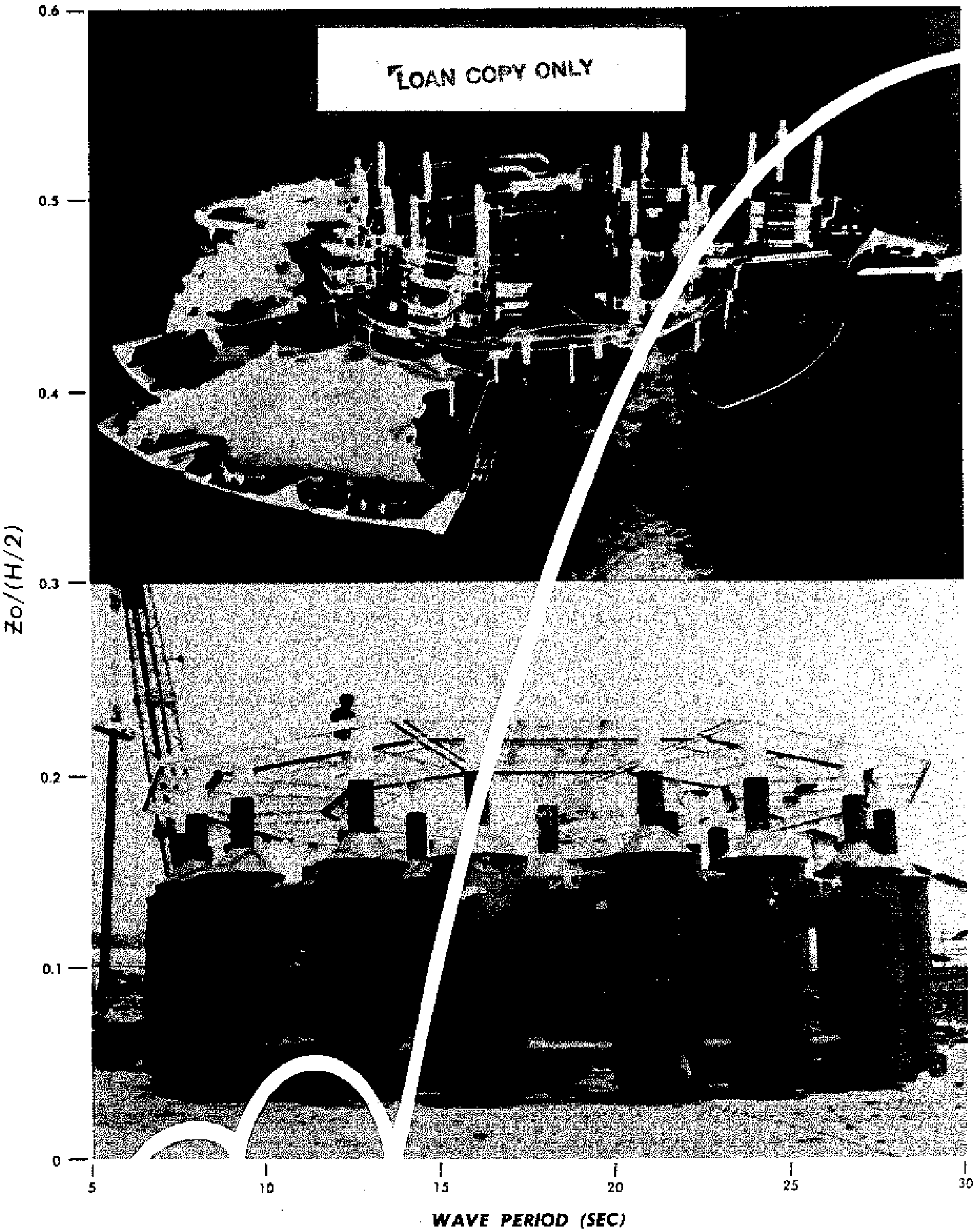
HAWAII'S FLOATING CITY

DEVELOPMENT PROGRAM

INTERNAL THERMAL AND HUMIDITY CONTROL



HAWAII'S FLOATING CITY DEVELOPMENT PROGRAM

Technical Report No. 3

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Internal Thermal and Humidity Control

by

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for

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ABSTRACT

This report is an investigation of some design aspects of a large floating platform to support urban activities in the Hawaiian Islands area. A brief account of weather and sea conditions in Hawaii is presented, with an analysis of human comfort requirements. Various types of air conditioning systems are described and choices are made for specific locations within the Floating City. Equations are presented and sample calculations are performed for determining cooling load, by the response factor method, for a superstructure and a buoyancy tank case. Preliminary air duct design is performed for both cases. Absorption-cycle refrigeration is discussed, and a flow diagram and calculations are given. A computer program which performs the response factor calculations is presented. Finally, several conclusions drawn from this work are reported.

AN INTRODUCTORY NOTE

During the time the research documented in this report was being conducted, Mr. Yamashita was pursuing his Master of Science degree at the University of Hawaii. His thesis, "A Calculation of Cooling Loads with Response Factors," developed the technique employed in this work, and this is its first professional application. Mr. Yamashita has been awarded his M.S. degree and is now a doctoral candidate. I believe the reader will agree that this is a thoroughly professional product and will understand the pleasure we take in Mr. Yamashita's association with the project.

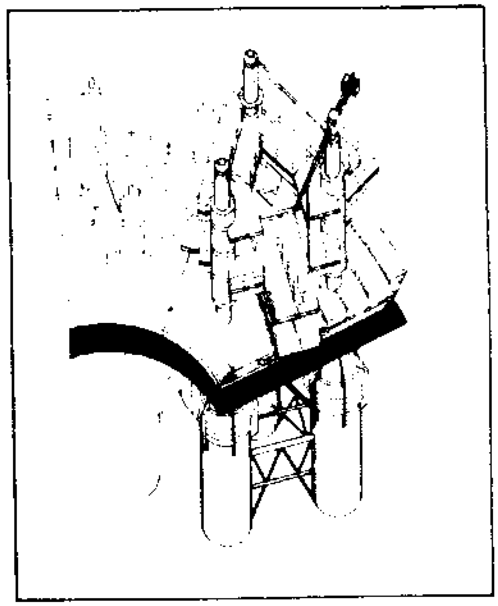
The following Oceanic Foundation personnel contributed to the study: Mr. Guy N. Rothwell provided the bulk of the technical guidance for this work; Ms. Bonnie M. Rhodes was responsible for editing the manuscript; Ms. Diane J. Henderson prepared the illustrations and final manuscript; Ms. Joyce Miller assisted with the early writing and typing.

Joe A. Hanson
Program Manager

TABLE OF CONTENTS

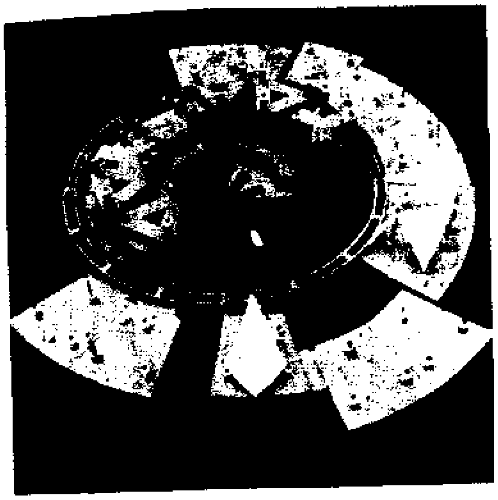
INTRODUCTION

Hawaii's Floating City Project had its formal inception in 1970, with the award of a grant by the State of Hawaii to the Department of Architecture, University of Hawaii, for a project to investigate the possibilities and opportunities that might lie in the construction of an urban center aboard a very large, deep-sea, floating platform. This work resulted in the conceptual design of a ring-shaped floating city with inner and outer rings consisting of up to thirty independently stable modules, all rigidly connected to form the final city, but separately removable for repair or renovation. The typical module would be supported from three or four vertically oriented, fully submerged buoyancy chambers, each topped by a structural column which would pass through the elevated main deck structure to form the core of a moderately high-rise superstructure building.



Typical module

The city was envisaged as housing a broad range of domestic, commercial, recreational, industrial and public activities, all served and supported by a complete suite of on-board public utilities, and linked with Honolulu and the world by a variety of transportation and communications services. In general, the industrial, utility and commercial services would utilize available space in the submerged buoyancy chambers. At the lowest levels would be found fuel and water storage, as well as sewage treatment facilities and variable ballast tanks. Above these would be power generation, air conditioning and desalination plants, as well as other machinery spaces. Also included would be warehousing and cold storage, maintenance and repair shops, and some commercial spaces, such as stores and offices. At the top of the buoyancy chambers, where windows on the sea could be provided, would be restaurants, apartments and places of public assembly.



Superstructure model

The main deck structure, located above the highest waves, would contain commercial and recreational facilities and transportation terminals. The weather surface of the main deck structure would be landscaped as a park. Finally, the superstructure would contain mainly living spaces, either apartments or hotel rooms.

Principal structural material was to be concrete. The city site was chosen as five miles south of Honolulu in the open ocean, where water depth is 600 meters.

In subsequent years, engineering studies of various aspects of this concept have been made, including environmental conditions of the site, hydrostatics and hydrodynamics of the city's form and feasibility of the modular approach. Studies for the current year include an analysis of concrete as a suitable material, a structural design investigation, a review of applicable transportation methods, a survey of potential construction sites and methods, and the study contained in this report.

Objectives, Scope, and Limits of the Present Study

The purpose of this study is to examine the requirements for internal environmental control, given the on-site environmental conditions and the configuration and projected occupancies of the city's enclosed spaces. Emphasis is placed on weather, air conditioning, duct design, and utilization of waste heat from the city's power plant. The report also develops a mathematical model for determining (and optimizing) the values of the various sources of heat gain, and suggests a total-energy approach to the design of the city's power plant, air conditioning plant, and freshwater distillation plant.

In view of the highly conceptual and schematic nature of the architectural design upon which this report (and others in this year's effort) are based, it has not been possible to make a detailed, realistic and quantitative inventory of occupancies, with their associated heat loads.

Neither has it been possible to provide more than a schematic representation of the thermal properties of the exterior covering materials of the main deck and superstructure. Therefore, reasonable assumptions have been taken in both categories. Thus, though the results of the computer runs appear reasonable to the extent that certain important conclusions can be drawn from them, the numerical values obtained are not meant to apply for any specific floating platform that may in the future be built. This report does, however, claim to present a convenient method by which such values may be calculated when they are required.

Contents of the Report

Section I describes the analysis of weather conditions in Hawaii based on statistical data. Criteria for human confort are also established for Hawaiian weather conditions.

Section II describes several types of air-conditioning systems and their applications, according to the economic and thermal requirements of various classes of interior space.

Section III is a theoretical investigation of heat gains for the superstructure and buoyancy tanks in the Floating City. Section IV outlines the proposed procedure for calculation of cooling load in relation to Section III.

Section V contains sample calculations of cooling load for a superstructure element and for a buoyancy tank. The major effort has been put on the computer simulations for the analysis of cooling load in the superstructure.

Section VI describes the utilization of waste heat from the city's power plant. An absorption cycle using waste steam is suggested for refrigeration.

Section VII contains the design of ducts for the superstructure and buoyancy tanks, yielding the physical dimensions of ducting in relation to other space requirements.

The summary and conclusions are presented in Section VIII.

I. CLIMATE AND COMFORT

A. Weather Data

Hawaii's thermal and climatic conditions are well characterized by its exposure to the prevailing trade winds and its mid-oceanic location. The influence of the surrounding waters upon climate is considerable. In general, the trade winds flow in an east to west direction. Owing to the trades, showers are very common and cloudless skies are rare. The trade winds provide natural ventilation for the Hawaiian islands, bringing in mildly warm air that has moved great distances across subtropical seas. During the winter months, the persistent North Pacific cell of high barometric pressure tends to be displaced and occasionally broken up, allowing frontal storms to reach the islands. Accompanying these storms are sharp changes in wind direction, increased wind velocity and moderate to heavy precipitation. However, air temperature changes are not drastic, being moderated by the great temperature stability of the surrounding ocean surface.

There is relatively slight variation in the length of the daylight period in Hawaii in contrast with other areas, as shown in Table 1.

Table 1 - Comparative daylight periods (1)

City	North latitude	Longest Day		Shortest Day	
		hr	min	hr	min
Anchorage	61	24	0	7	30
Seattle	48	17	20	9	30
St. Louis	39	16	0	10	20
Atlanta	34	15	30	10	50
Honolulu	21	14	10	11	40

An outstanding feature of the climate of Hawaii is the small annual temperature variation (Figure 1). In downtown Honolulu, the warmest

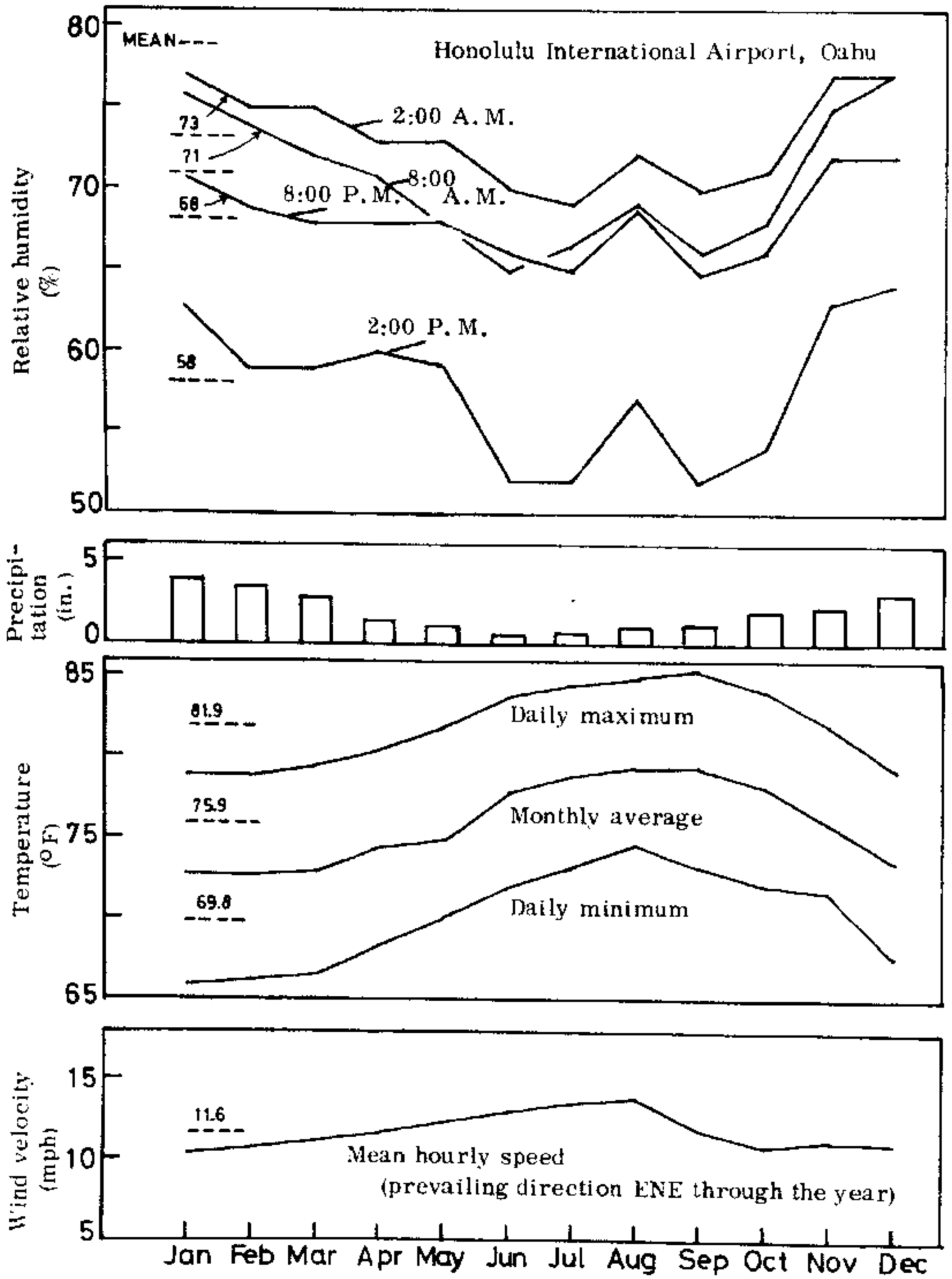


Figure 1 - Monthly average weather data in Honolulu (2).

month is August, with an average daily temperature of 78.4° F, which is only 6.5° F higher than that for the coldest month of February.

Comparison of mean temperature ranges among the different cities in the Hawaiian islands is presented in Table 2. It shows fairly similar temperature variations except for the data taken at Mauna Loa.

Table 2 - Mean temperature ranges in Hawaii (2)

Station	Elevation (ft)	Mean temperatures (°F) January	August
Hilo	40	71	76
Olaa	280	70	75
Mountain View	1,530	65	70
Mauna Loa	11,150	41	47
Honolulu Airport	7	73	78

August and September are the warmest months of the year throughout Hawaii. Figure 1 shows the monthly average temperatures at Honolulu Airport, which were taken for ten years from 1951 to 1960. In the same figure, the mean hourly speed of wind is plotted and the average of 11.6 mph is shown. Wind direction frequency is given in Table 3, as well as wind speed frequency, for the months of January and August.

Table 3 - Wind direction and speed in Honolulu (1, 2,)

Wind Direction Frequency January	August	Direction	Wind Speed Frequency Speed	January	August
50%	93%	NNE to E	0-12 mph	68%	38%
			13-24 mph	29%	58%

The average relative humidity in the coastal areas and along the mountain ranges is 70 to 80 percent, while that for the leeward areas is 60 to 90 percent. Figure 1 shows the fairly constant relative humidity in Honolulu, which is largely due to the influence of the surrounding ocean.

The Hawaiian Sugar Planters Association has measured the total solar radiation, which is the sum of direct and diffuse solar radiation. Data from 16 years of measurements are plotted in Figure 2 in conjunction with the average temperature of outside air taken at the Honolulu Airport.

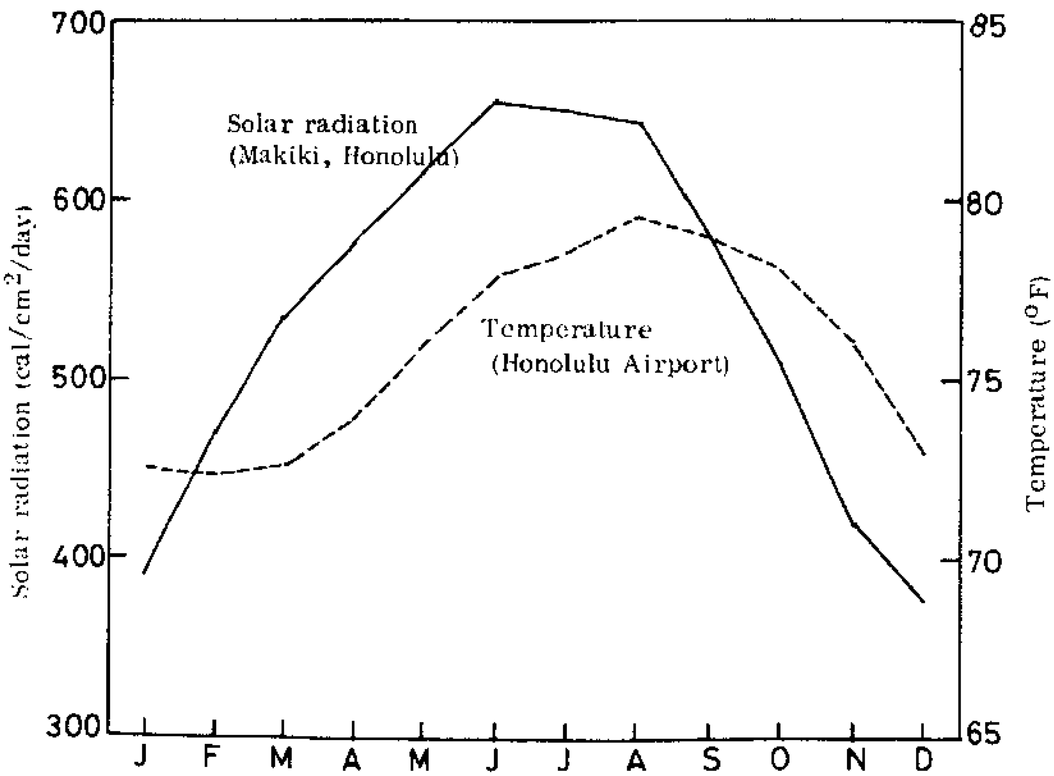


Figure 2 - Monthly averages of daily accumulation of solar radiation and outdoor temperature (3).

B. Human Response to Climatic Conditions

The major climatic elements that affect human comfort are air temperature, radiation, air movement and humidity. Minor elements are the chemical composition of air, its physical impurities, and its electrical properties. In general, ordinary temperature readings are not adequate to describe thermal environment. Air movement, humidity and radiation are all relevant as external thermal stimuli of the human body. They all affect the rate of heat loss of the body.

The most commonly used climatic scale is effective temperature (E. T.) introduced by Houghton and Yaglou in 1923 and again by Yaglou and Miller in 1925. Their index covers temperature, humidity and air movement without radiation. There have been numerous proposals on comfort conditions in many countries which have their own respective climatological and racial conditions. For a normally clothed American man at rest during the summer, Yaglou and Drinkers found that the comfort range is 66 to 75^{o} F (E. T.) with 71^{o} F(E. T.) as the optimum point.

Figure 3 shows the comfort zone for inhabitants in the U.S. temperate zone (40 deg N. lat.) and the estimated comfort zone for Hawaiian inhabitants by Olgray's method (4). He suggested that when the bioclimatic chart is applied to climatic regions other than 40 deg north latitude, the lower perimeter of the summer comfort line should be elevated about 3/4^{o} F for every 5 deg north latitude change toward the lower latitude. The upper perimeter may be raised proportionally, but not above 85^{o} F. In the same figure, temperature and humidity are plotted to indicate monthly variations.

Sugiyama (3) proposed the comfort curve that is applicable to Hawaii inhabitants (Figure 4). It shows the percentage of subjects who feel comfortable under the specific warmth index.

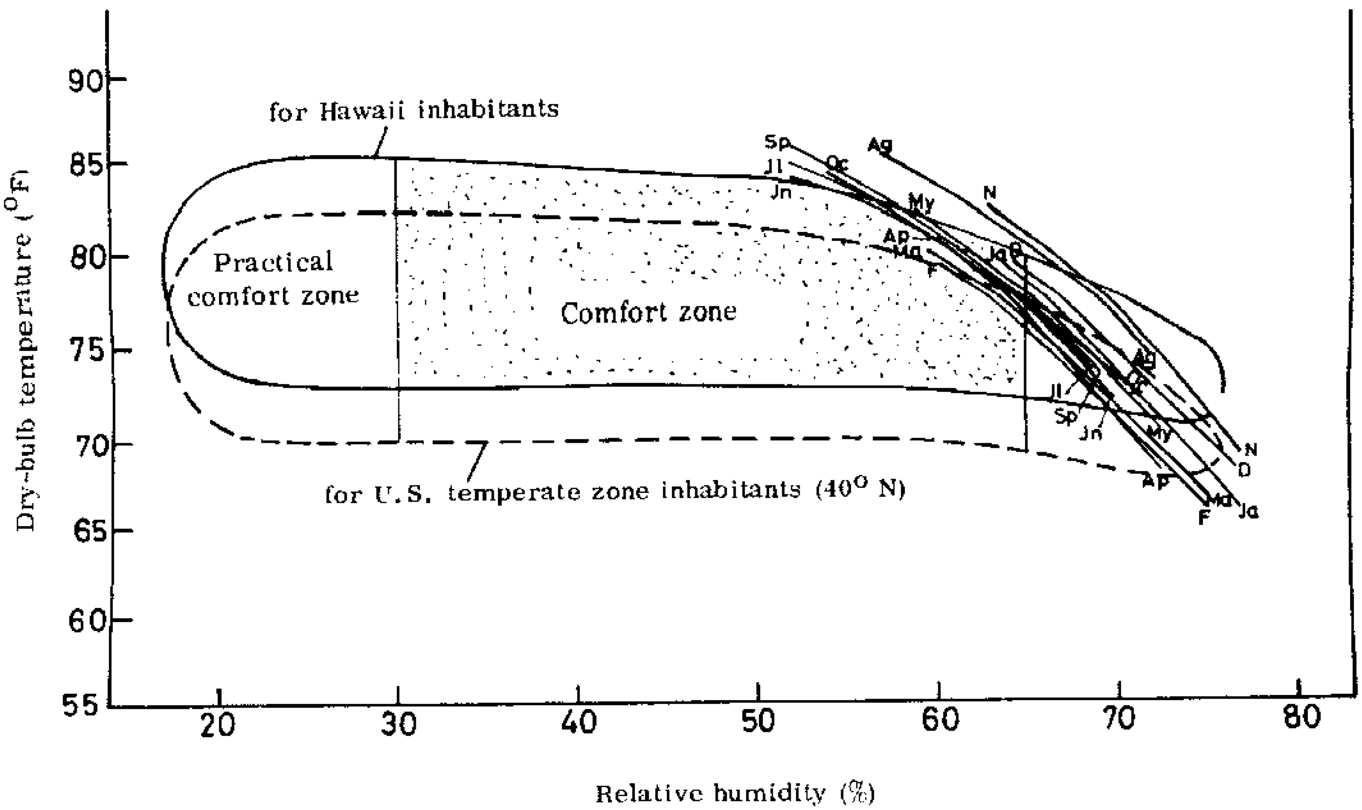


Figure 3 - Bioclimatic chart for Hawaii inhabitants and U.S. temperate zone inhabitants (4).

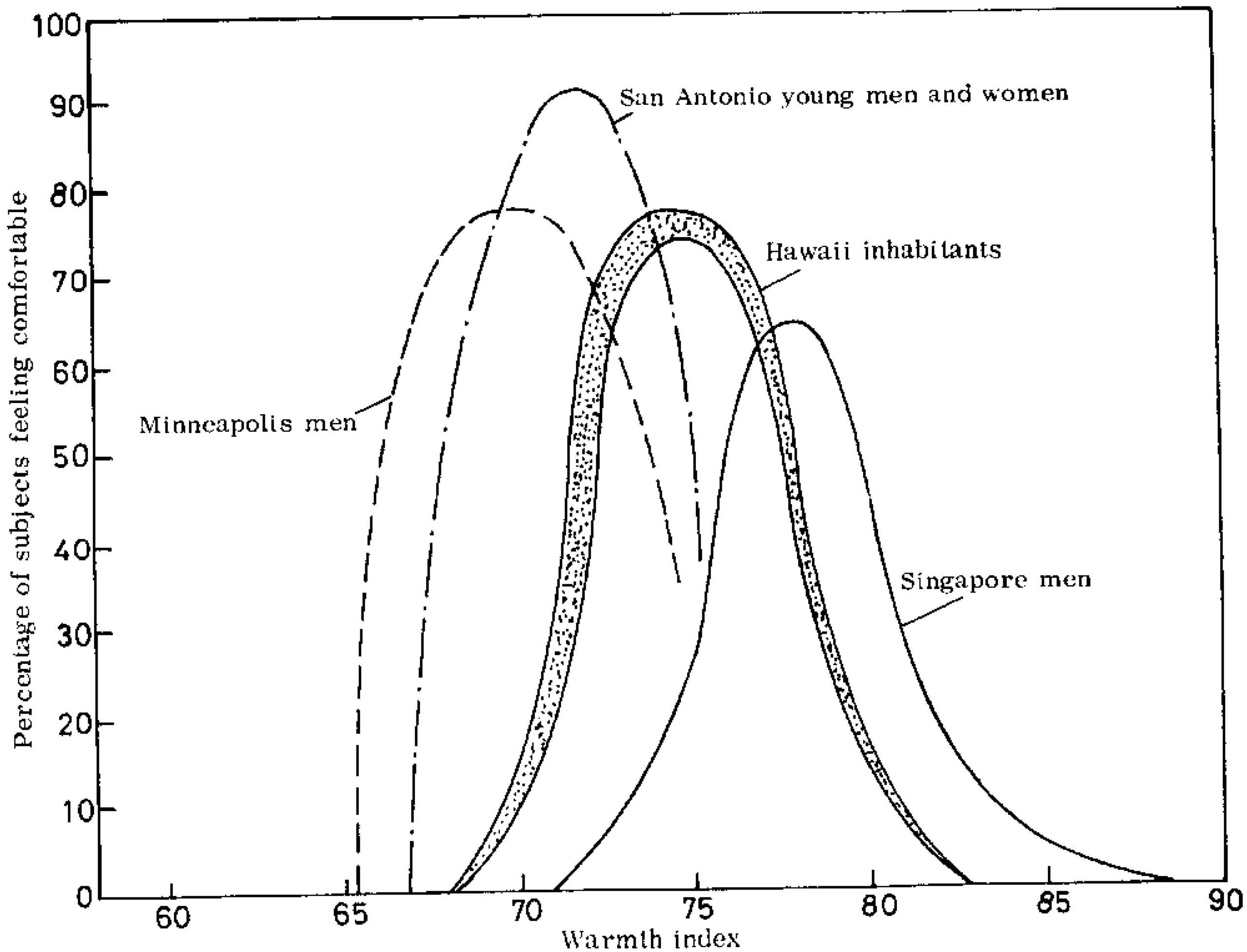


Figure 4 - Comfort graphs for different latitudes on a common scale of warmth.
Note: 1. Original data by C.G. Webb & Dept of Architecture, U. of Hawaii
2. Hawaii curve is assumption (3)

C. Sea Conditions

Vertical temperature distribution of seawater has not been measured around the Floating City site. However, measurement taken by Wyrtki (5,6), Patzert (7), et al., may provide the general characteristics of vertical temperature distribution in Hawaiian seawater. These are plotted in Figure 5. It is seen that the characteristics of vertical temperature distribution are fairly similar at different places in the Hawaiian area. This may permit us to assume that a modified and generalized curve for vertical temperature distribution in Hawaiian seawater is possible. Therefore, vertical temperature distribution of seawater at the Floating City site is represented by the previously assumed curve and is given in Figure 5.

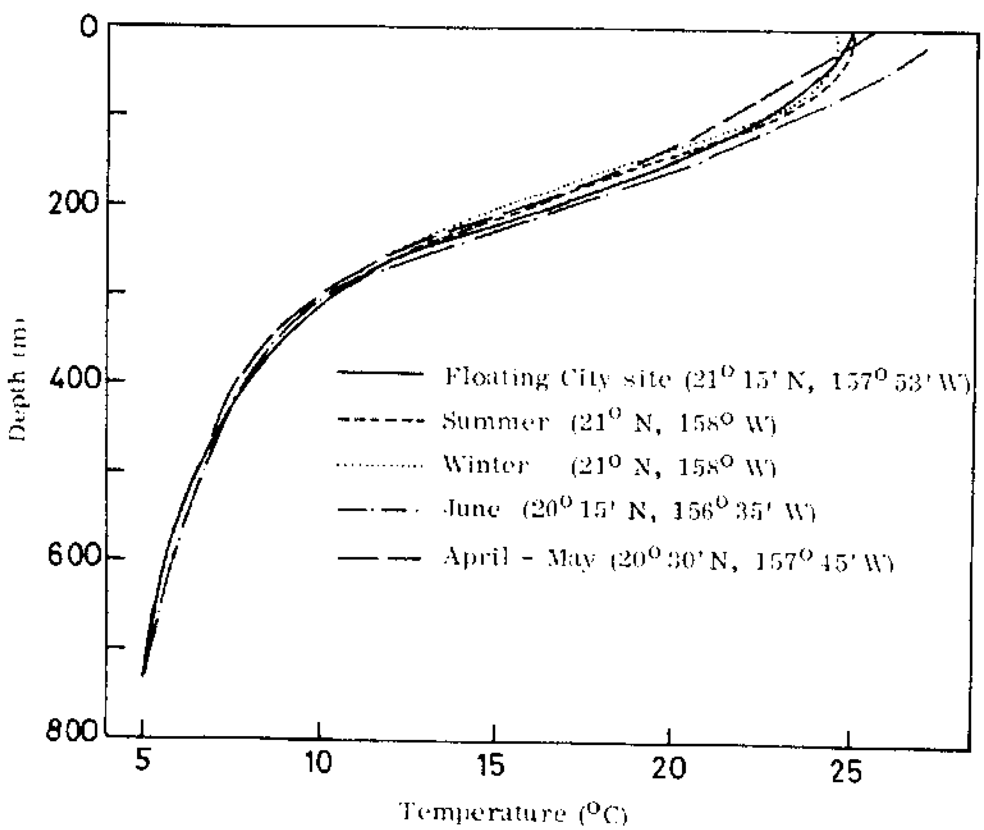


Figure 5 - Thermocline for Floating City site (5,6,7).

Surface temperature of Hawaiian seawater varies approximately from 25° C to 27° C throughout the year: 25° C to 26° C in winter, 26° C to 27° C in summer. In 1965, Wyrtki reported that the yearly temperature variation of the surface water is about 1.5° C.

Surface salinity of Hawaiian seawater ranges approximately from 34.5 °/oo to 35.0 °/oo (Figure 6). Salinity changes very slightly with depth; minimum salinity of 34.2 °/oo is obtained at a depth of about 500 to 600 meters in Hawaiian seawater.

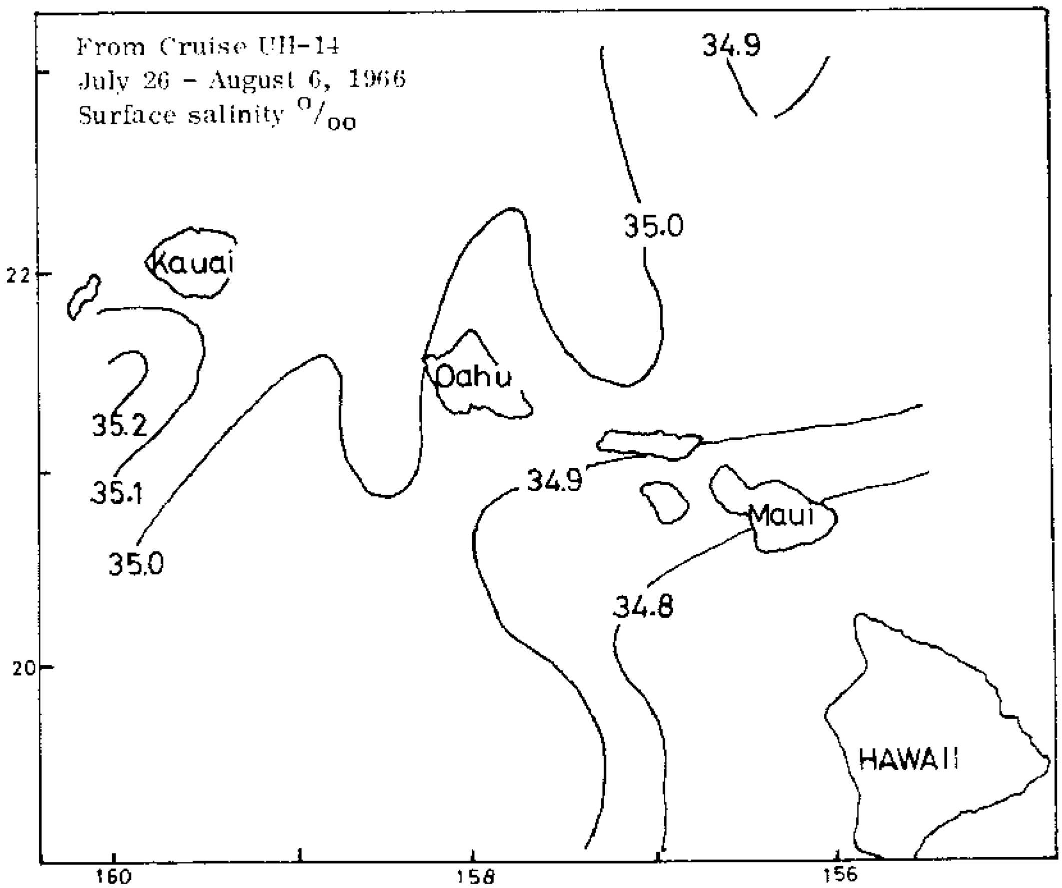


Figure 6 - Surface salinity of Hawaiian seawater (summer) (5).

According to the data taken by During in 1969 (8) at a depth of 25 meters in Hawaiian seawater, current velocity varies from 5 cm/sec to 50 cm/sec (Figure 7). Twenty cm/sec of current velocity is assumed to be an average for most of the time at a depth of 25 meters. However, further measurements relating temperature, depth and current velocity at the Floating City site are needed.

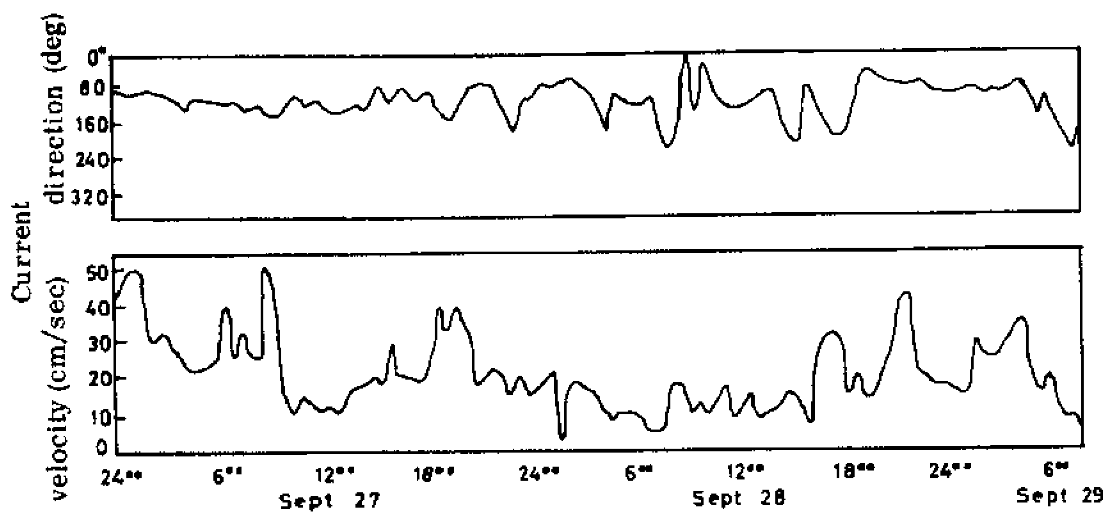


Figure 7 - Typical current velocity for Hawaiian seawater at a depth of 25 m (1969) (8).

II. AIR CONDITIONING SYSTEMS

The bioclimatic chart for Hawaii residents shown in the preceding section (Figure 3) suggests that air conditioning should rarely be needed for comfort reasons. However, the climatic data in the figure represent ambient conditions. In so compactly organized a structure as the Floating City, these ambient conditions could not be maintained throughout its interior without air conditioning unless the city could be designed so as to allow the passage of outside air through each compartment essentially unimpeded and unchanged, which is manifestly impossible. These data, therefore, establish a basis for the amount and kind of air conditioning the interior spaces will need to maintain them within comfort limits.

The American Society of Heating and Ventilating Engineers in 1929 defined comfort air conditioning as "the process of treating air so as to control simultaneously its temperature, humidity, cleanliness and distribution". The same definition applies today, with certain refinements. Comfort air conditioning, or climate control, means the maintenance of those atmospheric factors affecting human comfort. Specifically, it is the maintenance of the following variables:

- o Desired temperature
- o Acceptable humidity
- o Minimum number of atmospheric particles, including pollens and bacteria
- o Uniform air pattern and proper air motion.

A. Preliminary Planning

Preliminary design of an air-conditioning system requires an awareness of the esthetic, structural, and economic factors in the overall design from its very early stages. The machine room and air ducts require considerable floor area and space, which must be provided for during preliminary design.

In most respects, the air-conditioning requirements for a superstructure element on the Floating City will not differ from those of a high-rise office building in a similar climate. Therefore the following procedures used in air-conditioning system design for an office building are representative.

1. Selection of the air-conditioning system

Considerations of the equipment cost, operation cost, required space for the air-handling unit, structures, and zones within the building all contribute to the determination of a reasonable air-conditioning system. However, the final decision may rest with the designer, based on his experience and preference.

2. Placement of the machine room and air ducts in each zone

Once the air-conditioning system has been selected, a rough estimation of the required supply air is needed to determine the placement of machine rooms and ducts of appropriate sizes.

3. Determination of capacities required for heat source equipment

The capacities of the boiler and refrigerator can be determined by heat load estimations, so that the physical dimensions of the machine room may be designed accordingly.

The general procedures above may be further subdivided in order to determine more detailed procedures:

1. Detailed heat load estimations for each room in a building for the determination of peak load.

2. Selection of boilers and refrigerators.

3. Selection of air-handling devices such as cooling and heating coils.

4. Placement of the devices into the machine room for the determination of suitable machine room dimensions.

5. Selection of fans in accordance with the pressure drop in the ducts.

6. Sizing of the pipes in conjunction with the capacity and location of the boiler and refrigerator.

7. Determination of the capacity of the cooling means in accordance with the thermal performance of the boiler and refrigerator.

8. Placing of the apparatus, ducts and pipelines on the drawings.

9. Necessary corrections and/or revisions.

B. Various Systems

Table 4 shows typical characteristics of three types of air-conditioning systems that are applicable for office buildings. These are: 1) all-air-duct systems, 2) air-water systems, and 3) all-water systems. The first two are central station apparatus systems, while the all-water systems are primarily individual room or zone unit systems. Depending on the thermal requirements and function of the building, one may select the most efficient and economical system. Energy costs in Table 4 involve the sum of electric power and fuel for the refrigerator and boiler. The recommended floor area for these systems is 200,000 square feet or more.

Table 4 - Classification of various air conditioning systems

Classification	System	Energy cost	Equipment cost	Power cost	Individual control	Fresh air cooling	Space requirements
All-Air-Duct Systems	Single duct	Low	Medium	High	No	Possible	Large
	Multi-zone unit	High	Medium	High	Possible	Possible	
	High velocity dual duct	High	Medium	Maximum	Possible	Possible	Large
Air-Water Systems	Single duct reheat	Medium	High	High	Possible	Possible	Large
	Floor unit	Low	Medium	High	No	Possible	Maximum
	Induction unit (2-pipe)	Medium	Medium	Medium	Possible	No	Small
	Induction unit (3-pipe)	Low	High	Medium	Possible	No	Small
	Primary air fan-coil unit	Low	Maximum	Low	Possible	No	Small
All-Water Systems	Fan-coil unit	Low	Low	Low	Possible	No	Minimum
	Package	Low	Low	Low	Possible	No	Small

1. All-Air-Duct Systems

In this type of system, the air treatment and refrigeration plants are located some distance away from the conditioned space. Only the cooled or heated air is brought into the conditioned space through ducts. A central station cleans, humidifies, dehumidifies, cools and heats the air. Figure 8 shows a typical all-air-duct system. The space designated as "room" represents the entire space in a building, and the supply and return ducts represent the whole network of ducts.

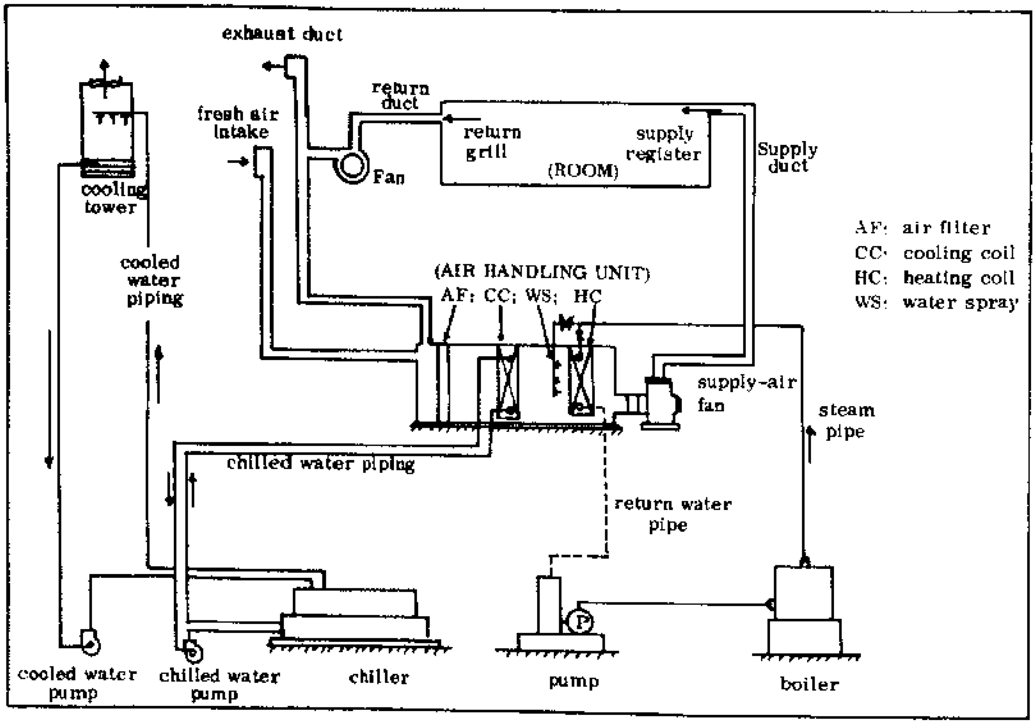


Figure 8 - All-air-duct system.

Single Duct System. This central station system (Figure 9) supplies a single stream of either hot or cold air into the conditioned spaces. Components of this system are mainly the air-handling unit, heat conveyance apparatus, heat sources and automatic control devices.

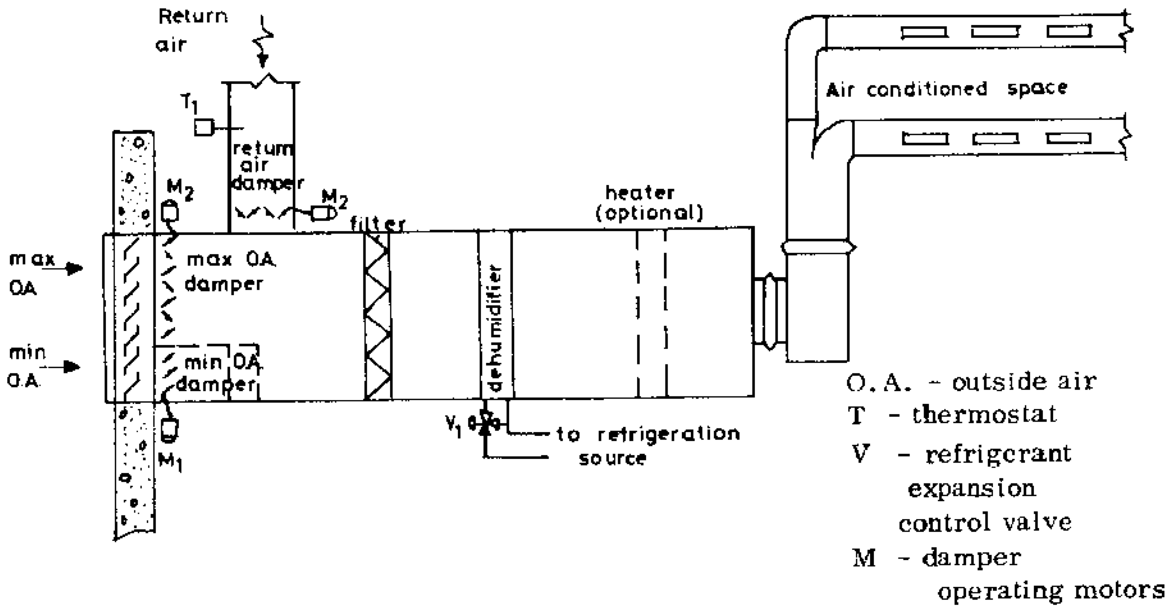


Figure 9 - Single duct system.

Merits of this system are as follows:

- Operation and maintenance can be done very easily, as the air-handling equipment is placed in a machine room. Dust can be eliminated completely from the supply air.

- Supply air quantity is large enough to provide sufficient ventilation in the room.

- Noise and vibration can be controlled easily.

- If the building is divided into a few zones, equipment cost becomes low in comparison with other all-air-duct systems.

Demerits are given as follows:

- o Large duct space is required if there are many zones.
- o Equipment cost is considerably higher than the all-water system.

Multi-Zone Unit System. Heating and cooling coils are placed in an air-handling unit so that the hot and cold air are built up separately by these coils. Mixing dampers are installed in each zone to adjust the quantity of hot and cold air. It operates in response to the thermal requirements of a room. Figure 10 illustrates a typical application of this system.

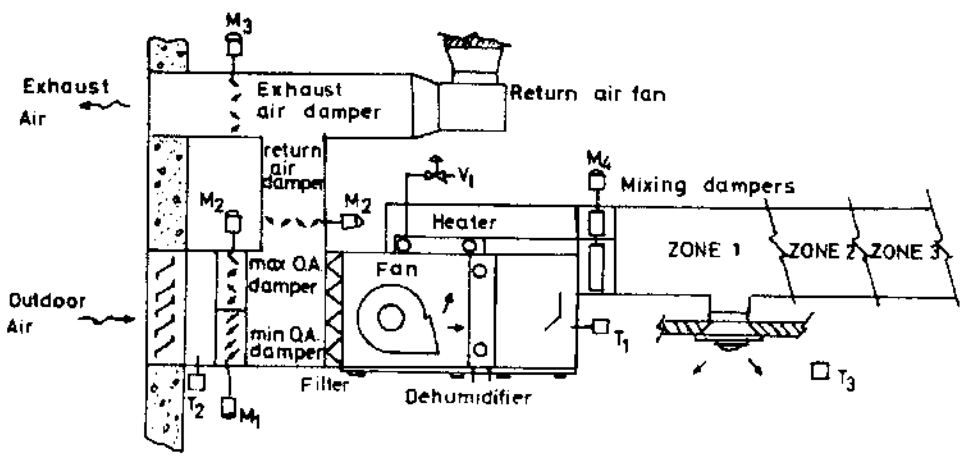


Figure 10 - Multi-zone unit system.

Merits of this system are given as follows:

- o It is useful in buildings that are divided into many different spaces and zones. This system offers excellent control of all spaces in accordance with their thermal requirements.
- o Space can be utilized for diverse purposes, as both the heating and cooling are available separately.
- o When the temperature difference between outside air and room air is large, outdoor air can be used for cooling or heating. This eliminates the operation cost of a boiler or refrigerator.

However, the multi-zone unit system requires large duct spaces if the building is divided into many different zones. In addition, boiler and refrigerator capacity must be large in order to deal with the mixing heat loss.

High Velocity Dual Duct System. This system supplies hot and cold air to each room separately through ducts. Air velocity is approximately 3,000 feet per minute or more in the ducts. Hot and cold air may be mixed in the air blenders which are installed and controlled by thermostats in each room (Figure 11). This system has characteristics similar to those of the multi-zone unit system, for both the hot and cold air are available separately. A small duct size is required for this system in comparison with other all-air-duct systems. This feature is a great advantage in tall buildings as it requires a fairly small space for ducting. Round ducts are commonly used in this system because of small frictional resistance and economy. Besides the excellent characteristics mentioned above, this system has other merits as follows:

- o Thermal response of the room is very quick as the air is a heat-transfer medium. Hence, control of room air is fairly easy.
- o No apparatus is exposed in the room.
- o Maintenance of the unit's air filters is unnecessary.

However, the heat loss due to mixing of hot and cold air is very large, especially in winter.

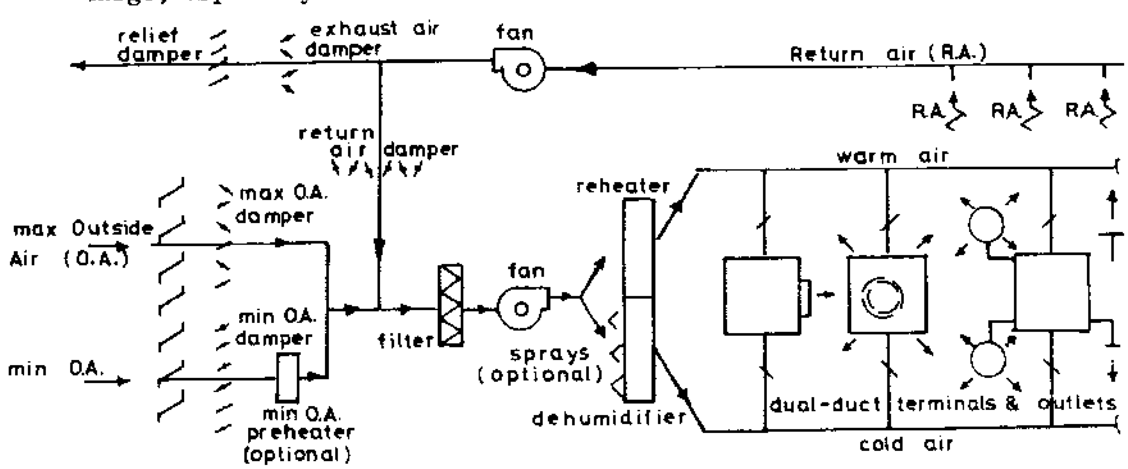


Figure 11 - Systematic diagram of high velocity dual duct system.

2. Air-Water Systems

This type of system has primary and secondary air treatment units. The primary air treatment plant is located some distance away from the conditioned space, as in the all-air-duct system. The function of the primary air treatment plant is identical to that of the all-air-duct system. The secondary air-handling unit is located in the room, ceiling, floor or corridor. Water is used to cool the media and to heat the coils in the air-handling unit. Local adjustment of room air is possible by using the secondary air-handling unit in accordance with the thermal requirements of the room and the condition of primary supply air into the room.

Single Duct Reheat System. This system is primarily the same as the single duct system except that the reheaters are installed in air diffusers or in branch ducts (Figure 12). Control valves for the steam or hot water are connected to the reheaters, and thermostats installed in the room regulate the control valves. Besides the merits described for the single-duct system, this system has another advantage in that it provides convenient zoning within a building.

However, the operation cost of the refrigerator increases in comparison to that for the single duct system due to the reheat load.

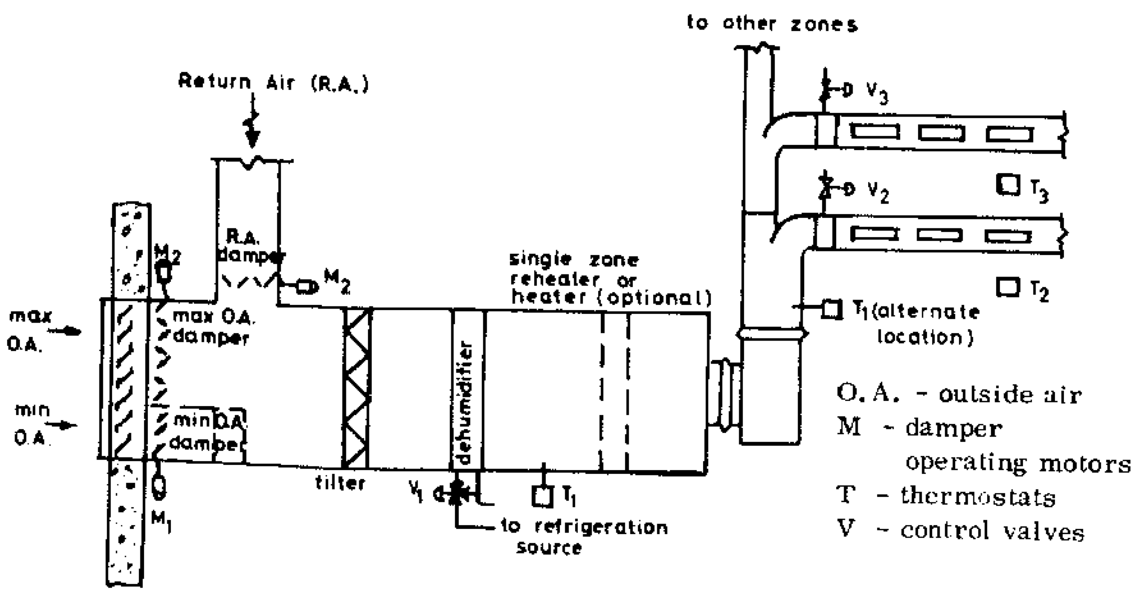


Figure 12 - Single duct reheat system.

Floor Unit System. The air-handling apparatus is placed on each floor in this system as shown in Figure 13. This system has merits such as:

- o Air conditioning can be done separately on each floor.
- o Horizontal duct size can be reduced.

However, the equipment cost is comparatively high and requires large spaces for air-handling equipment.

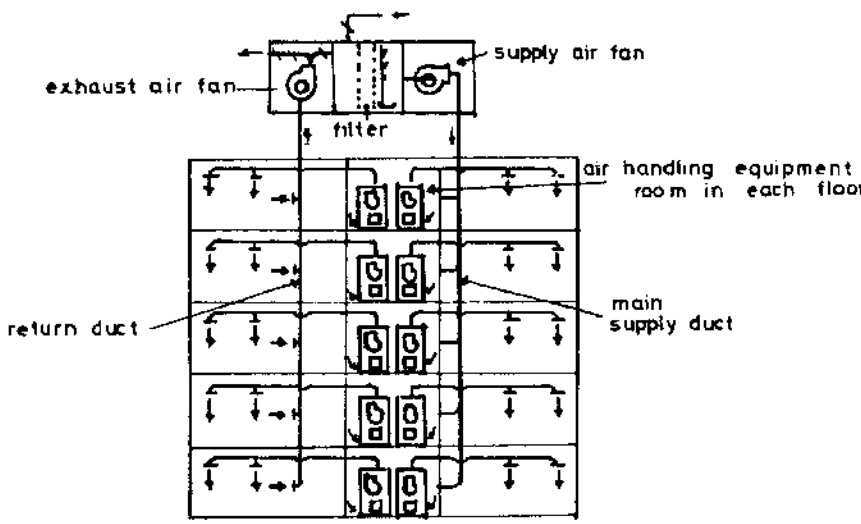


Figure 13 - Floor unit system.

Induction Unit System (2 Pipe). The induction unit system is a very interesting, compact and efficient, high-velocity air-conditioning method. Instead of carrying all of the air back to a central air-handling unit as in the dual duct system, room air is recirculated through a cabinet below a window. A small amount of fully conditioned outdoor air may be brought in through a single high-velocity duct. It flows through a jet after attenuation to induce room air circulation so that the room air and supply are mixed properly (Figure 14). Therefore, this system requires no air supply fans and no outdoor ventilation grills for fresh air. Chilled water and conditioned fresh air are supplied to the unit, and a two-way valve that is controlled by the room thermostat adjust the flow rate of

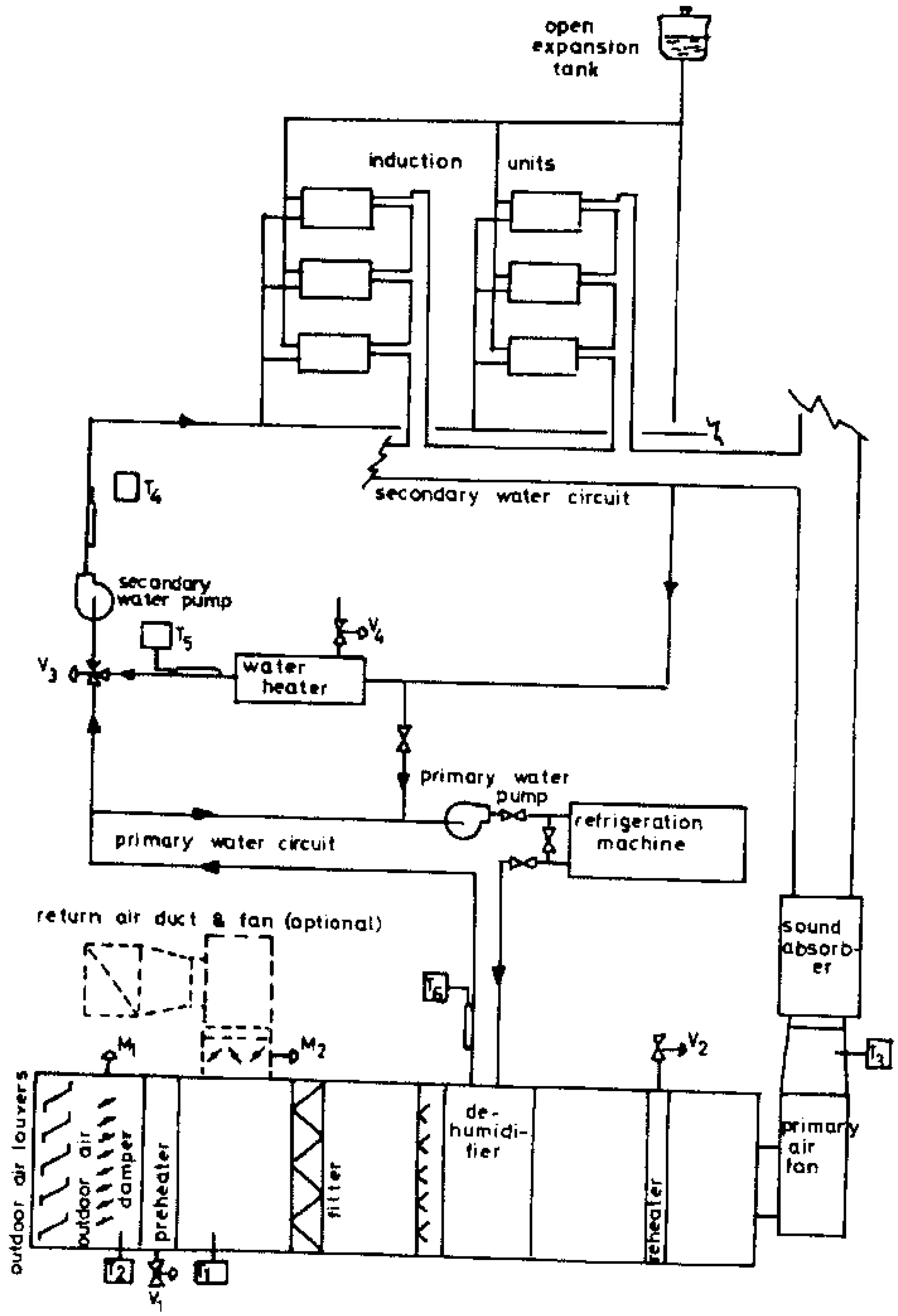


Figure 14 - Induction unit system (2 pipe).

chilled water. In general, heat loads from solar heat, people and lights are removed by chilled water, and the heat loads due to conduction, convection and radiation through the walls and roof are removed by conditioned air. As in the previous discussion, a smaller space is required in comparison with the dual duct system. However, heating and cooling energy offset each other in the same air-handling unit. Hence, operation cost increases. In addition, zoning of chilled water and conditioned fresh supply air is necessary.

Induction Unit System (3 Pipe). The 3-pipe induction unit system is more flexible than the 2-pipe system in that it supplies hot and chilled water simultaneously, while the 2-pipe system carries only hot or chilled water into an induction unit. According to the heating or cooling needs of a building, the return pipe is used in common for both hot and cold water return.

The flow rate of hot and chilled water is controlled by a sequence control valve (or three-way valve) operated by a room thermostat. This system has other merits in addition to those of the 2-pipe system: no zoning is required for either supply water or air, and mixing loss can be minimized when the return water pipes are divided into many zones. Hence, the operation cost can be reduced in comparison with the 2-pipe system.

On the other hand, as it requires a more complicated system, the equipment cost increases in comparison to the 2-pipe system.

Primary Air Fan-Coil Unit System. This system is similar to the induction unit system (Figure 15). The main difference is that conditioned air and chilled or hot water are supplied to the fan-coil unit instead of the induction unit. In turn, instead of inducing the room air, the supply air fan mixes the room and conditioned air properly. The most suitable applications for this system are multi-room buildings such as hotels, offices, hospitals and apartment houses. If we emphasize the importance of air-conditioning performance, this system may be the most suitable, but the initial cost of a fan-coil system is very high.

The fan-coil unit can be located along the perimeter of a building, and the primary air (conditioned air) is supplied to the fan-coil unit directly or is supplied to the room through ducts in the corridor (Figure 15). Application of this system is popular in Hawaii.

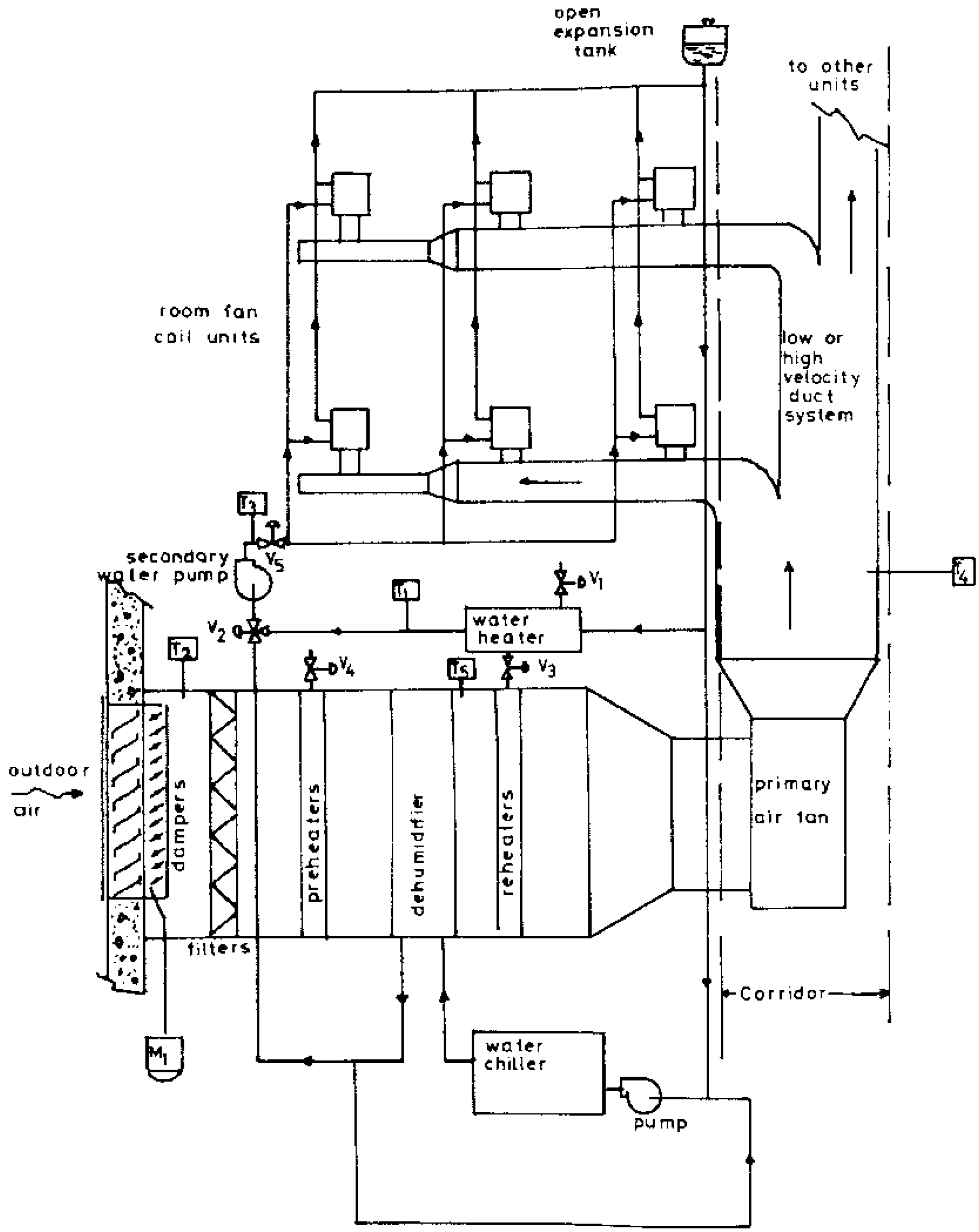


Figure 15 - Primary air fan-coil unit system.

Merits of this system are as follows:

- The system is ideally adapted for control of individual room air temperatures, as each fan-coil unit has integral heating and cooling coils.

- Motion and distribution of air in the room can be easily controlled by this system.

- Wherever the effect of cold radiation and convection from windows causes discomfort, this system is advantageous. Fan-coils along the perimeter of the windows supply the upward hot air and remove the effect of cold radiation and convection.

However, this system requires a relatively high equipment cost if the total floor area of a building is less than 200,000 square feet (i.e., a small or medium size office building).

3. <u>All-Water Systems</u>

The all-water systems are mainly fan-coil and packaged types of room terminals to which may be connected one or two water circuits. The cooling medium such as chilled water or brine may be supplied from a central station and circulated through the coils in the fan-coil or package unit terminal which is located in the room. Ventilation is obtained through an opening in the wall or by infiltration.

a. <u>Fan-Coil Unit System</u>.

This system is particularly applicable to multi-room buildings where large-sized duct work is impossible. It is not recommended for applications having high latent loads. Hotels, motels, hospitals, apartment houses and office buildings can use the system to advantage. The unit may be located under the window, over the closets, or in dropped ceilings.

There are two types of fan-coil unit systems: single piping (2 pipe), and multi-piping (3 pipe, 4 pipe). In the former, a single supply medium such as cold or hot water is available at each fan-coil unit, and a single return piping system is utilized. In the latter, hot and cold media are available at each fan-coil unit, and a single (3 pipe) or double (4 pipe) return piping system is utilized.

Single Piping (2 Pipe) System. This system (Figure 16) consists of central heating-cooling equipment and a fan-coil unit. The fan-coil unit system is designed to control individual space without connecting to the central air-handling station and duct. Either a mixture of fresh and return air or return air alone is supplied into the conditioned room. Although fresh air is generally supplied through a low pressure duct, it may be taken directly from the wall openings. This latter method costs less initially and gives greater flexibility in utilizing the system. However, wall openings for outdoor air are not generally recommended in multi-story buildings. Stack and wind effects may adversely affect the performance of the units. In some cases, infiltration air through windows and doors is sufficient for ventilation. The room air temperature can be adjusted by means of control switches for fan speed and water flow. Merits of the system are:

- It is suitable for individual control of room air temperature. As it has integral heating-cooling coils, adjustment of room air is quick and easy.
- Operation and equipment costs are comparatively low.
- Minimal duct work is required.

However, the quantity of supply air is not enough to humidify the room.

Multi-Piping (3 Pipe, 4 Pipe) System. This system provides hot and cold water through each fan-coil unit in the room (Figure 16). Each unit can be used for many different zones and functions by controlling the hot and cold water with control valves. Merits of the multi-piping system in addition to those for the single piping system are as follows:

- Quick thermal response is obtained through hot and cold water in the unit.
- Zoned piping and allied controls are eliminated due to the flexibility of using both hot and cold water.
- Complaints of occupants during the intermediate seasons are eliminated because of the availability of both heating and cooling.

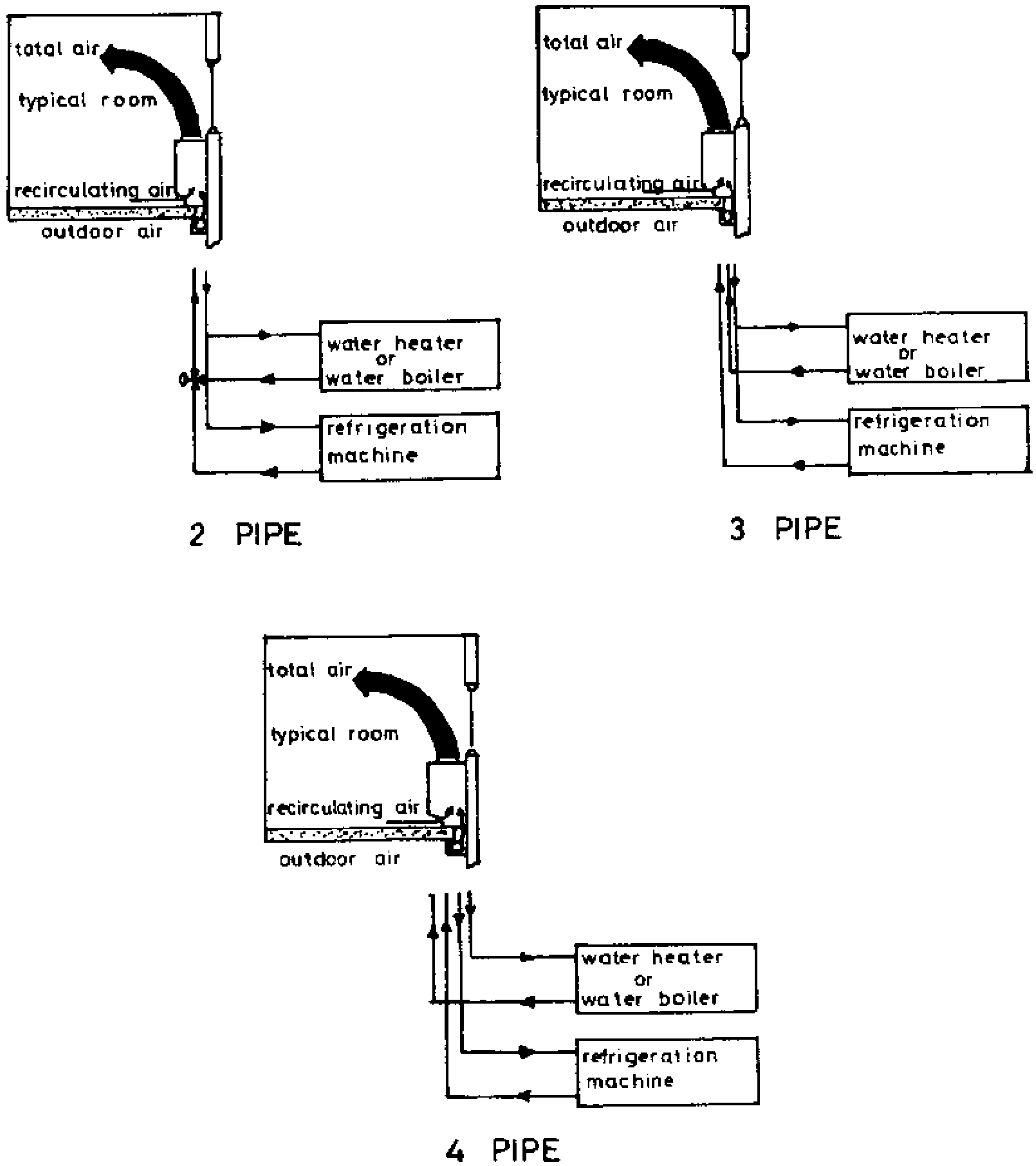


Figure 16 – Fan-coil unit system.

b. Packaged Air-Conditioners

Packaged air-conditioners, as in Figure 17, which are also called self-contained equipment, provide a complete heating and cooling system in the package unit. This unit is composed of compressor, condenser, fans, evaporator, heater, filter and controller. Commercial self-contained air conditioners are available in sizes up to 66 tons. Units of 2 to 7.5 tons are designed for the room usage and the larger units are located away from the rooms. Major refinements of this system are its low cost, compactness and easy operation. The packaged air conditioners are either air or water-cooled systems. The former has evaporative condensers while the latter has cooling towers. Merits of this system are as follows:

- The initial costs of packaged units are low due to less volume, and they can be installed easily with minimum disturbance to the occupants.
- The units are compact and easy to maintain because of their accessibility.

However, room air is not properly distributed and humidity control is insufficient. Additional expense is required to improve the performance of the system. Also, as maintenance is required fairly frequently, maintenance cost is comparatively high.

c. Room Air Conditioners

Packaged air conditioners with capacities of 0.5 to 2 tons are usually defined as room air conditioners. These are mainly air-cooled conditioners, but larger sizes may require water-cooled systems. Air-cooled units may be located on the wall, roof, window or any other place facing the outside air to reject the heat and to dispose of the condensed water from the cooling coil.

C. Applications

Typical applications for the various systems described above are shown in Table 5. Detailed explanations are found in the Handbook of Air Conditioning System Design (9), which was the source of most of the figures and tables in this section.

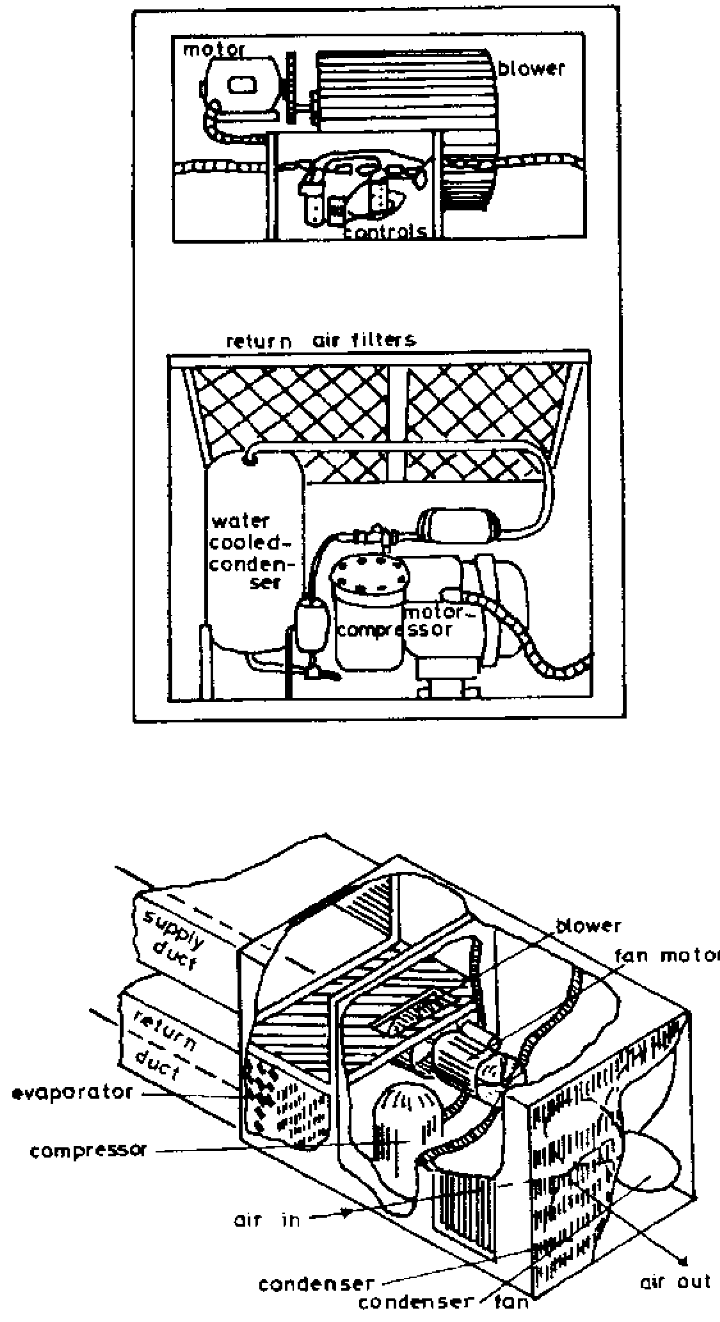


Figure 17 - Packaged air conditioners.

Table 5 - Systems and applications

APPLICATIONS			Individual Room or Zone Unit Systems				Central Station Apparatus Systems						
			DX Self-Contained		All-Water		All-Air					Air-Water	
			Room	Zone	Room Fan-Coil		Single Air Stream					Prim. Air Systems	
									Reheat				
			⅓ to 2 tons	2 tons and over	Recir. Air	With Outdoor Air	Variable Volume	Bypass	At Terminal	Zone in Duct	Multi-Zone Single Duct	Secndry. Water H-V H-P Induction	Room Fan-Coil with O.A.
		Page	(9–8)	(9–8)	(9–8)	(9–8)	(9–9)	(9–9)	(9–10)	(9–10)	(9–10)	(9–11)	(9–8)
Single-Purpose Occupancies	Residential Medium	(9–13)	x										
	Residential Large	(9–13)		x	x						x		
	Restaurants Medium	(9–13)		x						x			
	Restaurants Large	(9–13)		x				x	x	x	x		
	Variety & Spctly. Shops	(9–13)		x									
	Bowling Alleys	(9–14)		x				x					
	Radio and TV Studios Small	(9–14)		x				x		x	x		
	Radio and TV Studios Large	(9–14)		x				x		x	x		
	Country Clubs	(9–14)		x				x		x	x		
	Funeral Homes	(9–14)		x							x		
	Beauty Salons	(9–14)	x	x									
	Barber Shops	(9–15)	x	x									
	Churches	(9–15)		x				x			x		
	Theaters	(9–15)						x					
	Auditoriums	(9–15)						x					
	Dance and Roller Skating Pavilions	(9–15)		x			x	x					
	Factories (comfort)	(9–15)		x				x		x			
Multi-Purpose Occupancies	Office Buildings	(9–16)					x				x	x	
	Hotels, Dormitories	(9–18)			x	x						x	x
	Motels	(9–18)			x								
	Apartment Buildings	(9–18)				x						x	x
	Hospitals	(9–18)				x				x		x	
	Schools and Colleges	(9–19)				x	x	x	x	x			
	Museums	(9–20)								x	x		
	Libraries Standard	(9–20)		x				x		x	x		
	Libraries Rare Books	(9–20)		x						x			
	Department Stores	(9–19)						x					
	Shopping Centers	(9–19)		x				x			x		
	Laboratories Small	(9–20)		x			x			x	x		
	Laboratories Lge Bldg	(9–20)					x		x			x	
	Marine	(9–21)							x			x	

NOTES: 1. Systems checked for a particular application are the systems *most commonly used*. Economics and design objectives dictate the choice and deviations of systems listed above, other systems as listed in Note 2, and some entirely new systems.

2. There are several systems used on many of these applications when higher quality air conditioning is desired (often at higher expense). They are Dual-Duct (9–11), Dual Conduit (9–9), 3-pipe Induction and Fan-Coil (9–11), 4-pipe Induction and Fan-Coil, and Panel-Air (9–12).

3. Numbers in parentheses are page numbers of the text describing the particular system or application.

III. HEAT GAINS

The thermal factors affecting the superstructure and buoyancy tanks of the Floating City are numerous and, as these are intricately interrelated, the evaluation and analysis of these factors should be performed carefully.

Heat gain is defined as the rate at which heat enters into or is generated within a space. It can be influenced by weather, location of a building, time, construction materials, usage of a building, equipment, lighting and occupancy.

In this section, equations are developed for determining heat gains through the windows, walls and roof of the superstructure, as well as heat gains due to ventilation and infiltration of outside air. The heat generated by people, lights, and equipment within the building is briefly discussed. Finally, consideration is given to heat gains through the walls of the buoyancy tanks.

A. Heat Gain through Windows of Superstructure

At any instant, the total instantaneous heat gain through glass is the sum of solar radiation transmitted through glass and the heat flow due to convection from the inner glass surface and conduction caused by the temperature difference between outside and inside air. For glazing materials, heat gain Q_g is expressed in the 1972 ASHRAE Handbook (10) as

$$Q_g = (SC) \Big[\Big(I_{nd} \sum_{j=0}^{5} \tau_j \cos^j \eta \quad + 2\, I_d \sum_{j=0}^{5} \tau_j (j+2)^{-1}\Big)$$

$$+ \Big(I_{nd} \sum_{j=0}^{5} \alpha_j \cos^j \eta \; + 2\, I_d \sum_{j=0}^{5} \alpha_j (j+2)^{-1} (h_o)(h_o + h_i)^{-1}\Big)\Big]$$

$$+ U(\theta_o - \theta_i) \qquad (3\text{-}1)$$

$$= (SC)\ (SHGF) + U(\theta_o - \theta_i) \qquad (3\text{-}2)$$

SHGF is the abbreviation for solar heat gain factor, which is the rate of heat flow due to transmitted solar radiation and convection from the inner surface of glass; SC is the shading coefficient, defined as the ratio of solar heat gain through a fenestration under a specific set of conditions to the solar gain through a plain double-strength glass under the same set of conditions. The effect of reflected radiation is neglected in Equation 3-1, which must be modified in the case of high intensity of reflected radiation. Shading coefficients applicable to some of the widely used types of insulating glass and internal shading coefficients for curtains and venetian blinds are given in air-conditioning handbooks. Calculation of the solar heat gain factor is made following the ASHRAE's recommended procedures.

B. Heat Gain through Walls or Roof of Superstructure

Figure 18 illustrates the factors which affect the heat balance at the outside surface of a wall at time $n\Delta t$. The X_j, Y_j and Z_j are called response factors (8,11). The heat-balance equation at the outside surface is

$$\sum_{j=0}^{J} Y_j \theta_{i,n-j} + \alpha_s I_{t,n} - h_o(\theta_{s,n} - \theta_{o,n}) - \sum_{j=0}^{J} X_j \theta_{s,n-j} = 0 \qquad (3-3)$$

where α_s is the absorptivity of the wall surface. The value of J is to be selected arbitrarily, depending upon the degree of accuracy of calculations. Rearrangement of Equation 3-3 gives

$$\theta_{s,n} = (X_o + h_o)^{-1} \left[\alpha_s I_{t,n} - \sum_{j=1}^{J} X_j \theta_{s,n-j} + \sum_{j=0}^{J} Y_j \theta_{i,n-j} + h_o \theta_{o,n} \right] \qquad (3-4)$$

The inside air temperature $\theta_{i,n-j}$ is usually considered to be constant. By taking appropriate prior values of outside surface temperature $\theta_{s,n-j}$ to start the calculation, the value of $\theta_{s,n}$ can be determined by the above equation. By increasing the number n, accurate values of outside surface temperature can be found subsequently. For a constant inside air temperature $\theta_{i,n-j}$, the effective excitation of outside surface temperature is $(\theta_{s,n-j} - \theta_{i,n-j})$. Then the heat flux for the wall area A_w is

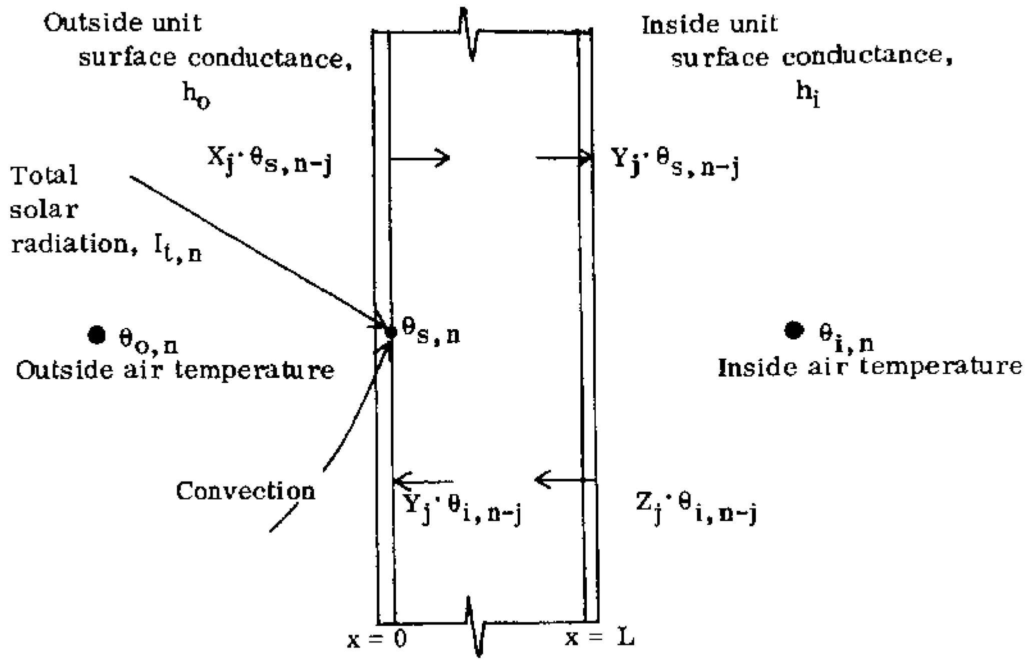


Figure 18 - Heat balance at outside surface.

$$Q_w = A_w \sum_{j=0}^{J} Y_j(\theta_{s,n-j} - \theta_{i,n-j}) \tag{3-5}$$

Physical interpretations of response factors X_j, Y_j and Z_j as depicted in Figure 19 are as follows:

X_j : Heat rate at time $t = j\Delta t$ at the surface $x = o$ in Btu/hr-ft^2-^{o}F when the unit temperature excitation is applied to the same surface.

Y_j : Heat rate through the wall at time $t = j\Delta t$ at $x = 0$ or $x = L$ in Btu/hr-ft^2-^{o}F when the unit temperature excitation is given to the surface at the opposite side of the wall. Y_j at $x = 0$ is the same as that at $x = L$ because of the reciprocity theorem.

Z_j : Heat rate at time $t = j\Delta t$ at the surface $x = L$ in Btu/hr-ft^2-^{o}F when the unit temperature excitation is given to the surface at $x = L$.

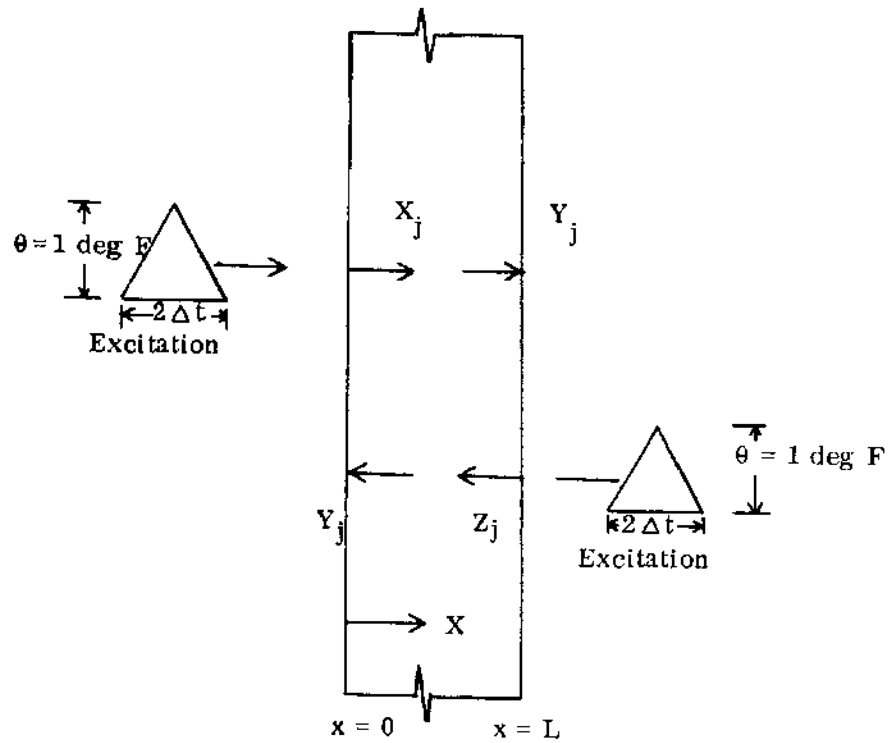


Figure 19 - Physical representation of response factors.

C. Heat Gain from Lights, Equipment and People

Lights, equipment and people are the main heat generating sources within a room. The precise prediction of the number of people and lights is not possible. In general, the number of people in a room and power consumption of lights and equipment are estimated according to the functions of the building. The heat generated per average person for various types of activities can be found in handbooks (9, 10).

D. Heat Gain from Ventilation and Infiltration

Comfort criteria require a certain amount of outside air to be supplied continuously through a ventilation duct to the conditioned space. The heat gain in Btu per hour due to ventilation at G cfm of outside air (10) is given by

$$Q_{vent} = \text{sensible gain} + \text{latent gain} \tag{3-6}$$

$$= G \times 60 \times 0.075 \times 0.245\,(\theta_o - \theta_i)$$

$$+ G \times 60 \times 0.075 \times 1076\,(W_o - W_i)$$

Infiltration of outside air into a building is affected largely by wind velocity, temperature difference between outside and inside air, and construction of the building. If N is the recommended rate of air change per hour for the room of V cu ft, the heat gain in Btu per hr due to infiltration (8,10) is

$$Q_{infilt} = 0.0183 \times N \times V(\theta_o - \theta_i) + 79.5 \times N \times V\,(W_o - W_i) \tag{3-7}$$

E. Heat Gain through Walls of Buoyancy Tank

Application of an overall heat transmission coefficient is a convenient way to calculate the heat gain through the walls of the buoyancy tank. By taking into consideration Figure 5, temperature of seawater is assumed to be constant at a specific depth. Therefore, the application of response factors is not practical, as the condition of heat flow is steady. In general, the heat flow through the wall due to conduction in cylindrical coordinates is expressed by

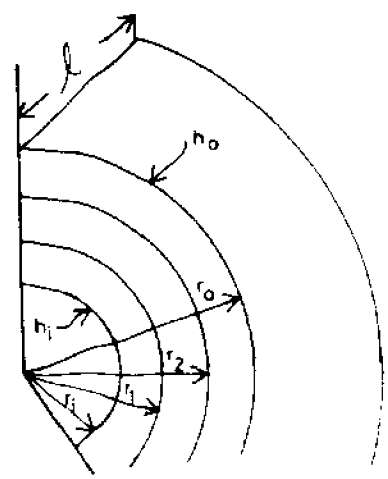


Figure 20 - Wall of cylindrical coordinates

$$Q = U(\theta_i - \theta_o)\,2\pi\ell \tag{3-8}$$

$$U = 1/\left[(h_i r_i)^{-1} + \sum_{n=1}^{N} \ln(\rho/k)_n + (h_o r_o)^{-1}\right] \tag{3-9}$$

$$\rho_1 = r_1/r_i \quad ; \quad \rho_2 = r_2/r_1 \quad ; \ldots \quad \rho_n = r_o/r_{n-1}$$

IV. COOLING LOADS

A. Heat Gain

The heat gains described in the preceding section are considered to be instantaneous heat gains in the new ASHRAE procedures for cooling load calculation. As we have seen, instantaneous heat gains may originate from a number of sources: solar radiation admitted through windows, heat transmitted through walls and roof, infiltration, ventilation, lighting, people, and machinery inside the space. The heat gains from different sources undergo different delays before they become cooling loads.

In the 1972 ASHRAE Handbook of Fundamentals (10), the instantaneous heat gain from each source is separated into a convective portion and a radiant portion. The radiant portion is first absorbed by the walls and by the contents in the space, and then transferred to the inside air over an extended period of time through the mechanism of convection. The total instantaneous cooling load is the sume of the convective portion and the average of the radiant portion over a period of time. The recommended period of time is 2 to 3 hours for lightweight construction and 6 to 7 hours for heavyweight, without a clear definition to differentiate lightweight from heavyweight.

B. Cooling Load Evaluation

Exact evaluation of the space cooling load involves the solution of simultaneous equations of heat balance for many parts of the building. Stephenson and Mitalas (12) demonstrated the exact method by formulating eight simultaneous equations for a room of simple configuration. Mitalas and Arsenault (13) later proposed a weighting factor method based on the assumption that the instantaneous heat gain and the corresponding component of the cooling load can be expressed in the form of a characteristic transfer function. Their weighting factors are the sets of transfer functions which relate the heat gains to cooling loads, being determined by the z-transfer method.

C. Response Factor Method

It would be advantageous to express weighting factors in terms of response factors. This idea was first introduced by Kimura and Stephenson (14). For 1 Btu/hr-ft^2 of radiant heat impinging upon a surface whose absorptivity is α, the heat flow through the surface is α Btu/hr-ft^2. Assume the heat transfer coefficient of the convective film to be h_i Btu/hr-ft^2-^oF. If the temperature at the outside surface of the convective film is α/h_i degrees higher than the temperature at the wall's surface, as depicted in Figure 21, the heat flow is also α Btu/hr-ft^2.

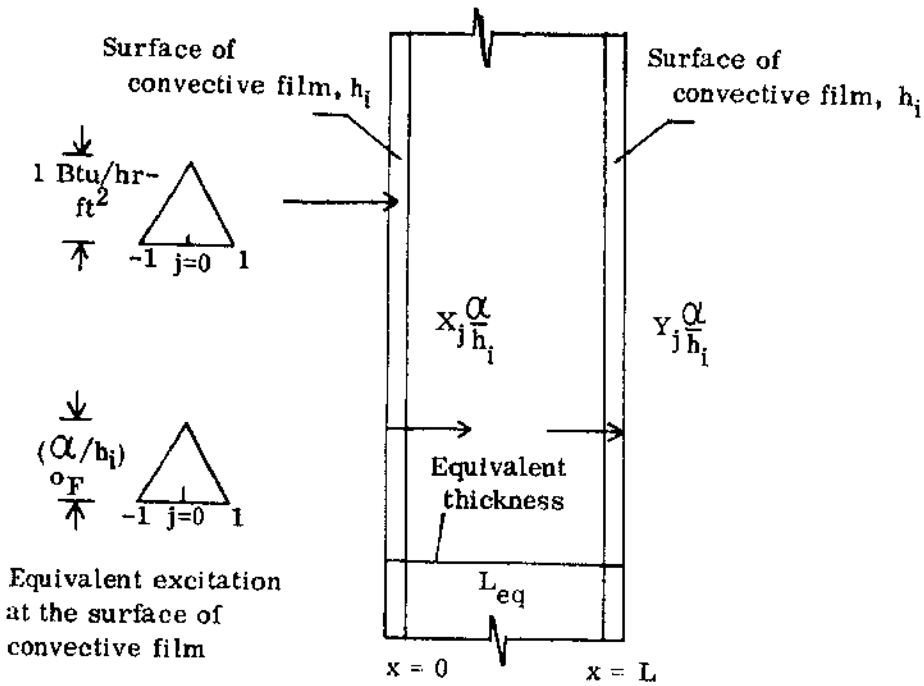


Figure 21 - Equivalent excitation for wall of equivalent thickness.

D. Equivalent Thickness Approximation

Hypothetically one may assign an equivalent excitation α/h_i degrees at the outside surface of the film to evaluate the transient flow of heat through the wall due to the unit radiant heat.

If the response factors X_j, Y_j and Z_j are evaluated for a solid wall of equivalent thickness L_{eq} together with the convective films as shown in Figure 21, the outside surfaces of the composite wall should be the exterior surfaces of the rilms. The magnitude of response factor Y_j is relatively small in comparison with X_j as shown in Figures 18 and 19, in which one also finds that X_j at $j = 0$ has a sign opposite to the sign of other X_j's and the sign of Y_j's to signify that the heat is first driven into the wall momentarily by the excitation at $j = 0$ and then flows out from both sides of the wall.

Referring to the definitions of X_j and Y_j in Section III, the amount of heat which remains in the space after 1 Btu/hr-ft^2 of radiative heat from an internal source impinges upon the wall, can be evaluated by the following two equations:

$$W_j = 1 - \frac{\alpha}{h_i}(X_j + Y_j) \simeq 1 - \frac{\alpha}{h_i} X_j, \text{ at } j = 0 \tag{4-1}$$

$$W_j = - \frac{\alpha}{h_i} X_j, \text{ at } j \geq 1 \tag{4-2}$$

The W_j is known as the weighting factor, which may appear in different versions by different ways of evaluation.

Takeda and Matsuo (15) studied the thermal characteristics of composite walls and found that the transient behavior of multi-layer walls can be approximated with an equivalent wall having a single layer. The thickness L_s of the equivalent wall for a composite wall of M layers is given by

$$L_s = \sum_{m=1}^{M} (C_p L/\rho)_m / (C_p/\rho)_k \tag{4-3}$$

where the subscript k is used to indicate the thermal properties of the equivalent wall.

To simplify the calculation of weighting factors, it is proposed here to use an equivalent thickness L_{eq} for all the walls in a room by extending Takeda and Matsuo's idea:

$$L_{eq} = \sum_{m=1}^{M} (A \cdot L_s)_n \Big/ \sum_{n=1}^{N} (A)_n \tag{4-4}$$

where A is the surface area of the wall with the respective thickness of L_S, and N is number of walls in a room. The degree of validity of Equation 4-4 has yet to be checked by other methods, and some empirical constants may have to be added to improve its applicability.

After the heat gains and weighting factors are found, the cooling load at time $n \Delta t$ can be calculated by the equation,

$$CL_n = \sum_{j=0}^{J} W_j HG_{n-j} + HC_n \tag{4-5}$$

where HG_{n-j} is the sum of radiative heat gains of a room at $j \Delta t$ hours prior to time $n \Delta t$, and HC_n is the sum of convective heat gains at time $n \Delta t$. The proportioning of radiative heat gains and convective heat gains from various heat sources has been recommended by ASHRAE (10).

The cooling loads for the superstructure and buoyancy tanks were calculated with the proposed concept of equivalent wall thickness. The results, as shown in Section V, appear reasonable in relation to the total heat gains.

V. CALCULATIONS OF COOLING LOAD FOR SUPERSTRUCTURE AND BUOYANCY TANK

Sample cooling load calculations have been made for the superstructure and for a buoyancy tank, to illustrate the application of the foregoing heat-flow model to the design of an environmental control system for the Floating City. The calculations were performed on an IBM 360/65 computer using a program written for this report in FORTRAN IV. Appendix B contains the program flow diagram and a brief description of the subroutines used in the calculations. The program listing and sample outputs are presented in Appendix C.

A. Superstructure

The top floor of the superstructure in the Floating City is used as the model for sample calculations. Figures 22, 23, and 24 show the floor plan, a typical room, and the elevation of the building.

For satisfactory control of indoor conditions in large buildings, zones of heating and air conditioning are usually established according to thermal requirements (16). The building shown in Figure 22 is divided into nine zones, labeled A through I. The cooling load of a typical room in each zone has been calculated and reported hereupon.

The structure is mainly made of lightweight concrete. Table 6 shows the composition of walls and roof, and the physical properties of the pertinent materials; in the same table are the equivalent thicknesses (L_{eq}), which are determined by Equation 4-4.

The following input data are required for the calculation of cooling load:

- o Physical properties of materials from Table 6
- o Weather data from Table 7
- o Input data given in Table 8

Table 8 summarizes the indoor design conditions; the assumed temperature of the outside surface to start the calculations; areas of outside walls, windows and roof; absorptivities of outside surfaces; volumes of rooms and orientation of walls.

Also shown in Table 8 are sample input values for the number of people and the power input from lights, both of which are functions of time of day, and these are further specified in Tables 9 and 10. The rate of heat gain per occupant is an input code selected from Table 11 according to type of building and degree of activity. Ventilation requirements, based on anticipated smoke levels, are selected from coded values in Table 12. Infiltration rate depends upon type of construction, and the input codes are selected from Table 13. Finally, the glazing material is selected from Table 14, which lists shading coefficients and U values for various types of glass. (In this case, input code 3 denotes 1/8-inch double strength glass).

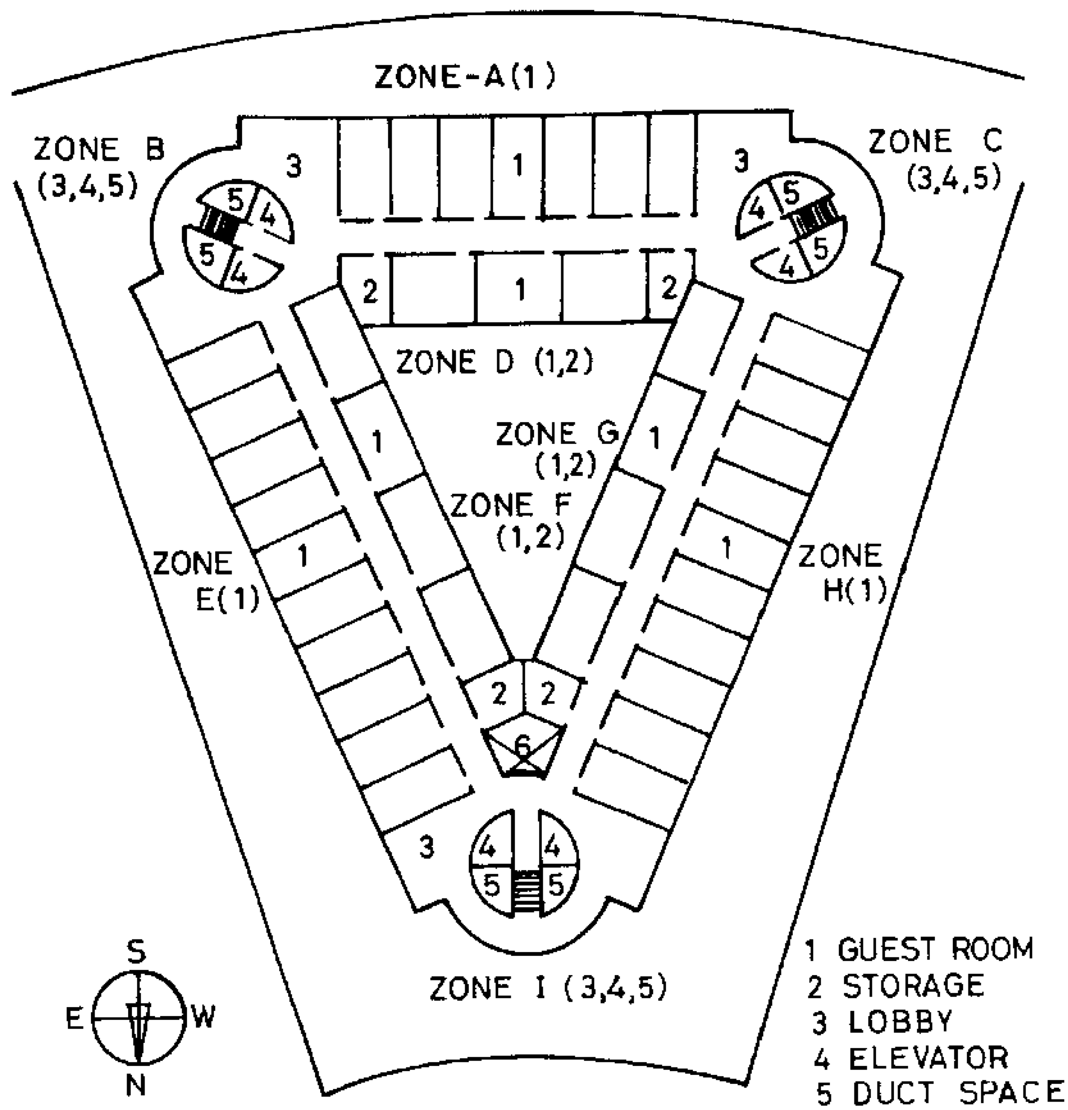


Figure 22 - Top floor plan of superstructure.

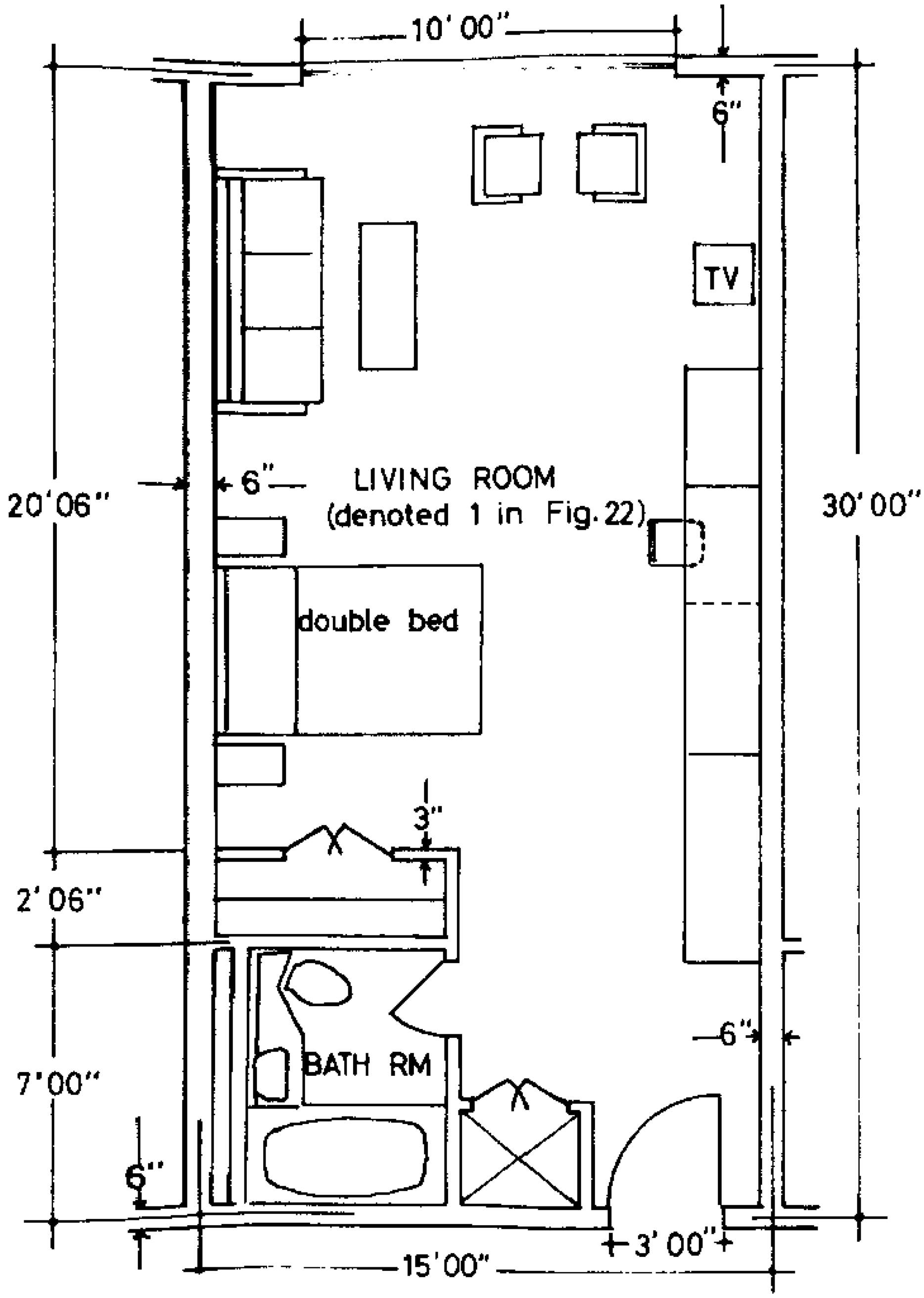


Figure 23 - Typical room of superstructure

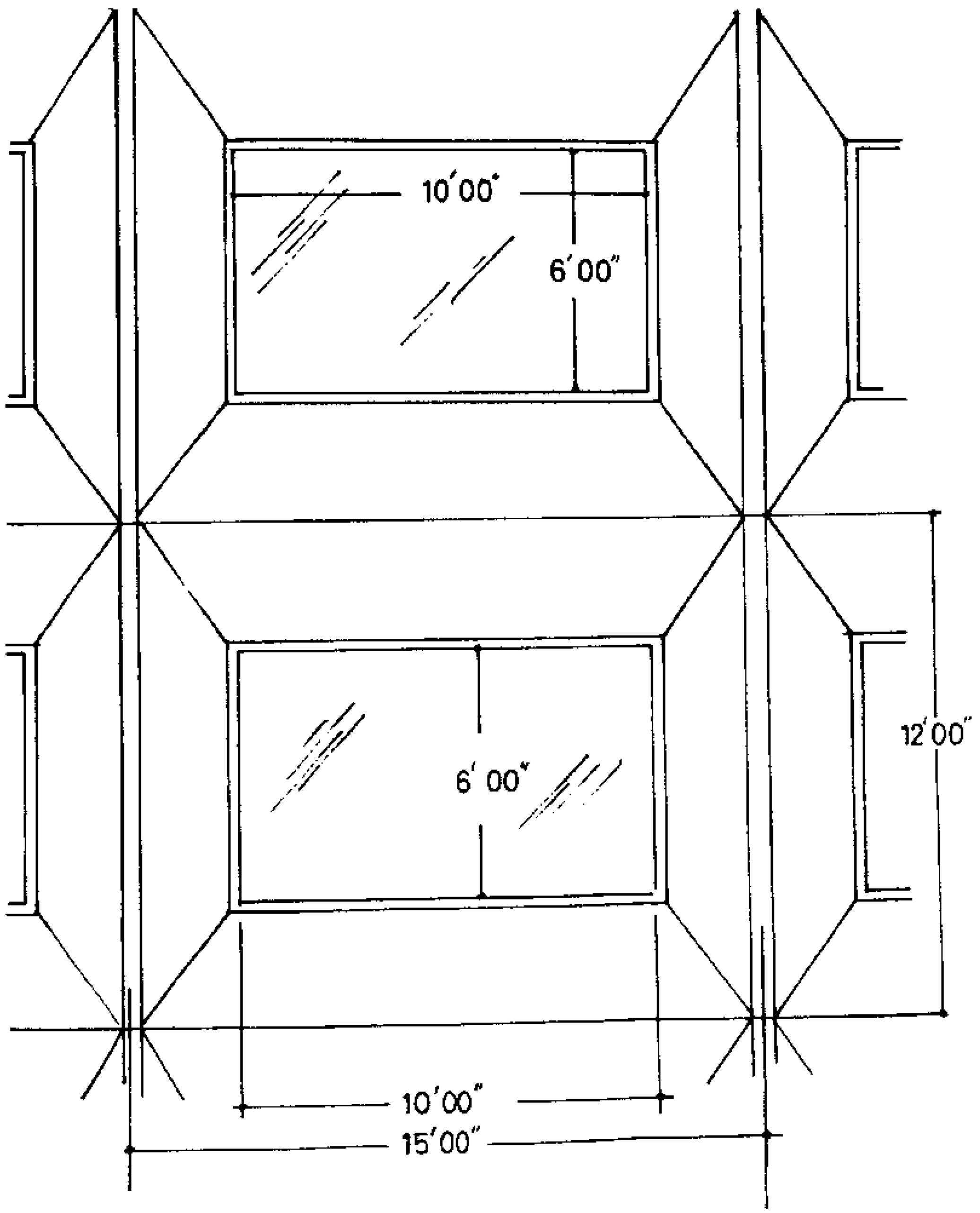


Figure 24 - Elevation of superstructure

Table 6 - Physical properties of wall and roof of superstructure

	DESCRIPTION	L	k	ρ	C_p	R	α
Wall	1" Stucco	0.0833	0.40	116.0	0.2	--	0.61
	6" Lightweight concrete	0.50	0.10	40.0	0.2	--	--
	1" Insulation	0.0833	0.025	2.0	0.2	--	--
	3/4" Plaster	0.0625	0.420	100.0	0.2	--	--
	Inside air film	0.0	0.0	0.0	0.0	0.50	--
	FNOL = 5.0, IREF = 1						
Roof	1" Waterproof cement mortar	0.0833	0.40	116.0	0.70	--	0.73
	3/8" Asphalt	0.0417	0.83	55.0	0.40	--	--
	6" Lightweight concrete	0.50	0.10	40.0	0.20	--	--
	12" Ceiling air space	0.0	0.0	0.0	0.0	1.0	--
	3/4" Acoustic tile	0.0625	0.035	30.0	0.20	--	--
	Inside air film	0.0	0.0	0.0	0.0	0.50	--
	FNOL = 6.0, IREF = 2						
T, a	Inside air film	0.0	0.0	0.0	0.0	0.50	--
	8.2" (L_{eq}) lightweight concrete	0.87	0.1	40.0	0.20	--	0.573
	Inside air film	0.0	0.0	0.0	0.0	0.50	--
	FNOL = 3.0						
T, b	Inside air film	0.0	0.0	0.0	0.0	0.50	--
	13.2" (L_{eq}) lightweight concrete	1.10	0.10	40.0	0.20	--	0.546
	Inside air film	0.0	0.0	0.0	0.0	0.50	--
	FNOL = 3.0						

Symbols:

T = Equivalent-thickness wall,
a = Zone A, D, E, F, G, and H,
b = Zone B, C, and I,
L_{eq} = Equivalent thickness, Eq. (4-4),
L = Thickness of layer (ft),
ρ = Density (lb/ft^3),
k = Thermal conductivity (Btu/hr-ft-^{o}F),
C_p = Specific heat (Btu/lb-^{o}F),
R = Resistance (hr-ft^2-^{o}F/Btu)
α = Absorptivity,
FNOL - Number of layers,
IREF - Kind of response factor.

Table 7 – Average hourly temperature and humidity on September 1 in downtown Honolulu (18)

Time (hr)	Temperature (°F)	Relative humidity (%)	Time (hr)	Temperature (°F)	Relative humidity (%)
0100	75.90	71.0	1300	88.80	51.5
0200	76.10	70.0	1400	88.60	52.0
0300	76.80	69.6	1500	87.90	54.0
0400	77.80	69.0	1600	86.90	56.8
0500	79.10	68.5	1700	85.60	59.5
0600	80.70	68.0	1800	84.00	61.5
0700	82.40	67.0	1900	82.40	63.3
0800	84.00	66.0	2000	80.70	65.0
0900	85.60	64.2	2100	79.10	66.5
1000	86.90	62.1	2200	77.80	68.0
1100	87.90	59.5	2300	76.80	69.0
1200	88.60	55.1	2400	76.10	70.0

Table 8 - Input for the sample calculation

Computer program label	Description	Value
TR, WR	Design air temperature (^{o}F) and specific humidity of the room, (lb/lb)	78.0, 0.0125
IDOY	Time of year, (Sept. 1)	244
NDAY	Length of calculation (days)	2
TDB	Initial surface temperature of wall and roof (^{o}F)	80.0
FO, FI	Outside and inside unit thermal conductances, ($Btu/hr-ft^2-^oF$)	6.0, 2.0
WW	Number of different delayed surfaces (wall and roof)	2.0
FNS	Number of spaces in the building (Zone A to I)	9.0
ARE	Outside wall and roof surface areas (ft^2)	
	Spaces 1 of Zone A, D, E, F, G, and H	60 (6)
	Space 3 of Zone B (ENE, E, ESE, S)	445, 295, 295, 445
	Space 3 of Zone C (S, WSW, W, WNW)	445, 295, 295, 445
	Space 3 of Zone I (N, ENE, WNW)	590, 445, 445
AAB	Absorptivities of outside wall and roof surface	
VOL	Spaces 1 of Zone A, D, E, F, G, and H	5400 (5)
	Space 1 of Zone D	6000
	Spaces 3 of Zone B, C, I	21672 (3)
IREF	Wall orientations as given in the note (first 8 numbers) and roof (last number)	
	0 = no wall, 1 = wall, 2 = roof	
	Zone A	000010002
	Zone B	000000002
	Zone C	000000002
	Zone D	100000002
	Zone E	010000002
	Zone F	000001002
	Zone G	000100002
	Zone H	000000012
	Zone I	000000002
ALFA	Absorptivity for the wall of equivalent thickness	
	Zone A, D, E, F, G, H	0.5725 (6)
	Zone B, C, I	0.5458 (3)
NOPP	Reference number of people	2
(NOP)	Number of people (Table 9)	
WAAT	Reference power input from lights (watt)	160
(WATT)	Power input from lights (Table 10)	
I50	Rate of heat gain per occupant (Table 11)	3
I51	Ventilation requirement (Table 12)	4
I52	Infiltration rate (Table 13)	1
I53	Type of glass (Table 14)	3
TTO	Dry-bulb temperature of outside air	
WWO	Specific humidity of outside air	

Note: Code for orientation of wall

Wall orientation	N	ENE	E	ESE	S	WSW	W	WNW
Wall azimuth (deg)	180	112.5	90	67.5	0	67.5	90	112.5

Table 9 - Number of occupants

Time of day	Zone A, D, E, F, G, H	Zone B, C, I
0100 to 0800	2	10
0900 to 1700	1	5
1800 to 2000	2	10
2100 to 2400	3	15

Table 10 - Power input from lights

Time of day	Zone A, D, E, F, G, H	Zone B, C, I
0100 to 0300	40 watt	400 watt
0400 to 0600	20	200
0700 to 1700	0	0
1800 to 2400	160	1600

Table 11 - Rate of heat gain per occupant (9)

Input Code (I50)	Heat Btu/hr	Typical application	Degree of activity
1	350	Theater -- evening	Seated at rest
2	400	Offices, Hotels, Apartments	Seated, very light work
3	450	Offices, Hotels, Retail store	Moderately active office work
4	500	Bank, Drug store	Standing, working slowly
5	550	Restaurant	Sedentary work
6	750	Factory	Light bench work
7	850	Dance hall	Moderate dancing
8	1000	Factory	Moderate heavy work
9	1450	Bowling alley, Factory	Heavy work

Table 12 - Ventilation requirements (10)

Input Code (I51)	Cfm/person Recommended	Application	Smoking
1	20	Average apartment	Some
2	7.5	Department store, Theater	None
3	10	Bank, Drug store	Occasional
4	30	Hotel rooms	Heavy
5	50	Meeting rooms	Very heavy
6	15	General offices	Some
7	12	Restaurant	Considerable

Table 13 - Infiltration rate (17)

Input Code (I52)	Number of air changes	Types of Construction	Degree
1	0.5	Concrete or steel, with steel sash	Good
2	1.0	Concrete or steel, with steel sash	Medium
3	1.5	Masonry, with steel sash	Medium
4	2.0	Masonry, with wooden sash	Medium

Table 14 - Shading coefficient and U-value of various glasses (10)

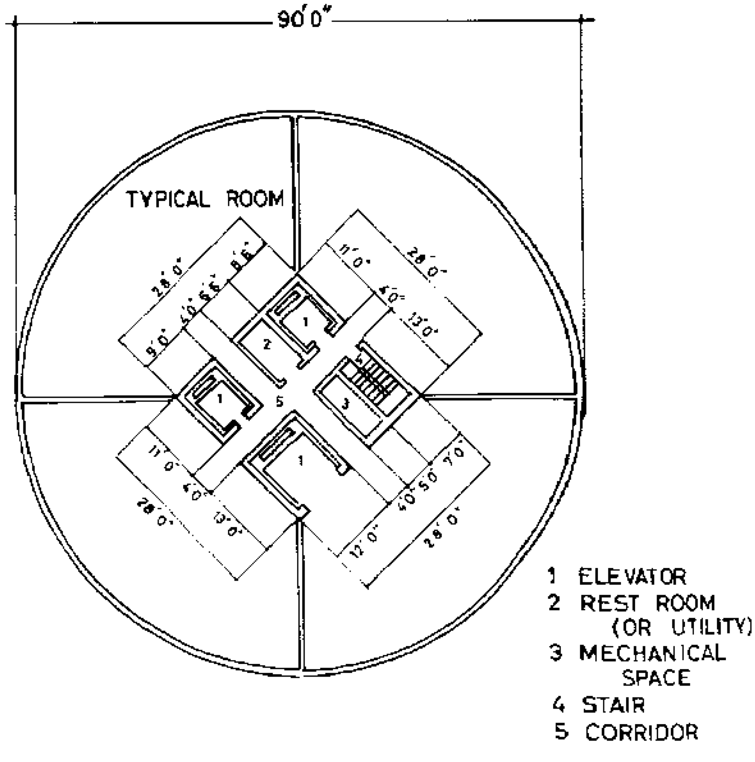
Input Code (I53)	Shading Coefficient	U-Value	Type of Glass
1	1.00	0.81	1/8", 3/32" of regular sheet glass
2	0.78	0.81	1/8" of grey sheet glass
3	0.90	0.81	1/8" of double strength glass
4	0.72	0.54	3/16" of heat absorbing glass

B. Buoyancy Tank

The floor plan and the elevation of the buoyancy tank are shown in Figures 25 and 26. Zoning is not necessary, for the entire tank has similar heat gain characteristics. The cooling load of a typical floor has been calculated by assuming a weighting factor of 1. Therefore, the heat gains and cooling load are identical in this case.

Eight different types of building materials (Table 15) were used for calculating the rate of heat flow through the wall in accordance with Equation 3-8. Finally, four different types of wall materials were selected for calculating cooling load.

Equations for heat gains due to ventilation and light are identical to those for the superstructure. Heat gains due to infiltration and total solar radiation transmitted and conducted through the glass are ignored, as the buoyancy tank is located in sea water.



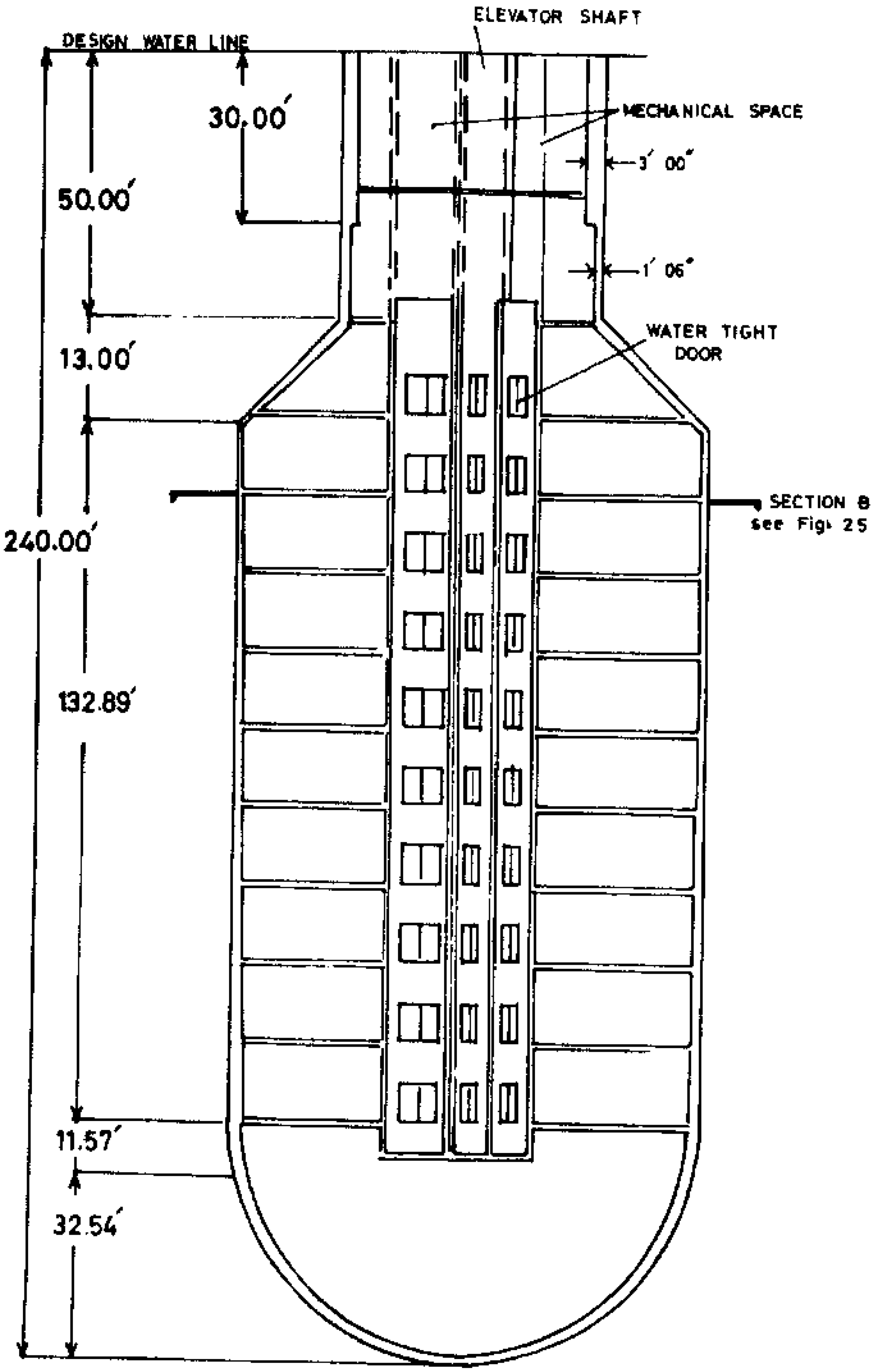


Figure 26 - Elevation of buoyancy tank.

Table 15 - Different types of wall configurations for the buoyancy tank and respective rates of heat flow

Description of Wall	Overall heat transmission coefficient ($Btu/hr-ft^2-^oF$) @ depth			Rate of heat flow ($Btu/hr-ft^2$) @ depth		
	70 ft	105 ft	140 ft	70 ft	105 ft	140 ft
a	0.397	0.310	0.253	0.556	0.527	0.886
b	0.364	0.296	0.244	0.510	0.503	0.854
c	0.396	0.309	0.253	0.554	0.525	0.886
d	0.356	0.284	0.236	0.498	0.483	0.826
e	0.292	0.233	0.200	0.409	0.396	0.700
f	0.259	0.219	0.200	0.363	0.372	0.662
g	1.973	1.967	0.189	2.762	3.344	6.864
h	0.675	0.481	1.961	0.945	0.818	1.684

Symbols:

a = bare concrete
b = 3/4" gypsum plaster + concrete (outside)
c = 3/8" steel plate + concrete
d = 3/8" steel plate + concrete + membrane (epoxy)
e = 3/4" gypsum plaster + 4" air space + concrete
f = 3/4" gypsum plaster + 4" air space + concrete + membrane (epoxy)
g = steel (1% carbon)
h = 3/4" plaster + 1/2" gypsum board + 4" air space + steel (outside)

Note: Thicknesses of steel and concrete of buoyancy tank vary in accordance with the depth from the design water level.

depth (ft)	70	105	140
concrete thickness (ft)	1.0	1.385	1.77
steel thickness (ft)	0.167	0.208	0.25

C. Results and Discussion

To facilitate the inspection of the results, the heat gains and cooling loads from the output of the computer program and the calculated values of cooling loads have been plotted.

Figure 27 illustrates the rates of heat gain through 1/8-inch double strength glass with a shading coefficient of 0.9 and an overall heat transmission coefficient of 0.61 Btu/hr-ft^2-^{o}F, plotted in accordance with orientations of windows. The glass facing east causes a high heat gain before noon due to direct solar radiation, and the glass facing west receives maximum heat flow in the afternoon. The heat gains of glass facing south and north are always symmetrical about noon time regardless of the time of year. It is surprising to note that the heat gain through a south glass is much lower than that of an east or west glass because of the relationships among solar angles. Selection of the glasses is one of the most important factors in the design of an air conditioning system. For example, selection of glass with a shading coefficient of 0.72 and an overall heat transmission coefficient of 0.54 Btu/hr-ft^2-^{o}F (Table 14) reduces the cooling loads of the superstructure to 60 to 70 percent of that for the superstructure with 1/8-inch double strength glass. This indicates that the selection of glass affects the cooling loads of a superstructure and the capacity of air-handling units as well. Hence, the glass with a smaller shading coefficient and smaller U-value reduces the operation cost of the air-handling units. On the other hand, it increases the initial cost of the superstructure. Therefore, the final decision depends upon the owner's interest and the function of the superstructure.

Cooling loads of a superstructure vary as a function of wall azimuth angles (wall orientations). The total radiative heat and solar heat gain factors at any orientation at any time of year may readily be calculated. Cooling loads for the entire superstructure are easily estimated by specifying the proper wall or window orientation. The note in Table 8 indicates the sample wall orientations.

Figures 28 to 36 show the heat gains from various sources and the total cooling load of typical rooms in the superstructure. The pattern of cooling loads in one zone is entirely different from that in another, which reinforces the concept of dividing the superstructure into several zones for the satisfactory control of indoor environment.

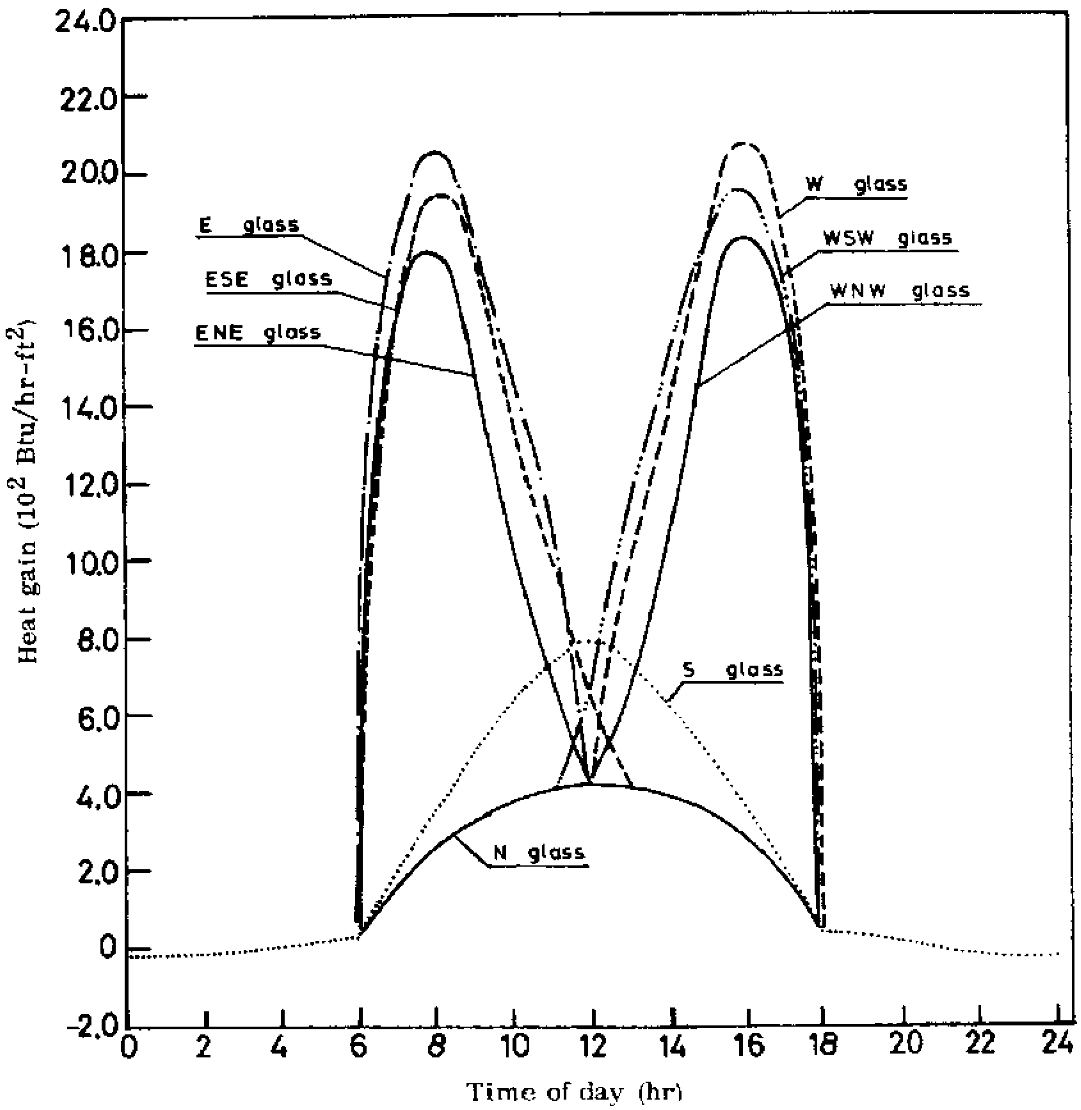


Figure 27 – Heat gain through glass.

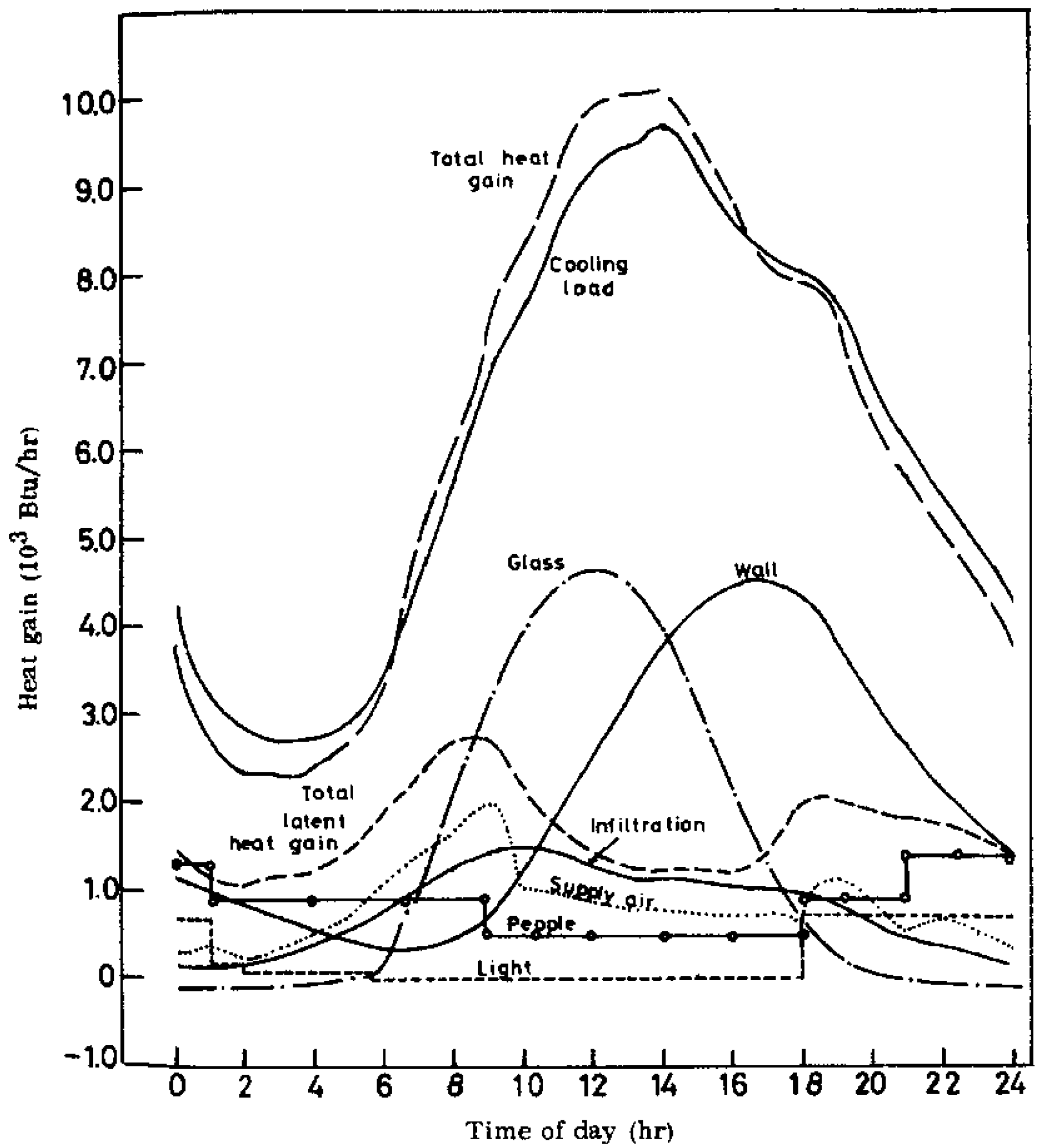


Figure 28 - Heat gains and cooling load of a typical room in superstructure zone A.

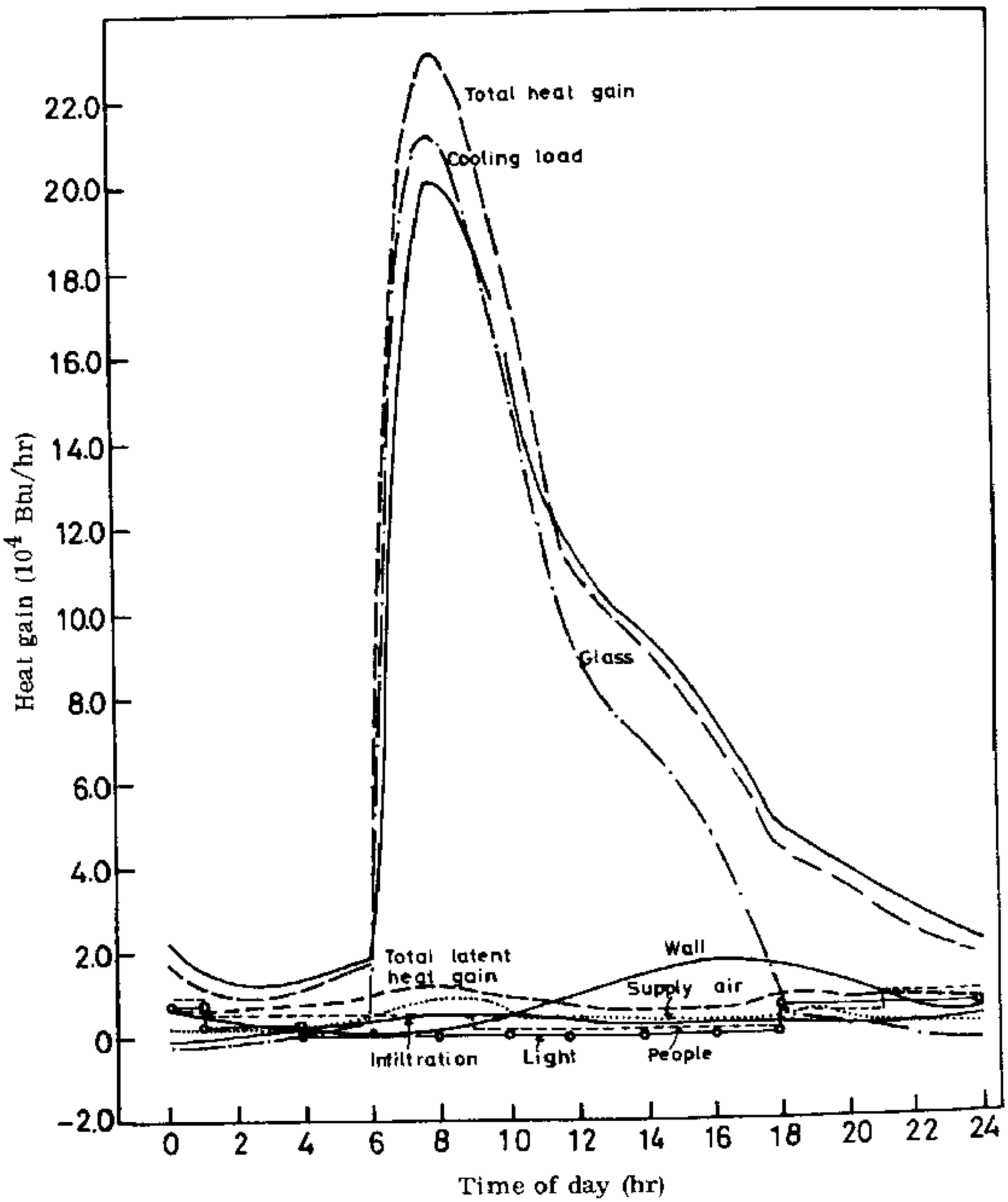


Figure 29 - Heat gains and cooling load of a typical room in superstructure zone B.

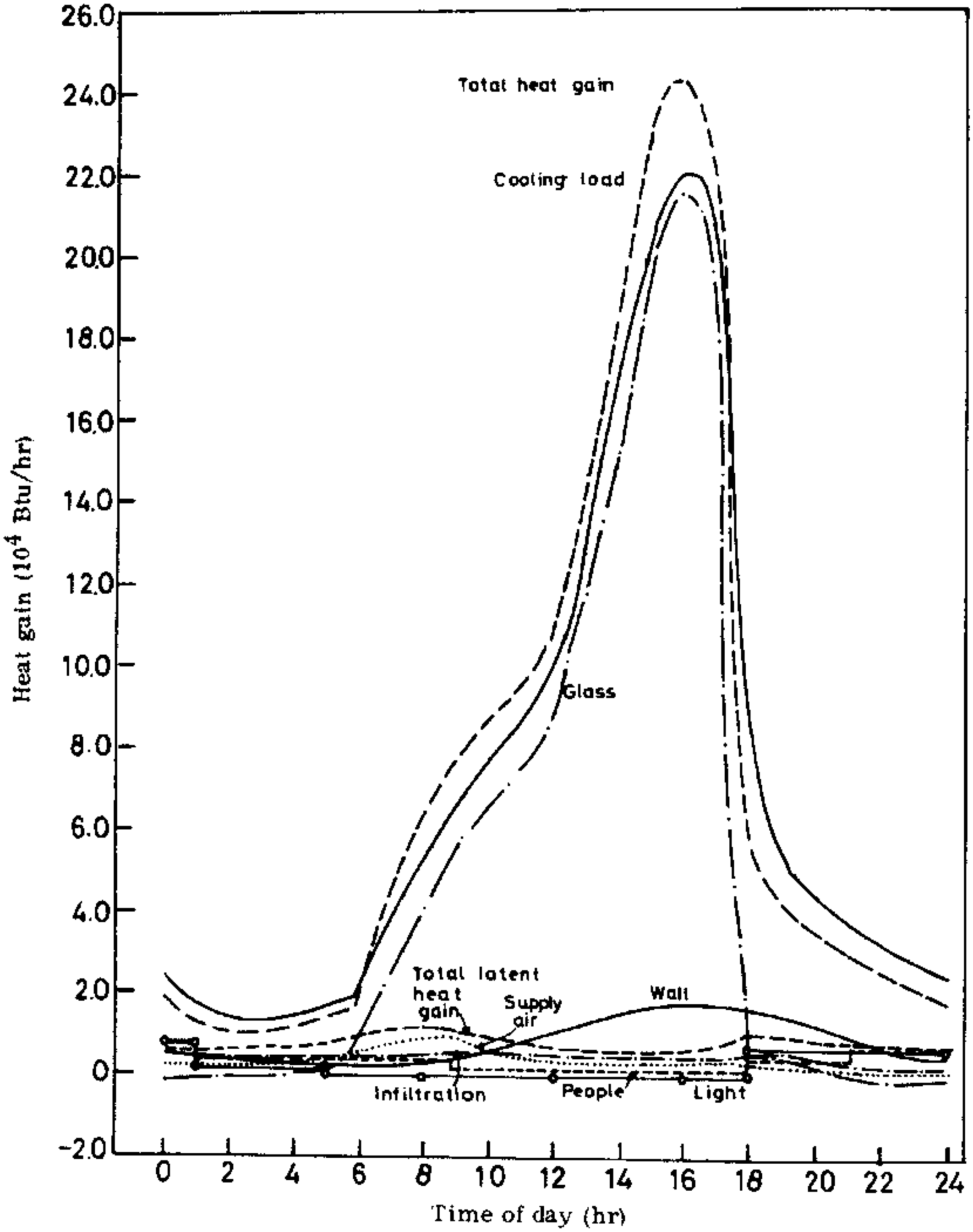


Figure 30 - Heat gains and cooling load of a typical room in superstructure zone C.

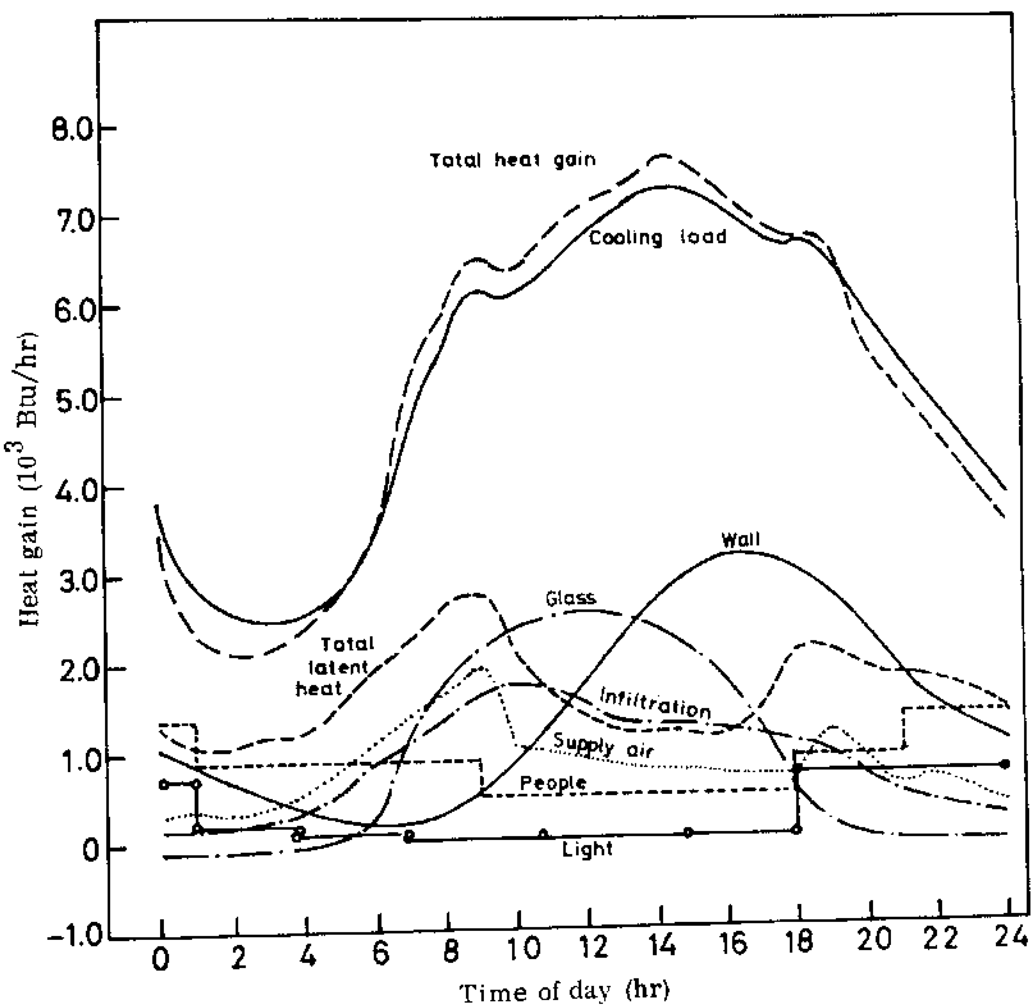


Figure 31 - Heat gains and cooling load of a typical room in superstructure zone D.

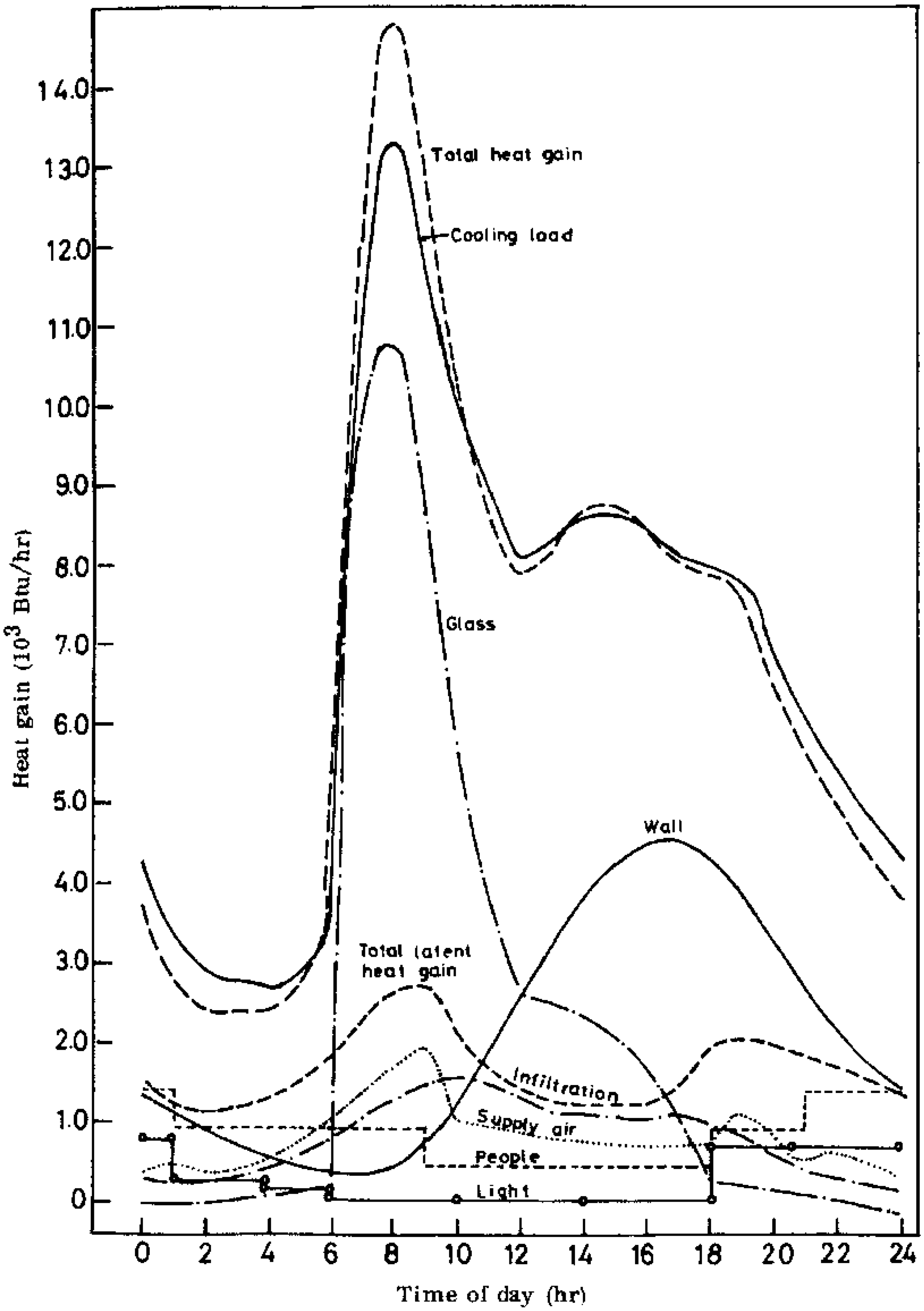


Figure 32 - Heat gains and cooling load of a typical room in superstructure zone E.

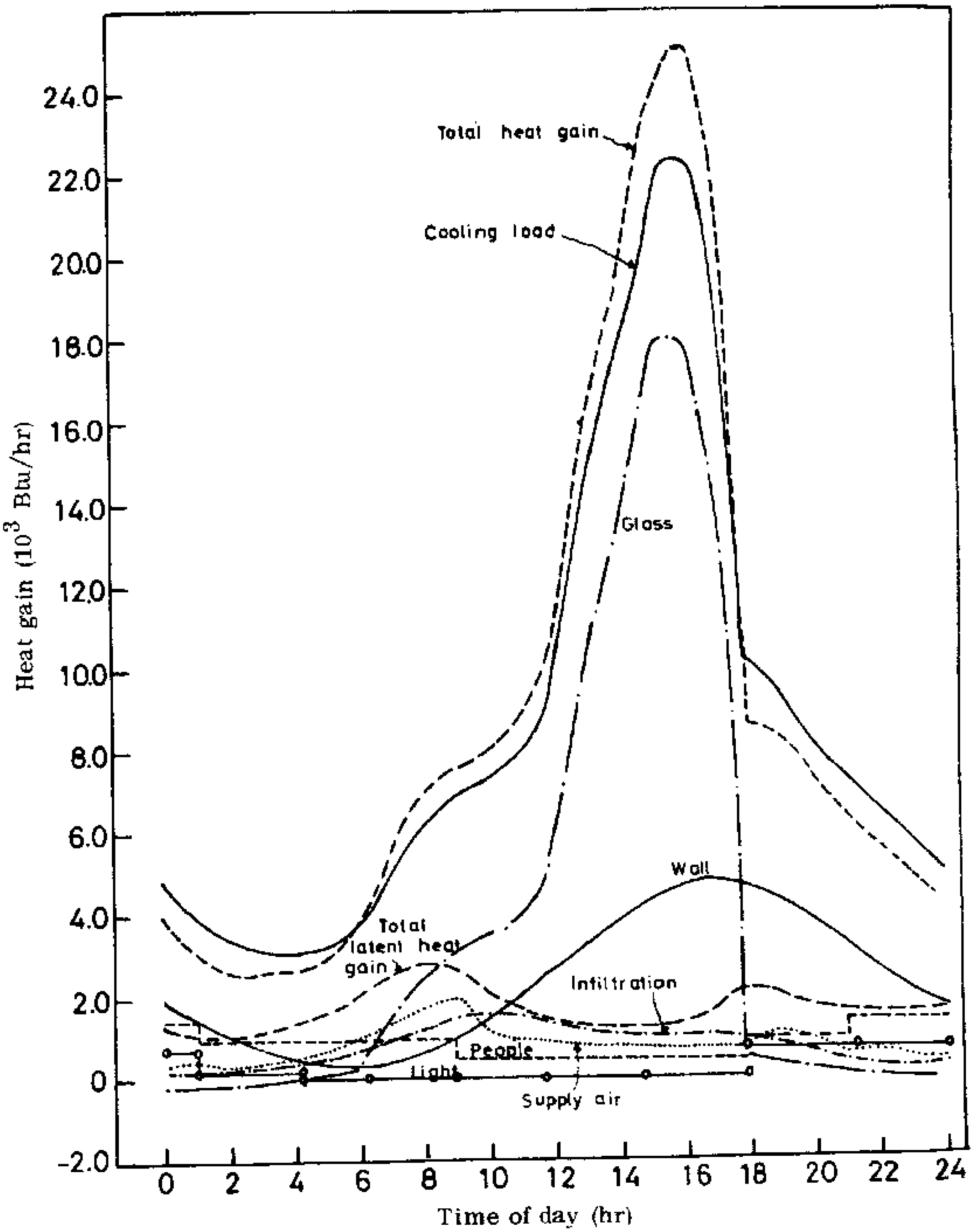


Figure 33 - Heat gains and cooling load of a typical room in superstructure zone F.

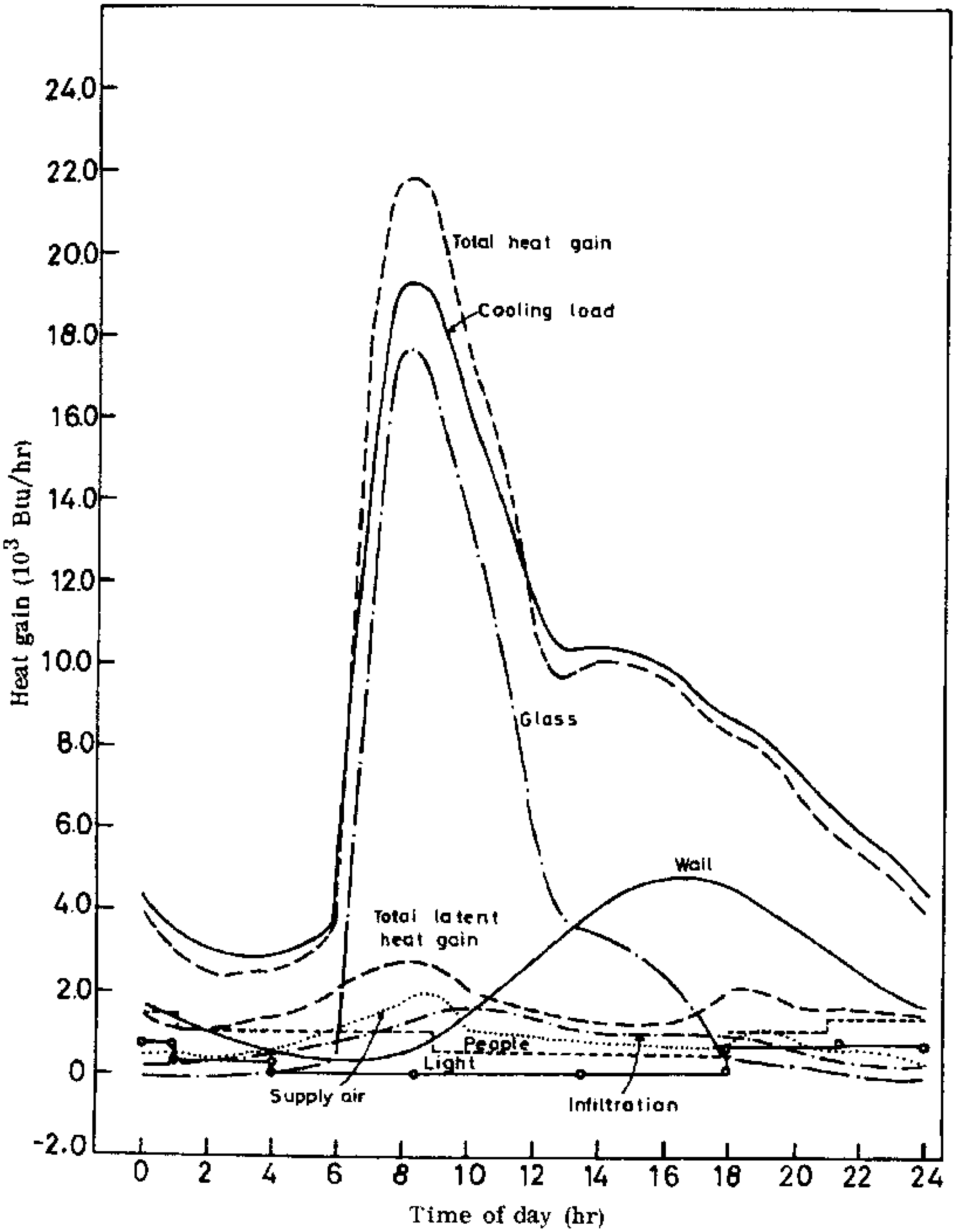


Figure 34 - Heat gains and cooling load of a typical room in superstructure zone G.

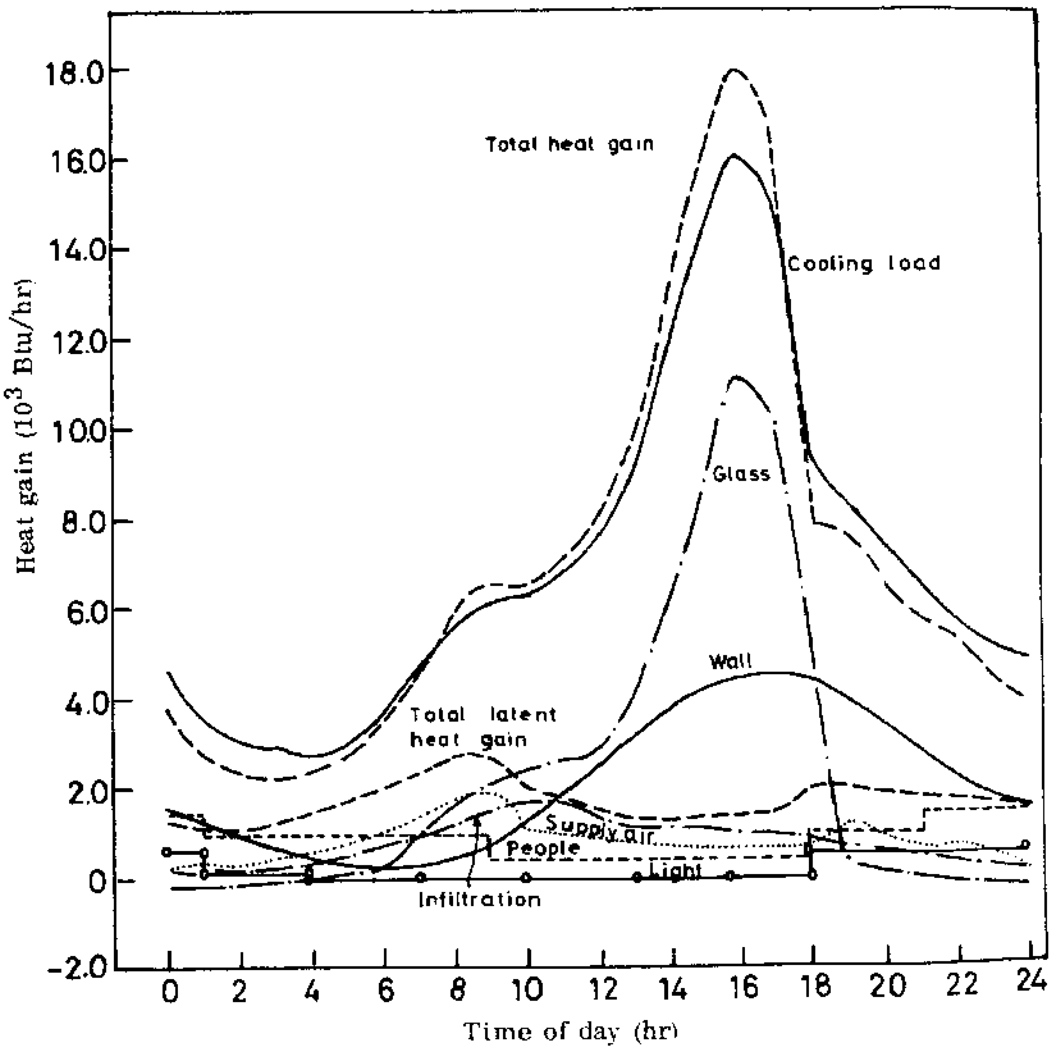


Figure 35 - Heat gains and cooling load of a typical room in superstructure zone H.

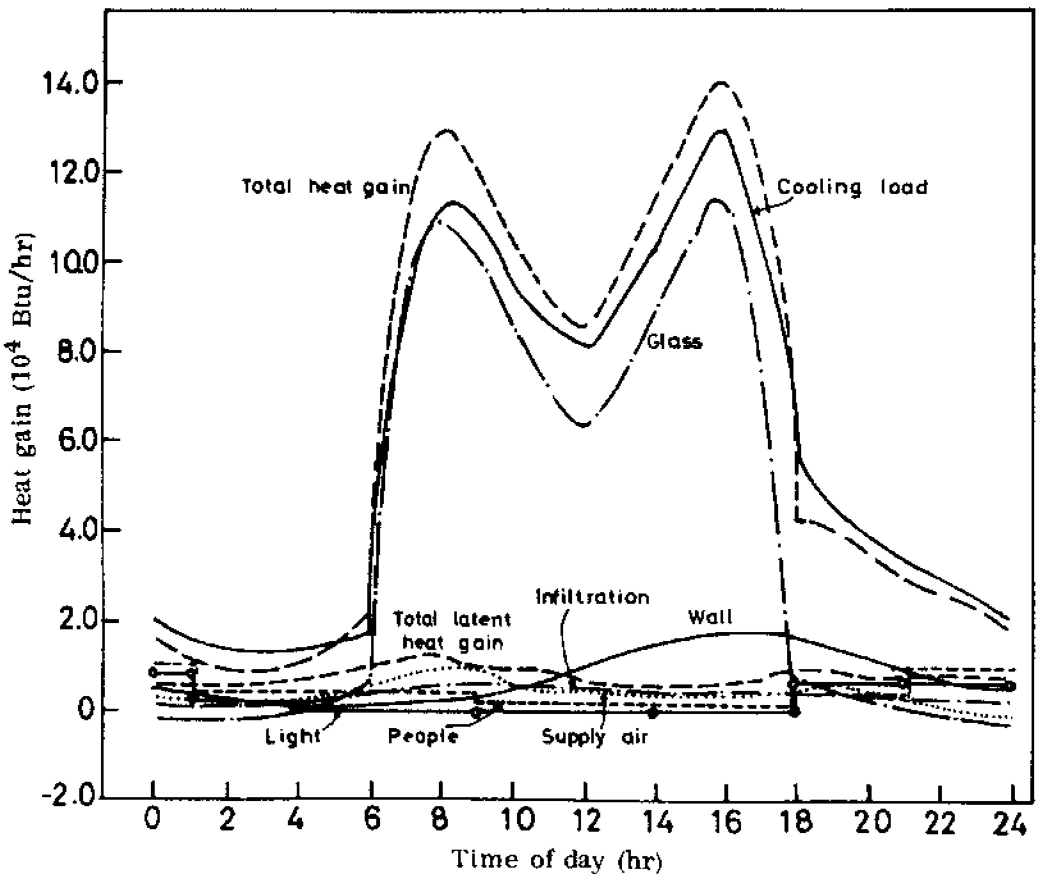


Figure 36 - Heat gains and cooling load of a typical room in superstructure zone I.

Figures 37 to 39 show the cooling load from various sources in a typical room in the buoyancy tank, in conjunction with the depth and the various wall materials. No zoning is required for a typical floor of a buoyancy tank. Heat transfer through buoyancy walls to the sea is not very significant. In fact, for steel, insulation is required.

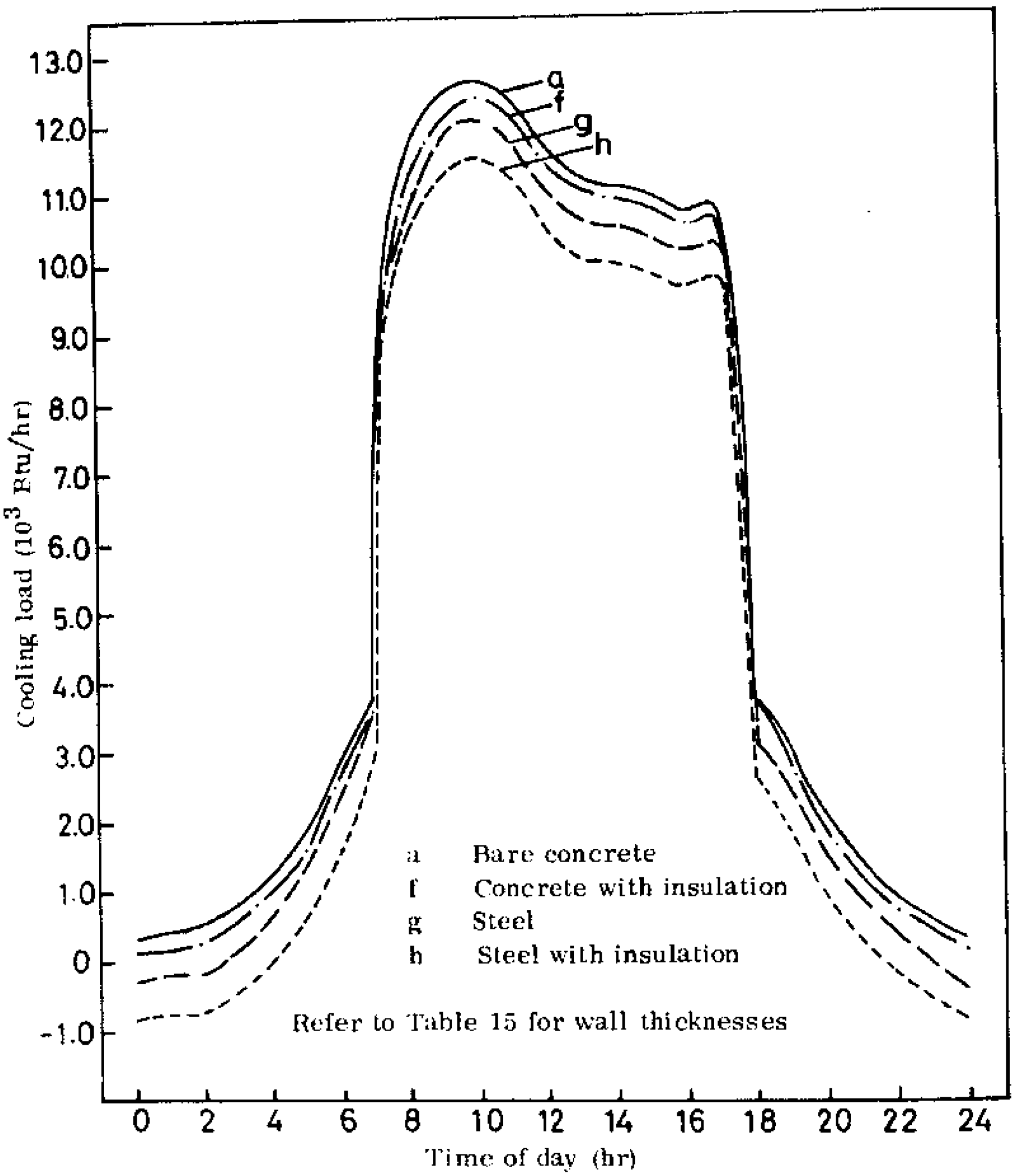


Figure 37 - Cooling load of a typical room in buoyancy tank at the depth of 70 feet from design water line.

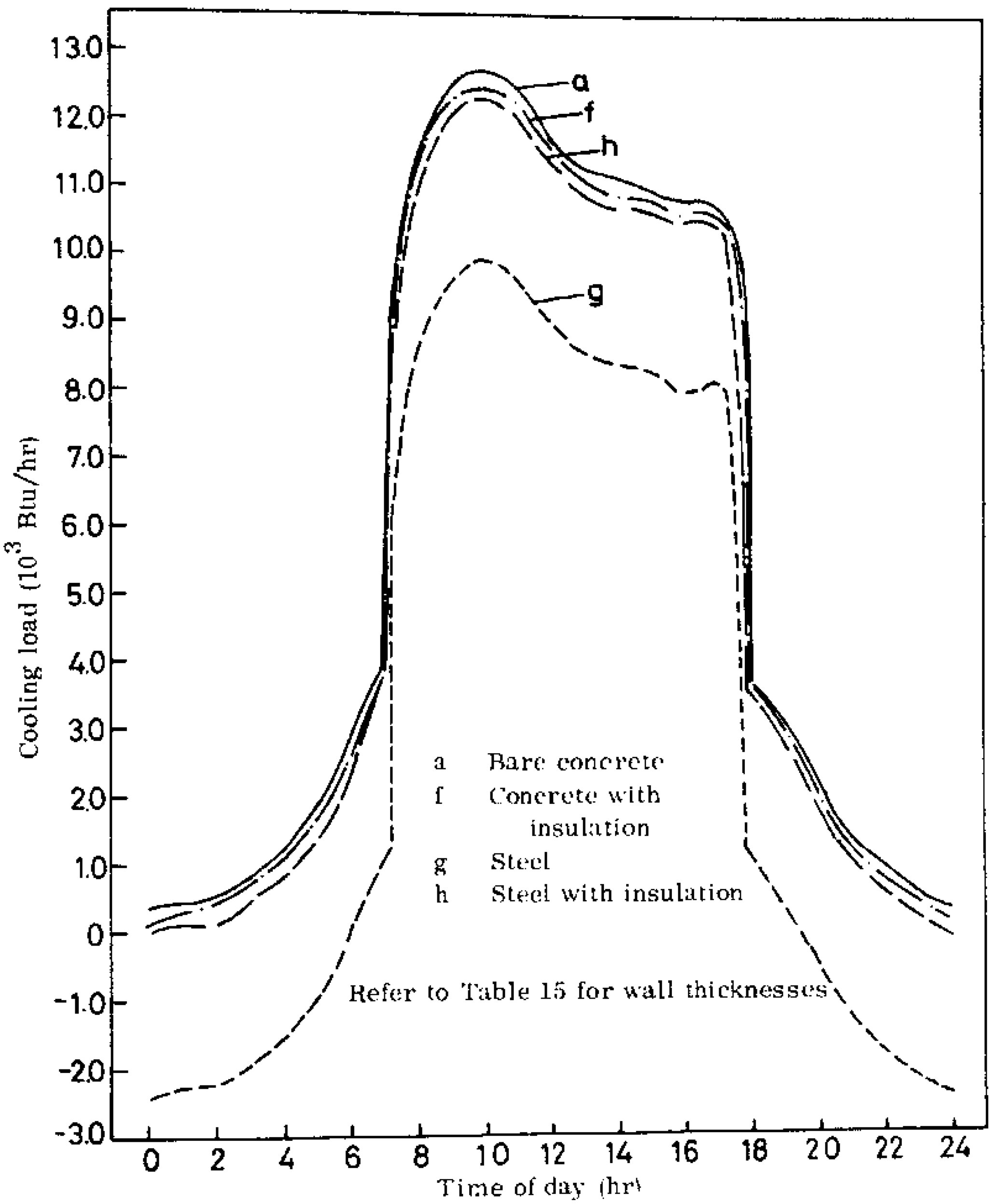


Figure 38 - Cooling load of a typical room in buoyancy tank at the depth of 105 feet from design water line.

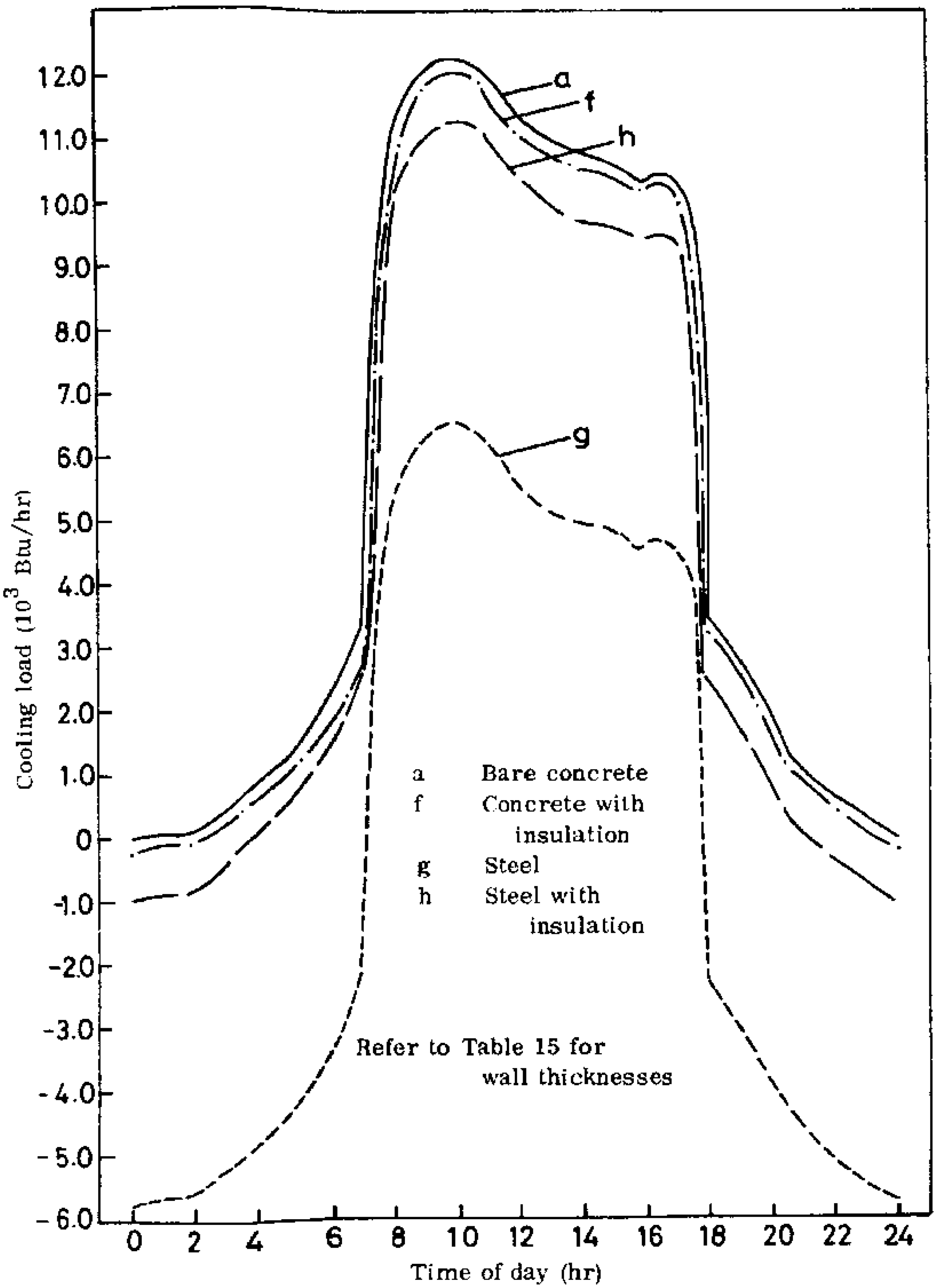


Figure 39 - Cooling load of a typical room in buoyancy tank at the depth of 140 feet from design water line.

VI. REFRIGERATION

A. Energy Considerations

Full-time air conditioning of a large occupied space often contributes a significant portion of the overall power cost for the facility. In the case of the Floating City, the increased size of the electrical power plant required to handle air conditioning demand, along with lighting, mechanical, and other uses of electricity, will have direct cost consequences for the hull itself, in increased fuel storage requirements.

It therefore seems especially important to consider the option of a total energy system. Any system that makes use of waste heat from the power plant to provide drive for air conditioning and refrigeration and for desalination of the city's drinking water will effect a gain in fuel economy. In this case, because the designer has complete control of power plant design, as well as other machinery, the total energy concept would seem to be conspicuously attractive.

B. Superstructure

Application of an absorption cycle for refrigeration is one of the convenient ways to utilize the exhaust heat from the city's power plant. Steam or hot water of approximately 200° F is fed into this cycle, and about 40° F chilled water may be obtained.

Calculations for refrigeration by absorption cycle are performed in accordance with the maximum cooling load of a typical floor, which occurs at 4 p.m. in the superstructure. Values for each zone are shown in Table 16. Water is used for the refrigerant, while lithium bromide serves as the absorbent. All the procedures for calculation of the refrigeration cycle in this section are found in the ASHRAE Handbook (10)

It is seen that the maximum cooling load of 87.43×10^4 Btu/hr should be removed by refrigeration. The refrigeration load of a typical floor is determined by

$$\text{Refrigeration load} = \frac{\text{Cooling load}}{1.2 \times 10^4} = \frac{87.43 \times 10^4}{1.2 \times 10^4} = 72.86 \text{ tons}$$

Table 16 - Cooling load of a typical floor in the superstructure at 1600

Zone	Cooling load of a room in each zone (10^4 Btu/hr)	Number of rooms in each Zone	Total cooling load of each Zone (10^4 Btu/hr)
A	0.85	7	5.95
B	7.5	1	7.50
C	21.8	1	21.80
D	0.70	3	2.10
E	0.84	10	8.40
F	2.22	4	8.88
G	1.00	4	4.00
H	1.60	10	16.00
I	12.8	1	12.80
Cooling load of a typical floor			$\sum$ 87.43

The cycle for a water-lithium bromide absorption refrigeration machine is shown in Figure 40. This cycle includes a liquid heat exchanger for the absorbent and refrigerant-absorbent streams. The design conditions listed below are selected for high performance. Figure 40 is a flow diagram for the cycle.

1. Refrigeration load: 72.86 tons/hr
2. Evaporator temperature: 40° F (this low temperature is desirable for achieving good dehumidification as well as cooling in air conditioning)
3. Absorbent outlet temperature: 90° F (The absorbent temperature should be kept at about 100° F or lower to reduce the danger of crystallization)
4. Condenser temperature: 110° F (This temperature is not critical and may be set higher than the absorber temperature to achieve better use of the cooling water)

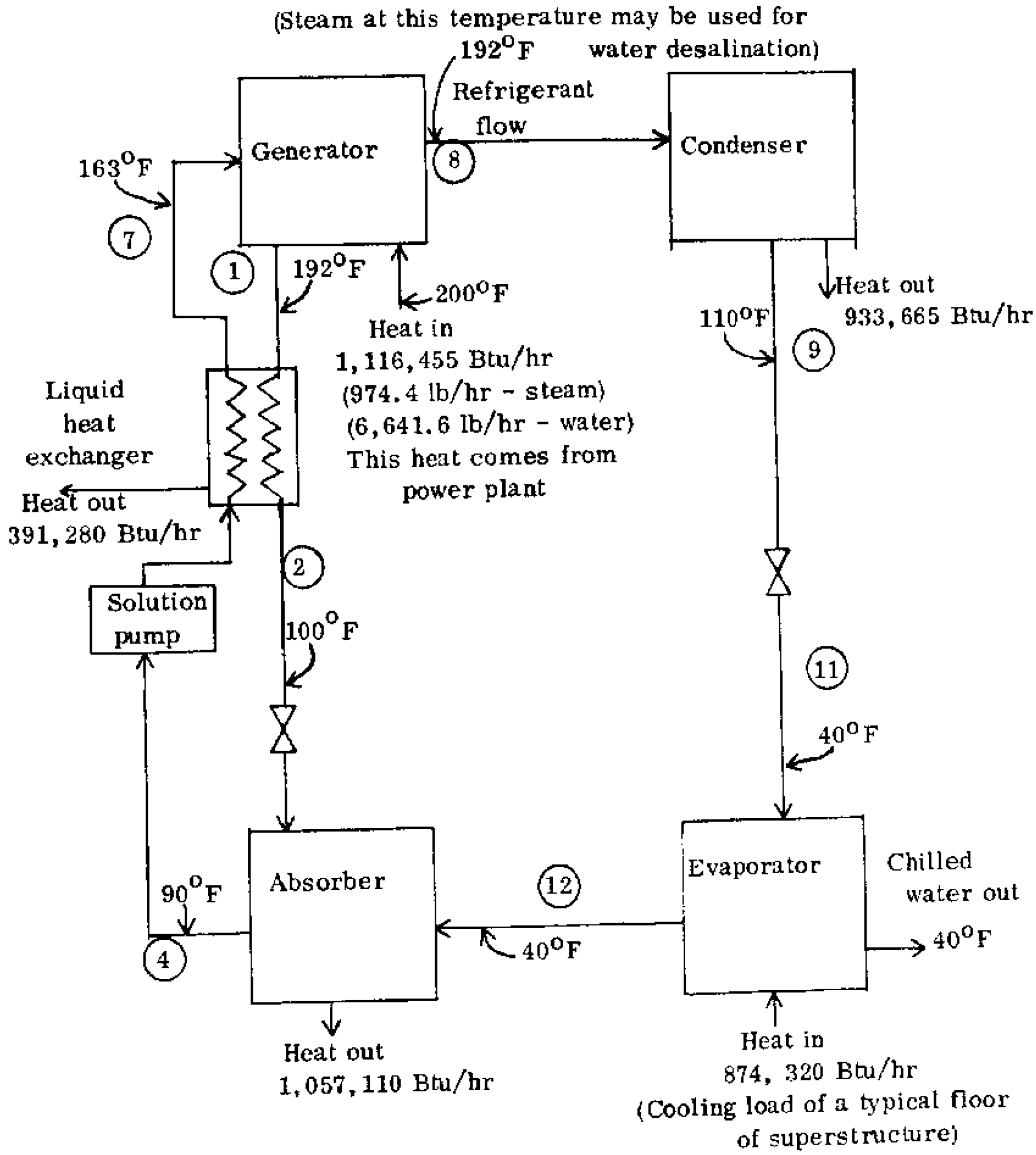


Figure 40 - Lithium bromide-water absorption refrigeration cycle for a typical floor of superstructure.

5. Generator temperature: 192^{o} F (This temperature is related to the condenser temperature in such a way that the absorbent will be in the feasible concentration range)

Pressure drops in this system are assumed to be negligible except for expansion devices. Table 17 is set up and a start is made at filling in the values. Pressures on the low side and high side are the water vapor pressures for the evaporator and condenser, respectively. Enthalpies for water and steam are found in steam tables. Concentrations and enthalpies for lithium bromide are found in the ASHRAE Handbook (10)

Table 17 - Conditions in lithium bromide cycle

Point	Temp (^{o}F)	Pressure (mm Hg)	Weight (Fraction LiBr)	Flow (lb/lb Refrigerant)	Enthalpy (Btu/lb)
1	192	66	0.61	11.2	-30
2	100	66	0.61	11.2	-70
4	90	6.3	0.56	12.2	-75
7	163	66	0.56	12.2	-38.3
8	192	66	0.0	1.0	1147
9	110	66	0.0	1.0	78
11	40	6.3	0.0	1.0	78
12	40	6.3	0.0	1.0	1079

Relative flow rates are determined from material balance as follows:

$$w_a/w_d = x_b/(x_a - x_b)$$

where w_a = flow rate of absorbent, lb/hr
w_b = flow rate of refrigerant, lb/hr
x_a = concentration of LiBr in absorbent, lb/lb solution
x_b = concentration of LiBr in refrigerant-absorbent solution, lb/lb solution

$$w_a/w_d = 0.56\ (0.61 - 0.56) = 11.2$$

$$w_b/w_d = (w_a/w_d) + 1 = 12.2$$

where w_b = flow rate of refrigerant-absorbent solution, lb/hr

The enthalpy of the refrigerant-absorbent solution leaving the liquid exchanger is calculated from energy balance as follows:

$$h_7 = h_4 + [(h_1 - h_2) \times w_a/w_b]$$

$$= -75 + [(-30) - (-70) \times 11.2/12.2]$$

$$= - 38.3 \text{ Btu/lb}$$

The temperature of 163° F, corresponding to this enthalpy, is found in the ASHRAE Handbook (10).

The refrigerant flow rate is calculated from an energy balance at the evaporator as follows:

$$w_d = \text{refrigerant load}/(h_{12} - h_{11})$$

$$= (1.2 \times 10^4 \times 72.86)/(1079 - 78)$$

$$= 873.4 \text{ lb/hr}$$

The absorbent and refrigerant-absorbent solution rates are next calculated by:

$$w_a = 11.2\ w_d = 9{,}782 \text{ lb/hr}$$

$$w_b = 12.2\ w_d = 10{,}656 \text{ lb/hr}$$

The net heat input to the generator is calculated from an energy balance as follows:

$$q_g = w_d \cdot h_g + w_a h_1 - w_b \cdot h_7$$

$$= 873.4\ (1147) + 9{,}782\ (-30) - (10{,}656)\ (-38.3)$$

$$= 1{,}116{,}455 \text{ Btu/hr}$$

Therefore, the flow rate of steam at 200^{o} F is obtained by

$$w_s = q_g/h_s = 1,116,455/1145.78 = 974.4 \text{ lb/hr}$$

and for hot water of 200^{o} F,

$$w_{H_2O} = q_g/h_{H_2O} = 1,116,455/168.1 = 6,641.6 \text{ lb/hr}$$

where w_s = flow rate of steam to generator, lb/hr
h_s = enthalpy of steam at 200^{o} F, Btu/lb
w_{H_2O} = flow rate of hot water to generator, lb/hr
h_{h_2O} = enthalpy of hot water at 200^{o}F, Btu/lb

The coefficient of performance on a net basis:

$$(COP) = \text{refrigeration load/net heat input to generator}$$

$$= 874,320 / 1,116,455 = 0.783$$

Heat transfer rates for the other components are:

Liquid heat exchanger

$$q_l = w_a(h_1 - h_2) = 9,782 [-30 - (-70)] = 391,280 \text{ Btu/hr}$$

Condenser

$$-q_c = w_d(h_8 - h_9) = 873.4 (1147 - 78) = 933,665 \text{ Btu/hr}$$

Absorber

$$-q_a = q_g + q_e + q_c$$
$$= 1,116,455 + 874,320 - 933,665 = 1,057,110 \text{ Btu/hr}$$

where q_e = refrigeration load.

Results are illustrated in Figure 40.

C. Buoyancy Tank

Following the same procedure and design conditions, the refrigeration process for a typical floor in the buoyancy tank has been calculated in accordance with the different depths and construction materials. Temperatures at each process are kept the same as those for the superstructure. Resultant values are presented in Table 18.

Table 18 - Heat transfer rate for buoyancy tank

Depth (ft)	Wall description*	Max. cooling load (Btu/hr)	Refrigeration load (tons)	Refrigerant flow rate (lb/hr)	Flow rate of steam (lb/hr)	Heat transfer rate			
						Generator (Btu/hr)	Heat exchanger (Btu/hr)	Condenser (Btu/hr)	Absorber (Btu/hr)
70	a	49,572	4.13	49.5	55.2	63,290	22,160	52,916	59,946
	f	50,300	4.19	50.2	56.0	64,159	22,480	53,664	60,795
	g	45,856	3.82	45.8	51.1	58,553	20,520	48,960	55,449
	h	48,112	4.01	48.1	53.4	61,483	21,560	51,419	58,176
105	a	49,684	4.14	49.6	55.3	63,382	22,240	53,022	60,044
	f	50,264	4.19	50.2	56.0	64,159	22,480	59,864	54,559
	g	39,092	3.26	39.0	43.5	49,854	17,480	46,502	42,444
	h	48,588	4.05	48.6	54.2	62,136	21,760	51,953	58,771
140	a	48,332	4.03	48.3	53.9	61,729	21,640	51,633	58,428
	f	49,172	4.10	49.2	54.9	62,882	22,040	52,595	59,459
	g	25,856	2.15	25.8	28.8	32,988	11,560	27,580	31,264
	h	45,332	3.78	45.3	50.5	57,929	20,280	48,426	54,835

*from Table 15

VII. DESIGN OF DUCT SIZE

A. Superstructure

Ducting is one of the most convenient ways to convey the conditioned or ventilation air into a room. Duct size changes in accordance with the function and cooling load of a building and the selection of the air conditioning system. Maximum cooling load differs from one room to another, due to the different orientation of the walls and windows, and the time of day. The application of a zoning system appears reasonable for the design of air ducts. A multi-zone unit system may be best suited for the superstructure, since it can distribute the conditioned air into many different zones in response to the individual thermal requirements of each room. The size of the duct should be based on the maximum cooling load of each zone.

The method used for the calculations of duct size in this section, proposed by the Handbook of Air Conditioning System Design (9), is commonly used. Necessary design conditions for the calculations are based on the values described in Section V. However, the important values for the calculations of duct size are given as follows:

- Air temperature of a room — 78^{o} F
- Specific humidity of a room — 0.0125 lb/lb
- Quantity of ventilation air for different zones in a typical floor (cfm_{oa})

Zone A	420 cfm
B	300
C	300
D	180
E	600
F	240
G	240
H	600
I	300

- Bypass factor (BF) — 0.15
  (the ratio between the quantity of air which doesn't contact the surface of cooling or heating coils and total amount of supplied air to these coils)

By following the procedures proposed by the Handbook of Air Conditioning System Design, calculations of the required quantity of supply air have been made. These are based on the time of day when the maximum cooling load occurs, which may be different for each zone.

For Zone A (at 2 p.m.):

Effective room sensible heat (ERSH)

$= \text{Room sensible heat (RSH)} + (\text{BF}) \times 1.08 \times \text{cfm}_{oa} \times (\theta_o - \theta_i)$

$= 8732 + (0.15) \times 1.08 \times 420 \times (88.60 - 78.0)$

$= 9,453$ Btu/hr

Effective room latent heat (ERLH)

$= \text{Room latent heat (RLH)} + (\text{BF}) \times 1.08 \times \text{cfm}_{oa} \times (W_o - W_i)$

$= 1268 + (0.15) \times 1.08 \times 420 \times (0.0154 - 0.0125)$

$= 1,268$ Btu/hr

Effective sensible heat factor (ESHF)

$\text{ERSH}/(\text{ERSH} + \text{ERLH}) = 9,453/(9,453 + 1,268) = 0.88$

Air-handling units should be kept dry mainly for the prevention of corrosion. Apparatus dew-point temperature (θ_{adp}) indicates the minimum allowable temperature to prevent the condensation of moist air. As the latent load for Zone A is not large in comparison with the sensible load for the same space, application of a dehumidification and cooling system is a suitable method (Figure 41).

Conditions of supply air change in accordance with the selection of the bypass factor (BF), outside and inside air conditions, mixing ratio between outside and inside air, selection of cooling coils and cooling load of a room. The mixing ratio, BF, and cooling coils may be selected arbitrarily to satisfy the thermal requirements of an individual room. Since Zone A is used for guest room purposes, a BF of 0.15 is recommended. The conditions of air at different points in Figure 41 can be found by the following formulas:

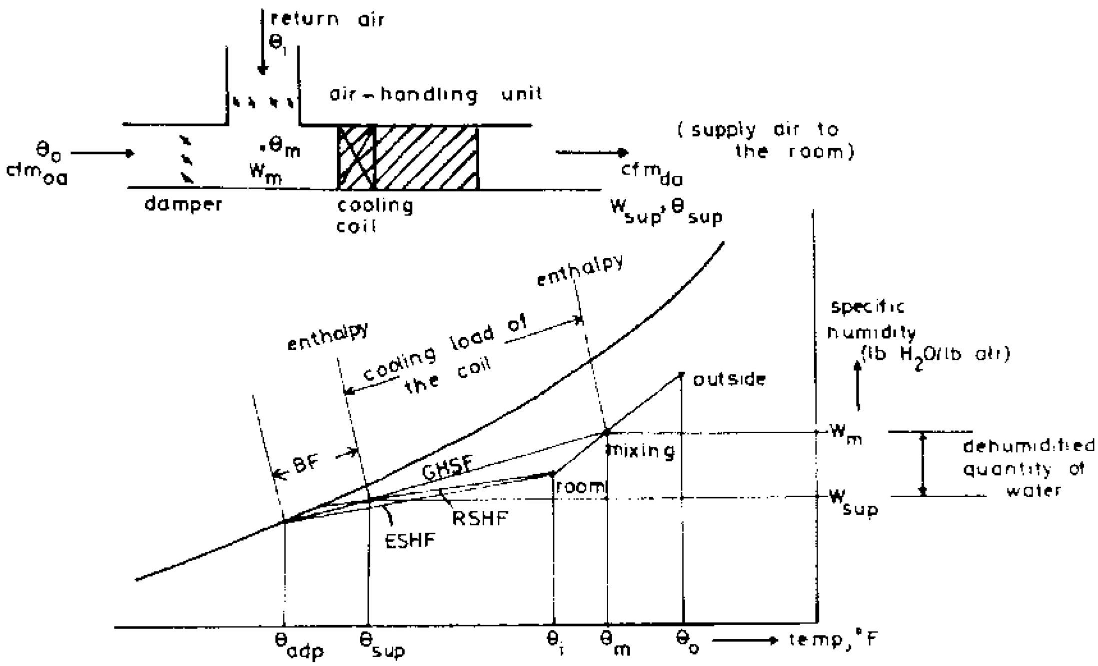


Figure 41 - Dehumidification and cooling system for the superstructure.

RSHF (Room sensible heat factor)

$$= \frac{RSH + (BF)\,[1.08 \times cfm_{oa}(\theta_o - \theta_i)]}{RSH + (BF)\,[1.08 \times cfm_{oa}(\theta_o - \theta_i)] - 0.68 \times cfm_{da}(W_i - W_{sup})}$$

$$= RSH/(RSH + RLH)$$

GSHF (Grand sensible heat factor)

$$= \frac{RSH + 1.08 \times cfm_{oa}(\theta_o - \theta_i)}{RSH + 1.08 \times cfm_{oa}\,(\theta_o - \theta_i) + RLH - 0.68 \times cfm_{oa}(W_o - W_i)}$$

$$= (RSH + OASH)/[(RSH + OASH) + (RLH + OALH)]$$

$$= TSH/(TSH + TLH)$$

To facilitate the above process, resultant values for Zone A are illustrated in Figure 42.

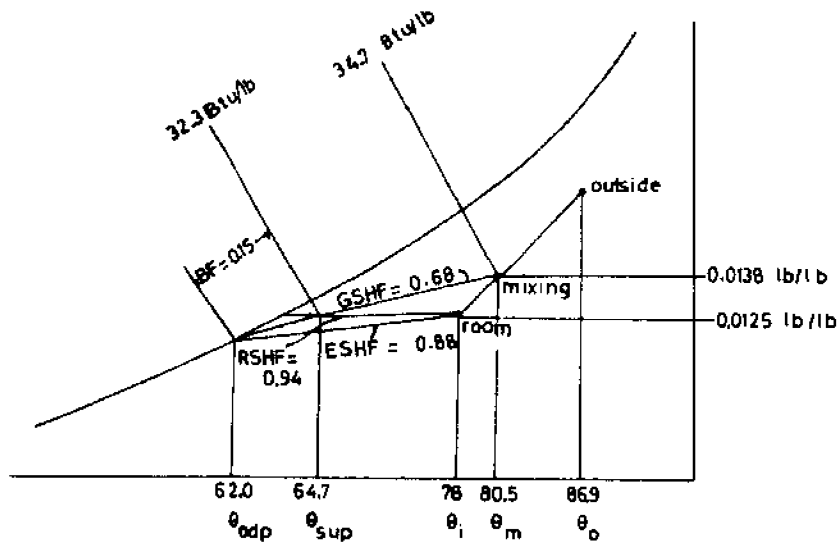


Figure 42 - Condition of air for Zone A.

The required quantity of supply air (cfm_{da}) for Zone A is estimated as follows·

$$cfm_{da} = ERSH/[1.08 \times (1 - BF)(\theta_i - \theta_{adp})]$$

$$= 9453/[1.08 \times (1 - 0.15)(78 - 62.0)]$$

$$= 686.5 \text{ cfm}$$

In the same manner, the required quantities of supply air for the other zones have been calculated and are shown in Table 19.

To fulfill the function of ducting in a practical manner, the system must be designed within acceptable ranges of the following factors: friction, velocity, sound level, heat and leakage losses. Upon considering these items, sound level, heat and leakage losses can be neglected by selecting the proper duct size and insulation materials. Leakage loss is unpredictable, as it depends on quality of workmanship. Application of low velocity, 1200 - 2500 fpm, is suited for commercial comfort air conditioning, such as in offices, hotels, schools, and apartment houses. High velocity is applicable for factory use. At a constant low pressure, such as 0.15 inch water gage at all terminals, the recommended velocity is found in

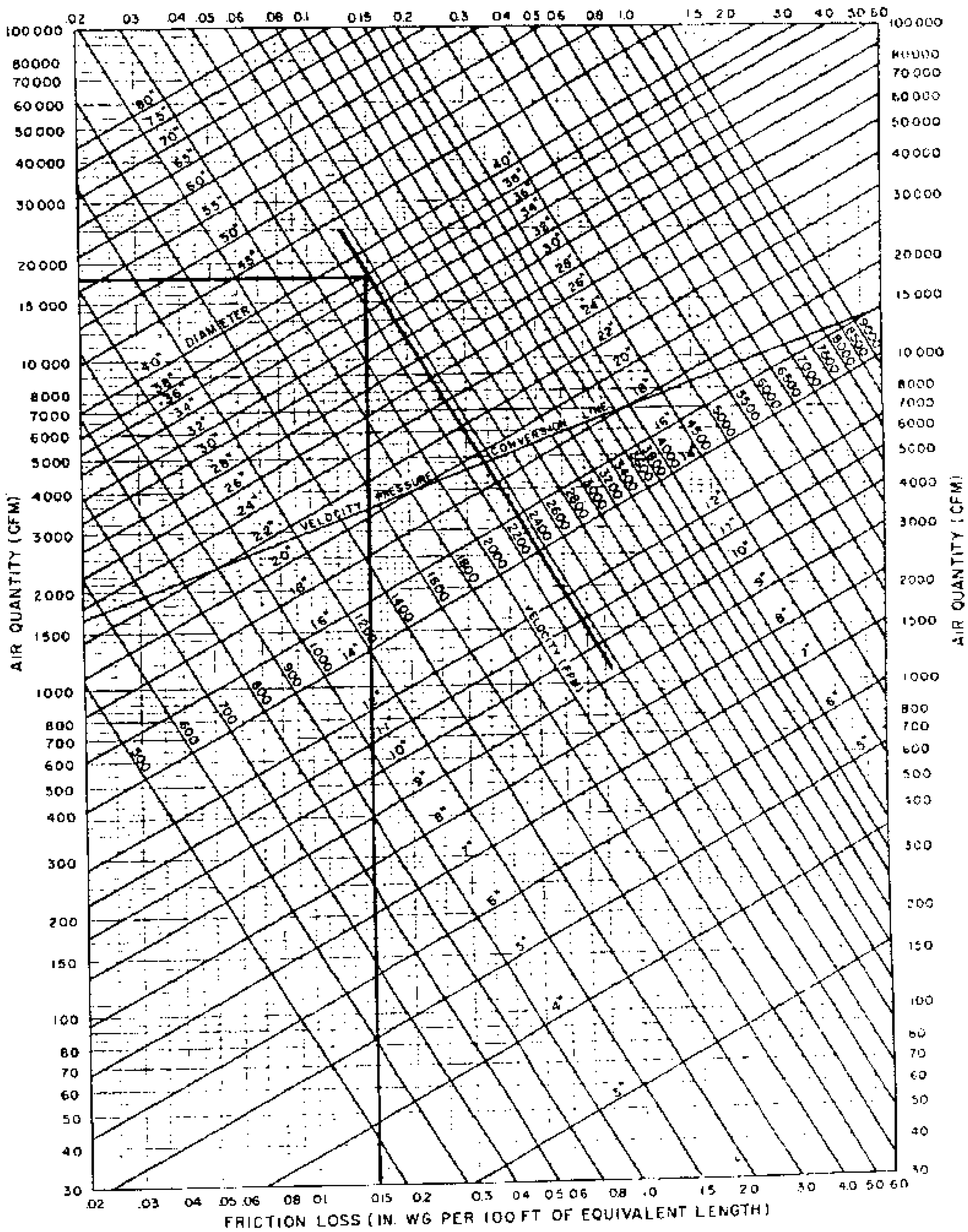


Figure 43 - Determination of air velocity in ducts for Zone A.

Table 19 – Conditions of air and the quantity of supply air for each zone in the superstructure

Zones	ERSH	ERLH	ERSH + ERLH	ERSH/ (ERSH+ERLH)	θ_{adp}	Required air quantity	Time of day
A	9,453	1,268	10,721	0.88	62	687	1400
B	190,092	12,200	202,292	0.94	64	14,791	0800
C	207,233	12,200	219,433	0.94	63.8	15,892	1600
D	6,214	1,305	7,519	0.83	62.5	437	1500
E	11,461	2,622	14,083	0.81	62.6	811	0800
F	16,911	2,622	19,533	0.87	63.3	1,253	0800
G	21,428	1,268	22,696	0.94	63.8	1,644	1600
H	15,817	1,268	17,085	0.93	63.6	1,197	1600
I	123,204	5,729	128,933	0.96	63.8	9,451	1600

conjunction with the quantity of supply air (Figure 43). In the process of determining the duct size, we must find the cfm percentage, duct area percentage, and duct area. They are estimated by the following procedures:

cfm percentage $\dfrac{\text{Local supply air quantity}}{\text{Initial supply air quantity}} \times 100$

Duct area percentage This is found in Table 20 in conjunction with cfm percentage

Duct area Initial duct area x duct area percentage

Duct sizes are then found in Table 21.

Therefore, the remaining important values for Zone A are estimated by following the above procedures. By taking into consideration Figure 43, an example calculation for Zone A of direction B – 1 is used for the determination of duct size. In addition, ducting for Zone A is considered to be one of the branches from the main duct, which takes care of

Table 20 - Percent section area in branches for maintaining equal friction

CFM CAPACITY %	DUCT AREA %	CFM CAPACITY %	DUCT AREA %	CFM CAPACITY %	DUCT AREA %	CFM CAPACITY %	DUCT AREA %
1	2.0	26	33.5	51	59.0	76	81.0
2	3.5	27	34.5	52	60.0	77	82.0
3	5.5	28	35.5	53	61.0	78	83.0
4	7.0	29	36.5	54	62.0	79	84.0
5	9.0	30	37.5	55	63.0	80	84.5
6	10.5	31	39.0	56	64.0	81	85.5
7	11.5	32	40.0	57	65.0	82	86.0
8	13.0	33	41.0	58	65.5	83	87.0
9	14.5	34	42.0	59	66.5	84	87.5
10	16.5	35	43.0	60	67.5	85	88.5
11	17.5	36	44.0	61	68.0	86	89.0
12	18.5	37	45.0	62	69.0	87	90.0
13	19.5	38	46.0	63	70.0	88	90.5
14	20.5	39	47.0	64	71.0	89	91.5
15	21.5	40	48.0	65	71.5	90	92.0
16	23.0	41	49.0	66	72.5	91	93.0
17	24.0	42	50.0	67	73.5	92	94.0
18	25.0	43	51.0	68	74.5	93	94.5
19	26.0	44	52.0	69	75.5	94	95.0
20	27.0	45	53.0	70	76.5	95	96.0
21	28.0	46	54.0	71	77.0	96	96.5
22	29.5	47	55.0	72	78.0	97	97.5
23	30.5	48	56.0	73	79.0	98	98.0
24	31.5	49	57.0	74	80.0	99	99.0
25	32.5	50	58.0	75	80.5	100	100.0

Zones A, C and D. Initial supply air quantity is the sum of these zones, namely 18,520 cfm.

Initial duct velocity = found from Figure 43 in conjunction with the friction loss of 0.15 in. water gage and the initial supply air quantity of 18,520 cfm

Initial duct area = 18,520/2,300 = 8.05 ft^2

cfm percentage = (3,500/18,520) x 100 = 18.9

Duct area percentage = 25.9 (found from Table 20 by interpolation of cfm capacity)

Duct area = initial duct area x duct area percentage

= 8.05 x 0.259 = 2.08 ft^2

Table 21 - Duct dimensions, section area, circular equivalent diameter, and duct class

SIDE	6		8		10		12		14		16		18		20		22	
	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.
10	.39	8.4	.52	9.8	.65	10.9												
12	.45	9.1	.62	10.7	.77	11.9	.94	13.1										
14	.52	9.8	.72	11.5	.91	12.9	1.09	14.2	1.28	15.3								
16	.59	10.4	.81	12.2	1.02	13.7	1.24	15.1	1.45	16.3	1.67	17.5						
18	.66	11.0	.91	12.9	1.15	14.5	1.40	16.0	1.63	17.3	1.87	18.5	2.12	19.7				
20	.72	11.5	.99	13.5	1.26	15.2	1.54	16.8	1.81	18.2	2.07	19.5	2.34	20.7	2.61	21.9		
22	.78	12.0	1.08	14.1	1.38	15.9	1.69	17.6	1.99	19.1	2.27	20.4	2.57	21.7	2.86	22.9	3.17	24.1
24	.84	12.4	1.16	14.6	1.50	16.6	1.83	18.3	2.14	19.8	2.47	21.3	2.78	22.6	3.11	23.9	3.43	25.1
26	.89	12.8	1.26	15.2	1.61	17.2	1.97	19.0	2.31	20.6	2.66	22.1	3.01	23.5	3.35	24.8	3.71	26.1
28	.95	13.2	1.33	15.6	1.71	17.7	2.09	19.6	2.47	21.3	2.86	22.9	3.25	24.4	3.60	25.7	4.00	27.1
30	1.01	13.6	1.41	16.1	1.82	18.3	2.22	20.2	2.64	22.0	3.06	23.7	3.46	25.2	3.89	26.7	4.27	28.0
32	1.07	14.0	1.48	16.5	1.93	18.8	2.36	20.8	2.81	22.7	3.25	24.4	3.68	26.0	4.12	27.5	4.55	28.9
34	1.13	14.4	1.58	17.0	2.03	19.3	2.49	21.4	2.96	23.3	3.43	25.1	3.89	26.7	4.37	28.3	4.81	29.7
36	1.18	14.7	1.65	17.4	2.14	19.8	2.61	21.9	3.11	23.9	3.63	25.8	4.09	27.4	4.58	29.0	5.07	30.5
38	1.23	15.0	1.73	17.8	2.25	20.3	2.76	22.5	3.27	24.5	3.80	26.4	4.30	28.1	4.84	29.8	5.37	31.4
40	1.28	15.3	1.81	18.2	2.33	20.7	2.88	23.0	3.43	25.1	3.97	27.0	4.52	28.8	5.07	30.5	5.62	32.1
42	1.33	15.6	1.86	18.5	2.43	21.1	2.98	23.4	3.57	25.6	4.15	27.6	4.71	29.4	5.31	31.2	5.86	32.8
44	1.38	15.9	1.95	18.9	2.52	21.5	3.11	23.9	3.71	26.1	4.33	28.2	4.90	30.0	5.55	31.9	6.12	33.5
46	1.43	16.2	2.01	19.2	2.61	21.9	3.22	24.3	3.85	26.7	4.49	28.7	5.10	30.6	5.76	32.5	6.37	34.2
48	1.48	16.5	2.09	19.6	2.71	22.3	3.35	24.8	4.03	27.2	4.65	29.2	5.30	31.2	5.97	33.1	6.64	34.9
50			2.16	19.9	2.81	22.7	3.46	25.2	4.15	27.6	4.84	29.8	5.51	31.8	6.19	33.7	6.87	35.5
52			2.22	20.2	2.91	23.1	3.57	25.6	4.30	28.1	5.00	30.3	5.72	32.4	6.41	34.3	7.14	36.0
54			2.29	20.5	2.98	23.4	3.71	26.1	4.43	28.5	5.17	30.8	5.90	32.9	6.64	34.9	7.38	36.8
56			2.38	20.9	3.09	23.8	3.83	26.5	4.55	28.9	5.31	31.2	6.08	33.4	6.87	35.5	7.62	37.4
58			2.43	21.1	3.19	24.2	3.94	26.9	4.68	29.3	5.48	31.7	6.26	33.9	7.06	36.0	7.87	38.0
60			2.50	21.4	3.27	24.5	4.06	27.3	4.84	29.8	5.65	32.2	6.50	34.5	7.26	36.5	8.13	38.6
64			2.64	22.0	3.46	25.2	4.24	27.9	5.10	30.6	5.91	33.1	6.87	35.5	7.71	37.6	8.59	39.7
68					3.63	25.8	4.49	28.7	5.37	31.4	6.26	33.9	7.18	36.3	8.12	38.6	9.03	40.7
72					3.83	26.5	4.71	29.4	5.69	32.3	6.60	34.8	7.54	37.2	8.50	39.5	9.52	41.8
76					4.09	27.4	4.91	30.0	5.86	32.8	6.83	35.4	7.95	38.2	8.90	40.4	9.98	42.8
80					4.15	27.6	5.17	30.8	6.15	33.6	7.22	36.4	8.29	39.0	9.21	41.1	10.4	43.8
84							5.41	31.5	6.41	34.5	7.54	37.2	8.55	39.6	9.75	42.3	10.8	44.6
88							5.58	32.0	6.64	34.9	7.87	38.0	8.94	40.5	10.1	43.1	11.2	45.4
92							5.79	32.6	6.91	35.6	8.12	38.6	9.39	41.5	10.4	43.8	11.7	46.3
96							5.90	33.0	7.14	36.2	8.40	39.3	9.70	42.2	10.8	44.5	12.1	47.2
100									7.40	36.9	8.50	39.5	9.80	42.5	11.3	45.5	12.3	47.6
104									7.60	37.4	8.90	40.5	10.3	43.5	11.6	46.2	13.0	48.6
108									7.90	38.0	9.20	41.2	10.6	44.0	12.0	47.0	13.4	49.6
112									8.10	38.6	9.[illegible]0	41.8	10.9	44.7	12.3	47.5	13.8	50.3
116											9.80	42.4	11.3	45.5	12.6	48.1	14.3	51.3
120											10.0	42.8	11.5	46.0	13.1	49.1	14.4	51.5
124											10.3	43.5	11.9	46.7	13.4	49.6	15.0	52.3
128											10.6	44.1	12.1	47.1	13.8	50.4	15.5	53.3
132													12.5	47.9	14.1	50.9	15.8	53.9
136													12.8	48.5	14.5	51.6	16.2	54.5
140													13.0	48.8	14.7	52.0	16.5	55.0
144													13.3	49.4	15.2	52.9	16.8	55.6

*Circular equivalent diameter (d_e). Calculated from $d_e = 1.3 \frac{(ab)^{.625}}{(a+b)^{.25}}$

†Large numbers in table are duct class.

Table 21 - Duct dimensions, section area, circular equivalent diameter, and duct class (cont.)

SIDE	24		26		28		30		32		34		36		38		40	
	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.	Area sq ft	Diam in.
10																		
12																		
14																		
16																		
18																		
20																		
22																		
24	3.74	26.2																
26	4.03	27.2	4.40	28.4														
28	4.33	28.2	4.74	29.5	5.10	30.6												
30	4.68	29.3	5.07	30.5	5.44	31.6	5.86	32.8										
32	4.94	30.1	5.37	31.4	5.79	32.6	6.23	33.8	6.68	35.0								
34	5.24	31.0	5.69	32.3	6.15	33.6	6.60	34.8	7.06	36.0	7.54	37.2						
36	5.58	32.0	5.94	33.0	6.52	34.6	6.99	35.8	7.46	37.0	7.95	38.2	8.46	39.4				
38	5.86	32.8	6.38	34.2	6.87	35.5	7.34	36.7	7.87	38.0	8.37	39.2	8.89	40.4	9.43	41.6		
40	6.15	33.6	6.71	35.1	7.22	36.4	7.71	37.6	8.29	39.0	8.81	40.2	9.34	41.4	9.89	42.6	10.5	43.8
42	6.45	34.4	7.03	35.9	7.58	37.3	8.12	38.6	8.68	39.9	9.21	41.1	9.80	42.4	10.4	43.6	11.0	44.8
44	6.75	35.2	7.34	36.7	7.91	38.1	8.50	39.5	9.07	40.8	9.61	42.0	10.3	43.4	10.8	44.6	11.4	45.8
46	7.03	35.9	7.63	37.4	8.25	38.9	8.85	40.3	9.48	41.7	10.1	43.0	10.7	44.3	11.3	45.6	11.9	46.8
48	7.30	36.6	7.95	38.2	8.59	39.7	9.25	41.2	9.89	42.6	10.5	43.9	11.1	45.2	11.8	46.5	12.4	47.8
50	7.58	37.3	8.25	38.9	8.90	40.4	9.61	42.0	10.3	43.5	10.9	44.8	11.6	46.1	12.2	47.4	13.0	48.8
52	7.87	38.0	8.55	39.6	9.25	41.2	9.98	42.8	10.7	44.3	11.4	45.7	12.1	47.1	12.7	48.3	13.5	49.7
54	8.16	38.7	8.85	40.3	9.61	42.0	10.4	43.6	11.0	45.0	11.8	46.5	12.6	48.0	13.2	49.2	14.0	50.6
56	8.42	39.3	9.16	41.0	9.94	42.7	10.7	44.3	11.4	45.8	12.2	47.3	13.0	48.8	13.7	50.1	14.5	51.5
58	8.63	39.8	9.48	41.7	10.3	43.4	11.0	45.0	11.8	46.6	12.6	48.1	13.4	49.6	14.2	51.0	15.0	52.4
60	8.89	40.4	9.75	42.3	10.5	44.0	11.4	45.8	12.2	47.3	13.0	48.9	13.8	50.4	14.6	51.8	15.5	53.3
64	9.43	41.6	10.3	43.5	11.2	45.4	12.1	47.2	12.9	48.7	13.8	50.4	14.7	52.0	15.5	53.4	16.5	55.0
68	9.98	42.8	10.9	44.7	11.8	46.6	12.8	48.4	13.7	50.2	14.6	51.8	15.6	53.5	16.5	55.0	17.5	56.6
72	10.4	43.8	11.5	45.9	12.4	47.8	13.5	49.7	14.4	51.5	15.4	53.2	16.4	54.9	17.4	56.5	18.3	58.0
76	10.8	44.9	12.0	47.0	13.1	49.0	14.1	50.8	15.1	52.7	16.2	54.6	17.3	56.3	18.3	57.9	19.3	59.5
80	11.5	46.0	12.6	48.0	13.7	50.1	14.7	52.0	15.8	53.9	17.0	55.8	18.1	57.6	19.2	59.3	20.3	61.0
84	12.0	46.9	13.2	49.2	14.2	51.1	15.4	53.2	16.5	55.0	17.7	57.0	18.9	58.9	20.1	60.7	21.2	62.4
88	12.5	47.9	13.7	50.1	14.8	52.2	16.1	54.3	17.3	56.3	18.5	58.2	19.7	60.1	20.9	62.0	22.1	63.7
92	12.9	48.7	14.2	51.1	15.5	53.4	16.7	55.4	18.0	57.4	19.2	59.4	20.5	61.3	21.8	63.2	23.0	65.0
96	13.3	49.5	14.8	52.2	15.9	54.0	17.2	56.2	18.6	58.5	19.7	60.2	21.1	62.2	22.7	64.5	24.0	66.3
100	13.9	50.6	15.0	52.5	16.7	55.3	17.9	57.3	19.2	59.4	20.6	61.5	21.6	63.0	23.4	65.5	24.8	67.5
104	14.6	51.8	15.8	53.9	17.1	56.0	18.6	58.5	19.9	60.5	21.4	62.6	22.7	64.5	24.1	66.5	25.6	68.5
108	14.8	52.1	16.2	54.6	17.6	56.8	19.2	59.4	20.5	61.4	22.0	63.5	23.5	65.7	24.8	67.5	26.5	69.7
112	15.1	52.7	16.8	55.5	18.3	58.0	19.7	60.1	21.1	62.3	22.5	64.3	24.5	67.0	25.7	68.7	27.1	70.5
116	15.8	53.9	17.3	56.4	18.9	58.9	20.3	61.1	22.0	63.6	23.5	65.7	24.8	67.5	26.2	69.4	28.2	71.9
120	16.2	54.6	17.8	57.1	19.4	59.6	20.9	62.0	22.7	64.5	24.2	66.7	26.1	69.2	27.2	70.6	29.0	73.0
124	16.6	55.2	18.4	58.1	19.8	60.3	21.6	63.0	23.2	65.4	25.2	68.0	26.5	69.8	28.2	71.9	29.8	74.0
128	17.1	56.0	18.8	58.8	20.3	61.1	22.3	64.0	23.7	66.0	25.6	68.6	27.3	70.8	28.7	72.6	30.2	74.5
132	17.4	56.5	19.3	59.5	20.8	61.8	22.6	64.4	24.5	67.0	26.3	69.5	28.2	72.0	29.8	74.0	32.0	76.6
136	17.9	57.3	19.7	60.2	21.4	62.7	23.0	65.0	25.1	67.9	26.9	70.3	28.7	72.6	30.5	74.8	32.6	77.3
140	18.5	58.2	20.3	61.0	22.3	64.0	24.1	66.5	25.9	69.0	27.5	71.1	29.4	73.5	31.5	76.0	33.4	78.3
144	18.8	58.7	20.6	61.5	22.7	64.5	24.8	67.5	26.3	69.5	28.2	72.0	29.9	74.1	32.0	76.6	34.0	79.0

*Circular equivalent diameter (d_e). Calculated from $d_e = 1.3\frac{(ab)^{.625}}{(a+b)^{.25}}$

†Large numbers in table are duct class.

Duct size = 22 x 16 (found from Table 21) which has a greater duct area than that for the direction B-1 of 2.08.

In the same manner, the other necessary values are estimated and presented in Table 22. Figure 44 shows the supply air distribution of Zones A, C and D.

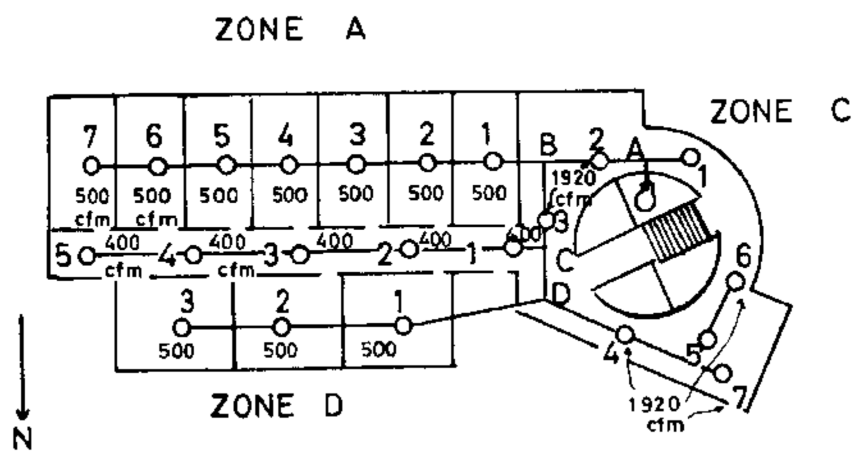


Figure 44 - Air supply diagram for Zones A, C and D.

Table 22 - Duct sizes for Zones A, C and D

Operation pressure for all terminals	0.15 inch water gage
Initial duct velocity	2,300 fpm
Duct area (18,520/2,300)	8.05 ft^2

Zone	Direction	Air Quantity (cfm)	cfm percentage (%)	Duct area percentage (%)	Duct area (ft^2)	Duct size width x height (inches)
	to A	18,520	100	100	8.05	60 x 22
	A - 1	1,920	10.4	16.9	1.36	16 x 14
	A - 2	16,600	89.6	92.0	7.41	56 x 22
	2 - B	14,680	79.3	84.2	6.78	50 x 22
C	B - 3	11,180	60.4	67.7	5.45	40 x 22
	3 - C	9,260	50.0	58.0	4.67	34 x 22
	C - D	7,260	39.2	56.2	4.52	32 x 22
	D - 4	5,760	31.1	39.1	3.15	22 x 12
	4 - 5	3,840	20.7	27.7	2.23	22 x 16
	5 - 6	1,920	10.4	16.9	1.36	16 x 16
	5 - 7	1,920	10.4	16.9	1.36	16 x 14
	B - 1	3,500	18.9	25.9	2.08	22 x 16
	1 - 2	3,000	16.2	23.2	1.87	18 x 16
	2 - 3	2,500	13.5	20.25	1.63	16 x 16
	3 - 4	2,000	10.8	17.3	1.39	18 x 12
A	4 - 5	1,500	8.1	13.2	1.06	14 x 12
	5 - 6	1,000	5.4	9.6	0.77	12 x 12
	6 - 7	500	2.7	4.9	0.39	12 x 6
	C - 1	2,000	10.8	17.3	1.39	18 x 12
	1 - 2	1,600	8.6	14.2	1.14	16 x 12
Corridor	2 - 3	1,200	6.5	11.25	0.91	12 x 12
	3 - 4	800	4.3	7.6	0.61	12 x 8
	4 - 5	400	2.2	3.9	0.31	12 x 6
	D - 1	1,500	8.1	13.2	1.06	14 x 12
D	1 - 2	1,000	5.4	9.6	0.77	12 x 12
	2 - 3	500	2.7	4.9	0.39	12 x 6

In the same manner, other necessary values are estimated for Zones B, E and F along with the necessary air supply diagram (Table 23 and Figure 45).

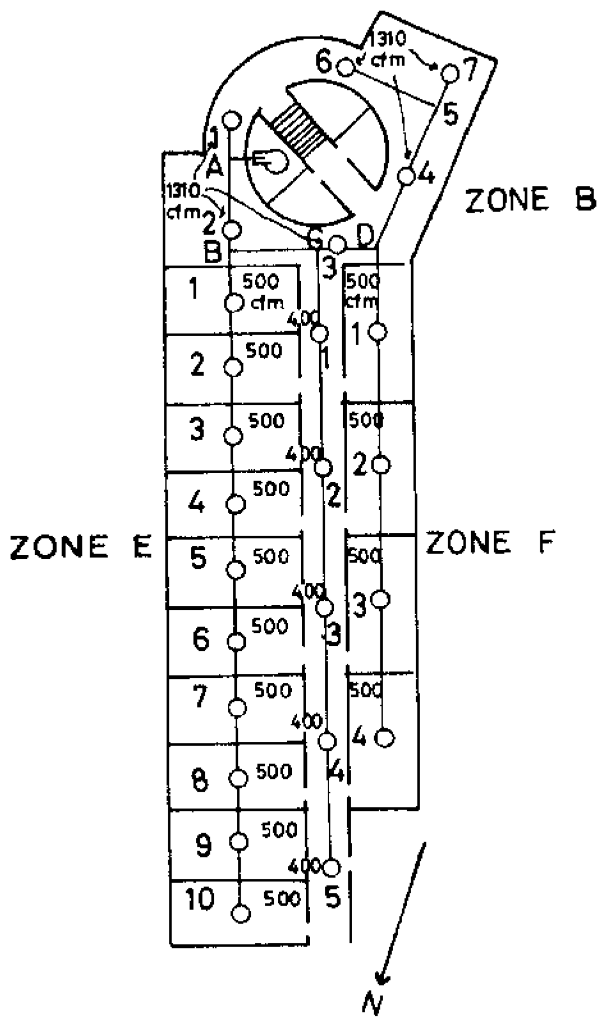


Figure 45 - Air supply diagram for Zones B, E and F.

Table 23 - Duct sizes for Zones B, E and F

Operation pressure for all terminals	0.15 inch water gage
Initial duct velocity	2,300 fpm
Quantity of supply air	17,460 cfm
Duct area (17,460/2,300)	7.59 ft^2

Zone	Direction	Air Quantity (cfm)	cfm percentage (%)	Duct area percentage (%)	Duct area (ft^2)	Duct size width x heigl (inches)
	to A	17,460	100	100	7.59	56 x 22
	A - 1	1,310	7.5	12.25	0.93	12 x 12
	A - 2	16,150	92.5	94.25	7.15	52 x 22
	2 - B	14,840	85.0	88.5	6.72	44 x 22
B	B - C	9,840	56.4	64.4	4.89	36 x 22
	C - 3	7,840	44.9	53.0	4.02	30 x 22
	3 - D	6,530	37.4	45.4	3.45	26 x 22
	D - 4	3,930	22.5	29.75	2.26	22 x 16
	4 - 5	2,620	15.0	21.5	1.63	16 x 16
	5 - 6	1,310	7.5	12.25	0.93	16 x 10
	5 - 7	1,310	7.5	12.25	0.93	16 x 10
	B - 1	5,000	28.6	36.25	2.75	24 x 18
	1 - 2	4,500	25.8	33.4	2.54	22 x 18
	2 - 3	4,000	22.9	30.4	2.31	20 x 18
	3 - 4	3,500	20.0	27.0	2.05	18 x 18
E	4 - 5	3,000	17.2	24.2	1.84	18 x 16
	5 - 6	2,500	14.3	20.8	1.58	16 x 16
	6 - 7	2,000	11.5	18.25	1.39	16 x 14
	7 - 8	1,500	8.6	13.9	1.06	14 x 14
	8 - 9	1,000	5.7	11.5	0.88	14 x 10
	9 - 10	500	2.9	5.3	0.40	10 x 10
	C - 1	2,000	11.5	18.25	1.39	16 x 14
	1 - 2	1,600	9.2	14.7	1.12	14 x 14
Corridor	2 - 3	1,200	6.9	11.4	0.87	14 x 10
	3 - 4	800	4.6	8.2	0.62	10 x 10
	4 - 5	400	2.3	4.1	0.31	10 x 6
	D - 1	2,000	11.5	18.25	1.39	16 x 14
F	1 - 2	1,500	8.6	13.9	1.06	14 x 14
	2 - 3	1,000	5.7	11.55	0.88	14 x 10
	3 - 4	500	2.9	5.3	0.40	10 x 10

In the same way, necessary values for Zones G, H and I are found as shown in Table 24. Figure 46 is the air supply diagram.

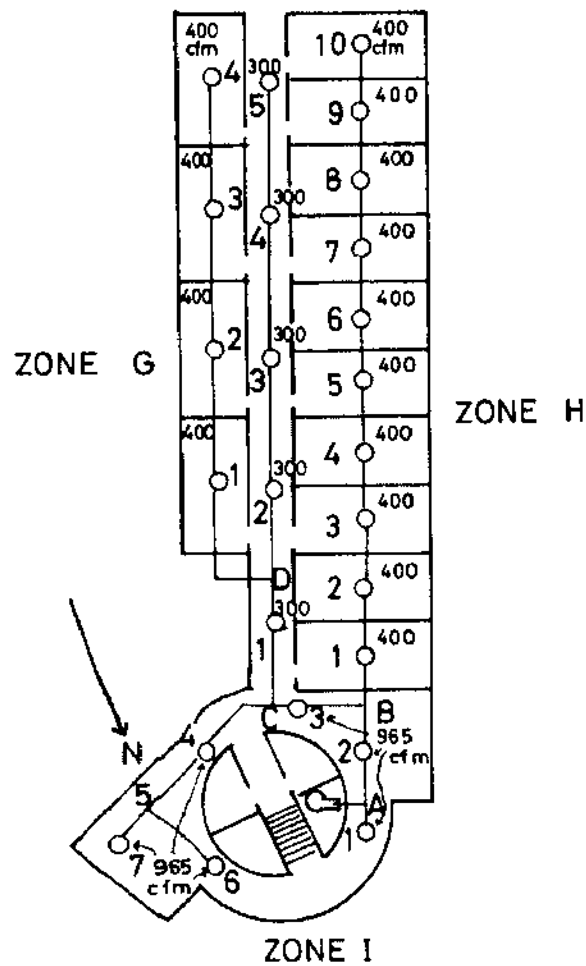


Figure 46 – Air supply diagram for Zones G, H and I.

Table 24 - Duct sizes for Zones G, H and I

Operation pressure for all terminals	0.15 inch
Initial duct velocity	2,150 fpn
Duct area (12,890/2,150)	6.0 ft^2

Zone	Direction	Air Quantity (cfm)	cfm percentage (%)	Duct area percentage (%)	Duct area (ft^2)
	to A	12,890	100	100	6.0
	A - 1	965	7.5	12.25	0.74
	A - 2	11,925	92.5	94.25	5.66
	2 - B	10,960	85.0	88.5	5.31
I	B - 3	6,960	54.0	71	4.26
	3 - C	5,995	46.5	54.5	3.27
	C - 4	2,895	22.5	29.75	1.79
	4 - 5	1,930	15.0	21.5	1.29
	5 - 6	965	7.5	12.25	0.74
	5 - 7	965	7.5	12.25	0.74
	B - 1	4,000	31.0	39.0	2.34
	1 - 2	3,600	27.9	35.5	2.13
	2 - 3	3,200	24.8	32.1	1.93
	3 - 4	2,800	21.7	28.8	1.73
H	4 - 5	2,400	18.6	25.6	1.54
	5 - 6	2,000	15.5	22.25	1.34
	6 - 7	1,600	12.4	18.9	1.13
	7 - 8	1,200	9.3	14.8	0.89
	8 - 9	800	6.2	10.7	0.64
	9 - 10	400	3.1	5.7	0.34
Corridor	C - 1	3,100	24.1	31.6	1.90
	1 - D	2,800	21.7	29.1	1.75
	D - 2	1,200	9.3	15.1	0.91
	2 - 3	900	7.0	11.5	0.69
	3 - 4	600	4.7	8.4	0.50
	4 - 5	300	2.3	4.1	0.25
	D - 1	1,600	12.4	18.9	1.13
G	1 - 2	1,200	9.3	15.1	0.91
	2 - 3	800	6.2	10.7	0.64
	3 - 4	400	3.1	5.7	0.34

B. Buoyancy Tank

Calculations for the determination of duct size for two zones in the buoyancy tank are based on the following design conditions:

o	Air temperature of a room	78^{o} F
o	Specific humidity of the rooms (W_i)	0.0125 lb/lb
o	Quantity of ventilation air	360 cfm
o	Bypass factor (BF)	0.05

(Application - residence, factory, wherever the high latent heat occurs)

Latent heat gain is fairly large in the buoyancy tank. The main heat gains depend on the heat from people and ventilation, both of which have high latent heat. This situation requires the use of cooling with dehumidification, and a reheat cycle to increase the efficiency of the air-conditioning system as well as to reduce the size of the air duct. This cycle is illustrated in Figure 47.

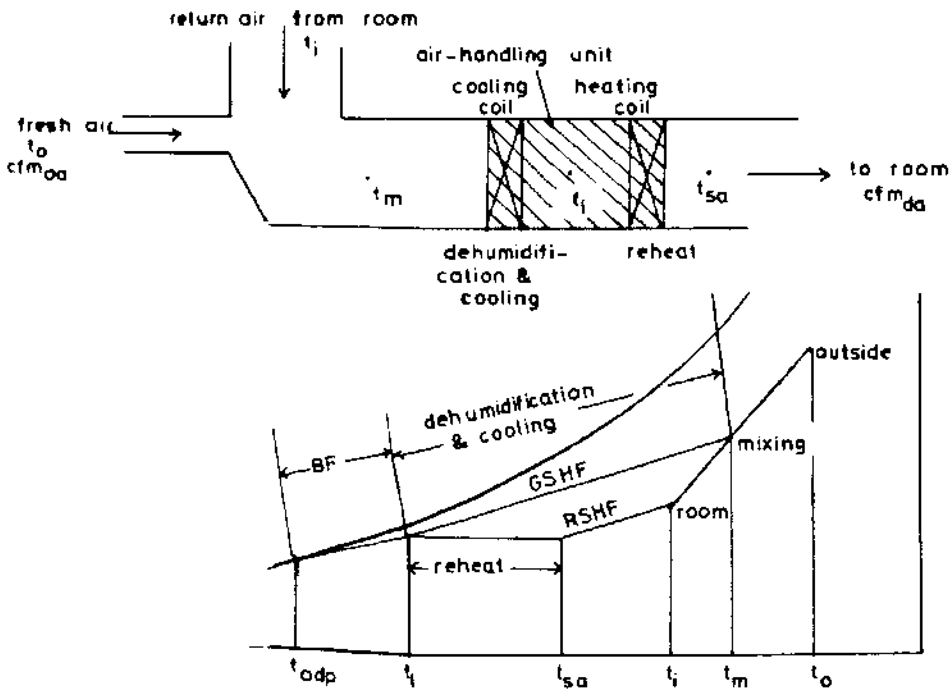


Figure 47 - Dehumidification with cooling and reheat.

The required air quantity is determined by the following procedu for bare concrete at 70 ft depth:

$$OASH = 1.08 \times cfm_{oa} (t_o - t_i) = 1.08 \times 360 (86.9 - 78) = 3,460 \text{ (Bt}$$

$$OALH = 0.68 \times cfm_{oa} (W_o - W_i)_{grain} = 0.68 \times 360 (120.5 - 86)$$

$$= 8,446 \text{ (Btu/hr)}$$

Assume a bypass factor of 0.05 because of high latent load

$$ESHF = \frac{RSH + (BF)(OASH)}{RSH + (BF)(OASH) + RLH + (BF)(OALH)}$$

$$= \frac{12,176 + 0.05(3,460)}{12,176 + 0.05(3,460) + 12,610 + (0.05)(8,446)} = \frac{12,3}{12,349 +}$$

$$= 0.487$$

where OASH = Sensible heat of outside air
OALH = Latent heat of outside air
cfm_{oa} = Ventilation (fresh) air from outside

When plotted on the psychrometric chart, this ESHF doesn't inte the saturation curve. Referring to the apparatus dew-point in Table 25 53° F is found in conjunction with the ESHF of 0.6.

Therefore, the amount of reheat (Btu/hr) required to produce an ESHF of 0.6 is

$$0.6 = \frac{12,349 + \text{reheat}}{12,349 + 13,032 + \text{reheat}}$$

reheat = 7,199 Btu/hr

Required air quantity is estimated by

$$cfm_{da} = \frac{ERSH}{1.08 \times (1 - BF)(t_i - t_{adp})} = \frac{12,349 + 7,199}{1.08\ (1 - 0.05)(78 - 53)}$$

$$= \frac{19,548}{25.65} = 762$$

Temperature of air entering the cooling coil (t_m) is given by

$$t_m = \frac{(cfm_{oa} \times t_o) + (cfm_{ra} \times t_i)}{cfm_{da}}$$

$$= \frac{(360 \times 86.9) + (402 \times 78)}{762}$$

$$= 82.2^{\circ} F$$

Leaving condition of air from cooling coil (t_l)

$$t_l = t_{adp} + BF\,(t_m - t_{adp})$$

$$= 53 + 0.05\,(78 - 53)$$

$$= 54.25^{\circ} F$$

Supply air temperature to the room (t_{sa})

$$t_{sa} = t_i - \frac{RSH}{1.08\,(cfm_{da})} = 78 - \frac{12,176}{1.08\,(762)} = 78 - 14.8$$

$$= 63.2^{\circ} F$$

Resultant values for this calculation (bare concrete at 70 ft depth) are illustrated in Figure 48. In the same manner, the other required air quantities are calculated and presented in Table 25.

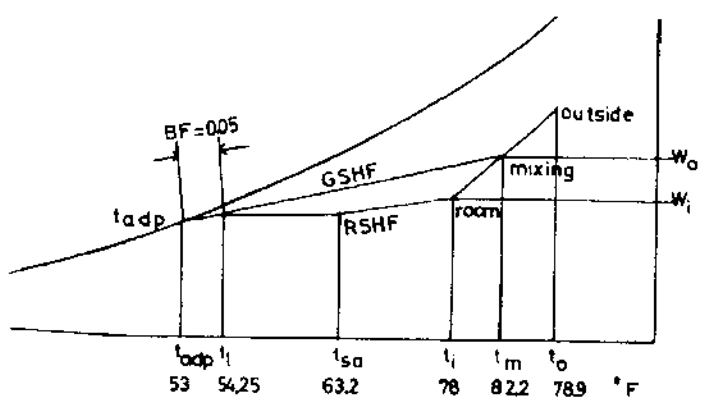


Figure 48 - Dehumidification with cooling, and reheat for buoyancy tank with bare concrete at depth of 70 feet.

Table 25 - Required air quantity for two rooms in a typical floor

Depth (ft)	Wall description*	RSH (Btu/hr)	RLH (Btu/hr)	OASH (Btu/hr)	OALH (Btu/hr)	ESHF	Corrected ESHF	t_{adp} (°F)	Reheat load (Btu/hr)	cfm_{da} (cfm)	t_{sa} (°F)
	a	12,176	12,610	3,460	8,446	0.487	0.6	53	7,199	762	63.2
70	f	12,540	12,610	3,460	8,446	0.494	0.6	53	7,745	784	63.2
	g	10,318	12,610	3,460	8,446	0.446	0.6	53	4,412	798	66.0
	h	11,446	12,610	3,460	8,446	0.471	0.6	53	6,104	691	62.7
	a	12,232	12,610	3,460	8,446	0.488	0.6	53	7,283	768	63.3
105	f	12,522	12,610	3,460	8,446	0.493	0.6	53	7,718	796	63.4
	g	6,936	12,610	3,460	8,446	0.353	0.6	53	(-661)	--	--
	h	11,684	12,610	3,460	8,446	0.476	0.6	53	6,461	715	62.9
	a	11,556	12,610	3,460	8,446	0.474	0.6	53	6,269	702	62.8
140	f	11,976	12,610	3,460	8,446	0.482	0.6	53	6,899	743	63.1
	g	318	12,610	3,460	8,446	0.036	0.6	53	(-10,588)	--	--
	h	10,056	12,610	3,460	8,446	0.440	0.6	53	4,019	556	61.3

* From Table 15

Table 25 shows that the amount of supply air required for a typical floor of the buoyancy tank at a depth of 70 ft is fairly similar for different wall configurations. At a depth of 105 ft, bare steel causes a lot of condensation on the inner surface of the wall. To maintain comfort conditions in a room, it may be necessary to install huge refrigeration machines and boilers. Hence the use of a bare steel wall for the buoyancy tank at this depth sounds impractical. However, the addition of insulation such as air space, gypsum board and/or plaster finish reduces this problem. The required quantity of supply air for a room with bare concrete walls is similar to that for steel with insulation. This means the applicability of steel with insulation is equivalent to that of bare concrete. Therefore the problem of condensation is no longer a disadvantage with a steel wall.

At a depth of 140 ft, the use of steel with insulation materials seems more economical than bare concrete in terms of the operation cost of the air-handling unit. Reheat load and required air quantity are the least among the four different configurations, because this configuration has the smallest total heat gain.

Following the same procedures described previously, design of duct size for a typical floor in the buoyancy tank is estimated for a depth of 105 ft. Bare concrete is selected as the sample material for the outside wall, and the necessary design conditions for the ducting are shown in Table 26. The supply air diagram is shown in Figure 49.

Table 26 - Duct sizes for buoyancy tank

Operation pressure for all terminals	0.15 inch wate
Initial duct velocity	1,300 fpm
Required air quantity to be supplied to buoyancy tank	1,800 cfm
Duct area (1,800/1,300)	1.38 ft^2

Direction	Air Quantity (cfm)	cfm percentage (%)	Duct area percentage (%)	Duct area (ft^2)	Duct width x (inc
to A	1,800	100	100	1.38	22 x
A - 1	900	50.0	58.0	0.80	14 x
1 - 2	750	41.7	49.7	0.58	10 x
2 - 3	600	33.3	41.3	0.46	10 x
3 - 4	450	25.0	32.5	0.35	10 x
4 - 5	300	16.7	23.7	0.23	10 x
5 - 6	150	8.3	13.2	0.11	10 x
A - 7	900	50.0	58.0	0.80	14 x
7 - 8	750	41.7	49.7	0.58	10 x
8 - 9	600	33.3	41.3	0.46	10 x
9 - 10	450	25.0	32.5	0.35	10 x
10 - 11	300	16.7	23.7	0.23	10 x
11 - 12	150	8.3	13.2	0.11	10 x

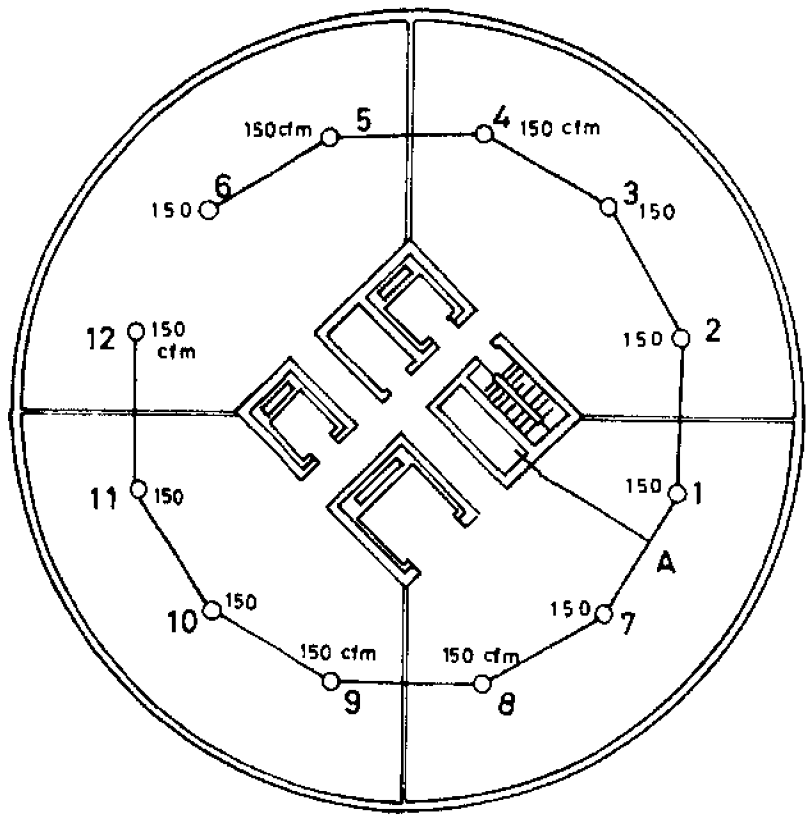


Figure 49 - Supply air diagram for buoyancy tank.

VIII. SUMMARY AND CONCLUSIONS

1. The climatic data show in Section I would indicate that refrigeration requirements for the superstructure would be relatively small because of the low internal heat gains and the relatively open structural plan. This was not found to be the case. Total heat gain is rather large and is dominated by sun loads on walls and glass, as shown in the curves in Section V.C. Proper attention to shading, reflective exterior finishes, etc., would significantly reduce total loads, perhaps leading to a condition where part-time air conditioning would suffice for the superstructure. The method used for investigating heat loads lends itself readily to an optimization of the thermal design by tracing the effect of each material or configuration change through the heat flow simulation. Thus, useful comparisons can be made between refrigeration and ventilating costs on the one hand and structural and finish costs, or even esthetic merit, on the other.

2. Analysis of heat flow in the exterior walls of the buoyancy chambers (Section V.C) shows that even though the area in contact with the sea is very large, its contribution to the heat budget of the city's underbody is not important. In fact, when uninsulated steel buoyancy chambers were examined, an <u>increase</u> in refrigeration load was required, accompanied by reheating, to limit condensation on the inside surface (Section VII, p. 98). It seems clear that, at least for Hawaii's climate, the fact of the city's immersion in the sea has no value thermally. Removal of the city's rejected heat will have to be obtained conventionally, i.e., by pumped cooling water.

3. For urban applications of very large floating platforms, comfort air conditioning will demand large amounts of energy. The use of a steam absorption cycle for refrigeration, followed by distillation desalination, appears to be an efficient way to minimize the city's total fuel bill. It is rendered feasible by the fact that the entire mechanical plant for the city can be designed at one time, in contrast to the usual urban case.

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APPENDIX A

LIST OF SYMBOLS

All the special uses of symbols are described wherever they occur.

A	Surface area, ft^2
A_{fl}	Surface area of floor, ft^2
A_w	Surface area of wall or roof, ft^2
a	Thermal diffusivity, ft^2/hr
CL	Cooling load, Btu/hr
C_p	Specific heat, Btu/lb-^oF
$F(t)$	Temperature excitation, ^oF
HC	Total convective heat gain, Btu/hr
HG	Total radiant heat gain, Btu/hr
h_i	Inside unit surface conductance,Btu/hr-ft^2-^oF
h_o	Outside unit surface conductance, Btu/hr-ft^2-^oF
I_d	Diffuse solar radiation, Btu/hr-ft^2
I_{nd}	Direct solar radiation, Btu/hr-ft^2
I_τ	Transmitted solar radiation, Btu/hr-ft^2
k	Thermal conductivity, Btu/hr-ft-^oF
ℓ	Length of wall, ft
L	Thickness of wall or roof, ft

L_{eq}	Equivalent thickness, ft
L_s	Thickness of equivalent single-layer wall, ft
Q, q	Rate of heat transfer, Btu/hr or Btu/hr-ft^2
r_o, r_i	Radius, ft
R	Resistance, hr-ft^2-^oF/Btu
SC	Shading coefficient
SHGF	Solar heat gain factor, Btu/hr-ft^2
t	Time, hr
U	Overall heat transmission coefficient, Btu/hr-ft^2-^oF
V	Volume of room air, ft^3
W_i	Specific humidity of room air, lb/lb
W_j	Weighting factors
W_o	Specific humidity of outside air, lb/lb
x	Distance, ft
X_j	Response factors, Btu/hr-ft^2-^oF
Y_j	Response factors, Btu/hr-ft^2-^oF
Z_j	Response factors, Btu/hr-ft^2-^oF
α	Absorptivity
α_j	Absorptances of double-strength glass
Δt	Time interval, hr
η	Incident angle, deg

θ	Temperature, $^{\circ}F$
θ_{fl}	Surface temperature of floor, $^{\circ}F$
θ_i	Temperature of room air, $^{\circ}F$
θ_o	Temperature of outside air, $^{\circ}F$
θ_s	Surface temperature of wall or roof, $^{\circ}F$
ρ	Density, lb/ft^3
τ_j	Transmittances of double-strength glass

APPENDIX B

PROGRAM FLOW DIAGRAM

The program flow diagram is presented in the following pages. The entire program was written in FORTRAN IV and the calculations were performed on the IBM 360/65 computer. The subroutines are described briefly below.

Subroutine SUN determines the intensities of direct, diffuse and total solar radiation. The heat gains through sunlit double-strength sheet glass can be calculated for different orientations at any time of the year.

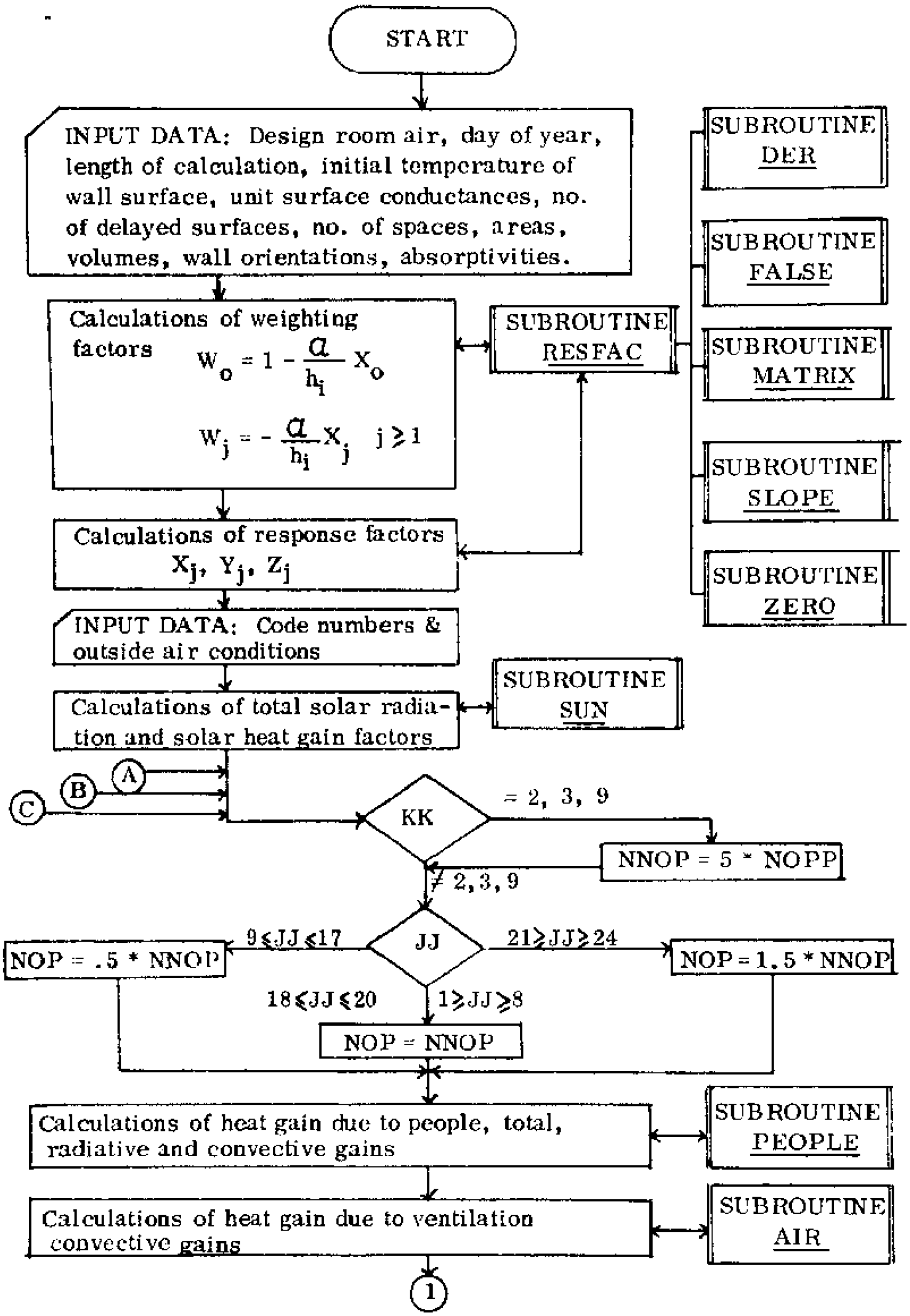
Subroutine RESFAC, in conjunction with subroutines SLOPE, FALSE, MATRIX, DER and ZERO, is for the calculation of response factors. Table 6 contains all the necessary input data for subroutine RESFAC. Subroutines for the calculation of response factors are taken from the computer program for analysis of energy utilization in postal facilities (18).

Subroutine AIR calculates the total heat gain due to ventilation in accordance with Equation 3-8 and data from Tables 7, 9 and 12.

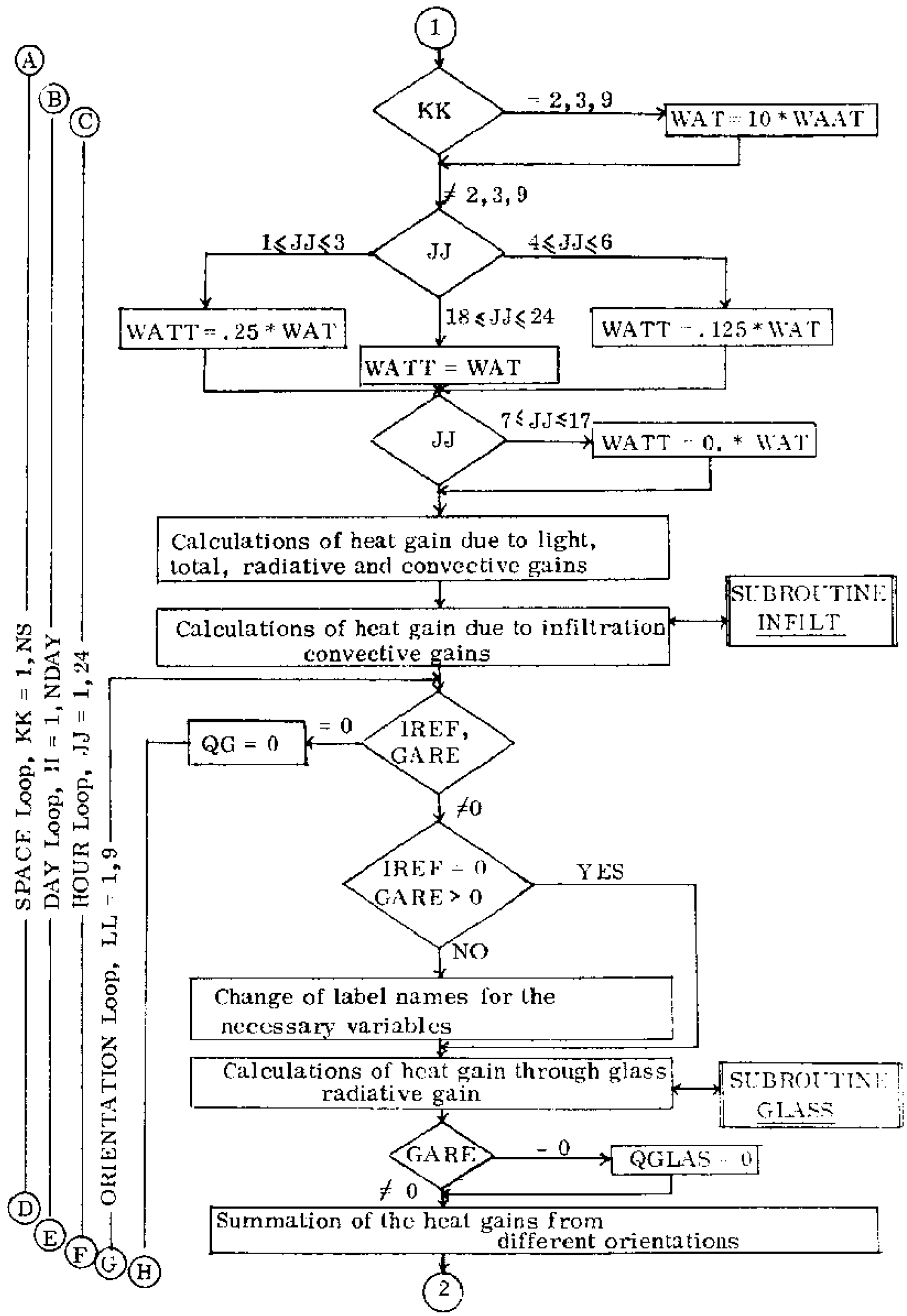
Subroutine INFILT calculates the heat gain due to infiltration of outside air through cracks in the windows and doors. Calculations are based on Equation 3-9, with the proper code numbers selected from Tables 7 and 13.

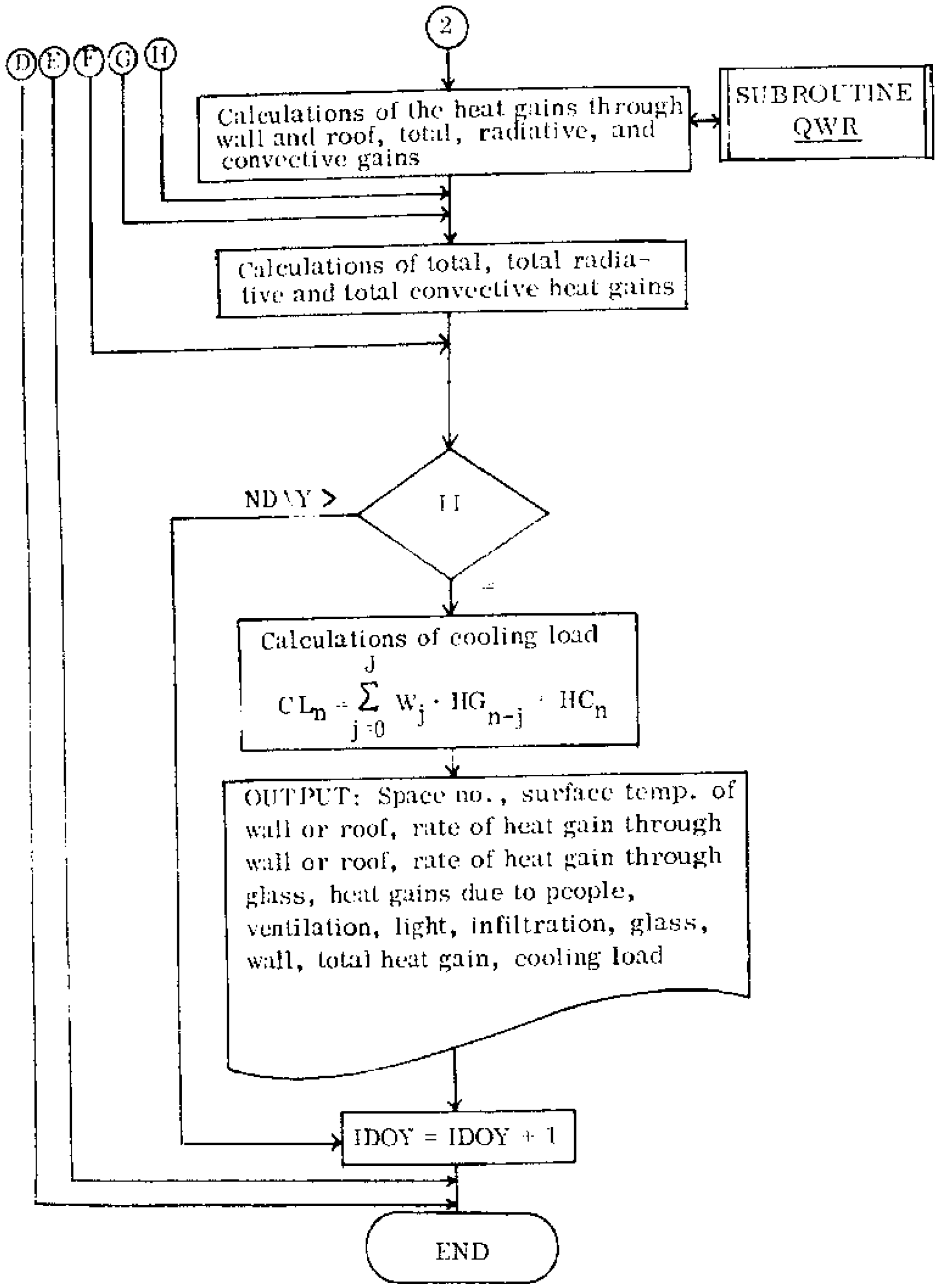
Subroutine GLASS calculates the heat gain through glass by using Equation 3-2 and Table 14. Subroutine QWR calculates the heat gain through the wall or roof by the response factor method. Calculations are based on Equation 3-5 with data from Tables 6 and 7.

Weighting factors, from Equations 4-1 and 4-2, and heat gain due to lights are programmed in the main program.



START
INPUT DATA: Design room air, day of year, length of calculation, initial temperature of wall surface, unit surface conductances, no. of delayed surfaces, no. of spaces, areas, volumes, wall orientations, absorptivities.
SUBROUTINE DER
SUBROUTINE FALSE
Calculations of weighting factors $W_o = 1 - \frac{a}{h_i} X_o$
$W_j = - \frac{a}{h_i} X_j \quad j \geq 1$
SUBROUTINE RESFAC
SUBROUTINE MATRIX
SUBROUTINE SLOPE
Calculations of response factors X_j, Y_j, Z_j
SUBROUTINE ZERO
INPUT DATA: Code numbers & outside air conditions
SUBROUTINE SUN
Calculations of total solar radiation and solar heat gain factors
A
B
C
KK
= 2, 3, 9
NNOP = 5 * NOPP
≠ 2,3,9
JJ
9≤JJ≤17
NOP = .5 * NNOP
21≥JJ≥24
NOP = 1.5 * NNOP
18≤JJ≤20
1≥JJ≥8
NOP = NNOP
Calculations of heat gain due to people, total, radiative and convective gains
SUBROUTINE PEOPLE
Calculations of heat gain due to ventilation convective gains
SUBROUTINE AIR
1


1
KK
= 2, 3, 9
WAT = 10 * WAAT
≠ 2, 3, 9
JJ
1≤JJ≤3
4≤JJ≤6
18≤JJ≤24
WATT = .25 * WAT
WATT = .125 * WAT
WATT = WAT
JJ
7≤JJ≤17
WATT = 0. * WAT
Calculations of heat gain due to light, total, radiative and convective gains
Calculations of heat gain due to infiltration convective gains
SUBROUTINE INFILT
IREF, GARE
= 0
QG = 0
≠0
IREF = 0
GARE > 0
YES
NO
Change of label names for the necessary variables
Calculations of heat gain through glass radiative gain
SUBROUTINE GLASS
GARE
= 0
QGLAS = 0
≠ 0
Summation of the heat gains from different orientations
2
SPACE Loop, KK = 1, NS
DAY Loop, II = 1, NDAY
HOUR Loop, JJ = 1, 24
ORIENTATION Loop, LL = 1, 9
A
B
C
D
E
F
G
H


2
D
E
F
G
H
Calculations of the heat gains through wall and roof, total, radiative, and convective gains
SUBROUTINE QWR
Calculations of total, total radiative and total convective heat gains
NDAY >
Calculations of cooling load
$CL_n = \sum_{j=0}^{J} W_j \cdot HG_{n-j} + HC_n$
OUTPUT: Space no., surface temp. of wall or roof, rate of heat gain through wall or roof, rate of heat gain through glass, heat gains due to people, ventilation, light, infiltration, glass, wall, total heat gain, cooling load
IDOY = IDOY + 1
END

APPENDIX C

PROGRAM LISTING AND SAMPLE OUTPUTS

```
$JOB         YOSHI YAMASHITA,TIME=1,PAGES=100 ,LINES=60
      DIMENSION RFX(100),RFY(100),RFZ(100),TTR(100),RX(10,100)
      DIMENSION RY(10,100),XQ1(100),YQ1(100),XQ2(100),YQ2(100)
      DIMENSION QX1(100),QX2(100),QY1(100),QY2(100),QG(24,9)
      DIMENSION TOTRAD(24,9),SHGF(24,9),TRAD(24),SHFAC(24)
      DIMENSION GO(24),WO(24),     TTO(24),WWO(24),TO(9,100)
      DIMENSION TS(100),        QGAIN(24,9),QP(24),QTOTAL(24)
      DIMENSION QAIR(24),QL(24),QINF(24),QOTOT(48),CCR(10)
      DIMENSION QQW(   24,9),TTS(24,9),QUS(24),QQG(24,9)
      DIMENSION ARE(10,9),GARE(10,9),AAB(10,9),VOL(10),IREF(10,9)
      DIMENSION ALFA(5),WT( 10,20),QLOAD(100),QTOT(100),H(100)
      DIMENSION QPR(24),QPC(24),QLF(24),QLC(24)
      DIMENSION QTOTR(48),QTOTC(48),QCONV(48),WCR(48),WCC(48)
C
C     READ AIR TEMPERATURE(DEG.F) AND SPECIFIC HUMIDITY(LB/LB)
C     OF THE ROOM
      READ(5,500) TR,WR
C
C     READ THE DAY OF YEAR
      READ(5,501) IDOY
C
C     READ THE LENGTH OF CALCULATION (DAYS)
      READ(5,502) NDAY
C
C     READ THE INITIAL SURFACE TEMPERATURE OF WALL AND ROOF(DEG.F)
      READ(5,504) TDB
C
      TS(1)=TDB
C
C     READ THE OUTSIDE AND INSIDE UNIT SURFACE CONDUCTANCES
C     (BTU/HR-FT*FT/F)
      READ(5,500) FO,FI
C
C     READ THE NUMBER OF DIFFERENT DELAYED SURFACES(WALL AND ROOF)
      READ(5,504) WW
      NOW=WW+0.1
C
C     READ THE NUMBER OF SPACES IN THE BUILDING(ZONE A TO I)
      READ(5,504) FNS
C
      NS=FNS+0.1
C
C     READ THE SURFACE AREA  OF THE OUTSIDE WALL AND ROOF(FT*FT),
C     SURFACE AREA OF OUTSIDE GLASS(FT*FT), AND THE ABSORPTIVITY
C     OF OUTSIDE WALL AND ROOF SURFACE
      READ(5,531) ((ARE(I,J),GARE(I,J),AAB(I,J),J=1,9),I=1,NS)
C
C     READ THE VOLUME OF A ROOM
      READ(5,504) (VOL(I),I=1,NS)
C
C     READ THE WALL ORIENTATIONS AS GIVEN IN THE NOTE IN TABLE III
      READ(5,530) ((IREF(I,J),J=1,9),I=1,NS)
C
C     READ THE ABSORPTIVITY FOR THE WALL OF EQUIVALENT THICKNESS
C     (BTU/HR-FT*FT), EQS.(4-1), (4-2)
      READ(5,500) (ALFA(J),J=3,4)
C
C     CALCULATION OF THE ROOM WEIGHTING FACTORS
      DO 33 I=3,4
      CALL RESFAC(R1,RFX,RFY)
      AA=ALFA(I)
      B=AA/FI
```

```
      WT(I,1)=1.0-B* RFX(1)
      DO 30 L=2,20
      WT(I,L)=-B* RFX(L)
   30 CONTINUE
   33 CONTINUE
      DO 100 I=1,NCW
C
      CALL RESFAC(R1,RFX,RFY)
C
      CCR(I)=R1
C
C     K DENOTES THE THE CAPITAL J IN EQUATION (3-5)
      K=20
      IIR(I)=K
      CCR(I)=R1
      NNOR=K-1
      UXI=0.
      UYI=0.
      DO 103 J=1,NNOR
      RX(I,J)=RFX(J)
      RY(I,J)=RFY(J)
      UXI=UXI+RFX(J)
      UYI=UYI+RFY(J)
  103 CONTINUE
      RX(I,K)=RFX(K)
      RY(I,K)=RFY(K)
      TDROP=TDB   -TR
      XQ1(I)=UXI*TDROP
      XQ2(I)=(UXI+RFX(K)/(1.0-R1))*TDROP
      YQ1(I)=UYI*TDROP
      YQ2(I)=(UYI+RFY(K)/(1.0-R1))*TDROP
  100 CONTINUE
      DO 115 J=1,100
      DO 116 I=1,9
  116 TO(I,J)=TDB
  115 CONTINUE
C
C     READ THE CODE NUMBERS AS SPECIFIED IN TABLES VI, VII,
C     VIII AND IX, REFERENCE NUMBERS OF PEOPLE AND POWER INPUT
C     FROM LIGHTS(WATT)
      READ(5,507) I50,I51,I52,I53,NOPP,WAAT
C
C     READ DRY-BULB TEMPERATURE(DEG.F) AND SPECIFIC HUMIDITY
C     (LB/LB) OF OUTSIDE AIR
      READ(5,508) (TTO(I),WWO(I),I=1,24)
C
      CALL SUN(IDOY,SHGF,TOTRAD)
C
C     SPACE LOOP STARTS FROM HERE
      DO 120 KK=1,NS
      MM=0
C
C     DAY LOOP STARTS FROM HERE
      DO 104 II=1,NDAY
C
C     HOUR LOOP STARTS FROM HERE
      DO 105 JJ=1,24
C
      TO=TTO(JJ)
      WO=WWO(JJ)
      NNOP=NOPP
      IF(KK.EQ.2.OR.KK.EQ.3.OR.KK.EQ.9) NNOP=5*NOPP
      IF(JJ.GE.21.AND.JJ.LE.24) NOP=1.5*NNOP
      IF(JJ.GE.1.AND.JJ.LE.8) NOP=NNOP
      IF(JJ.GE.9.AND.JJ.LE.17) NOP=0.5*NNOP
      IF(JJ.GE.18.AND.JJ.LE.20) NOP=NNOP
```

```
      C
            CALL PEOPLE(I50,NOP,QPP)
      C     QPR IS THE RADIATIVE PART OF HEAT GAIN DUE TO PEOPLE
      C     QPC IS THE CONVECTIVE PART OF HEAT GAIN DUE TO PEOPLE
      C     QP IS THE TOTAL HEAT GAIN DUE TO PEOPLE
      C
            QP(JJ)=QPP
            QPR(JJ)=0.4*QPP
            QPC(JJ)=0.2*QPP
      C
            CALL AIR(I51,NOP,TO,TR,WO,WR,QAR)
      C     HEAT GAIN DUE TO VENTILATION HAS NO RADIATIVE PART OF
      C     HEAT GAIN TO COOLING LOAD
      C
            QAIR(JJ)=QAR
            WAT=WAAT
            IF(KK.EQ.2.OR.KK.EQ.3.OR.KK.EQ.9) WAT=10.*WAAT
            IF(JJ.GE.18.AND.JJ.LE.24) WATT=WAT
            IF(JJ.GE.1.AND.JJ.LE.3) WATT=0.25*WAT
            IF(JJ.GE.4.AND.JJ.LE.6) WATT=0.125*WAT
            IF(JJ.GE.7.AND.JJ.LE.17) WATT=0.*WAT
      C
      C     CALCULATION OF HEAT GAIN DUE TO LIGHT
      C     QL IS THE TOTAL HEAT GAIN DUE TO LIGHT
      C     QLR IS THE RADIATIVE PART OF GAINED HEAT BY LIGHT
      C     QLC IS THE CONVECTIVE PART OF HEAT GAIN DUE TO LIGHT
            XLT=1.25*3.4*WATT
            QL(JJ)=XLT
            QLR(JJ)=0.5*XLT
            QLC(JJ)=0.5*XLT
      C
            V=VOL(KK)
      C
            CALL INFILT(I52,V,TO,TR,WO,WR,QINFL)
      C     HEAT GAIN DUE TO INFILTRATION HAS NO RADIATIVE
      C     PART OF HEAT GAIN TO COOLING LOAD
      C
            QINF(JJ)=QINFL
            QW=0.
            QGL=0.
      C
      C     LOOP OF WALL ORIENTATIONS STARTS FROM HERE
            DO 121 LL=1,9
      C
            IF(IREF(KK,LL).EQ.0.AND.GARE(KK,LL).EQ.0.) GO TO 20
            SHG=SHGF(JJ,LL)
            TTR=TCIRAD(JJ,LL)
            AB=AA3(KK,LL)
            AREA=ARE(KK,LL)
            GAREA=GARE(KK,LL)
            IF(IREF(KK,LL).EQ.0.AND.GARE(KK,LL).GT.0.) GO TO 15
            IRE=IREF(KK,LL)
            CR=CCR(IRE)
            IR=IIR(IRE)
            QX1(LL)=XQ1(IRE)
            QX2(LL)=XQ2(IRE)
            QY1(LL)=YQ1(IRE)
            QY2(LL)=YQ2(IRE)
      C
         15 CALL GLASS(I53,SHG ,TO, TR,QGLAS)
      C     HEAT GAIN DUE TO GLASS HAS NO CONVECTIVE PART
      C     OF HEAT GAIN TO COOLING LOAD
      C
            IF(GARE(KK,LL).EQ.0.) QGLAS=0.
            QGS(JJ,LL)=QGLAS
            QG(JJ,LL)=QGLAS*GAREA
```

```
      QGL=QGL+QG(JJ,LL)
      IF(LL.EQ.9) GQ(JJ)=QGL
      IF(IREF(KK,LL).EQ.0) GO TO 21
   17 DO 122 I=1,IR
      RRRX=RX(IRE,I)
      RFX(I)=RRRX
      RRRY=RY(IRE,I)
      RFY(I)=RRRY
      TTT=TD(LL,I)
      TS(I)=TTT
  122 CONTINUE
      GO TO 117
   20 QG(JJ,LL)=0.
      QQG(JJ,LL)=0.
   21 QQW(    JJ,LL)=0.
      TTS(JJ,LL)=0.
      QDS(LL)=0.
      GO TO 121
C
  117 CALL QWR(FO,RFX,RFY,TR,TO,TS,QDS(LL),AR,TTR,QX1(LL),
     +QX2(LL),QY1(LL),QY2(LL),IR,CK)
C     WQC IS THE CONVECTIVE PART OF HEAT GAIN THROUGH
C     WALL OR ROOF
C     WQR IS THE RADIATIVE PART OF HEAT GAIN THROUGH
C     WALL OR ROOF
C
      TTS(JJ,LL)=TS(1)
      DO 112 I=1,IR
      TD(LL,I)=TS(I)
  112 CONTINUE
      QQW(    JJ,LL)=QDS(LL)
      QW=QW+QDS(LL)*AREA
      WQ(JJ)=QW
      WQR(JJ)=0.6*WQ(JJ)
      WQC(JJ)=0.4*WQ(JJ)
  121 CONTINUE
C
C     QTOTAL DENOTES THE TOTAL HEAT GAIN
C     QTOTR DENOTES THE TOTAL RADIATIVE HEAT GAIN
C     QTOTC DENOTES THE TOTAL CONVECTIVE HEAT GAIN
C
      QTOTAL(JJ)=QP(JJ)+QAIR(JJ)+QL(JJ) +QINF(JJ)+GQ(JJ)+WQ(JJ)
      QTOTR(JJ)=QPR(JJ)+QLR(JJ)+ GQ(JJ)+WQR(JJ)
      QTOTC(JJ)=QPC(JJ)+QLC(JJ)+QAIR(JJ)+QINF(JJ)+WQC(JJ)
      QHEAT=QTOTR(JJ)
      MM=MM+1
      QQTOT(MM)=QHEAT
      QCONV(MM)=QTOTC(JJ)
  105 CONTINUE
      IF(II.LT.NDAY) GO TO 106
C
      DO 32 L=1,20
      IF(KK.EQ.2.OR.KK.EQ.3.OR.KK.EQ.9) GO TO 31
      W(L)=WT(3,L)
      GO TO 32
   31 W(L)=WT(4,L)
   32 CONTINUE
      DO 108 NN=25,48
      QQL=0.
      NNN=NN+1
C
C     CALCULATION OF COOLING LOAD GIVEN BY EQ.14-5)
      DO 107 M=1,15
      QQL=QQL+W(M)*QQTOT(NNN-M)
  107 CONTINUE
      QLOAD(NN)=QQL+QCONV(NN)
```

```
  108 CONTINUE
      WRITE(6,1001)
      WRITE(6,998) KK
      WRITE(6,1006)
      WRITE(6,1002) (JJ,(TTS(JJ,LL),LL=1,9),JJ=1,24)
      WRITE(6,1003)
      WRITE(6,1002)  (JJ,(QQW(JJ,LL),LL=1,9),JJ=1,24)
      WRITE(6,1001)
      WRITE(6,1004)
      WRITE(6,1002)  (JJ,(QQG(JJ,LL),LL=1,9),JJ=1,24)
      WRITE(6,1005)
      WRITE(6,1000) (JJ,QP(JJ),QAIR(JJ),QL(JJ),CINF(JJ),GQ(JJ),
     +WQ(JJ),QTOTAL(JJ),QLOAD(JJ+24),JJ=1,24)
      WRITE(6,1001)
  106 IDOY=IDOY+1
  104 CONTINUE
  120 CONTINUE
  500 FORMAT(2F10.4)
  501 FORMAT(I3)
  502 FORMAT(I1)
  504 FORMAT(F10.4)
  507 FORMAT(5I1, F10.2)
  508 FORMAT(6F10.5)
  530 FORMAT(9I1)
  531 FORMAT(9F8.2)
  998 FORMAT(1H ,1X,'SPACE NO = ',I4/ )
 1000 FORMAT(1X,I5,7F10.1,4X,F10.1)
 1001 FORMAT(1H1)
 1002 FORMAT(1X,I5,9F9.3)
 1003 FORMAT(1H-,'HEAT GAIN THROUGH WALLS (BTU/HR*FT**2)'/)
 1004 FORMAT(1H-,1X,'HEAT GAIN THROUGH GLASS (BTU/HR*FT**2)'/)
 1005 FORMAT(1H-,1X,'HEAT GAIN FROM VARIOUS SOURCES(BTU/HR)'//2X,
     +'TIME',5X,'PEOPLE',5X,'AIR',6X,'LIGHT',5X,'INFILT.',4X,
     +'GLASS',6X,'WALL',5X,'TOTAL',5X,'COOLING LOAD'/)
 1006 FORMAT(1H0,1X,'TEMPERATURE OF WALL SURFACE (DEG F)'/)
      STOP
      END

      SUBROUTINE RESFAC(K1,RFX,RFY)
      DOUBLE PRECISION F(10),BETA(10),ROOT(100),KK(100,3),DT,
     *M1,M4,H1,H2,W3,BETAX,BETAY,BETAZ,B1,B2,BP3,A,B,C,AA,BB,
     *FA,FB,FC,XX,YY,ZZ,XL(10),XK(10),D(10),SH(10),RES(10),
     *K0,K1,CC,X,Y,Z
      DIMENSION RFX(100),RFY(100),RFZ(100)
C     THIS SUBROUTINE CALCULATES RESPONSE FACTORS
   10 FORMAT(5F10.5)
   11 FORMAT(5D10.5)
      KARD=5
      DT=1.0
C     ++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++
C                        READING DATA
C     FNOL IS THE NUMBER OF LAYERS OF A MULTI-COMPONENT WALL
  110 READ(KARD,10)FNOL
C     ++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++
      M=FNOL+0.1
      DO  150   I=1,M
C     ++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++
C                        READING DATA
C     XL IS THICKNESS OF EACH LAYER
C     XK IS THEMAL CONDUCTIVITY OF EACH LAYER
C     D IS DENSITY OF EACH LAYER
C     SH IS SPECIFIC HEAT OF EACH LAYER
C     RES IS RESISTENCE OF EACH LAYER
      READ(KARD,11)XL(I),XK(I),D(I),SH(I),RES(I)
C     ++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++
      IF(XL(I)) 130,120,130
```

```
  120 R(I)=RES(I)
      BETA(I)=0.0
      GO TO  150
  130 R(I)=XL(I)/XK(I)
  140 BETA(I)=XL(I)*DSQRT(D(I)*SH(I)/XK(I))
  150 CONTINUE
  160 DO  180   I=1,M
      IF(XL(I)) 180,170,180
  170 RES(I)=0.0
  180 CONTINUE
      N=100
      IROOT=0
      DO  190   I=1,N
      ROOT(I)=0.0
  190 CONTINUE
C     ******************************************************************
C                        CALLING A SUB SUBROUTINE
      CALL  ZERO(R,BETA,RES,M,KO,K1,M1,M4)
C     ******************************************************************
      W1=30.0/DT
      W2=100.0/DT
      NN=8
C     ******************************************************************
C                        CALLING A SUB SUBROUTINE
      CALL  FALSE( R,BETA,RES,W1,W2,W3,B1,B2,BP3,M,NN)
C     ******************************************************************
      IROOT=1
      LAST=1
      J=2
  200 J=J-1
      ROOT(J)=W3
      KK(J,2)=1.0/BP3/W3/W3
      KK(J,1)=KK(J,2)*B1
      KK(J,3)=KK(J,2)*B2
  210 IF(LAST-1) 220,220,230
  220 W1=0.0001/DT
      GO TO  240
  230 W1=ROOT(LAST-1)
  240 W2=ROOT(LAST)
      W2=W2-0.00001/DT
      W1=W1+0.00001/DT
C     ******************************************************************
C                        CALLING A SUB SUBROUTINE
      CALL  SLOPE(R,BETA,RES,W1,W2,M,ICONT,LAST)
C     ******************************************************************
      GO TO ( 250, 310),ICONT
  250 NN=0
C     ******************************************************************
C                        CALLING A SUB SUBROUTINE
      CALL  FALSE(R,BETA,RES,W1,W2,W3,B1,B2,BP3,M,NN)
C     ******************************************************************
      DO  270   I=1,IROOT
      IF(W3-ROOT(I)) 260,260,270
  260 J=I+1
      GO TO  280
  270 CONTINUE
  280 IROOT=IROOT+1
      LAST=LAST+1
      JJ=IROOT+1
      IF(IROOT-N) 290,290,320
  290 DO  300   I=J,IROOT
      JJ=JJ-1
      ROOT(JJ)=ROOT(JJ-1)
      KK(JJ,1)=KK(JJ-1,1)
      KK(JJ,2)=KK(JJ-1,2)
      KK(JJ,3)=KK(JJ-1,3)
```

```
  300 CONTINUE
      GO TO  200
  310 IF(LAST) 320,320,210
  320 BETAX=K1+M4*K0
      BETAZ=K1+M1*K0
      DO  450    I=1,100
      A=0.0
      B=0.0
      C=0.0
      DO  340    J=1,IROOT
      IF(ROOT(J)*I*DT-30.0) 330,330,350
  330 BETAY=DEXP(-ROOT(J)*I*DT)
      A=A+KK(J,1)*BETAY
      B=B+KK(J,2)*BETAY
      C=C+KK(J,3)*BETAY
  340 CONTINUE
  350 A=(A+(K1+M4*K0)+DT*I*K0)/DT
      B=(B+K0*DT*I+K1)/DT
      C=(C+(K1+M1*K0)+DT*I*K0)/DT
      IF(I-2) 360,370,380
  360 AA=A
      BB=B
      CC=C
      GO TO  390
  370 AA=A-2.0*X
      BB=B-2.0*Y
      CC=C-2.0*Z
      GO TO  390
  380 AA=A-2.0*X+FA
      BB=B-2.0*Y+FB
      CC=C-2.0*Z+FC
C
C     RFX IS RESPONSE FACTORS, X
C     RFY IS RESPONSE FACTORS, Y
C     RFZ IS RESPONSE FACTORS, Z
  390 RFX(I)=AA
      RFY(I)=BB
      RFZ(I)=CC
      IF(I-2) 430,430,400
  400 IF(DABS(XX/FAA-AA/XX)-0.00001) 410,410,430
  410 IF(DABS(YY/FBB-BB/YY)-0.00001) 420,420,430
  420 IF(DABS(ZZ/FCC-CC/ZZ)-0.00001) 460,460,430
  430 IF(DABS(AA)-0.00000001) 460,460,440
  440 FA=X
      FB=Y
      FC=Z
      FAA=XX
      XX=AA
      FBB=YY
      FCC=ZZ
      YY=BB
      ZZ=CC
      X=A
      Y=B
      Z=C
  450 CONTINUE
  460 R1=DEXP(-DT*ROOT(1))
      RETURN
      END

      SUBROUTINE SLOPE(R,BETA,RES,W1,W2,M,ICNT,LAST)
      DOUBLE PRECISION R(10),BETA(10),W1,W2,W3,BP1,BP3,
     *FF(2,2),F(2,2),RES(10)
C     THIS SUBROUTINE CALCULATES THE SIGN OF THE DERIVATIVE
C     OF THE FUNCTION GIVEN BY D/DS*B(SN)
      IC=0
```

```
      J=1
      DELTA=(W2-W1)/20.0
      W3=W1+J*DELTA
C     ***************************************************************
C                        CALLING A SUB SUBROUTINE
      CALL DER(R,BETA,RES,W1,M,F,FF)
C     ***************************************************************
      B1=FF(1,2)
      BP1=F(1,2)
C     ***************************************************************
C                        CALLING A SUB SUBROUTINE
  100 CALL DER(R,BETA,RES,W3,M,F,FF)
C     ***************************************************************
      B3=FF(1,2)
      BP3=F(1,2)
      IF(B1) 110,110,120
  110 IF(B3) 130,130,210
  120 IF(B3) 210,210,130
  130 IF(BP1) 140,140,150
  140 IF(BP3) 170,170,160
  150 IF(BP3) 160,160,170
  160 IC=IC+1
  170 IF(IC-2) 180,210,210
  180 J=J+1
      IF(J-20) 190,190,200
  190 B1=B3
      BP1=BP3
      W3=W1+J*DELTA
      GO TO  100
  200 ICUNT=2
      LAST=LAST-1
      RETURN
  210 ICUNT=1
      W2=W3
      RETURN
      END

      SUBROUTINE FALSE(R,BETA,RES,W1,W2,W3,B1,B2,BP3,M,N)
      DOUBLE PRECISION R(10),BETA(10),F(2,2),FF(2,2),W1,
     *W2,W3,B1,B2,B3,BP3,A,B,C,FA,FB,FC,RES(10)
C
C     THIS SUBROUTINE CALCULATES ROOTS, SN, GIVEN IN
C     EQUATION (2-28)
   10 FORMAT(17H NO ROOT IN FALSE)
      KAGIT=6
      IF(N) 100,100,110
  100 N=1
  110 J=1
      BP3=W1
C     ***************************************************************
C                        CALLING A SUB SUBROUTINE
      CALL MATRIX(R,BETA,RES,W1,M,F)
C     ***************************************************************
      B1=F(1,2)
      B2=B1
  120 DELTA=(W2-W1)/N/20.0
  130 W3=W1+J*DELTA
      IF(W3-W2) 160,160,140
  140 N=N+1
      BP3=W1
      B2=B1
      IF(N-25) 150,150,300
  150 J=1
      GO TO  120
C     ***************************************************************
C                        CALLING A SUB SUBROUTINE
```

```
  160 CALL MATRIX(R,BETA,RES,W3,M,F)
C     ***************************************************************
      B3=F(1,2)
      IF(B1) 170,170,180
  170 IF(B3) 190,190,200
  180 IF(B3) 200,200,190
  190 J=J+1
      B2=B3
      BP3=W3
      GO TO  130
  200 A=BP3
      B=W3
      FA=B2
      FB=B3
  210 C=(A+B)/2.0
C     ***************************************************************
C                     CALLING A SUB SUBROUTINE
      CALL MATRIX(R,BETA,RES,C,M,F)
C     ***************************************************************
      FC=F(1,2)
      IF(FC) 220,290,250
  220 IF(FA) 230,230,240
  230 FA=FC
      A=C
      GO TO  280
  240 FB=FC
      B=C
      GO TO  280
  250 IF(FA) 260,260,270
  260 FB=FC
      B=C
      GO TO  280
  270 FA=FC
      A=C
  280 IF(DABS(A-B)-1.0D-14) 290,290,210
C     ***************************************************************
C                     CALLING A SUB SUBROUTINE
  290 CALL DER(R,BETA,RES,C,M,F,FF)
C     ***************************************************************
      W3=C
      BP3=F(1,2)
      B1=FF(2,2)
      B2=FF(1,1)
      RETURN
  300 WRITE(KAGIT,10)
      RETURN
      END

      SUBROUTINE MATRIX(RR,BETA,RES,W,M,F)
      DOUBLE PRECISION RR(10),BETA(10),F1(2,2),F2(2,2),
     *F(2,2),W,P,R,RES(10),TEMP
C
C     THIS SUBROUTINE CALCULATES THE VALUES OF A(SN), B(SN),
C     C(SN), D(SN)
      P=DSQRT(W)*BETA(1)
      R=RR(1)
      IF(P) 110,100,110
  100 F1(1,1)=1.0
      F1(1,2)=R
      F1(2,1)=0.0
      F1(2,2)=1.0
      GO TO  120
  110 F1(1,1)=DCOS(P)
      F1(1,2)=+R/P*DSIN(P)
      F1(2,1)=-P/R*DSIN(P)
      F1(2,2)=DCOS(P)
```

```
  120 IF(RES(1)) 130,140,130
  130 TEMP=1.0/(F1(1,2)*RES(1))
      F1(2,1)=(F1(2,1)*RES(1)+2.0*F1(1,1)-2.0)*TEMP
      F1(1,1)=(F1(1,1)*RES(1)+F1(1,2))*TEMP
      F1(2,2)=F1(1,1)
      F1(1,2)=F1(1,2)*RES(1)*TEMP
  140 IF(M-1) 170,150,170
  150 DO  160   L=1,2
      DO  160   LL=1,2
      F( L,LL)=F1(L,LL)
  160 CONTINUE
      GO TO  270
  170 DO  260   J=2,M
      P=DSQRT(W)*BETA(J)
      R=RR(J)
      IF(P) 190,180,190
  180 F2(1,1)=1.0
      F2(1,2)=P
      F2(2,1)=0.0
      F2(2,2)=1.0
      GO TO  200
  190 F2(1,1)=DCOS(P)
      F2(1,2)=+R/P*DSIN(P)
      F2(2,1)=-P/R*DSIN(P)
      F2(2,2)=DCOS(P)
  200 IF(RES(J)) 210,220,210
  210 TEMP=1.0/(F2(1,2)*RES(J))
      F2(2,1)=(F2(2,1)*RES(J)+2.0*F2(1,1)-2.0)*TEMP
      F2(1,1)=(F2(1,1)*RES(J)+F2(1,2))*TEMP
      F2(2,2)=F2(1,1)
      F2(1,2)=F2(1,2)*RES(J)*TEMP
  220 DO  230   L=1,2
      DO  230   LL=1,2
      F(L,LL)=0.0
  230 CONTINUE
      DO  240   L=1,2
      DO  240   LL=1,2
      DO  240   LLL=1,2
      F(L,LL)=F(L,LL)+F1(L,LLL)*F2(LLL,LL)
  240 CONTINUE
      DO  250   L=1,2
      DO  250   LL=1,2
      F1(L,LL)=F(L,LL)
  250 CONTINUE
  260 CONTINUE
  270 RETURN
      END

      SUBROUTINE DER(RR,BETA,RES,W,M,F,FF)
      DOUBLE PRECISION RR(10),BETA(10),F1(10,2,2),
     *F(2,2),FF(2,2),P,R,ALPHA,SQ,W,RES(10),TEMP,TEMP1,
     *F2(10,2,2),F3(10,2,2)
C
C     THIS SUBROUTINE CALCULATES THE DERIVATIVE  OF THE
C     FUNCTIONS GIVEN IN EQS.(2-33) TO (2-35)
      DO  140   I=1,M
      P=DSQRT(W)*BETA(I)
      R=RR(I)
      ALPHA=BETA(I)
      SQ=DSQRT(W)
      IF(P) 110,100,110
  100 F1(I,1,1)=1.0
      F1(I,1,2)=R
      F1(I,2,1)=0.0
      F1(I,2,2)=1.0
      F2(I,1,1)=0.0
```

```
      F2(I,1,2)=0.0
      F2(I,2,1)=0.0
      GO TO  120
  110 F1(I,1,1)=DCOS(P)
      F1(I,1,2)=R/P*DSIN(P)
      F1(I,2,1)=-P/R*DSIN(P)
      F1(I,2,2)=DCOS(P)
      F2(I,1,1)=+ALPHA*DSIN(ALPHA*SQ)/2.0/SQ
      F2(I,1,2)=-R*DCOS(ALPHA*SQ)/2.0/R+R*DSIN(ALPHA*SQ)/
     *ALPHA/2.0/SQ/SQ/SQ
      F2(I,2,1)=+ALPHA*ALPHA*DCOS(ALPHA*SQ)/2.0/R+
     *DSIN(ALPHA*SQ)/2.0/SQ*ALPHA/R
  120 F2(I,2,2)=F2(I,1,1)
      IF(RES(I)) 130,140,130
  130 TEMP=1.0/(F1(I,1,2)*RES(I))
      F1(I,2,1)=(F1(I,2,1)*RES(I)+2.0*F1(I,1,1)-2.0)*TEMP
      F1(I,1,1)=(F1(I,1,1)*RES(I)+F1(I,1,2))*TEMP
      F1(I,2,2)=F1(I,1,1)
      F1(I,1,2)=F1(I,1,2)*RES(I)*TEMP
      TEMP1=F2(I,1,2)*TEMP
      F2(I,2,1)=(F2(I,2,1)*RES(I)+2.0*F2(I,1,1))*TEMP-
     *F1(I,2,1)*TEMP1
      F2(I,1,1)=(F2(I,1,1)*RES(I)+F2(I,1,2))*TEMP-F1(I,1,1)*
     *TEMP1
      F2(I,2,2)=F2(I,1,1)
      F2(I,1,2)=F2(I,1,2)*RES(I)*TEMP-F1(I,1,2)*TEMP1
  140 CONTINUE
      IF(M-1) 170,150,170
  150 DO  160   K=1,2
      DO  160   L=1,2
      F(K,L)=F2(1,K,L)
      FF(K,L)=F1(1,K,L)
  160 CONTINUE
      GO TO  330
  170 DO  180   K=1,2
      DO  180   L=1,2
      F(K,L)=0.0
  180 CONTINUE
      DO  280   I=1,M
      DO  260   J=1,M
      DO  210   K=1,2
      DO  210   L=1,2
      IF(I-J) 190,200,190
  190 F3(J,K,L)=F1(J,K,L)
      GO TO  210
  200 F3(J,K,L)=F2(J,K,L)
  210 CONTINUE
      IF(J-1) 220,260,220
  220 DO  230   K=1,2
      DO  230   L=1,2
      FF(K,L)=0.0
  230 CONTINUE
      DO  240   K=1,2
      DO  240   L=1,2
      DO  240   N=1,2
      FF(K,L)=FF(K,L)+F3(J-1,K,N)*F3(J,N,L)
  240 CONTINUE
      DO  250   K=1,2
      DO  250   L=1,2
      F3(J,K,L)=FF(K,L)
  250 CONTINUE
  260 CONTINUE
      DO  270   K=1,2
      DO  270   L=1,2
      F(K,L)=FF(K,L)+F(K,L)
  270 CONTINUE
```

```
  280 CONTINUE
      DO  320    I=2,M
      DO  290    K=1,2
      DO  290    L=1,2
      FF(K,L)=0.0
  290 CONTINUE
      DO  300    K=1,2
      DO  300    L=1,2
      DO  300    N=1,2
      FF(K,L)=FF(K,L)+F1(I-1,K,N)*F1(I,N,L)
  300 CONTINUE
      DO  310    K=1,2
      DO  310    L=1,2
      F1(I,K,L)=FF(K,L)
  310 CONTINUE
  320 CONTINUE
  330 RETURN
      END

      SUBROUTINE ZERO(RR,BETA,RES,M,K0,K1,M1,M4)
      DOUBLE PRECISION RR(10),BETA(10),A,B,C,D,AA,BB,CC,DD,
     *F4,M1,M2,M3,M4,K0,K1,P,R,RES(10),F1,F2,F3
C     THIS SUBROUTINE CALCULATES THE VALUES OF DERIVATIVE IN
C     EQUATIONS (2-33) TO (2-35) AT POLE S=0
      K0=0.0
      DO  120    I=1,M
      IF(RES(I)) 110,100,110
  100 K0=K0+RR(I)
      GO TO  120
  110 K0=K0+RR(I)*RES(I)/(RR(I)+RES(I))
  120 CONTINUE
      K0=1.0/K0
      M1=0.0
      M2=0.0
      M3=0.0
      M4=0.0
      DO  260    I=1,M
      P=BETA(I)*BETA(I)
      R=RR(I)
      IF(I-1) 130,150,130
  130 A=1.0
      B=R
      C=0.0
      D=1.0
      IF(RES(1)) 140,190,140
  140 B=R*RES(1)/(P+RES(1))
      GO TO  190
  150 A=+P/2.0
      B=+R*P/6.0
      C=+P/R
      D=+P/2.0
      IF(RES(1)) 160,170,160
  160 A=RES(1)*P/2.0/(R+RES(1))
      B=(1.0-R/(R+RES(1)))*RES(1)*R*P/6.0/(R+RES(1))
      C=(RES(1)*P/R   +1.0*P   )/(R+RES(1))
      D=A
  170 IF(M-1) 190,180,190
  180 M1=A
      K1=-K0*K0*B
      M4=D
      GO TO  270
  190 DO  250    J=2,M
      P=BETA(J)*BETA(J)
      R=RR(J)
      IF(I-J) 200,220,200
  200 AA=1.0
```

```
      BB=R
      CC=0.0
      DD=1.0
      IF(RES(J)) 210,240,210
  210 BB=R*RES(J)/(R+RES(J))
      GO TO  240
  220 AA=+P/2.0
      BB=+P*R/6.0
      CC=+P/R
      DD=+P/2.0
      IF(RES(J)) 230,240,230
  230 AA=RES(J)*P/2.0/(R+RES(J))
      BB=(1.0-R/(R+RES(J)))*RES(J)*R*P/6.0/(R+RES(J))
      CC=(RES(J)*P/R  +1.0*P  )/(R+RES(J))
      DD=AA
  240 F1=A*AA+B*CC
      F2=A*BB+B*DD
      F3=C*AA+D*CC
      F4=C*BB+D*DD
      A=F1
      B=F2
      C=F3
      D=F4
  250 CONTINUE
      M1=M1+F1
      M2=M2+F2
      M3=M3+F3
      M4=M4+F4
  260 CONTINUE
      K1=-K0*K0*M2
  270 RETURN
      END

      SUBROUTINE SUN(IDOY,SHGF,TOTRAD)
      DIMENSION AN(9),AM(9)
      DIMENSION ALPHA(6),TAU(6),AZIM(8)
      DIMENSION TOTRAD(24,9),SHGF(24,9)
C
C     THIS SUBROUTINE CALCULATES THE DIRECT, DIFFUSE AND TOTAL
C     SOLAR RADIATION, AND THE SOLAR HEAT GAIN FACTORS AT
C     ANY TIME OF YEAR AND LOCATION
C     IDOY IS THE TIME OF YEAR
C     SHGF IS THE SOLAR HEAT GAIN FACTORS
C     TOTRAD IS THE TOTAL SOLAR RADIATION
C
  508 FORMAT(6F10.4)
  510 FORMAT (8F10.4)
C
C     ALPHA IS ABSORPTANCE OF THE DOUBLE-STRENGTH GLASS
C     TAU IS TRANSMITTANCE OF DOUBLE-STRENGTH GLASS
C     AZIM IS WALL AZIMUTH ANGLE
      READ(5,508)(ALPHA(J), J=1,6)
      READ(5,508) (TAU(J),J=1,6)
      READ(5,510)(AZIM(J), J=1,8)
      SHGA=0.0
      SHGT=0.0
      DO 7 J=1,6
      AJ=J -1
      SH=TAU(J)/(AJ+2.)
      SHGT=SHGT+SH
      SHA=ALPHA(J)/(AJ+2.)
    7 SHGA=SHGA+SHA
      DO 30 J=1,24
      DO 31 K=1,9
      TOTRAD(J,K) = 0.
   31 SHGF(J,K) =0.
```

```
      30 CONTINUE
C
C        R IS GROUND REFLECTIVITY
         R=0.2
C
C        CL IS THE NORTH LATITUDE IN RADIANS
         CL = 21.25*3.1415926/180.
C
C        IDOY INDICATES THE DAY OF A YEAR
         C1=COS(.01721*IDOY)
         S1=SIN(.01721*IDOY)
         C2=C1*C1-S1*S1
         S2=2.*S1*C1
         C3=C1*C2-S1*S2
         S3=C1*S2+S1*C2
C
C        DEL IS DECLINATION ANGLE IN RADINAS
         DEL =     .00527-.4001*C1-.003996*C2-.004240*C3+.0672*S1
         A   =     368.44+24.52*C1-1.14*C2-1.09*C3+.58*S1-
        *.18*S2+.29*S3
         B   =     .1717-.0344*C1+.0032*C2+.0024*C3-
        *.0043*S1-.0008*S3
         C   =     .0905-.0410*C1+.0073*C2+.0015*C3-.0034*S1+.0004*S2
        *-.0006*S3
         ANGLE = -TAN(CL)*TAN(DEL)
         SUNR = ARCOS(ANGLE)
         TANGLE = SUNR*12./3.1415926
         IF(ANGLE) 1,1,2
       1 SRT = TANGLE
         SST = 24.-SRT
         GO TO 15
       2 SST = 12.+TANGLE
         SRT = 12.-TANGLE
      15 TIME = SRT
       3 TIME = TIME+1.
         ITIME = TIME
         IF(ITIME.GT.13) GO TO 90
C
C        H GIVES THE HOUR ANGLE
         H=0.25*((12.-TIME)*60.)*3.14159/180.
C
C        COSZ, COSW AND COSS GIVES THE DIRECTIN COSINES OF SOLAR BEAM
         COSZ=SIN(CL)*SIN(DEL)+COS(CL)*COS(DEL)*COS(H)
         SINZ=(1.-COSZ**2)**0.5
         Z=ATAN2(SINZ,COSZ)
         ZZ=Z*180./3.14159
C
C         ALTI IS THE SOLAR ALTITUDE ANGLE 'BETA', IN DEGREES
         ALTI=90.-ZZ
         ALTITU=2.*ATAN(1.)-Z
         SINP=COS(DEL)*SIN(H)/COS(ALTITU)
         COSP=(1.-SINP**2)**0.5
C
C        PHI IS THE SOLAR AZIMUTH ANGLE 'PHI' IN DEGREES
         PHI=ATAN2(SINP,COSP)*180./3.14159
         COSW=COS(DEL)*SIN(H)
         COSS=-SQRT(1.-COSW**2-COSZ**2)
         COSH=COS(H)
C
C        T IS USED HERE AS A DUMMY ARGUMENT
         T=(SIN(DEL)/COS(DEL))/(SIN(CL)/COS(CL))
         IF(COSH.GT.T) COSS=-COSS
C
C        AIDN IS DIRECT SOLAR RADIATION
         AIDN= A   /EXP( B   /COSZ)
         TA=-((SIN(DEL)/COS(DEL))*(SIN(CL)/COS(CL)))
```

```
      IF(COSH.GT.TA) GO TO 17
      AIDN=0.0
   17 BL=0.0
      DO 18 J=1,9
C
C     J=9 DENOTES THE HORIZONTAL WALL
      IF(J.EQ.9) GO TO 19
      AZ=AZIM(J)*3.14159/180.
      AN(J)=COS(AZ)
      AM(J)=SIN(AZ)
      IF((J.EQ.6).OR.(J.EQ.7).OR.(J.EQ.8)) AM(J)=-AM(J)
      GO TO 20
   19 BL=1.0
      AN(J)=0.0
      AM(J)=0.0
C
C     COST IS THE COSINE OF THE INCIDENT ANGLE 'THETA'
   20 COST=BL*COSZ+AM(J)*COSW+AN(J)*CCSS
      AID=AIDN*COST
      IF(COST.LE.0.0) AID=0.0
C
C     Y IS CALLED DIRECT INTENSITY AND IS DEFINED AS THE RATIO OF SKY
C     VERTICAL SURFACE TO SKY DIFFUSE ON HORIZONTAL SURFACE
      Y=0.45
      IF(COST.GT.(-.2)) GO TO 21
      GO TO 22
   21 Y=0.55+0.437*COST+0.313*COST**2
C
C     AISD IS DIFFUSED SOLAR RADIATION
   22 AISD=AIDN*(C    *Y+R/2.0*(C    +COSZ))
      E=C    *AIDN
      IF(J.EQ.9) AISD=E
      SUMA=0.0
      SUMT=0.0
      DO 9 JJ=1,6
      SUM1=TAU(JJ)*COST**(JJ-1)
      SUMT=SUM1+SUMT
      SUM2=ALPHA(JJ)*COST**(JJ-1)
    9 SUMA=SUMA+SUM2
C
C     TRANS IS TRANSMITTED SOLAR HEAT THROUGH DOUBLE-STRENGTH GLASS
C     ABSORB IS ABSORBED SOLAR HEAT BY DOUBLE-STRENGTH GLASS
      TRANS=AID*SUMT+AISD*2.0*SHGT
      ABSORB=AID*SUMA+AISD*2.*SHGA
      ANI=0.267
      TOTRAD(ITIME,J) = AID+AISD
   18 SHGF(   ITIME,J)=TRANS+ANI*ABSORB
      DO 32 K=1,9,4
      TOTRAD(24-ITIME,K) = TOTRAD(ITIME,K)
      SHGF(   24-ITIME,K)=SHGF(   ITIME,K)
   32 CONTINUE
      DO 33 K=1,3
      TOTRAD(24-ITIME,9-K) = TOTRAD(ITIME,K+1)
      TOTRAD(24-ITIME,5-K) = TOTRAD(ITIME,K+5)
      SHGF(   24-ITIME,9-K)=   SHGF(   ITIME,K+1)
      SHGF(   24-ITIME,5-K)=   SHGF(   ITIME,K+5)
   33 CONTINUE
      GO TO 3
   90 RETURN
      END

      SUBROUTINE PEOPLE(I50,NOP,QPP)
C     THIS SUBROUTINE IS TO CALCULATE THE HEAT GAIN
C     DUE TO OCCUPANTS IN A ROOM
C     NOP IS NUMBER OF PEOPLE
C     QP IS COOLING LOAD DUE TO PEOPLE
```

```
      GO TO (10,20,30,40,50,60,70,80,90) , I50
   10 QPP    = 350*NOP
      RETURN
   20 QPP    = 400*NOP
      RETURN
   30 QPP    = 450*NOP
      RETURN
   40 QPP    = 500*NOP
      RETURN
   50 QPP    = 550*NOP
      RETURN
   60 QPP    = 750*NOP
      RETURN
   70 QPP    = 850*NOP
      RETURN
   80 QPP    = 1000*NOP
      RETURN
   90 QPP    = 1450*NOP
      RETURN
      END

      SUBROUTINE AIR(I51,NOP,TO,TR,WO,WR,QAR)
C     THIS SUBROUTINE CALCULATES THE HEAT GAINED BY VENTILATION
C     QAR IS THE TOTAL HEAT GAIN DUE TO VENTILATION
C     TO IS TEMPERATURE OF OUTSIDE AIR
C     WO IS SPECIFIC HUMIDITY OF OUTSIDE AIR
C     TR IS TEMPERATURE OF INSIDE AIR
C     WR IS SPECIFIC HUMIDITY OF INSIDE AIR
C     QAS IS SENSIBLE HEAT GAIN DUE TO VENTILATION
C     QAL IS LATENT HEAT GAIN DUE TO VENTILATION
      W = 0.01
      GO TO (20,30,40,50,60,70,80),I51
   20 CFM = 20*NOP
      GO TO 15
   30 CFM = 7.5*NOP
      GO TO 15
   40 CFM = 10*NOP
      GO TO 15
   50 CFM = 30*NOP
      GO TO 15
   60 CFM = 50*NOP
      GO TO 15
   70 CFM = 15*NOP
      GO TO 15
   80 CFM = 12*NOP
   15 QAS      = CFM*60.*0.075*(0.24+0.45*W)*(TO    -TR    )
      QAL      = CFM*60.*0.075*1076.*(WO    -WR    )
      QAR       = QAS    +QAL
      RETURN
      END

      SUBROUTINE INFILT(I52,V,TO,TR,WO,WR,QINFL)
C     THIS SUBROUTINE CALCULATES THE HEAT GAIN DUE TO INFILTRATION
C     XN IS THE RATE OF AIR CHANGE OF A ROOM PER HOUR
C     QSINF IS SENSIBLE HEAT GAIN DUE TO INFILTRATION
C     QLINF IS LATENT HEAT GAIN DUE TO INFILTRATION
C     QINFL IS TOTAL HEAT GAIN DUE TO INFILTRATION
C     TO IS TEMPERATURE OF OUTSIDE AIR
C     WO IS SPECIFIC HUMIDITY OF OUTSIDE AIR
C     TR IS TEMPERATURE OF INSIDE AIR
C     WR IS SPECIFIC HUMIDITY OF INSIDE AIR
C     V IS THE VOLUME OF A ROOM
      GO TO (10,20,30,40),I52
   10 XN=0.5
      GO TO 50
   20 XN = 1.0
```

```
      GO TO 50
   30 XN = 1.5
      GO TO 50
   40 XN = 2.0
   50 QSINF     = 0.018*XN*V*(TO    -TR    )
      QLINF     = 79.5*XN*V*(WO    -WR    )
      QINFL    = QSINF    +QLINF
      RETURN
      END

      SUBROUTINE GLASS(I53,SHGF,TO,TR,QGLAS)
C     THIS SUBROUTINE CALCULATES THE HEAT GAIN THROUGH GLASS
C     SC IS SHADING COEFFICIENT DESCRIBED IN SECTION III
C     SHGF IS THE SOLAR HEAT GAIN FACTOR
C     U IS OVERALL HEAT TRANSMISSION COEFFICIENT
C     QGLAS IS HEAT GAINED BY DOUBLE-STRENGTH GLASS
C     TO IS TEMPERATURE OF OUTSIDE AIR
C     TR IS TEMPERTURE OF INSIDE AIR
      GO TO(10,20,30,40),I53
   10 SC = 1.0
      U=0.81
      GO TO 50
   20 SC = 0.76
      U=0.81
      GO TO 50
   30 SC = 0.90
      U=0.81
      GO TO 50
   40 SC = 0.72
      U=0.54
   50 QGLAS       = SC*SHGF       +U*(TO    -TR    )
      RETURN
      END

      SUBROUTINE QWR(FO,X,Y,TR,TO,TS, Q ,ALPHA,TRAD,QX1,QX2,QY1,QY2,
     +IR ,CR)
C     THIS SUBROUTINE CALCULATES THE HEAT GAIN THROUGH WALL  OR
C     ROOF WHICH HAS DELAYED SURFACE
C     Q IS TRANSMITTED HEAT THROUGH WALL OR ROOF GIVEN BY
C     BY EQ.(3-5).
C     TS(I)  ARE THE WALL AND ROOF SURFACE TEMPERATURES
C     CR  IS COMMON RATIO
C     ALPHA  IS THE ABSORPTIVITY OF THE SURFACE OF THE MATERIAL
C     FO  IS THE OUTSIDE SURFACE FILM COEFFICIENT
C     TRAD  IS THE TOTAL SOLAR RADIATION IMPINGING UPON THE
C     WALL OR ROOF SURFACE.
      DIMENSION X(100),Y(100),TS(100)
      XSUM=0.
      YSUM=0.
      T1=TS(1)
      J= IR-1
      DO 10 I=2,J
      T2=TS(I)
      TS(I)=T1
      TDIFF=T1-TR
      T1=T2
      XSUM=XSUM+X(I)*TDIFF
      YSUM=YSUM+Y(I)*TDIFF
   10 CONTINUE
      DIFF=T2-TR
      QX2=CR*(QX2-QX1)+XSUM+X( IR)*DIFF
      QY2=CR*(QY2-QY1)+YSUM+Y(IR)*DIFF
      QY1=YSUM
      QX1=XSUM
      TS(1)=(X(1)*TR+FO*TO+ALPHA*TRAD-QX2)/(X(1)+FO)
      Q   =QY2+Y(1)*(TS(1)-TR)
      RETURN
      END
```

SPACE NO = 1

TEMPERATURE OF WALL SURFACE (DEG F)

1	0.0	0.0	0.0	0.0	76.793	0.0	0.0	0.0	80.012
2	0.0	0.0	0.0	0.0	76.683	0.0	0.0	0.0	79.109
3	0.0	0.0	0.0	0.0	77.013	0.0	0.0	0.0	78.701
4	0.0	0.0	0.0	0.0	77.680	0.0	0.0	0.0	78.705
5	0.0	0.0	0.0	0.0	78.653	0.0	0.0	0.0	79.083
6	0.0	0.0	0.0	0.0	79.923	0.0	0.0	0.0	79.908
7	0.0	0.0	0.0	0.0	83.336	0.0	0.0	0.0	87.258
8	0.0	0.0	0.0	0.0	88.031	0.0	0.0	0.0	96.725
9	0.0	0.0	0.0	0.0	93.350	0.0	0.0	0.0	106.790
10	0.0	0.0	0.0	0.0	98.105	0.0	0.0	0.0	116.201
11	0.0	0.0	0.0	0.0	101.250	0.0	0.0	0.0	123.329
12	0.0	0.0	0.0	0.0	103.585	0.0	0.0	0.0	129.169
13	0.0	0.0	0.0	0.0	103.900	0.0	0.0	0.0	131.830
14	0.0	0.0	0.0	0.0	103.177	0.0	0.0	0.0	132.559
15	0.0	0.0	0.0	0.0	100.479	0.0	0.0	0.0	129.619
16	0.0	0.0	0.0	0.0	96.111	0.0	0.0	0.0	123.334
17	0.0	0.0	0.0	0.0	91.495	0.0	0.0	0.0	115.265
18	0.0	0.0	0.0	0.0	87.017	0.0	0.0	0.0	103.926
19	0.0	0.0	0.0	0.0	84.507	0.0	0.0	0.0	97.436
20	0.0	0.0	0.0	0.0	82.529	0.0	0.0	0.0	92.633
21	0.0	0.0	0.0	0.0	80.802	0.0	0.0	0.0	88.827
22	0.0	0.0	0.0	0.0	79.351	0.0	0.0	0.0	85.776
23	0.0	0.0	0.0	0.0	78.178	0.0	0.0	0.0	83.334
24	0.0	0.0	0.0	0.0	77.284	0.0	0.0	0.0	81.410

HEAT GAIN THROUGH WALLS (BTU/HR*FT**2)

1	0.0	0.0	0.0	0.0	1.066	0.0	0.0	0.0	2.178
2	0.0	0.0	0.0	0.0	0.900	0.0	0.0	0.0	1.731
3	0.0	0.0	0.0	0.0	0.745	0.0	0.0	0.0	1.392
4	0.0	0.0	0.0	0.0	0.606	0.0	0.0	0.0	1.149
5	0.0	0.0	0.0	0.0	0.483	0.0	0.0	0.0	0.976
6	0.0	0.0	0.0	0.0	0.381	0.0	0.0	0.0	0.858
7	0.0	0.0	0.0	0.0	0.305	0.0	0.0	0.0	0.822
8	0.0	0.0	0.0	0.0	0.255	0.0	0.0	0.0	0.990
9	0.0	0.0	0.0	0.0	0.244	0.0	0.0	0.0	1.608
10	0.0	0.0	0.0	0.0	0.286	0.0	0.0	0.0	2.662
11	0.0	0.0	0.0	0.0	0.395	0.0	0.0	0.0	4.016
12	0.0	0.0	0.0	0.0	0.572	0.0	0.0	0.0	5.477
13	0.0	0.0	0.0	0.0	0.802	0.0	0.0	0.0	6.873
14	0.0	0.0	0.0	0.0	1.064	0.0	0.0	0.0	8.086
15	0.0	0.0	0.0	0.0	1.327	0.0	0.0	0.0	8.992
16	0.0	0.0	0.0	0.0	1.564	0.0	0.0	0.0	9.513
17	0.0	0.0	0.0	0.0	1.752	0.0	0.0	0.0	9.561
18	0.0	0.0	0.0	0.0	1.869	0.0	0.0	0.0	9.092
19	0.0	0.0	0.0	0.0	1.902	0.0	0.0	0.0	8.111
20	0.0	0.0	0.0	0.0	1.854	0.0	0.0	0.0	6.912
21	0.0	0.0	0.0	0.0	1.743	0.0	0.0	0.0	5.538
22	0.0	0.0	0.0	0.0	1.592	0.0	0.0	0.0	4.429
23	0.0	0.0	0.0	0.0	1.422	0.0	0.0	0.0	3.513
24	0.0	0.0	0.0	0.0	1.244	0.0	0.0	0.0	2.774