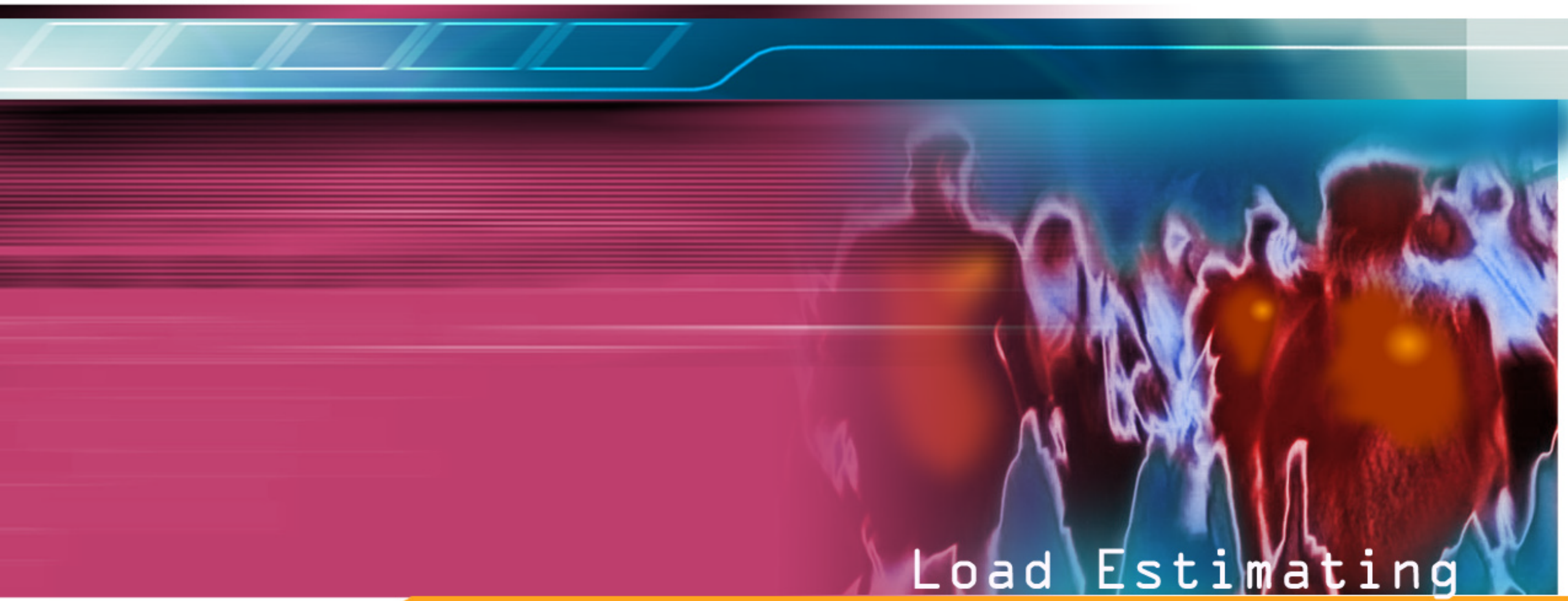
PART 1



Carrier

CONTENTS PART 1 WATER AND DX SYSTEMS

SYSTEM DESIGN MANUAL

SUMMARY OF PART ONE

This part of the System Design Manual presents data and examples to guide the engineer when preparing practical cooling and heating load estimates.

After the load has been determined, the "Applied Psychrometrics" chapter will bridge the gap between the load estimate and equipment selection.

The text of this Manual is offered as a general guide for the use of industry and of consulting engineers in designing systems. Judgment is required for application to specific installation, and Carrier is not responsible for any uses made of this text.

CHAPTER 1. BUILDIGN SURVEY AND LOAD ESTIMATE

The primary function of air conditioning is to maintain conditions that are (1) conducive to human comfort, or (2) required by a product, or process within a space. To perform this function, equipment of the proper capacity must be installed and controlled throughout the year. The equipment capacity is determined by the *actual* instantaneous peak load requirements; type of control is determined by the conditions to be maintained during peak and partial load. Generally, it is impossible to measure either the actual peak or the partial load in any given space; these loads must be estimated. It is for this purpose that the data contained in Part 1 has been compiled.

Before the load can be estimated, it is imperative that a comprehensive survey be made to assure accurate evaluation of the load components. If the building facilities and the actual instantaneous load within a given mass of the building are carefully studied, an economical equipment selection and system design can result, and smooth, trouble free performance is then possible.

The heat gain or loss is the amount of heat instantaneously coming into or going out of the space. *The actual load is defined as that amount of heat which is instantaneously added or removed by the equipment.* The instantaneous heat gain and the actual load on the equipment will rarely be equal, because of the thermal inertia or storage effect of the building structures surrounding a conditioned space.

Chapter 2, 4, 5, 6, and 7 contain the data from which the instantaneous heat gain or loss is estimated. *Chapter 3* provides the data and procedure for applying storage factors to the appropriate heat gains to result in the actual load. *Chapter 8* provides the bridge between the load estimate and the equipment selection. It furnishes the procedure for establishing the criteria to fulfill the conditions required by a given project.

The basis of the data and its use, with examples, are included in each chapter with the tables and charts; also an explanation of how each of the heat gains and the loads manifest themselves.

BUILDING SURVEY

SPACE CHARACTERISTICS AND HEAT LOAD SOURCES

An accurate survey of the load components of the space to be air conditioned is a basic requirement for a realistic estimate of cooling and heating loads. *The completeness and accuracy of this survey is the very foundation of the estimate, and its importance can not be overemphasized.* Mechanical and architectural drawings, complete field sketches and, in some cases, photographs of important aspects are part of a good survey. The following physical aspects must be considered:

1. *Orientation of building* - Location of the space to be air conditioned with respect to:
   a) Compass points-sun and wind effects.
   b) Nearby permanent structures-shading effects.
   c) Reflective surfaces-water, sand, parking lots, etc.
2. *Use of space(s)* – Office, hospital, department store, specialty shop, machine shop, factory, assembly plant, etc.
3. *Physical dimensions of space(s)* - Length, width, and height.
4. *Ceiling height* - Floor to floor height, floor to ceiling, clearance between suspended ceiling and beams.
5. *Columns and beams* - Size, depth, also knee braces.
6. *Construction materials* - Materials and thickness of walls, roof, ceiling, floors and partitions, and their relative position in the structure.
7. *Surrounding conditions* - Exterior color of walls and roof, shaded by adjacent building or sunlit. Attic space - unvented or vented, gravity or forced ventilation. Surrounding spaces conditioned or unconditioned-temperature of non-conditioned adjacent spaces, such as furnace or boiler room, and kitchens. Floor on ground, crawl space, basement.
8. *Windows* - Size and location, wood or metal sash, single or double hung. Type of shading device. Dimensions of reveals and overhangs.
9. *Doors* - Location, type, size, and frequency of use.
10. *Stairways, elevators, and escalators* - Location, temperature of space if open to unconditioned area. Horsepower of machinery, ventilated or not.
11. *People* - Number, duration of occupancy, nature of activity, any special concentration. At times, it is required to estimate the number

of people on the basis of square feet per person, or on average traffic.

12. *Lighting* - Wattage at peak. Type-incandescent, fluorescent, recessed, exposed. If the lights are recessed, the type of air flow over the lights, exhaust, return or supply, should be anticipated. At times, it is required to estimate the wattage on a basis of watts per sq ft, due to lack of exact information.
13. *Motors* – Location, nameplate and brake horsepower, and usage. The latter is of great significance and should be carefully evaluated.
    The power input to electric motors is not necessarily equal to the rated horsepower divided by the motor efficiency. Frequently these motors may be operating under a continuous overload, or may be operating at less than rated capacity. It is always advisable to measure the power input wherever possible. This is especially important in estimates for industrial installations where the motor machine load is normally a major portion of the cooling load.
14. *Appliances, business machines, electronic equipment* – Location, rated wattage, steam or gas consumption, hooded or unhooded, exhaust air quantity installed or required, and usage.
    Greater accuracy may be obtained by measuring the power or gas input during times of peak loading. The regular service meters may often be used for this purpose, provided power or gas consumption not contributing to the room heat gain can be segregated.
    Avoid pyramiding the heat gains from various appliances and business machines. For example, a toaster or a waffle iron may not be used during the evening, or the fry kettle may not be used during morning, or not all business machines in a given space may be used at the same time.
    Electronic equipment often requires individual air conditioning. The manufacturer's recommendation for temperature and humidity variation must be followed, and these requirements are often quite stringent.
15. *Ventilation* – Cfm per person, cfm per sq ft, scheduled ventilation (agreement with purchaser), see Chapter 6. Excessive smoking or odors, code requirements. Exhaust fans-type, size, speed, cfm delivery.
16. *Thermal storage* – Includes system operating schedule (12, 16 or 24 hours per day) specifically during peak outdoor conditions, permissible temperature swing in space during a design day, rugs on floor, nature of surface materials enclosing the space (see Chapter 3).
17. *Continuous or intermittent operation* – Whether system be required to operate every business day during cooling season, or only occasionally, such as churches and ballrooms. If intermittent operation, determine duration of time available for precooling or pulldown.

LOCATION OF EQUIPMENT AND SERVICE

The building survey should also include information which enables the engineer to select equipment location, and plan the air and water distribution systems. The following is a guide to obtaining this information:

1. *Available spaces* – Location of all stairwells, elevator shafts, abandoned smokestacks, pipe shafts, dumbwaiter shafts, etc., and spaces for air handing apparatus, refrigeration machines, cooling towers, pumps, and services (also see Item 5).
2. *Possible obstructions* – Locations of all electrical conduits, piping lines, and other obstructions or interferences that may be in the way of the duct system.
3. *Location of all fire walls and partitions* – Requiring fire dampers (also see Item 16).
4. *Location of outdoor air intakes* – In reference to street, other buildings, wind direction, dirt, and short-circuiting of unwanted contaminants.
5. *Power service* – Location, capacity, current limitations, voltage, phases and cycle, 3 or 4 wire; how additional power (if required) may be brought in and where.
6. *Water service* – Location, size of lines, capacity, pressure, maximum temperature.
7. *Steam service* – Location, size, capacity, temperature, pressure, type of return system.
8. *Refrigeration, brine or chilled water* (if furnished by customer) – Type of system, capacity, temperature, gpm, pressure.
9. *Architectural characteristics of space* – For selection of outlets that will blend into the space design.
10. *Existing air conveying equipment and ducts* – For possible reuse.

11. *Drains* – Location and capacity, sewage disposal.
12. *Control facilities* – Compressed air source and pressure, electrical.
13. *Foundation and support* – Requirements and facilities, strength of building.
14. *Sound and vibration control requirements* – Relation of refrigeration and air handling apparatus location to critical areas.
15. *Accessibility for moving equipment to the final location* – Elevators, stairways, doors, accessibility from street.
16. *Codes, local and national* – Governing wiring, drainage, water supply, venting of refrigeration, construction of refrigeration and air handling apparatus rooms, ductwork, fire dampers, and ventilation of buildings in general and apparatus rooms in particular.

AIR CONDITIONING LOAD ESTIMATE

The air conditioning load is estimated to provide the basis for selecting the conditioning equipment. It must take into account the heat coming into the space from outdoors on a design day, as well as the heat being generated within the space. A design day is defined as:

1. A day on which the dry-and wet-bulb temperatures are peaking simultaneously *(Chapter 2, "Design Conditions").*
2. A day when there is little or no haze in the air to reduce the solar heat *(Chapter 4, "Solar Heat Gain Thru Glass").*
3. All of the internal loads are normal *(Chapter 7, "Internal and System Heat Gain").*

The time of peak load can usually be established by inspection, although, in some cases, estimates must be made for several different times of the day.

Actually, the situation of having all of the loads peaking at the same time will very rarely occur. To be realistic, various diversity factors must be applied to some of the load components; refer to *Chapter 3, "Heat Storage, Diversity, and Stratification."*

The infiltration and ventilation air quantities are estimated as described in *Chapter 6.*

Fig. 1 illustrates an air conditioning load estimate form and is designed to permit systematic load evaluation. This form contains the references identified to the particular chapters of data and tables required to estimate the various load components.

OUTDOOR LOADS

The loads from outdoors consist of:

1. *The sun rays entering windows* – *Table 15*, *pages 44-49*, and *Table 16, page 52*, provide data from which the solar heat gain through glass is estimated.
   The solar heat gain is usually reduced by means of shading devices on the inside or outside of the windows; factors are contained in *Table 16*. In addition to this reduction, all or part of the window may be shaded by reveals, overhangs, and by adjacent buildings. *Chart 1, page 57*, and *Table 18, page 58*, provide an easy means of determining how much the window is shaded at a given time.
   A large portion of the solar heat gain is radiant and will be partially stored as described in *Chapter 3*. *Tables 7 thru 11, pages 30-34*, provide the storage factors to be applied to solar heat gains in order to arrive at the actual cooling load imposed on the air conditioning equipment. These storage factors are applied to *peak solar heat gains* obtained from *Table 6, page 29*, with overall factors from *Table 16*, *page 52*.
2. *The sun rays striking the walls and roof*- These, in conjunction with the high outdoor air temperature, cause heat to flow into the space. *Tables 19 and 20, pages 62 and 63*, provide equivalent temperature differences for sunlit and shaded walls and roofs. *Tables 21*, *22, 23, 24, 25, 27, and 28, pages 66-72*, provide the transmission coefficients or rates of heat flow for a variety of roof and wall constructions.
3. *The air temperature outside the conditioned space* – A higher ambient temperature causes heat to flow thru the windows, partitions, and floors. *Tables 25 and 26, pages 69 and 70*, and *Tables 29 and 30, pages 73 and 74*, provide the transmission coefficients. The temperature differences used to estimate the heat flow thru these structures are contained in the notes after each table.
4. *The air vapor pressure* – A higher vapor pressure surrounding conditioned space causes water vapor to flow thru the building materials. This load is significant only in low dewpoint applications. The data required to estimate this load is contained in Table 40, page 84. In comfort applications, this load is neglected.

CHAP REF	TABLE REFERENCES			
	ITEM	AREA OR QUANTITY	SUN GAIN OR TEMP. DIFF.	FACTOR
	SOLAR GAIN—GLASS			
3 & 4	GLASS — WITH STORAGE	SQ FT ×	TBLS 6&7, 8, 9, 10 OR 11 PP 29-34 ×	TBLS 16, 17 PP 52-54
	GLASS — WITH STORAGE	SQ FT ×	×	
	GLASS — WITHOUT STORAGE	SQ FT ×	TBL 15 PP 44-49 ×	TBL 15 CORR PP 44-49
	GLASS — WITHOUT STORAGE	SQ FT ×	×	
	SKYLIGHT	SQ FT ×	×	
	SOLAR & TRANS. GAIN—WALLS & ROOF			
5	WALL	SQ FT ×	TBL 19 P 62 ×	TBLS 21, 22, 23, 24 OR 25 PP 66-69
	WALL	SQ FT ×	×	
	WALL	SQ FT ×	×	
	WALL	SQ FT ×	×	
	ROOF—SUN	SQ FT ×	TBL 20 P 63 ×	TBLS 27, 28 71, 72
	ROOF—SHADED	SQ FT ×	×	
	TRANS. GAIN—EXCEPT WALLS & ROOF			
	ALL GLASS	SQ FT ×	NOTE 1 ×	TBL 33 P 76
	PARTITION	SQ FT ×	NOTES, PP 69, 70 ×	TBLS 25, 26 PP 69, 70
	CEILING	SQ FT ×	NOTES PP 73, 74 ×	TBL 29 OR 30 PP 73, 74
	FLOOR	SQ FT ×	×	
6	INFILTRATION	NOTE 4 CFM ×	NOTE 1 ×	1.08
	INTERNAL HEAT			
3 & 7	PEOPLE	PEOPLE ×		TBLS 14, 48 PP 38, 100
	POWER	HP OR KW ×		TBL 53 P 105
	LIGHTS	WATTS × 3.4 ×		TBLS 12, 14, 49 PP 35, 38, 101
	APPLIANCES, ETC.	TBLS 50-52 PP 101-103 ×		CORR BELOW TBLS 50-52
	ADDITIONAL HEAT GAINS	TBLS 54-57 PP 107-109 ×		
2 & 3			(TEMP SWING)	SUB TOTAL
	STORAGE	SQ FT ×	TBL 14 P 38 × (−	TBL 13 P 37)
				SUB TOTAL
	SAFETY FACTOR	P 113 %		
				ROOM SENSIBLE HEAT ■
7	SUPPLY DUCT HEAT GAIN (CHART 3 P 110) % + SUPPLY DUCT LEAK. LOSS (P 110) % + FAN H. P. (TBL 59 P 111) %			
8	OUTDOOR AIR	NOTE 3 CFM ×	NOTE 1 F ×	P 121 BF × 1.08
				EFFECTIVE ROOM SENSIBLE HEAT ■
	LATENT HEAT			
6	INFILTRATION	NOTE 4 CFM ×	NOTE 2	GR/LB × 0.68
3 & 7	PEOPLE	PEOPLE ×	TBLS 14, 48; PP 38, 100	
	STEAM	P 107 LB/HR × 1050		
	APPLIANCES, ETC.	TBLS 50-52 PP 101-103 ×	CORR BELOW TBLS	
	ADDITIONAL HEAT GAINS	TBL 58 P 109		
5	VAPOR TRANS.	SQ FT × 1/100 ×	NOTE 2 GR/LB ×	TBL 40 P 84
				SUB TOTAL
7	SAFETY FACTOR	P 113 %		
				ROOM LATENT HEAT
	SUPPLY DUCT LEAKAGE LOSS	P 110 %		
8	OUTDOOR AIR	NOTE 3 CFM ×	NOTE 2 GR/LB ×	P 121 BF × 0.68
				EFFECTIVE ROOM LATENT HEAT
				EFFECTIVE ROOM TOTAL HEAT ■
	OUTDOOR AIR HEAT			
	SENSIBLE:	NOTE 3 CFM ×	NOTE 1 F × (1 − P 121 BF)	× 1.08
	LATENT:	NOTE 3 CFM ×	NOTE 2 GR/LB × (1 − P 121 BF)	× 0.68
7	RETURN DUCT HEAT GAIN (CHART 3 P 110) % + RETURN DUCT LEAK. GAIN (P 112) % + HP PUMP (TBL 60 P 113) % + DEHUM. & PIPE LOSS (P 113) %			SUB TOTAL
				GRAND TOTAL HEAT ■

CHAP REF	TABLE REFERENCES					
	ESTIMATE FOR	LOCAL TIME / SUN TIME	PEAK LOAD	LOCAL TIME / SUN TIME		
	HOURS OF OPERATION					
2	CONDITIONS	DB	WB	% RH	DP	GR/LB
	OUTDOOR (OA)	TBLS 1-3	PP 10-19			
	ROOM (RM)	TBLS 4, 5	PP 20, 22, 23			
	DIFFERENCE		X X X	X X X	X X X	

OUTDOOR AIR (CHAP REF 6)

VENTILATION
______ PEOPLE × (TBL 45 P 97) ______ CFM/PERSON = ______
______ SQ FT × ______ CFM/SQ FT = ______
CFM VENTILATION ■

INFILTRATION
SWINGING / REVOLVING DOORS ______ PEOPLE × (TBL 41 P 90) ______ CFM/PERSON = ______
OPEN DOORS ______ DOORS × ______ CFM/DOOR = ______
EXHAUST FAN ______ TBLS 46, 47; P 98
CRACK ______ FEET × TBL 44 P 95 ______ CFM/FT = ______
CFM INFILTRATION ■ TBL 42 P 92

CFM OUTDOOR AIR THRU APPARATUS ■ NOTE 3 ______ CFM_{OA}

APPARATUS DEWPOINT (CHAP REF 8)

ESHF: EFFECTIVE SENS HEAT FACTOR = $\dfrac{\text{EFFECTIVE ROOM SENS. HEAT}}{\text{EFFECTIVE ROOM TOTAL HEAT}}$ = ______
TBL 65 P 145, OR PSYCH CHART, FIG 33 P 116

ADP: INDICATED ADP = ______ F SELECTED ADP = ______ F

DEHUMIDIFIED AIR QUANTITY

TEMP. RISE: $(1 - \underset{\text{P 121}}{\_\_\_}\ BF) \times (T_{RM}\_\_\_F - T_{ADP}\_\_\_F) = \_\_\_F$

DEHUM. CFM: $\dfrac{\text{EFFECTIVE ROOM SENS. HEAT}}{1.08 \times \_\_\_ F_{TEMP.\ RISE}} = \_\_\_ CFM_{DA}$

OUTLET TEMP. DIFF.: $\dfrac{\text{ROOM SENS. HEAT}}{1.08 \times \_\_\_ CFM_{DA}} = \_\_\_ F_{(RM-OUTLET\ AIR)}$*

SUPPLY AIR QUANTITY

SUPPLY CFM: $\dfrac{\text{ROOM SENS. HEAT}}{1.08 \times \_\_\_ F_{DESIRED\ DIFF}} = \_\_\_ CFM_{SA}$

BYPASS CFM: $\_\_\_ CFM_{SA} - \_\_\_ CFM_{DA} = \_\_\_ CFM_{BA}$

RESULTING ENT & LVG CONDITIONS AT APPARATUS

EDB: $T_{RM}\_\_\_F + \dfrac{CFM_{OA}}{CFM\dagger}\ (\text{P 125}) \times (T_{OA}\_\_\_F - T_{RM}\_\_\_F) = T_{EDB}\_\_\_F$

LDB: $T_{ADP}\_\_\_F + \underset{\text{P 121}}{\_\_\_}\ BF \times (T_{EDB}\_\_\_F - T_{ADP}\_\_\_F) = T_{LDB}\_\_\_F$

FROM PSYCH. CHART: T_{EWB} ______ F, T_{LWB} ______ F

NOTES

1. USE DRY-BULB (DB) TEMPERATURE DIFFERENCE FROM TOP OF ESTIMATE FORM
2. USE MOISTURE CONTENT (GR/LB) DIFFERENCE FROM TOP OF ESTIMATE FORM
3. NORMALLY, USE "CFM VENTILATION" FOR "CFM OUTDOOR AIR." HOWEVER, WHEN INFILTRATION IS TO BE OFFSET, REFER TO PAGE 92 TO DETERMINE "CFM OUTDOOR AIR."
4. WHEN INFILTRATION IS NOT TO BE OFFSET, AND "CFM VENTILATION" IS LESS THAN "CFM INFILTRATION," THEN THE EXCESS INFILTRATION IS ACCOUNTED FOR HERE.

*IF THIS ΔT IS TOO HIGH, DETERMINE SUPPLY CFM FOR DESIRED DIFFERENCE BY SUPPLY AIR QUANTITY FORMULA.

†WHEN BYPASSING A MIXTURE OF OUTDOOR AND RETURN AIR, USE SUPPLY CFM. WHEN BYPASSING RETURN AIR ONLY, USE DEHUMIFIED CFM.

With Carrier Masthead Form E20. Without Carrier Masthead Form E5024.

FIG. 1-AIR CONDITIONING LOAD ESTIMATE

5. *The wind blowing against a side of the building*- Wind causes the outdoor air that is higher in temperature and moisture content to infiltrate thru the cracks around the doors and windows, resulting in localized sensible and latent heat gains. All or part of this infiltration may be offset by air being introduced thru the apparatus for ventilation purposes. *Chapter 6* contains the estimating data.
6. *Outdoor air usually required for ventilation purposes* – Outdoor air is usually necessary to flush out the space and keep the odor level down. This ventilation air imposes a cooling and dehumidifying load on the apparatus because the heat and/or moisture must be removed. Most air conditioning equipment permits some outdoor air to bypass the cooling surface (see *Chapter 8*). This bypassed outdoor air becomes a load within the conditioned space, similar to infiltration; instead of coming thru a crack around the window, it enters the room thru the supply air duct. The amount of bypassed outdoor air depends on the type of equipment used as outlined in *Chapter 8. Table 45, page 97*, provides the data from which the ventilation requirements for most comfort applications can be estimated.

The foregoing is that portion of the load on the air conditioning equipment that originates outside the space and is common to all applications.

INTERNAL LOADS

Chapter 7 contains the data required to estimate the heat gain from most items that generate heat within the conditioned space. The internal load, or heat generated within the space, depends on the character of the application. Proper diversity and usage factor should be applied to all internal loads. As with the solar heat gain, some of the internal gains consist of radiant heat which is partially stored (as described in *Chapter 3*), thus reducing the load to be impressed on the air conditioning equipment.

Generally, internal heat gains consist of some or all of the following items:

1. *People* – The human body thru metabolism generates heat within itself and releases it by radiation, convection, and evaporation from the surface, and by convection and evaporation in the respiratory tract. The amount of heat generated and released depends on surrounding temperature and on the activity level of the person, as listed in *Table 48, page 100*.
2. *Lights* – Illuminants convert electrical power into light and heat (refer to *Chapter 7*). Some of the heat is radiant and is partially stored (see *Chapter 3*).
3. *Appliances* – Restaurants, hospitals, laboratories, and some specialty shops (beauty shops) have electrical, gas, or steam appliances which release heat into the space. *Tables 50 thru 52, pages 101-103*, list the recommended heat gain values for most appliances when not hooded. If a positive exhaust hood is used with the appliances, the heat gain is reduced.
4. *Electric calculating machines* – Refer to manufacturer's data to evaluate the heat gain from electric calculating machines. Normally, not all of the machines would be in use simultaneously, and, therefore, a usage or diversity factor should be applied to the full load heat gain. The machines may also be hooded, or partially cooled internally, to reduce the load on the air conditioning system.
5. *Electric motors* – Electric motors are a significant load in industrial applications and should be thoroughly analyzed with respect to operating time and capacity before estimating the load (see *Item 13* under "*Space Characteristics and Heat Load Sources*"). It is frequently possible to actually measure this load in existing applications, and should be so done where possible. *Table 53*, *page 105*, provides data for estimating the heat gain from electric motors.
6. *Hot pipes and tanks* – Steam or hot water pipes running thru the air conditioned space, or hot water tanks in the space, add heat. In many industrial applications, tanks are open to the air, causing water to evaporate into the space. *Tables 54 thru 58, pages 107-109* provide data for estimating the hear gain from these sources.
7. *Miscellaneous sources* – There may be other sources of heat and moisture gain within a space, such as escaping steam (industrial cleaning devices, pressing machines, etc.), absorption of water by hygroscopic material (paper, textiles, etc.); see *Chapter 7.*

In addition to the heat gains from the indoor and outdoor sources, the air conditioning equipment and duct system gain or lose heat. The fans and pumps required to distribute the air or water thru the system add heat;

heat is also added to supply and return air ducts running thru warner or hot spaces; cold air may leak out of the supply duct and hot air may leak into the return duct. The procedure for estimating the heat gains from these sources in percentage of room sensible load, room latent load, and grand total heat load is contained in *Chart 3, page 110, and Tables 59 and 60, pages 111-113.*

HEATING LOAD ESTIMATE

The heating load evaluation is the foundation for selecting the heating equipment. Normally, the heating load is estimated for the winter design temperatures (*Chapter 2*) usually occurring at night; therefore, no credit is taken for the heat given off by internal sources (people, lights, etc.). This estimate must take into account the heat loss thru the building structure surrounding the spaces and the heat required to offset the outdoor air which may infiltrate and/or may be required for ventilation. *Chapter 5* contains the transmission coefficients and procedures for determining heat loss. *Chapter 6* contains the data for estimating the infiltration air quantities. *Fig. 2* illustrates a heating estimate form for calculating the heat loss in a building structure.

Another factor that may be considered in the evaluation of the heating load is temperature swing. Capacity requirements may be reduced when the temperature within the space is allowed to drop a few degrees during periods of design load. This, of course, applies to continuous operation only. *Table 4, page 20*, provides recommended inside design conditions for various applications, and *Table 13, page 37*, contains the data for estimating the possible capacity reduction when operating in this manner.

The practice of drastically lowering the temperature to 50 F db or 55 F db when the building is unoccupied precludes the selection of equipment based on such capacity reduction. Although this type of operation may be effective in realizing fuel economy, *additional* equipment capacity is required for pickup. In fact, it may be desirable to provide the additional capacity, even if continuous operation is contemplated, because of pickup required after forced shutdown. It is, therefore, evident that the use of storage in reducing the heating load for the purpose of equipment selection should be applied with care.

HIGH ALTITUDE LOAD CALCULATIONS

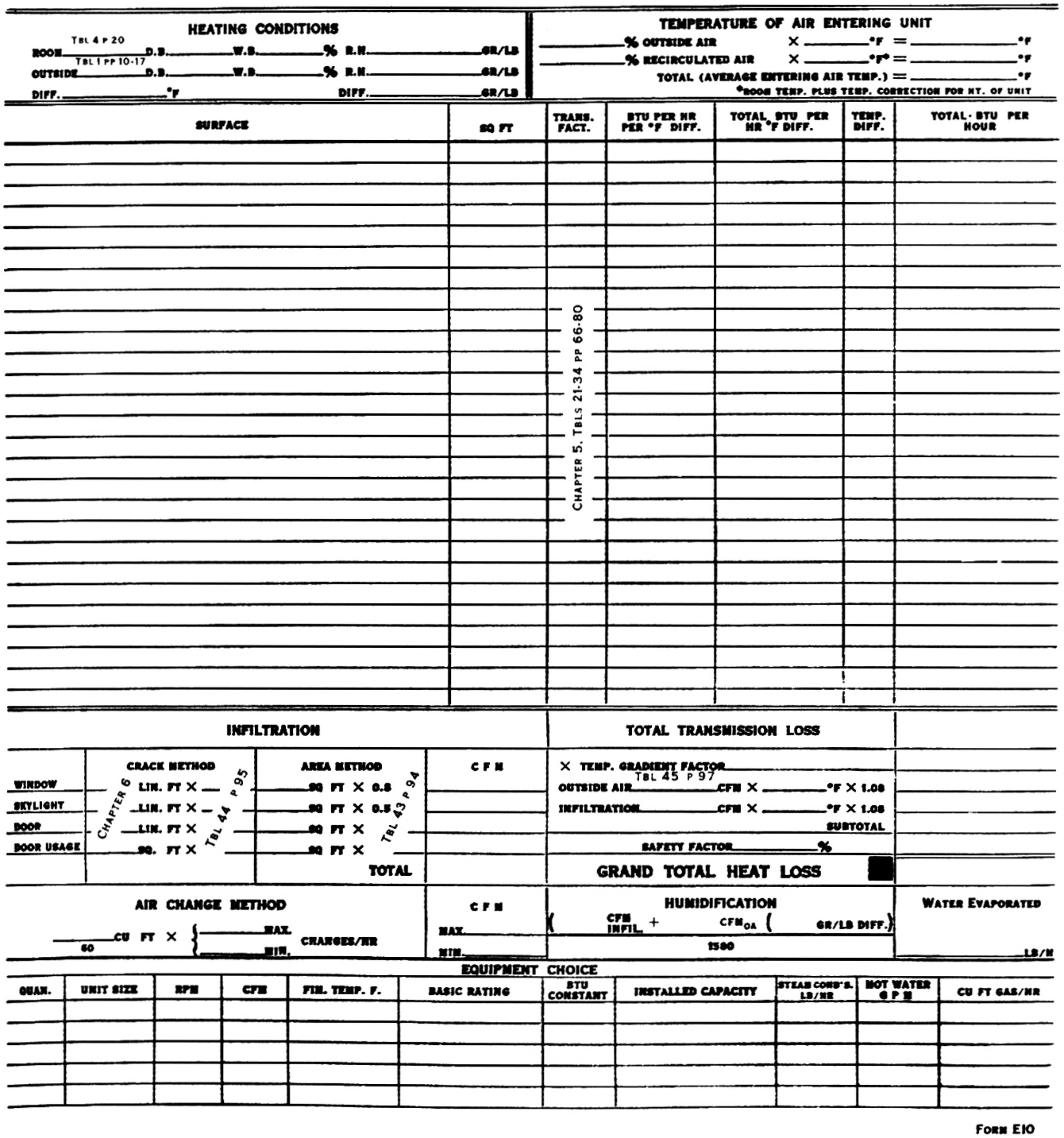
Since air conditioning load calculations are based on pounds of air necessary to handle a load, a decrease in density means an increase in cfm required to satisfy the given sensible load. The weight of air required to meet the latent load is decreased because of the higher latent load capacity of the air at higher altitudes (greater gr per lb per degree difference in dewpoint temperature). For the same dry-bulb and percent relative humidity, the wetbulb temperature decreases (except at saturation) as the elevation above sea level increases.

The following adjustments are required for high altitude load calculations (see *Chapter 8, Table 66, page 148*):

1. Design room air moisture content must be adjusted to the required elevation.
2. Standard load estimating methods and forms are used for load calculations, except that the factors affecting the calculations of volume and sensible and latent heat of air must be multiplied by the relative density at the particular elevation.
3. Because of the increased moisture content of the air, the effective sensible heat factor must be corrected.

EQUIPMENT SELECTION

After the load is evaluated, the equipment must be selected with capacity sufficient to offset this load. The air supplied to the space must be of the proper conditions to satisfy both the sensible and latent loads estimated. *Chapter 8*, "*Applied Psychrometrics*," provides procedures and examples for determining the criteria from which the air conditioning equipment is selected (air quantity, apparatus dewpoint, etc.).



HEATING CONDITIONS

	TBL 4 P 20			
ROOM	D.B.	W.B.	% R.H.	GR/LB
	TBL 1 PP 10-17			
OUTSIDE	D.B.	W.B.	% R.H.	GR/LB
DIFF.	°F		DIFF.	GR/LB

TEMPERATURE OF AIR ENTERING UNIT

% OUTSIDE AIR	×	°F	=	°F
% RECIRCULATED AIR	×	°F*	=	°F
TOTAL (AVERAGE ENTERING AIR TEMP.)			=	°F

*ROOM TEMP. PLUS TEMP. CORRECTION FOR HT. OF UNIT

SURFACE	SQ FT	TRANS. FACT.	BTU PER HR PER °F DIFF.	TOTAL BTU PER HR °F DIFF.	TEMP. DIFF.	TOTAL BTU PER HOUR
		CHAPTER 5, TBLS 21-34 PP 66-80				

INFILTRATION

	CRACK METHOD (CHAPTER 6, TBL 44 P 95)	AREA METHOD (TBL 43 P 94)	CFM
WINDOW	LIN. FT ×	SQ FT × 0.8	
SKYLIGHT	LIN. FT ×	SQ FT × 0.5	
DOOR	LIN. FT ×	SQ FT ×	
DOOR USAGE	SQ. FT ×	SQ FT ×	
		TOTAL	

TOTAL TRANSMISSION LOSS

× TEMP. GRADIENT FACTOR (TBL 45 P 97)	
OUTSIDE AIR ___ CFM × ___ °F × 1.08	
INFILTRATION ___ CFM × ___ °F × 1.08	
SUBTOTAL	
SAFETY FACTOR ___ %	
GRAND TOTAL HEAT LOSS	

AIR CHANGE METHOD

___ CU FT × { ___ MAX. / ___ MIN. } CHANGES/HR ÷ 60

CFM: MAX. ___ MIN. ___

HUMIDIFICATION

(CFM INFIL. + CFM OA) (GR/LB DIFF.) ÷ 1580

WATER EVAPORATED ___ LB/H

EQUIPMENT CHOICE

QUAN.	UNIT SIZE	RPM	CFM	FIN. TEMP. F.	BASIC RATING	BTU CONSTANT	INSTALLED CAPACITY	STEAM COND'S. LB/HR	HOT WATER GPM	CU FT GAS/HR

FORM E10

FIG. 2- HEATING LOAD ESTIMATE

CHAPTER 2. DESIGN CONDITIONS

This chapter presents the data from which the outdoor design conditions are established for various localities and inside design conditions for various applications. The design conditions established determine the heat content of air, both outdoor and inside. They directly affect the load on the air conditioning equipment by influencing the transmission of heat across the exterior structure and the difference in heat content between the outdoor and inside air. For further details, refer to *Chapters 5 and 6.*

OUTDOOR DESIGN CONDITIONS – SUMMER AND WINTER

The outdoor design conditions listed in *Table 1* are the industry accepted design conditions as published in ARI Std. 530-56 and the 1958 ASHAE Guide. The conditions, as listed, permit a choice of outdoor dry-bulb and wet-bulb temperatures for different types of applications as outlined below.

NORMAL DESIGN CONDITIONS – SUMMER

Normal design conditions are recommended for use with *comfort and industrial cooling applications* where it is occasionally permissible to exceed the design room conditions. These outdoor design conditions are the *simultaneously occurring dry-bulb and wet-bulb temperatures and moisture content,* which can be expected to be exceeded a few times a year for short periods. The dry-bulb is exceeded more frequently than the wet-bulb temperature. And usually when the wet-bulb is lower than design.

When cooling and dehumidification (dehydration) are performed separately with these types of applications, use the normal design dry-bulb temperature for selecting the sensible cooling apparatus; use a moisture content corresponding to the normal design wet-bulb temperature and 80 % rh for selecting the dehumidifier (dehydrator)

Daily range is the average difference between the high and low dry-bulb temperatures for a 24-hr period on a design day. This range varies with local climate conditions.

MAXIMUM DESIGN CONDITIONS-SUMMER

Maximum summer design conditions are recommended for *laboratories and industrial applications* where exceeding the room design conditions for even short periods of time can be detrimental to a product or process.

The maximum design dry-bulb and wet-bulb temperatures are simultaneous peaks (not individual peaks). The moisture content is an individual peak, and is listed only for use in the selection of separate cooling and dehumidifying systems for closely controlled spaces. Each of these conditions can be expected to be exceeded no more than 3 hours in a normal summer.

NORMAL DESIGN CONDITIONS – WINTER

Normal winter design conditions are recommended for use with all *comfort and industrial heating applications.* The outdoor dry-bulb temperature can be expected to go below the listed temperatures a few times a year, normally during the early morning hours. The annual degree days listed are the sum of all the days in the year on which the daily mean temperature falls below 65 F db, times the number of degrees between 65 F db and the daily mean temperature.

TABLE 1—OUTDOOR DESIGN CONDITIONS—SUMMER AND WINTER

STATE AND CITY	NORMAL DESIGN COND.—SUMMER July at 3:00 PM			AVG. DAILY RANGE	MAXIMUM DESIGN COND.—SUMMER July at 3:00 PM			NORMAL DESIGN COND. WINTER		WIND DATA Avg. Velocity and Prevailing Direction		Elevation Above Sea Level (ft)	Latitude (deg)
	Dry-Bulb (F)	Wet-Bulb (F)	Moisture Content* (gr/lb of dry air)	Dry-Bulb (F)	Dry-Bulb (F)	Wet-Bulb (F)	Moisture Content† (gr/lb of dry air)	Dry-Bulb (F)	Annual Degree Days	Summer	Winter		
ALABAMA													
Anniston	95	78	117.5	19				5	2806			733	34
Birmingham	95	78	117.5	19	99			10	2611	5.0 S	8.0 N	694	34
Mobile	95	80	131	12	95	82	155.6	15	1566	9.0 SW	9.9 N	10	31
Montgomery	95	78	117.5	15				10	2071		7.5 NW	293	32
ARIZONA													
Flagstaff	90	65	81	26	90			−10	7242		7.7 SW	6,894	35
Phoenix	105	76	94	30	113	78	126.9	25	1441	5.0 W	5.4 E	1,108	33
Tucson	105	72	77	30				25		5.0 W	5.2 NW	2,376	32
Winslow	100	70	85					−10				4,853	35
Yuma	110	78	93	30				30	1036		6.7 N	146	33
ARKANSAS													
Fort Smith	95	76	104.5	16	103			10	3226	7.0 E	8.3 E	448	35
Little Rock	95	78	117.5	16	103	83	145.5	5	3009	6.0 NW	8.3 NW	324	35
CALIFORNIA													
Bakersfield	105	70	54	25				25				499	35
El Centro	110	78	94									43	33
Eureka	90	65	52					30	4758	7.0 N	7.3	132	41
Fresno	105	74	76	35	110	75	95.9	25	2403	8.0 NW	5.4 NW	287	37
Laguna Beach				9	82	70	103.0					10	34
Long Beach	90	70	78	14								47	34
Los Angeles	90	70	78	14	94			35	1391	6.0 SW	6.4 NE	261	34
Oakland	85	65	60	17	94	68	99.3	30				17	38
Montague								0				2,635	42
Pasadena	95	70	70										34
Red Bluff	100	70	62									305	40
Sacramento	100	72	73	18				30	2680		7.2 SE	116	39
San Bernadino	105	72	65										34
San Diego	85	68	75	10	88	74	78.4	35	1596	7.0 W	6.3 NW	26	33
San Francisco	85	65	60	17				35	3137	12.0 W	7.5 N	17	38
San Jose	91	70	76.5					25	2823			100	37
Williams				40	110	80	74.4					86	39
COLORADO													
Denver	95	64	60	25	99	68	89.4	−10	5839	7.0 S	7.5 S	5,221	40
Durango	95	65	70									6,558	37
Fort Collins								−30					41
Grand Junction	95	65	62	24	102	68	86.2	−15	5613	6.0 SE	4.4 NW	4,587	39
Pueblo	95	65	63	25				−20	5558		7.9 NW	4,770	38
CONNECTICUT													
Bridgeport	95	75	99	14				0				9	41
Hartford	93	75	102	16	94	82		0	6113	7.0 S	8.7 NW	58	42
New Haven	95	75	99	14	95			0	5880	7.0 S	9.4 N	23	41
Waterbury								−15					42
DELAWARE													
Wilmington	95	78	117.5	15				0		10.0 SW	NW	134	40
DIST. OF COLUMBIA													
Washington	95	78	117.5	18	99	84	155.6	0	4561	5.0 S	7.8 NW	72	39
FLORIDA													
Apalachicola	95	80	131					25	1252	5.0 SW	8.4	23	30
Jacksonville	95	78	117.5	17	99	82	150.5	25	1185	8.0 SW	9.0 NE	18	30
Key West	98	78	112.5					45	59	9.0 SE	10.6 NE	23	25
Miami	91	79	131	12	92	81	150.5	35	185	7.0 SE	10.1 E	11	26
Pensacola	95	78	117.5	12				20	1281		10.9 N	408	31
Tampa	95	78	117.5	14	95			30	571	6.0 NE	8.6 NE	25	28
Tallahassee								25	1463		N	68	30

*Corresponds to dry-bulb and wet-bulb temperatures listed, and is corrected for altitude of city.

†Corresponds to peak dewpoint temperature, corrected for altitude.

TABLE 1—OUTDOOR DESIGN CONDITIONS—SUMMER AND WINTER (Contd)

STATE AND CITY	NORMAL DESIGN COND.—SUMMER July at 3:00 PM			AVG. DAILY RANGE	MAXIMUM DESIGN COND.—SUMMER July at 3:00 PM			NORMAL DESIGN COND. WINTER		WIND DATA Avg. Velocity and Prevailing Direction		Elevation Above Sea Level (ft)	Latitude (deg)
	Dry-Bulb (F)	Wet-Bulb (F)	Moisture Content* (gr/lb of dry air)	Dry-Bulb (F)	Dry-Bulb (F)	Wet-Bulb (F)	Moisture Content† (gr/lb of dry air)	Dry-Bulb (F)	Annual Degree Days	Summer	Winter		
GEORGIA													
Atlanta	95	76	109.5	18	101	82	150.5	10	2985	7.0 NW	11.7 NW	975	34
Augusta	98	76	100	18				10	2306		6.5 NW	195	34
Brunswick	95	78	117.5										31
Columbus	98	76	100										33
Macon	95	78	117.5	18				15	2338	5.0 S	6.7 NW	408	33
Savannah	95	78	117.5	17	99			20	1635	8.0 SW	9.5 NW	42	32
IDAHO													
Boise	95	65	54.5	31	109	71	92.6	−10	5678	5.0 NW	9.1 SE	2,705	44
Lewiston	95	65	44	28				5	5109		4.1 E	763	46
Pocatello	95	65	61	28	100			−5	6741		8.9 SE	4,468	43
Twin Falls								−10			W		42
ILLINOIS													
Cairo	98	78	112.5					0	3957		9.8	319	37
Chicago	95	75	99	19	104	80	140.6	−10	6282	10.0 NE	12.0 SW	594	42
Danville								−5			NW		40
Moline	96	76	103	22	103	83	155.6	−10				594	41
Peoria	96	76	103	20	100			−10	6004	8.0 S	8.3 S	602	41
Springfield	98	77	106	20				−10	5446		11.9 NW	603	40
INDIANA													
Evansville	95	78	117.5	19	102	82	150.5	0	4410	7.0 SW	9.7 S	388	38
Fort Wayne	95	75	99	20	100			−10	6232	8.0 SW	10.4 SW	777	41
Indianapolis	95	76	104.5	18	99			−10	5458	9.0 SW	11.3 S	715	40
South Bend								−5			SW	773	42
Terre Haute	95	78	124									1,146	40
IOWA													
Cedar Rapids								−5					
Davenport	95	78	117.5	18				−15	6252		10.5 NW	648	42
Des Moines	95	78	123	18	102			−15	6375	6.0 SW	10.1 NW	800	42
Dubuque	95	78	117.5					−20	6820		7.1	740	43
Fort Dodge								−20					42
Keokuk	95	78	117.5					−10	5663		8.2 SW	637	41
Sioux City	95	78	124	19	102			−20	6905	10.0 S	11.5 NW	1,111	43
Waterloo								−15					43
KANSAS													
Concordia	95	78	125	20				−10	5425		7.7 S	1,425	39
Dodge City	95	78	132	21	106			−10	5069		10.6	2,522	38
Salina					111			−15			NW	1,226	39
Topeka	100	78	109.5	19				−10	5075	10.0 S	9.2 S	991	39
Wichita	100	75	98	21	110	79	126.9	−10	4644	11.0 S	12.4 S	1,300	38
KENTUCKY													
Lexington								0	4792		13.3 SW	989	38
Louisville	95	78	117.5	22	99			0	4417	7.0 SW	9.8 SW	459	38
LOUISIANA													
Alexandria								20			N	89	32
New Orleans	95	80	131	13	95	83	161.2	20	1203	6.0 SW	8.6 N	9	30
Shreveport	100	78	109.5	15	102	83	150.5	20	2132	5.0 S	8.8 SE	197	33
MAINE													
Augusta	90	73	95	13								362	45
Bangor	90	73	95	13									45
Bar Harbor								−15			NW		44
Belfast								−5					44
Eastport	90	70	78	13				−10	8445	7.0 S	12.6 W	100	45
Millinocket								−15					46
Presque Isle									9644		NW		47
Portland	90	73	95	13	93			−5	7377	7.0 S	10.4 NW	47	44
Rumford								−20					44

*Corresponds to dry-bulb and wet-bulb temperatures listed, and is corrected for altitude of city.

†Corresponds to peak dewpoint temperature, corrected for altitude.

TABLE 1—OUTDOOR DESIGN CONDITIONS—SUMMER AND WINTER (CONT.)

STATE AND CITY	NORMAL DESIGN COND.—SUMMER July at 3:00 PM			AVG. DAILY RANGE	MAXIMUM DESIGN COND.—SUMMER July at 3:00 PM			NORMAL DESIGN COND. WINTER		WIND DATA Avg. Velocity and Prevailing Direction		Elevation Above Sea Level (ft)	Latitude (deg)
	Dry-Bulb (F)	Wet-Bulb (F)	Moisture Content* (gr/lb of dry air)	Dry-Bulb (F)	Dry-Bulb (F)	Wet-Bulb (F)	Moisture Content† (gr/lb of dry air)	Dry-Bulb (F)	Annual Degree Days	Summer	Winter		
MARYLAND													
Baltimore	95	78	117.5	18	99			0	4487	6.0 SW	8.2 NW	14	39
Cambridge								5			NW		39
Cumberland	95	75	99	18									39
Frederick								−5			NW		40
Frostburg								−5			W		40
Salisbury								10			NW		40
MASSACHUSETTS													
Amherst								−10			NW		42
Boston	92	75	104	13	96	78	135.9	0	5936	9.0 SW	12.4 W	14	42
Fall River								−10					42
Fitchburg	93	75	102	17				−10	6743	W	NW	402	43
Lowell								−15					43
Nantucket	95	75	99					0			14.8	45	41
New Bedford								0					42
Plymouth								−5			W		42
Springfield	93	75	102	17				−10		9.0 SW		199	42
Worcester	93	75	102	17				0				625	42
MICHIGAN													
Alpena	95	75	99					−10	8278		11.0 SW	615	45
Big Rapids								−15			NW		43
Detroit	95	75	99	19	101	79	135.9	−10	6560	10.0 SW	12.0 SW	619	42
Escanaba								−15	8777		9.5 NW		46
Flint	95	75	99	20				−10		W	W	766	43
Grand Rapids	95	75	99	20	98			−10	6702	8.0 W	12.1 NW	638	43
Kalamazoo								−5			W		42
Lansing	95	75	104	20				−10	7149		9.8 SW	861	43
Ludington								−10	7458		11.9 W		44
Marquette	93	73	90	20	96			−10	8745		10.6 NW	652	47
Saginaw	95	75	99									601	43
Sault Ste Marie								−20	9307		8.9 SE	724	47
MINNESOTA													
Alexandria								−25			NW		
Duluth	93	73	96	19				−25	9723	13.4 SW	13.4 SW	1,128	47
Minneapolis	95	75	103	17	102			−20	7966	10.0 S	11.3 NW	839	45
St. Cloud								−25					46
St. Paul	95	75	99	17	103	79	131.1	−20	7975	8.0 SE	9.5 NW	719	45
MISSISSIPPI													
Jackson				21	103	83	155.6	15		5.0 SW	7.7 SE	316	32
Meridian	95	79	124	21				10	2330	4.0 SW	6.3 N	410	32
Vicksburg	95	78	117.5	21	96			10	2069	6.0 SW	8.3	226	32
MISSOURI													
Columbia	100	78	109.5	19				−10	5070		8.9 SW	739	39
Kansas City	100	76	106.5	19	109	79	135.9	−10	4962	9.0 S	10.3 NW	741	39
Kirksville				19	108	82	150.5			SW		969	40
St. Louis	95	78	117.5	20	108	81	135.9	0	4596	9.0 S	11.8 S	465	39
St. Joseph								−10	5596		9.3 NW	817	40
Springfield				18	98	79	135.9	−10	4569	8.0 S	10.9 SE	1,301	37
MONTANA													
Billings	90	66	70	20	104			−25	7213		12.4 W	3,119	46
Butte								−20			NW	5,538	46
Great Falls								−20			SW	3,687	48
Havre	95	70	82	20				−30	8416	7.0 E	9.4 SW	2,498	49
Helena	95	67	71	20	97	70	77.4	−20	7930	7.0 SW	7.4 SW	4,090	47
Kalispell	95	65	56					−20	8032		5.2	3,004	48
Miles City								−35	7591		5.6 S	2,629	47
Missoula	95	66	49	20				−20	7604		E	3,205	47

*Corresponds to dry-bulb and wet-bulb temperatures listed, and is corrected for altitude of city.

†Corresponds to peak dewpoint temperature, corrected for altitude.

TABLE 1—OUTDOOR DESIGN CONDITIONS—SUMMER AND WINTER (CONT.)

STATE AND CITY	NORMAL DESIGN COND.—SUMMER July at 3:00 PM Dry-Bulb (F)	Wet-Bulb (F)	Moisture Content* (gr/lb of dry air)	AVG. DAILY RANGE Dry-Bulb (F)	MAXIMUM DESIGN COND.—SUMMER July at 3:00 PM Dry-Bulb (F)	Wet-Bulb (F)	Moisture Content† (gr/lb of dry air)	NORMAL DESIGN COND. WINTER Dry-Bulb (F)	Annual Degree Days	WIND DATA Avg. Velocity and Prevailing Direction Summer	Winter	Elevation Above Sea Level (ft)	Latitud (deg)
NEBRASKA													
Grand Island								−20				1,856	41
Lincoln	95	78	124	20	106			−10	5980	9.0 S	10.6 S	1,180	41
Norfolk								−15		NW	NW		42
North Platte	95	78	135	26	104	76	74.4	−20	6384	6.0 S	7.9 W	2,805	41
Omaha	95	78	123	20	108	80	131.1	−10	6095	8.0 S	9.7 NW	978	41
Valentine	95	78	135	20				−25	7197		9.2	2,627	43
York								−15			NW		
NEVADA													
Las Vegas	115	75	76	40				20			S	1,882	36
Reno	95	65	62	41	102	66	66.9	−5	5621	7.0 SW	6.0 W	4,493	40
Tonopah								5	5812		9.9 SE	5,421	38
Winnemucca	95	65	62	40				−15	6357	7.0 SW	8.1 NE	4,293	42
NEW HAMPSHIRE													
Berlin								−25					45
Concord	90	73	95	14				−15	7400	5.0 NW	6.2 NW	289	43
Keene								−20			NW		43
Manchester	90	73	95	14	92							171	43
Portsmouth	90	73	95	14									43
NEW JERSEY													
Atlantic City	95	78	117.5	14				5	5015	13.0 SW	15.8 NW	8	39
Bloomfield	95	75	99	14								125	41
Camden				14	102	82	145.5	0		10.0 SW		30	40
East Orange	95	75	99	14								173	41
Jersey City	95	75	99					0			NW		41
Newark	95	75	99	14	99	81	140.6	0	5500	13.0 SW	17.1 NW	10	41
Paterson	95	75	99	14	95					13.0 SW		10	41
Sandy Hook								0	5369		16.1		41
Trenton	95	78	117.5	14	96			0	5256	9.0 SW	10.9 NW	56	40
NEW MEXICO													
Albuquerque	95	70	94.5	26	98	68	95.9	0	4517	8.0 SW	7.3 N	5,101	35
Roswell	95	70	87	25				−10	3578	6.0 S	7.1 S	3,643	32
Santa Fe	90	65	80	30	90			0	6123	6.0 SE	7.1 NE	7,000	36
NEW YORK													
Albany	93	75	102	18	97	78	131.1	−10	6648	7.0 S	10.5 S	19	43
Binghamton	95	75	103.5					−10	6818		6.8 NW	915	42
Buffalo	93	73	90	18	93	77	126.9	−5	6925	12.0 SW	17.1 W	604	43
Canton	90	73	95					−25	8305	8.0	10.5	458	43
Cortland								−10			NW		43
Glens Falls								−15			W		43
Ithaca								−15	6914		11.3 NW		42
Jamestown								−10			SW		42
Lake Placid								−20			W		44
New York City	95	75	99	14	100	81	145.5	0	5280	13.0 S	16.8 NW	10	41
Ogdensburg								−20			SW		45
Oneonta								−15			SW		43
Oswego	93	73	90					−10	7186		12.1 S	363	43
Rochester	95	75	102	18	95			−5	6772	8.0 SW	9.6 W	543	43
Schenectady	93	75	102	18								235	43
Syracuse	93	75	102	18	96			−10	6899	9.0 S	11.2 S	400	43
Watertown								−15			SW		44
NORTH CAROLINA													
Asheville	93	75	114.5	19	93			0	4236	6.0 NW	9.5 NW	2,192	36
Charlotte	95	78	117.5	16				10	3224	5.0 SW	7.3 SW	809	35
Greensboro	95	78	123.5	15				10	3849		7.9 SW	896	37
Raleigh	95	78	117.5	15	98	82	155.6	10	3275	6.0 SW	7.9 SW	345	36
Wilmington	95	78	117.5	15	95	81	150.5	15	2420	7.0 SW	9.4 SW	6	34

*Corresponds to dry-bulb and wet-bulb temperatures listed, and is corrected for altitude of city.

†Corresponds to peak dewpoint temperature, corrected for altitude.

TABLE 1—OUTDOOR DESIGN CONDITIONS—SUMMER AND WINTER (CONT.)

STATE AND CITY	NORMAL DESIGN COND.—SUMMER July at 3:00 PM			AVG. DAILY RANGE	MAXIMUM DESIGN COND.—SUMMER July at 3:00 PM			NORMAL DESIGN COND. WINTER		WIND DATA		Elevation Above Sea Level (ft)	Latitude (deg)
	Dry-Bulb (F)	Wet-Bulb (F)	Moisture Content* (gr/lb of dry air)	Dry-Bulb (F)	Dry-Bulb (F)	Wet-Bulb (F)	Moisture Content† (gr/lb of dry air)	Dry-Bulb (F)	Annual Degree Days	Avg. Velocity and Prevailing Direction Summer	Avg. Velocity and Prevailing Direction Winter		
NORTH DAKOTA													
Bismarck	95	73	95.5	19	103			−30	8937	9.0 NW	9.1 NW	1,670	47
Devils Lake	95	70	77					−30	10104		10.1 W	1,481	48
Fargo	95	75	104.5	19				−25			10.9 NW	900	47
Grand Forks								−25	9871		NW	832	48
Williston	95	73	96.5					−35	9301	8.0 SE	8.6 W	1,919	48
OHIO													
Akron	95	75	99	19				−5				104	41
Cincinnati	95	78	117.5	22	106	81	145.5	0	4990	7.0 SW	8.5 SW	553	39
Cleveland	95	75	99	19	101	79	135.9	0	6144	11.0 S	14.7 SW	651	42
Columbus	95	76	104.5	23	95			−10	5506	9.0 SW	11.6 SW	724	40
Dayton	95	78	123	23	99			0	5412	8.0 SW	11.1 SW	900	40
Lima								−5					41
Sandusky	95	75	99					0	6095		11.0	608	42
Toledo	95	75	99	19	99			−10	6269	10.0 SW	12.1 SW	589	42
Youngstown	95	75	99	19								1,186	41
OKLAHOMA													
Ardmore								10			N	762	34
Bartlesville								−10			N		37
Oklahoma City	101	77	108	21	104			0	3670	10.0 S	11.5 S	1,254	35
Tulsa	101	77	101.5		106	79	140.6	0		10.0 S	N	804	36
OREGON													
Baker	90	66	71	19				−5	7197		5.6 SE	3,501	44
Eugene	90	68	67	19				−15				366	44
Medford	95	70	76	19								1,428	42
Pendleton								−15			W	1,494	46
Portland	90	68	67	19	99	70	103.0	10	4353	6.0 NW	7.3 S	30	46
Roseburg	90	66	57	19						4.0 N		523	42
Wamic								0			W		45
PENNSYLVANIA													
Altoona	95	75	99	14				−5				1,469	40
Bethlehem								−5					41
Erie	93	75	102	18				−5	6363	9.0 S	13.6 SW	670	42
Harrisburg	95	75	99	14				0	5412		7.6 NW	339	40
New Castle								0			NW		41
Oil City	95	75	99	18									42
Philadelphia	95	78	117.5	14	97			0	4739	10.0 SW	11.0 NW	26	40
Pittsburgh	95	75	105	14	98	79	126.9	0	5430	9.0 SW	11.6 W	1,248	40
Reading	95	75	99					0	5232		9.0	311	40
Scranton	95	75	99	14	95			−5	6218	6.0 SW	7.6 SW	746	41
Warren								−15			NW		41
Williamsport								−5			NW	525	42
RHODE ISLAND													
Block Island	95	75	99						5897		20.6 NW	46	41
Pawtucket	93	75	102	14									41
Providence	93	75	102	14				0	5984	10.0 NW	12.1 NW	8	42
SOUTH CAROLINA													
Charleston	95	78	117.5	17	98	82	155.6	15	1866	10.0 SW	10.5 SW	9	33
Columbia	95	75	99	17				10	2488		8.0 SW	401	34
Greenville	95	76	104.5	17				10	3059	7.0 NE	8.4	982	35
SOUTH DAKOTA													
Huron	95	75	106	19	106	76	126.9	−20	7940	10.0 SE	10.7 NW	1,282	44
Rapid City	95	70	85	22	103	71	95.9	−20	7197	7.0 W	8.0 W	3,231	44
Sioux Falls	95	75	99	20				−20			NW	1,427	43

*Corresponds to dry-bulb and wet-bulb temperatures listed, and is corrected for altitude of city.
†Corresponds to peak dewpoint temperature, corrected for altitude.

TABLE 1—OUTDOOR DESIGN CONDITIONS—SUMMER AND WINTER (CONT.)

STATE AND CITY	NORMAL DESIGN COND.—SUMMER July at 3:00 PM			AVG. DAILY RANGE	MAXIMUM DESIGN COND.—SUMMER July at 3:00 PM			NORMAL DESIGN COND. WINTER		WIND DATA Avg. Velocity and Prevailing Direction		Elevation Above Sea Level (ft)	Latitude (deg)
	Dry-Bulb (F)	Wet-Bulb (F)	Moisture Content* (gr/lb of dry air)	Dry-Bulb (F)	Dry-Bulb (F)	Wet-Bulb (F)	Moisture Content† (gr/lb of dry air)	Dry-Bulb (F)	Annual Degree Days	Summer	Winter		
TENNESSEE													
Chattanooga	95	76	104.5	18	98			10	3238	6.0 SW	7.7 NW	689	35
Johnson City								0			W		36
Knoxville	95	75	103.5	17	100	79	135.9	0	3658	6.0 SW	7.2 SW	921	36
Memphis	95	78	117.5	18	103	83	155.6	0	3090	7.0 SW	9.3 W	271	35
Nashville	95	78	117.5	17	98			0	3613	8.0 W	9.8 NW	485	36
TEXAS													
Abilene	100	74	93					15	2573	9.0 S	10.1 S	1,748	32
Amarillo	100	72	91.6	22	101	75	110.4	-10	4196	11.0 S	12.1 SW	3,657	35
Austin	100	78	109.5	19				20	1679		8.3 N	625	31
Brownsville	95	80	131	20	96	80	150.5	30	628	9.0 SE	10.4 SE	35	26
Corpus Christi	95	80	131					20	965	13.0 SE	11.0 SE	21	28
Dallas	100	78	109.5	21	105	80	135.9	0	2367	8.0 S	10.6 NW	460	33
Del Rio	100	78	115					15	1501	10.0 SE	8.0 SE	1,020	29
El Paso	100	69	73	23	101	72	106.6	10	2532	9.0 E	9.0 NW	3,720	32
Fort Worth	100	78	109.5	21				10	2355	10.0	10.5 NW	708	33
Galveston	95	80	131	14				20	1174	9.0 S	11.2 SE	6	29
Houston	95	80	131	14	100	81	150.5	20	1315	8.0 S	10.5 SE	52	30
Palestine	100	78	109.5					15	2068		8.0	555	32
Port Arthur	95	79	124					20	1532		10.7	64	30
San Antonio	100	78	109.5	19	102	83	166.4	20	1435	7.0 SE	8.3 NE	646	29
UTAH													
Modena	95	65	66	25	97	66	80.3	-15	6598	11.0 SW	9.0	5,479	38
Logan								-15					42
Ogden								-10			S	4,446	41
Salt Lake City	95	65	61	25	102	68	89.4	-10	5650	7.0 S	7.8 SE	4,222	41
VERMONT													
Bennington								-10					43
Burlington	90	73	95	17	91			-10	8051	8.0 S	11.6 S	308	44
Rutland	90	73	95	17				-20					43
VIRGINIA													
Cape Henry	95	78	117.5					10	3538		14.0	24	37
Lynchburg	95	75	99	16	99			5	4068		8.1	386	37
Norfolk	95	78	117.5	16	95			15	3364	11.0 S	12.1 N	11	37
Richmond	95	78	117.5	16	98			15	3922	6.0 SW	8.1 SW	162	38
Roanoke	95	76	111.5	16				0	4075		8.2 W	1,194	38
WASHINGTON													
North Head	85	65	60					20	5367		16.1	199	
Seattle	85	65	60	17	86	70	99.3	15	4815	7.0 N	9.8 SE	14	48
Spokane	93	65	54.5	28	106	68	71.9	-15	6318	7.0 SW	6.2 SW	1,879	48
Tacoma	85	64	55.5	17				15	5039		8.0	279	47
Tatoosh Island								15	5857		18.9	110	48
Walla Walla	95	65	47.5	28	105			-10	4910		5.4 S	952	46
Wenatchee	90	65	52	20									48
Yakima	95	65	48	20				5	5585		4.1	1,160	47
WEST VIRGINIA													
Bluefield	95	75	99	16									37
Charleston	95	75	99	16	102			0		4.0 SW	W	603	38
Elkins								-10	5800		6.2 W	2,006	39
Huntington	95	76	104.5	16				-5			W		38
Martinsburg								-5				540	39
Parkersburg	95	75	99	16	98			-10	4928	4.0 SE	7.2 SW	615	39
Wheeling	95	75	99	14				-5					40

*Corresponds to dry-bulb and wet-bulb temperatures listed, and is corrected for altitude of city.
†Corresponds to peak dewpoint temperature, corrected for altitude.

TABLE 1—OUTDOOR DESIGN CONDITIONS—SUMMER AND WINTER (CONT.)

STATE AND CITY	NORMAL DESIGN COND.—SUMMER July at 3:00 PM			AVG. DAILY RANGE	MAXIMUM DESIGN COND.—SUMMER July at 3:00 PM			NORMAL DESIGN COND. WINTER		WIND DATA Avg. Velocity and Prevailing Direction		Elevation Above Sea Level (ft)	Latitude (deg)
	Dry-Bulb (F)	Wet-Bulb (F)	Moisture Content* (gr/lb of dry air)	Dry-Bulb (F)	Dry-Bulb (F)	Wet-Bulb (F)	Moisture Content† (gr/lb of dry air)	Dry-Bulb (F)	Annual Degree Days	Summer	Winter		
WISCONSIN													
Ashland								−20			SW		42
Eau Claire								−20			NW	885	45
Green Bay	95	75	99	14	99	79	131.1	−20	7931	8.0 S	10.5 SW	589	45
La Crosse	95	75	99	17	100	83	161.2	−25	7421	6.0 S	9.3 S	673	44
Madison	95	75	103.5	18	96			−15	7405	8.0 SW	10.1 NW	938	43
Milwaukee	95	75	99	14	99			−15	7079	9.0 SW	12.1 W	619	43
WYOMING													
Casper								−20			SW	5,321	43
Cheyenne	95	65	68.5	28				−15	7536	9.0 S	13.3 NW	6,139	42
Lander	95	65	66	28				−18	8243	5.0 SW	3.9	5,448	44
Sheridan					102			−30	7239	5.0 NW	4.9 NW	3,773	45
CANADA													
PROVINCE AND CITY													
ALBERTA													
Calgary	90	66	71					−29	9520	9.7	10.1	3,540	51
Edmonton	90	68	77					−33	10320	8.9	7.6	2,219	54
Grand Prairie								−39			7.9	2,190	55
Lethbridge								−32	8650		15.0	3,018	50
McMurray								−42				1,216	57
Medicine Hat	90	65						−35	8650	9.1	9.0	2,365	50
BRITISH COLUMBIA													
Estevan Point								17			9.9	20	49
Fort Nelson								−38			3.7	1,230	59
Penticton								−6				1,121	50
Prince George								−32	9500		7.2	2,218	54
Prince Rupert								8	6910		8.0	170	54
Vancouver	80	67	78					11	5230		7.7	22	49
Victoria								15	5410		12.3	228	48
MANITOBA													
Brandon								−32	10930			1,200	50
Churchill								−42	16810		14.7	115	59
The Pas								−39			6.4	894	54
Winnipeg	90	71	83.5					−29	10630	11.5	12.0	786	50
NEW BRUNSWICK													
Campbellton								−11				42	48
Fredericton	90	75	107					−6	8830		9.2	164	46
Moncton								−8	8700		14.9	248	46
Saint John								−3	8380	7.9	13.8	119	45
NEWFOUNDLAND													
Corner Brook								−1	9210			40	49
Gander								−3	9440		17.2	482	49
Goose Bay								−26	12140		10.3	144	53
Saint Johns								1	8780		19.3	463	48
NORTHWEST TERRITORIES													
Aklavik								−46	17870			30	68
Fort Norman								−42	16020			300	65
Frobisher								−47				68	
Resolute								−42			9.2	56	
Yellowknife								−47				682	62

*Corresponds to dry-bulb and wet-bulb temperatures listed, and is corrected for altitude of city.

†Corresponds to peak dewpoint temperature, corrected for altitude.

TABLE 1—OUTDOOR DESIGN CONDITIONS—SUMMER AND WINTER (CONT.)

CANADA PROVINCE AND CITY	NORMAL DESIGN COND.—SUMMER July at 3:00 PM Dry-Bulb (F)	Wet-Bulb (F)	Moisture Content* (gr/lb of dry air)	AVG. DAILY RANGE Dry-Bulb (F)	MAXIMUM DESIGN COND.—SUMMER July at 3:00 PM Dry-Bulb (F)	Wet-Bulb (F)	Moisture Content† (gr/lb of dry air)	NORMAL DESIGN COND. WINTER Dry-Bulb (F)	Annual Degree Days	WIND DATA Avg. Velocity and Prevailing Direction Summer	Winter	Elevation Above Sea Level (ft)	Latitude (deg)
NOVA SCOTIA													
Halifax	90	75	107					4	7570	6.6	9.6	83	45
Sydney								1	8220	9.9	13.1	197	46
Yarmouth								7	7520		13.5	136	44
ONTARIO													
Fort William								−24	10350	8.4	9.6	644	48
Hamilton								0	6890			303	43
Kapuskasing								−30	11790		10.0	752	49
Kingston								−11	7810		10.8	340	44
Kitchener								−3	7380			1,100	43
London								−1			11.9	912	43
North Bay								−20		9.6	11.3	1,210	46
Ottawa	90	75	107					−15	8830	8.9	11.1	339	45
Peterborough								−11				648	44
Souix Lookout								−33			8.5	1,227	50
Sudbury								−17				837	47
Timmins								−26				1,100	48
Toronto	93	75	102					0	7020	8.1	14.1	379	43
Windsor								3			12.3	637	42
Sault Ste. Marie	93	75	102									635	47
PRINCE EDWARD ISLAND													
Charlottetown								−3	8380	8.7	11.3	74	46
QUEBEC													
Arvida								−19	10440		8.2	375	
Knob Lake								−40				1,605	55
Mont Joli								−11			13.3	150	48
Montreal	90	75	107					−9	8130	9.9	12.3	187	46
Port Harrison								−39			13.4	66	58
Quebec City	90	75	107					−12	9070	9.0	12.4	296	47
Seven Islands								−20				190	50
Sherbrooke								−12	8610		8.2	620	45
Three Rivers								−13				50	46
SASKATCHEWAN													
Prince Albert								−41	11430		4.9	1,414	53
Regina	90	71	92.5					−34	10770	12.4	12.1	1,884	50
Saskatoon	90	70	81					−37	10960	10.7	9.7	1,645	52
Swift Current								−33	9660		14.6	2,677	50
YUKON TERRITORY													
Dawson								−56	15040			1,062	64
Whitehorse								−43			8.7	2,289	61

*Corresponds to dry-bulb and wet-bulb temperatures listed, and is corrected for altitude of city.

†Corresponds to peak dewpoint temperature, corrected for altitude.

CORRECTIONS TO OUTDOOR DESIGN CONDITIONS FOR TIME OF DAY AND TIME OFYEAR

The normal design conditions for summer, listed in *Table 1,* are applicable to the month of July at about 3:00 P.M. Frequently, the design conditions at other times of the day and other months of the year must be known.

Table 2 lists the approximate corrections on the dry-bulb and wet-bulb temperatures from 8 a.m. to 12 p.m. based on the average daily range. The dry-bulb corrections are based on analysis of weather data, and the wet-bulb corrections assume a relatively constant dewpoint throughout the 24-hr period.

Table 3 lists the approximate corrections of the dry-bulb and wet-bulb temperatures from March to November, based on the yearly range in dry-bulb temperature (summer normal design dry-bulb minus winter normal design dry-bulb temperature). These corrections are based on analysis of weather data and are applicable only to the cooling load estimate.

Example 1 – Corrections to Design Conditions

Given:

A comfort application in New York City.

Find:

The approximate dry-bulb and wet-bulb temperatures at 12:00 noon in October.

Solution:

Normal design conditions for New York in July at 3:00 p.m. are 95 F db, 75 F wb *(Table 1).*

Daily range in New York City is 14 F db.

Yearly range in New York City = 95-0 = 95 F db.

Correction for time of day (12 noon) from *Table 2:*

Dry-bulb = -5 F

Wet-bulb = -1 F

Correction for time of year (October) from *Table 3:*

Dry-bulb = -16 F

Wet-bulb = -8 F

Design conditions at 12 noon in October (approximate) :

Dry-bulb = 95-5-16 = 74 F

Wet-bulb = 75-1- 8 = 66 F

INSIDE COMFORT DESIGN CONDITIONS-SUMMER

The inside design conditions listed in *Table 4* are recommended for types of applications listed. These conditions are based on experience gathered from many applications, substantiated by ASHAE tests.

The optimum or deluxe conditions are chosen where costs are not of prime importance and for comfort applications in localities having summer outdoor design dry-bulb temperatures of 90 F or less. Since all of the loads (sun, lights, people, outdoor air, etc.) do not peak simultaneously for any prolonged periods, it may be uneconomical to design for the optimum conditions.

TABLE 2—CORRECTIONS IN OUTDOOR DESIGN TEMPERATURES FOR TIME OF DAY

(For Cooling Load Estimates)

DAILY RANGE OF TEMPERATURE* (F)	DRY- OR WET-BULB	SUN TIME									
		AM			PM						
		8	10	12	2	3	4	6	8	10	12
10	Dry-Bulb	−9	−7	−5	−1	0	−1	−2	−5	−8	−9
	Wet-Bulb	−2	−2	−1	0	0	0	−1	−1	−2	−2
15	Dry-Bulb	−12	−9	−5	−1	0	−1	−2	−6	−10	−14
	Wet-Bulb	−3	−2	−1	0	0	0	−1	−1	−3	−4
20	Dry-Bulb	−14	−10	−5	−1	0	−1	−3	−7	−11	−16
	Wet-Bulb	−4	−3	−1	0	0	0	−1	−2	−3	−4
25	Dry-Bulb	−16	−10	−5	−1	0	−1	−3	−8	−13	−18
	Wet-Bulb	−4	−3	−1	0	0	0	−1	−2	−3	−5
30	Dry-Bulb	−18	−12	−6	−1	0	−1	−4	−10	−15	−21
	Wet-Bulb	−5	−3	−1	0	0	0	−1	−3	−4	−6
35	Dry-Bulb	−21	−14	−7	−1	0	−1	−6	−12	−18	−24
	Wet-Bulb	−6	−4	−2	0	0	0	−1	−3	−5	−7
40	Dry-Bulb	−24	−16	−8	−1	0	−1	−7	−14	−21	−28
	Wet-Bulb	−7	−4	−2	0	0	0	−2	−4	−6	−9
45	Dry-Bulb	−26	−17	−8	−2	0	−2	−8	−16	−24	−31
	Wet-Bulb	−7	−5	−2	0	0	−1	−2	−4	−8	−10

*The daily range of dry-bulb temperature is the difference between the highest and lowest dry-bulb temperature during a 24-hour period on a typical design day. (See *Table 1* for the value of daily range for a particular city).

Equation: Outdoor design temperature at any time = Outdoor design temperature from *Table 1* + Correction from above table.

TABLE 3—CORRECTIONS IN OUTDOOR DESIGN CONDITIONS FOR TIME OF YEAR

(For Cooling Load Estimates)

YEARLY RANGE OF TEMPERA-TURE(F)*	DRY- OR WET-BULB	TIME OF YEAR								
		March	April	May	June	July	August	Sept.	Oct.	Nov.
120	Dry-Bulb	−39	−22	−11	−4	0	0	−9	−24	−44
	Wet-Bulb	−23	−12	− 5	−2	0	0	−4	−13	−27
115	Dry-Bulb	−33	−22	−11	−4	0	0	−8	−20	−36
	Wet-Bulb	−18	−11	− 5	−2	0	0	−4	−10	−21
110	Dry-Bulb	−30	−20	−11	−4	0	0	−6	−17	−31
	Wet-Bulb	−15	−10	− 5	−2	0	0	−3	− 8	−16
105	Dry-Bulb	−30	−20	−11	−4	0	0	−6	−17	−29
	Wet-Bulb	−15	−10	− 5	−2	0	0	−3	− 8	−14
100	Dry-Bulb	−29	−19	−10	−3	0	0	−6	−16	−27
	Wet-Bulb	−14	−10	− 5	−2	0	0	−3	− 8	−14
95	Dry-Bulb	−29	−19	−10	−3	0	0	−6	−16	−27
	Wet-Bulb	−14	−10	− 5	−2	0	0	−3	− 8	−14
90	Dry-Bulb	−29	−19	−10	−3	0	0	−6	−16	−26
	Wet-Bulb	−14	−10	− 5	−2	0	0	−3	− 8	−14
85	Dry-Bulb	−29	−19	− 9	−3	0	0	−5	−16	−25
	Wet-Bulb	−14	−10	− 5	−2	0	0	−3	− 8	−14
80	Dry-Bulb	−24	−16	− 8	−3	0	0	−4	−12	−20
	Wet-Bulb	−13	− 9	− 4	−2	0	0	−2	− 6	−11
75	Dry-Bulb	−14	− 9	− 4	−1	0	0	−3	− 7	−15
	Wet-Bulb	− 7	− 5	− 2	0	0	0	−2	− 4	− 8
70	Dry-Bulb	−13	− 9	− 4	−1	0	0	−2	− 7	−14
	Wet-Bulb	− 6	− 4	− 2	0	0	0	−1	− 4	− 6
65	Dry-Bulb	−11	− 8	− 4	−1	0	0	−2	− 6	−12
	Wet-Bulb	− 6	− 4	− 2	0	0	0	−1	− 3	− 6
60	Dry-Bulb	− 9	− 7	− 3	−1	0	0	−2	− 5	−10
	Wet-Bulb	− 4	− 3	− 2	0	0	0	−1	− 3	− 5
55	Dry-Bulb	− 6	− 5	− 3	−1	0	0	−2	− 4	− 8
	Wet-Bulb	− 3	− 3	− 2	0	0	0	−1	− 2	− 4
50	Dry-Bulb	− 5	− 4	− 3	−1	0	0	−2	− 4	− 7
	Wet-Bulb	− 3	− 2	− 1	0	0	0	−1	− 2	− 3

*Yearly range of temperature is the difference between the summer and winter normal design dry-bulb temperatures (*Table 1*).

Equation: Outdoor design temperature = Outdoor design temperature from *Table 1* + Corrections from above table.

The commercial inside design conditions are recommended for general comfort air conditioning applications. Since a majority of people are comfortable at 75 F or 76 F db and around 45% to 50% rh, the thermostat is set to these temperatures, and these conditions are maintained under partial loads. As the peak loading occurs (outdoor peak dry-bulb and wet-bulb temperatures, 100% sun, all people and lights, etc.), the temperature in the space rises to the design point, usually 78 F db.

If the temperature in the conditioned space is forced to rise, heat will be stored in the building mass. Refer to *Chapter 3, "Heat Storage, Diversity and Stratification,"* for a more complete discussion of heat storage. With summer cooling, the temperature swing used in the calculation of storage is the difference between the design temperature and the normal thermostat setting.

The range of summer inside design conditions is provided to allow for the most economical selection of equipment. Applications of inherently high sen-sible heat factor (relatively small latent load) usually result in the most economical equipment selection if the higher dry-bulb temperatures and lower relative humidities are used. Applications with low sensible heat factors (high latent load) usually result in more economical equipment selection if the lower dry-bulb temperatures and higher relative humidities are used.

INSIDE COMFORT DESIGN CONDITIONS-WINTER

For winter season operation, the inside design conditions listed in *Table 4* are recommended for general heating applications. With heating, the temperature swing (variation) is below the comfort condition at the time of peak heating load (no people, lights, or solar gain, and with the minimum outdoor temperature). Heat stored in the building structure during partial load (day) operation reduces the required equipment capacity for peak load operation in the same manner as it does with cooling.

TABLE 4—RECOMMENDED INSIDE DESIGN CONDITIONS*—SUMMER AND WINTER

TYPE OF APPLICATION	SUMMER					WINTER				
	Deluxe		Commercial Practice			With Humidification			Without Humidification	
	Dry-Bulb (F)	Rel. Hum. (%)	Dry-Bulb (F)	Rel. Hum. (%)	Temp. Swing† (F)	Dry-Bulb (F)	Rel. Hum. (%)	Temp. Swing‡ (F)	Dry-Bulb (F)	Temp. Swing‡ (F)
GENERAL COMFORT Apt., House, Hotel, Office Hospital, School, etc.	74-76	50-45	77-79	50-45	2 to 4	74-76	35-30	−3 to −4	75-77	−4
RETAIL SHOPS (Short term occupancy) Bank, Barber or Beauty Shop, Dept. Store, Supermarket, etc.	76-78	50-45	78-80	50-45	2 to 4	72-74	35-30**	−3 to −4	73-75	−4
LOW SENSIBLE HEAT FACTOR APPLICATIONS (High Latent Load) Auditorium, Church, Bar, Restaurant, Kitchen, etc.	76-78	55-50	78-80	60-50	1 to 2	72-74	40-35	−2 to −3	74-76	−4
FACTORY COMFORT Assembly Areas, Machining Rooms, etc.	77-80	55-45	80-85	60-50	3 to 6	68-72	35-30	−4 to −6	70-74	−6

*The room design dry-bulb temperature should be reduced when hot radiant panels are adjacent to the occupant and increased when cold panels are adjacent, to compensate for the increase or decrease in radiant heat exchange from the body. A hot or cold panel may be unshaded glass or glass block windows (hot in summer, cold in winter) and thin partitions with hot or cold spaces adjacent. An unheated slab floor on the ground or walls below the ground level are cold panels during the winter and frequently during the summer also. Hot tanks, furnaces or machines are hot panels.

†Temperature swing is above the thermostat setting at peak summer load conditions.

‡Temperature swing is below the thermostat setting at peak winter load conditions (no lights, people or solar heat gain).

**Winter humidification in retail clothing shops is recommended to maintain the quality texture of goods.

INSIDE INDUSTRIAL DESIGN CONDITIONS

Table 5 lists typical temperatures and relative humidities used in preparing, processing, and manufacturing various products, and for storing both raw and finished goods. These conditions are only typical of what has been used, and my vary with applications. They may also vary as changes occur in processes, products, and knowledge of the effect of temperature and humidity. In all cases, the temperature and humidity conditions and the permissible limits of variations on these conditions should be established by common agreement with the customer.

Some of the conditions listed have no effect on the product or process other than to increase the efficiency of the employee by maintaining comfort conditions. This normally improves workmanship and uniformity, thus reducing rejects and production cost. In some cases, it may be advisable to compromise between the required conditions and comfort conditions to maintain high quality commensurate with low production cost.

Generally, specific inside design conditions are required in industrial applications for one or more of the following reasons:

1. A constant temperature level is required for close tolerance measuring, gaging, machining, or grinding operations, to prevent expansion and contraction of the machine parts, machined products and measuring devices. Normally, a constant temperature is more important than the temperature level. A constant relative humidity is secondary in nature but should not go over 45% to minimize formation of heavier surface moisture film. Non-hygroscopic materials such as metals, glass, plastics, etc., have a property of capturing water molecules within the microscopic surface crevices, forming an invisible, non-continuous surface film. The density of this film increases when relative

humidity increases. Hence, this film must, in many instances, be held below a critical point at which metals may etch, or the electric resistance of insulating materials is significantly decreased.

2. Where highly polished surfaces are manufactured or stored, a constant relative humidity and temperature is maintained, to minimize increase is maintained, to minimize increase in surface moisture film. The temperature and humidity should be at, or a little below, the comfort conditions to minimize perspiration of the operator. Constant temperature and humidity may also be required in machine rooms to prevent etching or corrosion of the parts of the machines. With applications of this type, if the conditions are not maintained 24 hours a day, the starting of air conditioning after any prolonged shutdown should be done carefully: (1) During the summer, the moisture accumulation in the space should be reduced before the temperature is reduced; (2) During the winter, the moisture should not be introduced before the materials have a chance to warm up if they are cooled during shutdown periods.
3. Control of relative humidity is required to maintain the strength, pliability, and regain of hydroscopic materials, such as textiles and paper. The humidity must also be controlled in some applications to reduce the effect of static electricity. Development of static electric charges is minimized of 55% or higher.
4. The temperature and relative humidity control are required to regulate the rate of chemical or biochemical reactions, such as drying of Varnishes or sugar coatings, preparation of synthetic fibers or chemical compounds, fermentation of yeast, etc. Generally, high temperatures with low humidities increase drying rates; high temperatures increase the rate of chemical reaction, and high temperatures and relative humidities increase such processes as yeast fermentations.
5. Laboratories require precise control of both temperature and relative humidity or either. Both testing and quality control laboratories are frequently designed to maintain the ASTM Standard Conditions* of 73.4 F db and 50% rh.
6. With some industrial applications where the load is excessive and the machines or materials do not benefit from controlled conditions, it may be advisable to apply spot cooling for the relief of the workers. Generally, the conditions to be maintained by this means will be above normal comfort.

*Published in ASTN pamphlet dated 9-29-48. These conditions have also been approved by the Technical Committee on Standard Temperature and Relative Humidity Conditions of the FSB (Federal Specifications Board) with one variation: FSB permits ±4%, whereas ASTM requires ±2% permissable humidity tolerance.

TABLE 5—TYPICAL INSIDE DESIGN CONDITIONS—INDUSTRIAL

(Listed conditions are only typical; final design conditions are established by customer requirements)

INDUSTRY	PROCESS	DRY-BULB (F)	RH (%)
ABRASIVE	Manufacture	75-80	45-50
BAKERY	Dough Mixer	75-80	40-50
	Fermenting	75-82	70-75
	Proof Box	92-96	80-85
	Bread Cooler	70-80	80-85
	Cold Room	40-45	—
	Make-up Rm.	78-82	65-70
	Cake Mixing	95-105	—
	Crackers & Biscuits	60-65	50
	Wrapping	60-65	60-65
	Storage—		
	Dried Ingred.	70	55-65
	Fresh Ingred.	30-45	80-85
	Flour	70-75	50-65
	Shortening	45-70	55-60
	Sugar	80	35
	Water	32-35	—
	Wax Paper	70-80	40-50
BREWERY	Storage—		
	Hops	30-32	55-60
	Grain	80	60
	Liquid Yeast	32-34	75
	Lager	32-35	75
	Ale	40-45	75
	Fermenting Cellar—		
	Lager	40-45	75
	Ale	55	75
	Racking Cellar	32-35	75
CANDY—CHOCOLATE	Candy Centers	80-85	40-50
	Hand Dipping Rm.	60-65	50-55
	Enrobing Rm.	75-80	55-60
	Enrobing—		
	Loading End	80	50
	Enrober	90	13
	Stringing	70	40-50
	Tunnel	40-45	DP — 40
	Packing	65	55
	Pan Specialty Rm.	70-75	45
	General Storage	65-70	40-50
CANDY—HARD	Mfg.	75-80	30-40
	Mixing & Cooling	75-80	40-45
	Tunnel	55	DP — 55
	Packing	65-75	40-45
	Storage	65-75	45-50
	Drying—Jellies, Gums	120-150	15
	Cold Rm.—Marshmallow	75-80	45-50
CHEWING GUM	Mfg.	77	33
	Rolling	68	63
	Stripping	72	53
	Breaking	74	47
	Wrapping	74	58

INDUSTRY	PROCESS	DRY-BULB (F)	RH (%)
CERAMICS	Refractory	110-150	50-90
	Molding Rm.	80	60-70
	Clay Storage	60-80	35-65
	Decal & Decorating	75-80	45-50
CEREAL	Packaging	75-80	45-50
COSMETICS	Mfg.	65-70	—
DISTILLING	Storage—		
	Grain	60	35-40
	Liquid Yeast	32-34	
	Mfg.	60-75	45-60
	Aging	65-72	50-60
ELECTRICAL PRODUCTS	Electronic & X-ray Coils & Trans. Winding	72	15
	Tube Assem.	68	40
	Electrical Inst. Mfg. & Lab.	70	50-55
	Thermostat Assem. & Calib.	76	50-55
	Humidistat Assem. & Calib.	76	50-55
	Close Tol. Assem.	72	40-45
	Meter Assem. Test	74-76	60-63
	Switchgear—		
	Fuse & Cut-Out Assem.	73	50
	Cap. Winding	73	50
	Paper Storage	73	50
	Conductor Wrapping	75	65-70
	Lightning Arrestor	68	20-40
	Circuit Brkr. Assem. & Test	76	30-60
	Rectifiers—		
	Process Selenium & Copper Oxide Plates	74	30-40
FURS	Drying	110	—
	Shock Treatment	18-20	—
	Storage	40-50	55-65
GLASS	Cutting	Comfort	
	Vinyl Lam. Rm.	55	15
LEATHER	Drying—		
	Veg. Tanned	70	75
	Chrome Tanned	120	75
	Storage	50-60	40-60
LENSES—OPTICAL	Fusing	Comfort	
	Grinding	80	50
MATCHES	Mfg.	72-74	50
	Drying	70-75	40
	Storage	60-62	50
MUNITIONS	Metal Percussion Elements—		
	Drying Parts	190	—
	Drying Paints	110	—
	Black Powder Drying	125	—
	Condition & Load Powder Type Fuse	70	40
	Load Tracer Pellets	80	40

TABLE 5—TYPICAL INSIDE CONDITIONS—INDUSTRIAL (Contd)

(Listed conditions are only typical; final design conditions are established by customer requirements)

INDUSTRY	PROCESS	DRY-BULB (F)	RH (%)
PHARMACEUTICAL	Powder Storage		
	Before Mfg.	70-80	30-35
	After Mfg.	75-80	15-35
	Milling Rm.	80	35
	Tablet Compressing	70-80	40
	Tablet Coating	80	35
	Effervescent—		
	Tablet & Powder	90	15
	Hypodermic Tablet	75-80	30
	Colloids	70	30-50
	Cough Syrup	80	40
	Glandular Prod.	78-80	5-10
	Ampule Mfg.	80	35
	Gelatin Capsule	78	40-50
	Capsule Storage	75	35-40
	Microanalysis	Comfort	
	Biological Mfg.	80	35
	Liver Extract	70-80	20-30
	Serums	Comfort	
	Animal Rm.	Comfort	
PHOTO MATERIAL	Drying	20-125	40-80
	Cutting & Packing	65-75	40-70
	Storage—		
	Film Base, Film Paper, Coated Paper	70-75	40-65
	Safety Film	60-80	45-50
	Nitrate Film	40-50	40-50
PLASTIC	Mfg.—		
	Thermo Setting Compounds	80	25-30
	Cellophane	75-80	45-65
PLYWOOD	Hot Press—Resin	90	60
	Cold Press	90	15-25
PRECISION MACHINING	Spectrographic Anal.	Comfort	
	Gear Matching & Assem.	75-80	35-40
	Storage—		
	Gasket	100	50
	Cement & Glue	65	40
	Machinings		
	Gaging, Assem. Adjusting Precision Parts	Comfort	
	Honing	75-80	35-45
PRINTING	Multicolor Litho.		
	Pressroom	75-80	46-48
	Stockroom	73-80	49-51
	Sheet & Web Print.	Comfort	
	Storage, Folding, etc.	Comfort	
REFRIGERATION EQUIPMENT	Valve Mfg.	75	40
	Compressor Assem.	70-76	30-45
	Refrigerator Assem.	Comfort	
	Testing	65-82	47
RUBBER DIPPED GOODS	Mfg.	90	—
	Cementing	80	25-30
	Surgical Articles	75-90	25-30
	Storage Before Mfg.	60-75	40-50
	Lab. (ASTM Std.)	73.4	50
TEXTILES	Cotton		
	Opening & Picking	70-75	55-70
	Carding	83-87	50-55
	Drawing & Roving	80	55-60
TEXTILES (cont.)	Cotton, cont.		
	Ring Spinning		
	Conventional	80-85	60-70
	Long Draft	80-85	
	Frame Spinning	80-85	55-60
	Spooling, Warping	78-80	60-65
	Weaving	78-80	70-85
	Cloth Room	75	65-70
	Combing	75	55-65
	Linen		
	Carding, Spinning	75-80	60
	Weaving	80	80
	Woolens		
	Pickers	80-85	60
	Carding	80-85	65-70
	Spinning	80-85	50-60
	Dressing	75-80	60
	Weaving—		
	Light Goods	80-85	55-70
	Heavy	80-85	60-65
	Drawing	75	50-60
	Worsteds		
	Carding, Combing, & Gilling	80-85	60-70
	Storage	70-85	75-80
	Drawing	80-85	50-70
	Cap Spinning	80-85	50-55
	Spooling, Winding	75-80	55-60
	Weaving	80	50-60
	Finishing	75-80	60
	Silk		
	Prep. & Dressing	80	60-65
	Weaving & Spinning	80	65-70
	Throwing	80	60
	Rayon		
	Spinning	80-90	50-60
	Throwing	80	55-60
	Weaving		
	Regenerated	80	50-60
	Acetate	80	55-60
	Spun Rayon	80	80
	Picking	75-80	50-60
	Carding, Roving, Drawing	80-90	50-60
	Knitting		
	Viscose or Cuprammonium	80-85	65
	Synthetic Fiber Prep. & Weaving		
	Viscose	80	60
	Celanese	80	70
	Nylon	80	50-60
TOBACCO	Cigar & Cigarette Mfg.	70-75	55-65
	Softening	90	85-88
	Stemming & Stripping	75-85	75
	Storage & Prep.	78	70
	Conditioning	75	75
	Packing & Shipping	75	60

CHAPTER 3. HEAT STORAGE, DIVERSITY AND STRATIFICATION

The normal load estimating procedure has been to evaluate the instantaneous heat gain to a space and to assume that the equipment will remove the heat at this rate. Generally, it was found that the equipment selected on this basis was oversized and therefore capable of maintaining much lower room conditions than the original design. Extensive analysis, research and testing have shown that the reasons for this are:

1. Storage of heat in the building structure.
2. Non-simultaneous occurrence of the peak of the individual loads (diversity).
3. Stratification of heat, in some cases.

This chapter contains the data and procedures for determining the load the equipment is actually picking into account the above factors. Application of these data to the appropriate individual heat gains results in the actual cooling load.

The actual cooling load is generally considerable below the peak total instantaneous heat gain, thus requiring smaller equipment to perform a specific job. In addition, the air quantities and/or water quantities are reduced, resulting in a smaller overall system. Also, as brought out in the tables, if the equipment is operated somewhat longer during the peak load periods, and/of the temperature in the space is allowed to rise a few degrees at the peak periods during cooling operation, a further reduction in required capacity results. The smaller system operating for longer periods at times of peak load will produce a lower first cost to the customer with commensurate lower demand charges and lower operating costs. It is a well-known fact that equipment sized to more nearly meet the requirements results in a more efficient, better operating system. Also, if a smaller system is selected, and is based on extended periods of operation at the peak load, it results in a more economical and efficient system at a partially loaded condition.

Since, in most cases, the equipment installed to perform a specific function is smaller, there is less margin for error. This requires more exacting engineering including air distribution design and system balancing.

With multi-story, multi-room application, it is usually desirable to provide some flexibility in the air side or room load to allow for individual room control, load pickup, etc. Generally, it is recommended that the full reduction from storage and diversity be taken on the overall refrigeration or building load, with some degree of conservatism on the air side or room loads. This degree should be determined by the engineer from project requirements and customer desires. A system so designed, full reduction on refrigeration load and less than full reduction on air side or room load, meets all of the flexibility requirements, except at time of peak load. In addition, such a system has a low owning and operating cost.

STORAGE OF HEAT IN BUILDING STRUCTURES

The instantaneous heat gain in a typical comfort application consists of sun, lights, people, transmission thru walls, roof and glass, infiltration and ventilation air and, in some cases, machinery, appliances, electric calculating machines, etc. A large portion of this instantaneous heat gain is radiant heat which does not become an instantaneous load on the equipment, because it must strike a solid surface and be absorbed by this surface before becoming a load on the equipment. The breakdown on the various instantaneous heat gains into radiant heat and convected heat is approximately as follows:

HEAT GAIN SOURCE	RADIANT HEAT	CONVECTIVE HEAT
Solar, without inside blinds	100%	-
Solar, with inside blinds	58%	42%
Fluorescent Lights	50%	50%
Incandescent Lights	80%	20%
People*	40%	20%
Transmission†	60%	40%
Infiltration and Ventilation	-	100%
Machinery or Appliances‡	20-80%	80-20%

*The remaining 40% is dissipated as latent load.

†Transmission load is considered to be 100% convective load. This load is normally a relatively small part of the total load, and for simplicity is considered to be the instantaneous load on the equipment.

‡The load from machinery or appliances varies, depending upon the temperature of the surface. The higher the surface temperature, the greater the radiant heat load.

CONSTANT SPACE TEMPERATURE AND EQUIPMENT OPERATING PERIODS

As the radiant heat from sources shown in the above table strikes a solid surface (walls, floor, ceiling, etc.), it is absorbed, raising the temperature at the surface of the material above that inside the material and the air adjacent to the surface. This temperature difference causes heat flow into the material by conduction and into the air by convection. The heat conducted away from the surface is stored, and theheat convected from the surface becomes an instantaneous cooling load. The portion of radiant heat being stored depends on the ratio of the resistance to heat flow into the material and the resistance to heat flow into the air film. With most construction materials, the resistance to heat flow into the material is much lower than the air resistance; therefore, most of the radiant heat will be stored. However, as this process of absorbing radiant heat continues, the material becomes warmer and less capable of storing more heat.

The highly varying and relatively sharp peak of the instantaneous solar heat gain results in a large part of it being stored at the time of peak solar heat gain, as illustrated in *Fig. 3.*

The upper curve in *Fig. 3* is typical of the *solar heat gain* for a west exposure, and the lower curve is the actual cooling load that results in an average construction application with the space temperature held constant. The reduction in the peak heat gain is approximately 40% and the peak load lags the peak heat gain by approximately 1 hour. The cross-hatched areas *(Fig. 3)* represent the Heat Stored and the Stored Heat Removed from the construction. Since all of the heat coming into a space must be removed, these two areas are equal.

The relatively constant light load results in a large portion being stored just after the lights are turned on, with a decreasing amount being stored the longer the lights are on, as illustrated in *Fig. 4.*

The upper and lower curves represent the instantaneous heat gain and actual cooling load from *fluorescent lights* with a constant space temperature. The cross-hatched areas are the Heat Stored and the Stored Heat Removed from the construction. The dotted line indicates the actual cooling load for the first day if the lights are on longer than the period shown.

Figs. 3 and 4 illustrate the relationship between the instantaneous heat gain and the actual cooling load in average construction spaces. With light construction, less heat is stored at the peak (less storage capacity available), and with heavy construction, more heat is stored at the peak (more storage capacity available), as shown in *Fig. 5.* This aspect affects the extent of zoning required in the design of a system for a given building; the lighter the building construction, the more attention should be given to zoning.

The upper curve of *Fig. 5* is the instantaneous solar heat gain while the three lower curves are the actual cooling load for *light, medium and heavy construction* respectively, with a constant temperature in the space.

One more item that significantly affects the storage of heat is the operating period of the air conditioning equipment.All of the curves shown in*Figs.3, 4 and 5* illustrate the actual cooling load for 24-hour operation.If the equipment is shut down after 16 hours of operation, some of the stored heat remains in the building construction. This heat must be removed (heat in must equal heat out) and will appear as a pulldown load when the equipment is turned on the next day, as illustrated in *Fig. 6.*

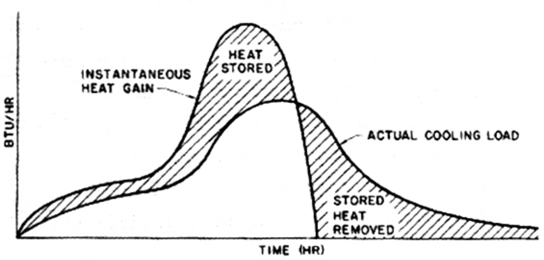


FIG. 3-ACTUAL COOLING LOAD, SOLAR HEAT GAIN, WEST EXPOSURE, AVERAGE CONSTRUCTION

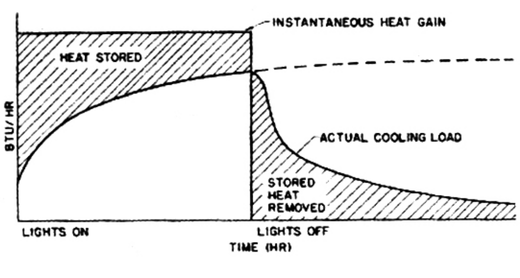


FIG. 4- ACTUAL COOLING LOAD FROM FLUORESCENT LIGHTS, AVERAGE CONSTRUCTION

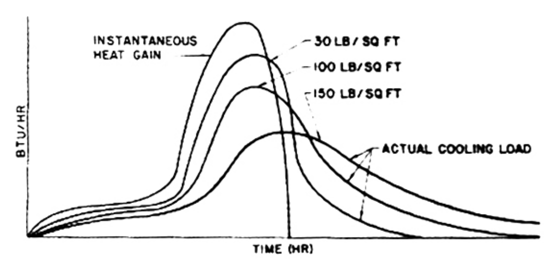


FIG. 5-ACTUAL COOLING LOAD, SOLAR HEAT GAIN, LIGHT, MEDIUM AND HEAVY CONSTRUCTION

Adding the pulldown load to the cooling load for that day results in the actual cooling load for *16-hour operation,* as illustrated in *Fig. 7.*

The upper curve represents the instantaneous heat gain and the lower curve the *actual cooling load* for that day with a constant temperature maintained within the space during the operating period of the equipment. The dotted line represents the additional cooling load from the heat left in the building construction. The temperature in the space rises during the shutdown period from the nighttime transmission load and the stored heat, and is brought back to the control point during the pulldown perios.

Shorter periods of operation increase the pulldown load because more stored heat is left in the building construction when the equipment is shut off. *Fig. 8* illustrates the *pulldown load for 12-hour operation.*

Adding this pulldown load to the cooling load for that day results in the actual cooling load for *12-hour operation,* as illustrated in *Fig. 9.*

The upper and lower solid curves are the instantaneous heat gain and the actual cooling load in average construction space with a constant temperature maintained during the operating period. The cross-hatched areas again represent the Heat Stored and the Stored Heat Removed from the construction.

The *light load (fluorescent)* is shown in *Fig. 10 for 12- and 16-hour operation* with a constant space temperature (assuming 10-hour operation of lights).

Basis of Tables 7 thru 12

Storage Load Factors,
Solar and light Heat Gain
12-, 16-, and 24-hour Operation,
Constant Space Temperature

These tables are calculated, using a procedure developed from a series of tests in actual buildings. These tests were conducted in office buildings, supermarkets, and residences throughout this country.

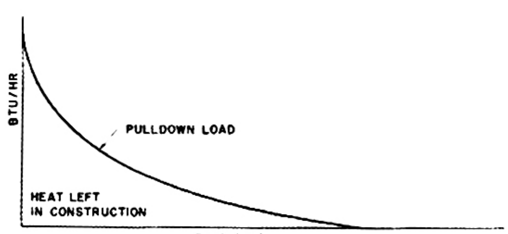


FIG. 6-PULLDOWN LOAD, SOLAR HEAT GAIN, WEST EXPOSURE, 16-HOUR OPERATION

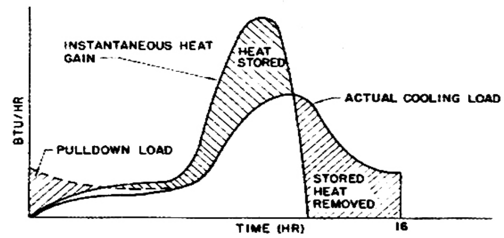


FIG. 7-ACTUAL COOLING LOAD, SOLAR HEAT GAIN, WEST EXPOSURE, 16-HOUR OPERATION

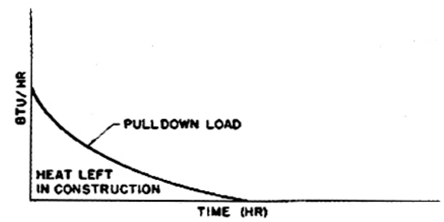


FIG. 8-PULLDOWN LOAD, SOLAR HEAT GAIN, WEST EXPOSURE, 12-HOUR OPERATION

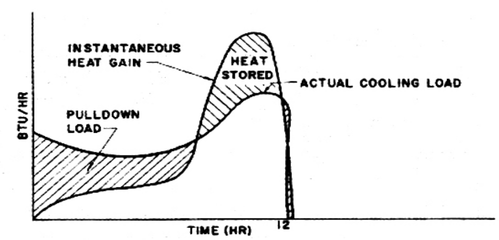


FIG. 9-ACTUAL COOLING LOAD, SOLAR HEAT GAIN, WEST EXPOSURE, 12-HOUR OPERATION

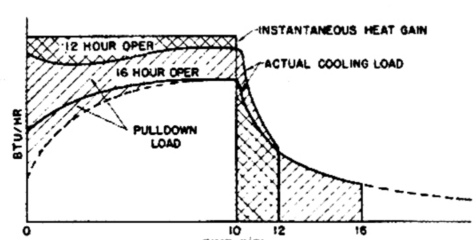


FIG. 10-ACTUAL COOLING LOAD FROM FLUORESCENT LIGHTS, 12-AND 16-HOUR OPERATION

The magnitude of the storage effect is determined largely by the thermal capacity or heat holding capacity of the materials surrounding the space. The thermal capacity of a material is the weight times the specific heat of the material. Since the specific heat of most construction material is approximately 0.20 Btu/ (lb) (F), the thermal capacity is directly proportional to the weight of the material. Therefore, the data in the tables is based on weight of the materials surrounding the space, per square foot of floor area.

Use of Tables 7 thru 12

Storage Load Factors,

Solar and Light Heat Gain

12-, 16-, and 24-hour Operation,

Constant Space Temperature

Table 7 thru 11 are used to determine the actual coolingload from the solar heat gain with a constant temperature maintained within the space for different types of construction and periods of operation. With both the 12- and 16-hour factors, the starting time is assumed to be 6 a.m. suntime (7 a.m. Daylight Saving Time). The weight per sq ft of types of construction are listed in *Tables 21 thru 33, pages 66-76.*

The actual cooling load is determined by multiplying the storage load factor from these tables for any or all times by the peak solar heat gain for the particular exposure, month and latitude desired. *Table 6* is a compilation of the peak solar heat gains for each exposure, month and latitude. These values are extracted from *Table 15, page 44.* The peak solar heat gain is also to be multiplied by either or both the applicable over-all factor for shading devices *(Table 16, page 52)* and the corrections listed under *Table 6.* Reduction in solar heat gain from the shading of the window by reveals and/or overhang should also be utilized.

Example 1 – Actual Cooling Load, Solar Heat Gain

Given:

A 20 ft X 20 ft X 8 ft outside office room with 6-inch sand aggregate concrete floor, with a floor tile finish, 21/2-inch solid sand plaster partitions, no suspended ceiling, and a 12-inch common brick outside wall with 5/8-inch sand aggregate plaster finish on inside surface. A 16 ftX5 ft steel sash window with a white venetian blind is in the outside wall and the wall faces west.

Find:

A. The actual cooling load from ths solar heat gain in July at 4 p.m., 40° North latitude with the air conditioning equipment operating 24 hours during the peak load periods and a constant temperature maintained within the room.

B. The cooling load at 8 p.m. for the same conditions.

Solution:

The weight per sq ft of floor area of this room (values obtained from *Chapter 5*) is:

$$\text{Outside wall} = \frac{(20x8) - (16x5)}{20x20} \text{ X126lb/sq ft}$$

(Table 21, page 66)

= 25.2 lb/sq ft floor area

$$\text{Partitions} = \tfrac{1}{2}\ \frac{20x8x3}{20x20} \text{ X22lb/sq ft}$$

(Table 26, page 70)

= 13.2 lb/sq ft floor area

$$\text{Floor} = \tfrac{1}{2}\ \frac{20x20}{20x20} \text{ X59lb/sq ft}$$

(Table 29, page 73)

= 29.5 lb/sq ft floor area

$$\text{Ceiling} = \tfrac{1}{2}\ \frac{20x20}{20x20} \text{ X59lb/sq ft}$$

(Table 29, page 73)

= 29.5 lb/sq ft floor area

NOTE: One-half of he partition, floor and ceiling thickness is used, assuming that the spaces above and below are conditioned and are utilizing the other halves for storage of heat.

Total weight per sq ft of floor area

= 25.2 + 13.2 + 29.5 + 29.5 = 97.4 lb/sq ft.

The overall factor for the window with the white venetian blind is 0.56 *(Table 16, page 52)* and the correction for steel sash = 1/.85.

A. Storage factor, 4 p.m. = 0.66 *(Table 7)*

The peak solar heat gain for a west exposure in July at 40° North latitude = 164 Btu/(hr)(sq ft), *(Table 6).*

Actual cooling load

$$= (5 \times 16 \times 164 \times .56 \times \frac{1}{.85}) \times 0.66 = 5700 \quad \text{Btu/hr}$$

B. Storage factor, 8 p.m. = .20 *(Table 7)*

Actual cooling load

$$= (5 \times 16 \times 164 \times .56 \times \frac{1}{.85}) \times .20 = 1730$$

Table 12 is used to determine the actual cooling load from the heat gain from lights. These data may also be used to determine the actual cooling load from:

1. People – except in densely populated areas such as auditoriums, theaters, etc. The radiant heat exchange from the body is reduced in situations like this because there is relatively

less surface available for the body to radiate to.

2. Some appliances and machines that operate periodically, with hot exterior surfaces such as ovens, dryers, hot tanks, etc.

NOTE: For Items 1 and 2 above, use values listed for fluorescent exposed lights.

Example 2 – Actual Cooling Load, Lights and People

Given:

The same room as in *Example 1* with a light heat gain of 3 watts per sq ft of floor area not including ballast, exposed fluorescent lights and 4 people. The room temperature to be maintained at 78 F db with 24-hour operation during the peak load periods.

Find:

The actual cooling load at 4 p.m. (with the lights turned on as the people arrive at 8 a.m.).

Solution:

The time elapsed after the lights are turned on is 8 hours (8 a.m. to 4 p.m.).

Storage load factor = .87 *(Table 12)*.

Sensible heat gain from people = 215 Btu/hr *(Table 48, page 100)*

Actual cooling load

$= [(3 \times 3.4 \times 1.25 \times 20 \times 20) + (4 \times 215)] \times .87$

= 5190 Btu/hr.

TABLE 6-PEAK SOLAR HEAT GAIN THRU ORDINARY GLASS*

Btu/(hr)(sq ft)

NORTH LAT.	MONTH	EXPOSURE NORTH LATITUDE N†	NE	E	SE	S	SW	W	NW	Horiz	MONTH	SOUTH LAT.
0°	June	59	156	147	42	14	42	147	156	226	Dec	0°
	July & May	48	153	152	52	14	52	152	153	233	Nov & Jan	
	Aug & April	25	141	163	79	14	79	163	141	245	Oct & Feb	
	Sept & March	10	118	167	118	14	118	167	118	250	Sept & March	
	Oct & Feb	10	79	163	141	34	141	163	79	245	Aug & April	
	Nov & Jan	10	52	152	153	67	153	152	52	233	July & May	
	Dec	10	42	147	156	82	156	147	42	226	June	
10°	June	40	153	155	55	14	55	155	153	243	Dec	10°
	July & May	30	148	158	66	14	66	158	148	247	Nov & Jan	
	Aug & April	13	130	163	94	14	94	163	130	250	Oct & Feb	
	Sept & March	10	103	164	127	28	127	164	103	247	Sept & March	
	Oct & Feb	10	66	155	149	73	149	155	66	230	Aug & April	
	Nov & Jan	9	37	143	161	106	101	143	37	210	July & May	
	Dec	9	28	137	163	120	163	137	28	202	June	
20°	June	26	154	160	73	14	73	160	154	250	Dec	20°
	July & May	19	138	163	85	14	85	163	138	251	Nov & Jan	
	Aug & April	11	118	165	113	26	113	165	118	247	Oct & Feb	
	Sept & March	10	87	163	140	65	140	163	87	233	Sept & March	
	Oct & Feb	9	52	147	160	111	160	147	52	208	Aug & April	
	Nov & Jan	8	26	128	164	141	164	128	26	180	July & May	
	Dec	8	18	121	167	149	167	121	18	170	June	
30°	June	20	139	161	90	21	90	161	139	250	Dec	30°
	July & May	16	131	164	100	30	100	164	131	246	Nov & Jan	
	Aug & April	11	108	165	129	63	129	165	108	235	Oct & Feb	
	Sept & March	9	90	158	152	105	152	158	90	212	Sept & March	
	Oct & Feb	8	39	135	163	145	163	135	39	179	Aug & April	
	Nov & Jan	7	16	116	162	159	162	116	16	145	July & May	
	Dec	6	12	105	162	163	162	105	12	131	June	
40°	June	17	133	162	111	54	111	162	133	237	Dec	40°
	July & May	15	127	164	125	69	125	164	127	233	Nov & Jan	
	Aug & April	11	102	162	146	102	146	162	102	214	Oct & Feb	
	Sept & March	9	58	149	162	140	162	149	58	183	Sept & March	
	Oct & Feb	7	35	122	163	162	163	122	35	129	Aug & April	
	Nov & Jan	5	12	100	156	166	156	100	12	103	July & May	
	Dec	5	10	86	148	165	148	86	10	85	June	
50°	June	16	126	164	135	93	135	164	126	220	Dec	50°
	July & May	14	117	163	143	106	143	163	117	211	Nov & Jan	
	Aug & April	11	94	158	157	138	157	158	94	185	Oct & Feb	
	Sept & March	8	58	138	163	158	163	138	58	148	Sept & March	
	Oct & Feb	5	29	105	157	167	157	105	29	94	Aug & April	
	Nov & Jan	4	9	64	127	153	127	64	9	53	July & May	
	Dec	3	7	47	116	141	116	47	7	40	June	
		5	SE	E	NE	N	NW	W	SW	Horiz		
		EXPOSURE SOUTH LATITUDE										

Solar Gain Correction	Steel Sash or No Sash	Haze	Altitude	Dewpoint	Dewpoint	South Lat
	X 1/.85 or 1.17	-15% (Max)	+0.7% per 1000 ft	Above 67 F -7% per 10 F	Below 67 F +7% per 10 F	Dec or Jan +7%

* Abstracted from Table 15, page 43.

†Solar heat gain on North exposure (inNorth Latitudes) or on South exposure (in South latitudes) consists primarily of diffuse radiation which is essentially constant throughout the day. The solar heat gain values for this exposure are the average for the 12 hr period (6 a.m. to 6 p.m.). The storage factors in Tables 7 thru 11 assume that the solar heat gain on the North (or South) exposure is constant.

TABLE 7-STORAGE LOAD FACTORS, SOLAR HEAT GAIN THRU GLASS
WITH INTERNAL SHADE*
24 Hour Operation, Constant Space Temperature†

EXPOSURE (North Lat)	WEIGHT§ (lb per sq ft of floor area)	SUN TIME																								EXPOSURE (South Lat)
		AM							PM													AM				
		6	7	8	9	10	11	12	1	2	3	4	5	6	7	8	9	10	11	12	1	2	3	4	5	
Northeast	150 & over	.47	.58	.54	.42	.27	.21	.20	.19	.18	.17	.16	.14	.12	.09	.08	.07	.06	.06	.05	.05	.04	.04	.04	.03	Southeast
	100	.48	.60	.57	.46	.30	.24	.20	.19	.17	.16	.15	.13	.11	.08	.07	.06	.05	.05	.04	.04	.03	.03	.02	.02	
	30	.55	.76	.73	.58	.36	.24	.19	.17	.15	.13	.12	.11	.07	.04	.02	.02	.01	.01	0	0	0	0	0	0	
East	150 & over	.39	.56	.62	.59	.49	.33	.23	.21	.20	.18	.17	.15	.12	.10	.09	.08	.08	.07	.06	.05	.05	.05	.04	.04	East
	100	.40	.58	.65	.63	.52	.35	.24	.22	.20	.18	.16	.14	.12	.09	.08	.07	.06	.05	.05	.04	.04	.03	.03	.02	
	30	.46	.70	.80	.79	.64	.42	.25	.19	.16	.14	.11	.09	.07	.04	.02	.02	.01	.01	0	0	0	0	0	0	
Southeast	150 & over	.04	.28	.47	.59	.64	.62	.53	.41	.27	.24	.21	.19	.16	.14	.12	.11	.10	.09	.08	.07	.06	.06	.05	.05	Northeast
	100	.03	.28	.47	.61	.67	.65	.57	.44	.29	.24	.21	.18	.15	.12	.10	.09	.08	.07	.06	.05	.05	.04	.04	.03	
	30	0	.30	.57	.75	.84	.81	.69	.50	.30	.20	.17	.13	.09	.05	.04	.03	.02	.01	0	0	0	0	0	0	
South	150 & over	.06	.06	.23	.38	.51	.60	.66	.67	.64	.59	.42	.24	.22	.19	.17	.15	.13	.12	.11	.10	.09	.08	.07	.07	North
	100	.04	.04	.22	.38	.52	.63	.70	.71	.69	.59	.45	.26	.22	.18	.16	.13	.12	.10	.09	.08	.07	.06	.06	.05	
	30	.10	.21	.43	.63	.77	.86	.88	.82	.56	.50	.24	.16	.11	.08	.05	.04	.02	.02	.01	.01	0	0	0	0	
Southwest	150 & over	.08	.08	.09	.10	.11	.24	.39	.53	.63	.66	.61	.47	.23	.19	.18	.16	.14	.13	.11	.10	.09	.08	.08	.07	Northwest
	100	.07	.08	.08	.08	.10	.24	.40	.55	.66	.70	.64	.50	.26	.20	.17	.15	.13	.11	.10	.09	.08	.07	.06	.05	
	30	.03	.04	.06	.07	.09	.23	.47	.67	.81	.86	.79	.60	.26	.17	.12	.08	.05	.04	.03	.02	.01	.01	0	0	
West	150 & over	.08	.09	.09	.10	.10	.10	.10	.18	.36	.52	.63	.65	.55	.22	.19	.17	.15	.14	.12	.11	.10	.09	.08	.07	West
	100	.07	.08	.08	.09	.09	.09	.09	.18	.36	.54	.66	.68	.60	.25	.20	.17	.15	.13	.11	.10	.08	.07	.06	.05	
	30	.03	.04	.06	.07	.08	.08	.08	.19	.42	.65	.81	.85	.74	.30	.19	.13	.09	.06	.05	.03	.02	.02	.01	0	
Northwest	150 & over	.08	.09	.10	.10	.10	.10	.10	.10	.16	.33	.49	.61	.60	.19	.17	.15	.13	.12	.10	.09	.08	.08	.07	.06	Southwest
	100	.07	.08	.09	.09	.10	.10	.10	.10	.16	.34	.52	.65	.64	.23	.18	.15	.12	.11	.09	.08	.07	.06	.06	.05	
	30	.03	.05	.07	.08	.09	.09	.10	.10	.17	.39	.63	.80	.79	.28	.18	.12	.09	.06	.04	.03	.02	.02	.01	0	
North and Shade	150 & over	.08	.37	.67	.71	.74	.76	.79	.81	.83	.84	.86	.87	.88	.29	.26	.23	.20	.19	.17	15	.14	.12	.11	.10	South and Shade
	100	.06	.31	.67	.72	.76	.79	.81	.83	.85	.87	.88	.90	.91	.30	.26	.22	.19	.16	.15	.13	.12	.10	.09	.08	
	30	0	.25	.74	.83	.88	.91	.94	.96	.96	.98	.98	.99	.99	.26	.17	.12	.08	.05	.04	.03	.02	.01	.01	.01	

Equation: Cooling Load, Btu/hr = [Peak solar heat gain, Btu/(hr) (sq ft), (*Table 6*)]
× [Window area, sq ft]
× [Shade factor, Haze factor, etc., (*Chapter 4*)]
× [Storage factor, (above Table at desired time)]

* Internal shading device is any type of shade located on the inside of the glass.

†These factors apply when maintaining a CONSTANT TEMPERATURE in the space during the operating period. Where the temperature is allowed to swing, additional storage will result during peak load periods. Refer to Table 13 for applicable storage factors.

§ Weight per sq ft of floor-

$$\text{Room on Bldg Exterior (One or more outside walls)} = \frac{\text{(Weight of Outside Walls, lb)} + \tfrac{1}{2}\text{(Weight of Partitions, Floor and Ceiling, lb)}}{\text{Floor Area in Room, sq ft}}$$

$$\text{Room in Bldg Interior (No outside walls)} = \frac{\tfrac{1}{2}\text{(Weight of Partitions, Floor and Ceiling, lb)}}{\text{Floor Area in Room, sq ft}}$$

$$\text{Basement Room (Floor on ground)} = \frac{\text{(Weight of Outside Walls, lb)} + \text{(Weight of Floor, lb)} + \tfrac{1}{2}\text{(Weight of Partitions and Ceiling, lb)}}{\text{Floor Area in Room, sq ft}}$$

$$\text{Entire Building or Zone} = \frac{\text{(Weight of Outside Wall, Partitons, Floors, Ceilings, Structural Members and Supports,lb)}}{\text{Air Conditioned Floor Area, sq ft}}$$

With rug on floor-Weight of floor should be multiplied by 0.50 to compensate for insulating effect of rug.

Weights per sq ft of common types of construction are contained in *Tables 21 thru 33, pages 66 thru 76.*

TABLE 8-STORAGE LOAD FACTORS, SOLAR HEAT GAIN THRU GLASS

WITH BARE GLASS OR WITH EXTERNAL SHADE‡
24 Hour Operation, Constant Space Temperature†

EXPOSURE (North Lat)	WEIGHT§ (lb per sq ft of floor area)	SUN TIME																								EXPOSURE (South Lat)
		AM							PM													AM				
		6	7	8	9	10	11	12	1	2	3	4	5	6	7	8	9	10	11	12	1	2	3	4	5	
Northeast	150 & over	.17	.27	.33	.33	.31	.29	.27	.25	.23	.22	.20	.19	.17	.15	.14	.12	.11	.10	.09	.08	.07	.07	.06	.06	Southeast
	100	.19	.31	.38	.39	.36	.34	.27	.24	.22	.21	.19	.17	.16	.14	.12	.10	.07	.08	.07	.06	.05	.05	.04	.03	
	30	.31	.56	.65	.61	.46	.33	.26	.21	.18	.16	.14	.12	.09	.06	.04	.03	.02	.01	.01	.01	0	0	0	0	
East	150 & over	.16	.26	.34	.39	.40	.38	.34	.30	.28	.26	.23	.22	.20	.18	.16	.14	.13	.12	.10	.09	.08	.08	.07	.06	East
	100	.16	.29	.40	.46	.46	.42	.36	.31	.28	.25	.23	.20	.18	.15	.14	.12	.11	.09	.08	.08	.06	.06	.05	.04	
	30	.27	.50	.67	.73	.68	.53	.38	.27	.22	.18	.15	.12	.09	.06	.04	.03	.02	.01	.01	.01	.01	0	0	.01	
Southeast	150 & over	.08	.14	.22	.31	.38	.43	.44	.43	.39	.35	.32	.29	.26	.23	.21	.19	.16	.15	.13	.12	.11	.10	.09	.08	Northeast
	100	.05	.12	.23	.35	.44	.49	.51	.47	.41	.36	.31	.27	.24	.21	.18	.16	.14	.12	.10	.09	.08	.08	.06	.06	
	30	0	.18	.40	.59	.72	.77	.72	.60	.44	.32	.23	.18	.14	.09	.07	.05	.03	.02	.01	.01	.01	0	0	0	
South	150 & over	.10	.10	.13	.20	.28	.35	.42	.48	.51	.51	.48	.42	.37	.33	.29	.26	.23	.21	.19	.17	.15	.14	.13	.12	North
	100	.07	.06	.12	.20	.30	.39	.48	.54	.58	.57	.53	.45	.37	.31	.27	.23	.20	.18	.16	.14	.12	.11	.10	.08	
	30	0	0	.12	.29	.48	.64	.75	.82	.81	.75	.61	.42	.28	.19	.13	.09	.06	.04	.03	.02	.01	.01	0	0	
Southwest	150 & over	.11	.10	.10	.10	.10	.14	.21	.29	.36	.43	.47	.46	.40	.34	.30	.27	.24	.22	.20	.18	.16	.14	.13	.12	Northwest
	100	.09	.09	.08	.09	.09	.14	.22	.31	.42	.50	.53	.51	.44	.35	.29	.26	.22	.19	.17	.15	.13	.12	.11	.09	
	30	.02	.03	.05	.06	.08	.12	.34	.53	.68	.78	.78	.68	.46	.29	.20	.14	.09	.07	.05	.03	.02	.02	.01	.01	
West	150 & over	.12	.11	.11	.10	.10	.10	.10	.13	.19	.27	.36	.42	.44	.38	.33	.29	.26	.23	.21	.18	.16	.15	.13	.12	West
	100	.09	.09	.09	.09	.09	.09	.10	.12	.19	.30	.40	.48	.51	.42	.35	.30	.25	.22	.19	.16	.14	.13	.11	.09	
	30	.02	.03	.05	.08	.07	.07	.08	.14	.29	.49	.67	.76	.75	.53	.33	.22	.15	.11	.08	.05	.04	.03	.02	.01	
Northwest	150 & over	.10	.10	.10	.47	.10	.10	.10	.10	.12	.17	.25	.34	.39	.34	.29	.26	.23	.20	.18	.16	.14	.13	.12	.10	Southwest
	100	.08	.09	.09	.57	.09	.09	.09	.09	.11	.19	.29	.40	.46	.40	.32	.26	.22	.19	.16	.14	.13	.11	.10	.08	
	30	.02	.04	.05	.82	.08	.09	.10	.10	.13	.27	.48	.65	.73	.49	.31	.21	.16	.10	.07	.05	.04	.03	.02	.01	
North and Shade	150 & over	.16	.23	.33		.47	.52	.57	.61	.66	.69	.72	.74	.59	.52	.46	.42	.37	.34	.31	.27	.25	.23	.21	.17	South and Shade
	100	.11	.33	.44		.57	.62	.66	.70	.74	.76	.79	.80	.60	.51	.44	.37	.32	.29	.27	.23	.21	.18	.16	.13	
	30	0	.48	.66		.82	.87	.91	.93	.95	.97	.98	.98	.52	.34	.24	.16	.11	.07	.05	.04	.02	.02	.01	.01	

Equation: Cooling Load, Btu/hr = [Peak solar heat gain, Btu/(hr) (sq ft), (*Table 6*)]
× [Window area, sq ft]
× [Shade factor, Haze factor, etc., (*Chapter 4*)]
× [Storage factor, (above Table at desired time)]

‡Bare glass-Any window with no inside shading device. Windows with shading devices on the outside or shaded by external projections are considered bare glass.

†These factors apply when maintaining a CONSTANT TEMPERATURE in the space during the operating period. Where the temperature is allowed to swing, additional storage will result during peak load periods. Refer to Table 13 for applicable storage factors.

§ Weight per sq ft of floor-

$$\text{Room on Bldg Exterior (One or more outside walls)} = \frac{\text{(Weight of Outside Walls, lb)} + \tfrac{1}{2}\text{(Weight of Partitions, Floor and Ceiling, lb)}}{\text{Floor Area in Room, sq ft}}$$

$$\text{Room in Bldg Interior (No outside walls)} = \frac{\tfrac{1}{2}\text{(Weight of Partitions, Floor and Ceiling, lb)}}{\text{Floor Area in Room, sq ft}}$$

$$\text{Basement Room (Floor on ground)} = \frac{\text{(Weight of Outside Walls, lb)} + \text{(Weight of Floor, lb)} + \tfrac{1}{2}\text{(Weight of Partitions and Ceiling, lb)}}{\text{Floor Area in Room, sq ft}}$$

$$\text{Entire Building or Zone} = \frac{\text{(Weight of Outside Wall, Partitons, Floors, Ceilings, Structural Members and Supports,lb)}}{\text{Air Conditioned Floor Area, sq ft}}$$

With rug on floor-Weight of floor should be multiplied by 0.50 to compensate for insulating effect of rug.
Weights per sq ft of common types of construction are contained in *Tables 21 thru 33, pages 66 thru 76.*

TABLE 9-STORAGE LOAD FACTORS, SOLAR HEAT GAIN THRU GLASS

WITH INTERNAL SHADING DEVICE*
16 Hour Operation, Constant Space Temperature†

EXPOSURE (North Lat)	WEIGHTS (lb per sq ft of floor area)	SUN TIME AM							PM									EXPOSURE (South Lat)
		6	7	8	9	10	11	12	1	2	3	4	5	6	7	8	9	
Northeast	150 & over	.53	.64	.59	.47	.31	.25	.24	.22	.18	.17	.16	.14	.12	.09	.08	.07	Southeast
	100	.53	.65	.61	.50	.33	.27	.22	.21	.17	.16	.15	.13	.11	.08	.07	.06	
	30	.56	.77	.73	.58	.36	.24	.19	.17	.15	.13	.12	.11	.07	.04	.02	.02	
East	150 & over	.47	.63	.68	.64	.54	.38	.27	.25	.20	.18	.17	.15	.12	.10	.09	.08	East
	100	.46	.63	.70	.67	.56	.38	.27	.24	.20	.18	.16	.14	.12	.09	.08	.07	
	30	.47	.71	.80	.79	.64	.42	.25	.19	.16	.14	.11	.09	.07	.04	.02	.02	
Southeast	150 & over	.14	.37	.55	.66	.70	.68	.58	.46	.27	.24	.21	.19	.16	.14	.12	.11	Northeast
	100	.11	.35	.53	.66	.72	.69	.61	.47	.29	.24	.21	.18	.15	.12	.10	.09	
	30	.02	.31	.57	.75	.84	.81	.69	.50	.30	.20	.17	.13	.09	.05	.04	.03	
South	150 & over	.19	.18	.34	.48	.60	.68	.73	.74	.64	.59	.42	.24	.22	.19	.17	.15	North
	100	.16	.14	.31	.46	.59	.69	.76	.70	.69	.59	.45	.26	.22	.18	.16	.13	
	30	.12	.23	.44	.64	.77	.86	.88	.82	.56	.50	.24	.16	.11	.08	.05	.04	
Southwest	150 & over	.22	.21	.20	.20	.20	.32	.47	.60	.63	.66	.61	.47	.23	.19	.18	.16	Northwest
	100	.20	.19	.18	.17	.18	.31	.46	.60	.66	.70	.64	.50	.26	.20	.17	.15	
	30	.08	.08	.09	.09	.10	.24	.47	.67	.81	.86	.79	.60	.26	.17	.12	.08	
West	150 & over	.23	.23	.21	.21	.20	.19	.18	.25	.36	.52	.63	.65	.55	.22	.19	.17	West
	100	.22	.21	.19	.19	.17	.16	.15	.23	.36	.54	.66	.68	.60	.25	.20	.17	
	30	.12	.10	.10	.10	.10	.10	.09	.19	.42	.65	.81	.85	.74	.30	.19	.13	
Northwest	150 & over	.21	.21	.20	.19	.18	.18	.17	.16	.16	.33	.49	.61	.60	.19	.17	.15	Southwest
	100	.19	.19	.18	.17	.17	.16	.16	.15	.16	.34	.52	.65	.23	.18	.15	.12	
	30	.12	.11	.11	.11	.11	.11	.11	.10	.17	.39	.63	.80	.79	.28	.18	.12	
North and Shade	150 & over	.23	.58	.75	.79	.80	.80	.81	.82	.83	.84	.86	.87	.88	.39	.35	.31	South and Shade
	100	.25	.46	.73	.78	.82	.82	.83	.84	.85	.87	.88	.89	.90	.40	.34	.29	
	30	.07	.22	.69	.80	.86	.93	.94	.95	.97	.98	.98	.99	.99	.35	.23	.16	

Equation: Cooling Load, Btu/hr = [Peak solar heat gain, Btu/(hr) (sq ft), (*Table 6*)]
× [Window area, sq ft]
× [Shade factor, Haze factor, etc., (*Chapter 4*)]
× [Storage factor, (above Table at desired time)]

*Internal shading device is any type of shade located on the inside of the glass.

†These factors apply when maintaining a CONSTANT TEMPERATURE in the space during the operating period. Where the temperature is allowed to swing, additional storage will result during peak load periods. Refer to Table 13 for applicable storage factors.

§ Weight per sq ft of floor-

$$\text{Room on Bldg Exterior (One or more outside walls)} = \frac{\text{(Weight of Outside Walls, lb)} + \tfrac{1}{2}\,\text{(Weight of Partitions, Floor and Ceiling, lb)}}{\text{Floor Area in Room, sq ft}}$$

$$\text{Room in Bldg Interior (No outside walls)} = \frac{\tfrac{1}{2}\,\text{(Weight of Partitions, Floor and Ceiling, lb)}}{\text{Floor Area in Room, sq ft}}$$

$$\text{Basement Room (Floor on ground)} = \frac{\text{(Weight of Outside Walls, lb)} + \text{(Weight of Floor, lb)} + \tfrac{1}{2}\,\text{(Weight of Partitions and Ceiling, lb)}}{\text{Floor Area in Room, sq ft}}$$

$$\text{Entire Building or Zone} = \frac{\text{(Weight of Outside Wall, Partitons, Floors, Ceilings, Structural Members and Supports,lb)}}{\text{Air Conditioned Floor Area, sq ft}}$$

With rug on floor-Weight of floor should be multiplied by 0.50 to compensate for insulating effect of rug.
Weights per sq ft of common types of construction are contained in *Tables 21 thru 33, pages 66 thru 76.*

TABLE 10-STORAGE LOAD FACTORS, SOLAR HEAT GAIN THRU GLASS

WITH BARE GLASS OR WITH EXTERNAL SHADE‡
16 Hour Operation, Constant Space Temperature†

EXPOSURE (North Lat)	WEIGHT§ (lb per sq ft of floor area)	SUN TIME AM 6	7	8	9	10	11	12	PM 1	2	3	4	5	6	7	8	9	EXPOSURE (South Lat)
Northeast	150 & over	.28	.37	.42	.41	.38	.36	.33	.31	.23	.22	.20	.19	.17	.15	.14	.12	Southeast
	100	.28	.39	.45	.45	.41	.39	.31	.27	.22	.21	.19	.17	.16	.14	.12	.10	
	30	.33	.57	.66	.62	.46	.33	.26	.21	.18	.16	.14	.12	.09	.06	.04	.03	
East	150 & over	.29	.38	.44	.48	.48	.46	.41	.36	.28	.26	.23	.22	.20	.18	.16	.14	East
	100	.27	.38	.48	.54	.52	.48	.41	.35	.28	.25	.23	.20	.18	.15	.14	.12	
	30	.29	.51	.68	.74	.69	.53	.38	.27	.22	.18	.15	.12	.09	.06	.04	.03	
Southeast	150 & over	.24	.29	.35	.43	.49	.53	.53	.51	.39	.35	.32	.29	.26	.23	.21	.19	Northeast
	100	.19	.24	.33	.44	.52	.57	.57	.53	.41	.36	.31	.27	.24	.21	.18	.16	
	30	.03	.20	.41	.60	.73	.77	.72	.60	.44	.32	.23	.18	.14	.09	.07	.05	
South	150 & over	.33	.31	.32	.37	.43	.49	.55	.60	.57	.51	.48	.42	.37	.33	.29	.26	North
	100	.27	.24	.28	.34	.42	.50	.58	.60	.60	.57	.53	.45	.37	.31	.27	.23	
	30	.06	.04	.15	.31	.49	.65	.75	.82	.81	.75	.61	.42	.28	.19	.13	.09	
Southwest	150 & over	.35	.32	.30	.28	.26	.28	.30	.37	.43	.47	.46	.40	.34	.30	.27	.24	Northwest
	100	.31	.28	.25	.24	.22	.26	.33	.40	.46	.50	.53	.51	.44	.35	.29	.26	
	30	.11	.10	.10	.09	.10	.14	.35	.54	.68	.78	.78	.68	.46	.29	.20	.14	
West	150 & over	.38	.34	.32	.28	.26	.25	.23	.25	.26	.27	.36	.42	.44	.38	.33	.29	West
	100	.34	.31	.28	.25	.23	.22	.21	.21	.23	.30	.40	.48	.51	.43	.35	.30	
	30	.17	.14	.13	.11	.11	.10	.10	.15	.29	.49	.67	.76	.75	.53	.33	.22	
Northwest	150 & over	.33	.30	.28	.26	.24	.23	.22	.20	.18	.17	.25	.34	.39	.34	.29	.26	Southwest
	100	.30	.28	.25	.23	.22	.20	.19	.17	.17	.19	.29	.40	.46	.40	.32	.26	
	30	.18	.14	.12	.12	.12	.12	.12	.11	.13	.27	.48	.65	.73	.49	.31	.21	
North and Shade	150 & over	.31	.57	.64	.68	.72	.73	.73	.74	.74	.75	.76	.78	.78	.59	.52	.46	South and Shade
	100	.30	.47	.60	.67	.72	.74	.77	.78	.79	.80	.81	.82	.83	.60	.51	.44	
	30	.04	.07	.53	.70	.78	.84	.88	.91	.93	.95	.97	.98	.99	.62	.34	.24	

Equation: Cooling Load, Btu/hr = [Peak solar heat gain, Btu/(hr) (sq ft), (*Table 6*)]
× [Window area, sq ft]
× [Shade factor, Haze factor, etc., (*Chapter 4*)]
× [Storage factor, (above Table at desired time)]

‡Bare glass-Any window with no inside shading device. Windows with shading devices on the outside or shaded by external projections are considered bare glass.

†These factors apply when maintaining a CONSTANT TEMPERATURE in the space during the operating period. Where the temperature is allowed to swing, additional storage will result during peak load periods. Refer to Table 13 for applicable storage factors.

§ Weight per sq ft of floor-

$$\text{Room on Bldg Exterior (One or more outside walls)} = \frac{\text{(Weight of Outside Walls, lb)} + \frac{1}{2}\text{ (Weight of Partitions, Floor and Ceiling, lb)}}{\text{Floor Area in Room, sq ft}}$$

$$\text{Room in Bldg Interior (No outside walls)} = \frac{\frac{1}{2}\text{ (Weight of Partitions, Floor and Ceiling, lb)}}{\text{Floor Area in Room, sq ft}}$$

$$\text{Basement Room (Floor on ground)} = \frac{\text{(Weight of Outside Walls, lb)} + \text{(Weight of Floor, lb)} + \frac{1}{2}\text{ (Weight of Partitions and Ceiling, lb)}}{\text{Floor Area in Room, sq ft}}$$

$$\text{Entire Building or Zone} = \frac{\text{(Weight of Outside Wall, Partitons, Floors, Ceilings, Structural Members and Supports,lb)}}{\text{Air Conditioned Floor Area, sq ft}}$$

With rug on floor-Weight of floor should be multiplied by 0.50 to compensate for insulating effect of rug.
Weights per sq ft of common types of construction are contained in *Tables 21 thru 33, pages 66 thru 76.*

TABLE 11-STORAGE LOAD FACTORS, SOLAR HEAT GAIN THRU GLASS

12 Hour Operation, Constant Space Temperature†

EXPOSURE (North Lat)	WEIGHT§ (lb per sq ft of floor area)	SUN TIME																								EXPOSURE (South Lat)
		AM							PM													AM				
		6	7	8	9	10	11	12	1	2	3	4	5	6	7	8	9	10	11	12	1	2	3	4	5	
Northeast	150 & over	.59	.67	.62	.49	.33	.27	.25	.24	.22	.21	.20	.17	.34	.42	.47	.45	.42	.39	.36	.33	.30	.29	.26	.25	Southeast
	100	.59	.68	.64	.52	.35	.29	.24	.23	.20	.19	.17	.15	.35	.45	.50	.49	.45	.42	.34	.30	.27	.26	.23	.20	
	30	.62	.80	.75	.60	.37	.25	.19	.17	.15	.13	.12	.11	.40	.62	.69	.64	.48	.34	.27	.22	.18	.16	.14	.12	
East	150 & over	.51	.66	.71	.67	.57	.40	.29	.26	.25	.23	.21	.19	.36	.44	.50	.53	.53	.50	.44	.39	.36	.34	.30	.28	East
	100	.52	.67	.73	.70	.58	.40	.29	.26	.24	.21	.19	.16	.34	.44	.54	.58	.57	.51	.44	.39	.34	.31	.28	.24	
	30	.53	.74	.82	.81	.65	.43	.25	.19	.16	.14	.11	.09	.36	.56	.71	.76	.70	.54	.39	.28	.23	.18	.15	.12	
Southeast	150 & over	.20	.42	.59	.70	.74	.71	.61	.48	.33	.30	.26	.24	.34	.37	.43	.50	.54	.58	.57	.55	.50	.45	.41	.37	Northeast
	100	.18	.40	.57	.70	.75	.72	.63	.49	.34	.28	.25	.21	.29	.33	.41	.51	.58	.61	.61	.56	.49	.44	.37	.33	
	30	.09	.35	.61	.78	.86	.82	.69	.50	.30	.20	.17	.13	.14	.27	.47	.64	.75	.79	.73	.61	.45	.32	.23	.18	
South	150 & over	.28	.25	.40	.53	.64	.72	.77	.77	.73	.67	.49	.31	.47	.43	.42	.46	.51	.56	.61	.65	.66	.65	.61	.54	North
	100	.26	.22	.38	.51	.64	.73	.79	.79	.77	.65	.51	.31	.44	.37	.39	.43	.50	.57	.64	.68	.70	.68	.63	.53	
	30	.21	.29	.48	.67	.79	.88	.89	.83	.56	.50	.24	.16	.28	.19	.25	.38	.54	.68	.78	.84	.82	.76	.61	.42	
Southwest	150 & over	.31	.27	.27	.26	.25	.27	.50	.63	.72	.74	.69	.54	.51	.44	.40	.37	.34	.36	.41	.47	.54	.57	.60	.58	Northwest
	100	.33	.28	.25	.23	.23	.35	.50	.64	.74	.77	.70	.55	.53	.44	.37	.35	.31	.33	.39	.46	.55	.62	.64	.60	
	30	.29	.21	.18	.15	.14	.27	.50	.69	.82	.87	.79	.60	.48	.32	.25	.20	.17	.19	.39	.56	.70	.80	.79	.69	
West	150 & over	.63	.31	.28	.27	.25	.24	.22	.29	.46	.61	.71	.72	.56	.49	.44	.39	.36	.33	.31	.31	.35	.42	.49	.54	West
	100	.67	.33	.28	.26	.24	.22	.20	.28	.44	.61	.72	.73	.60	.52	.44	.39	.34	.31	.29	.28	.33	.43	.51	.57	
	30	.77	.34	.25	.20	.17	.14	.13	.22	.44	.67	.82	.85	.77	.56	.38	.28	.22	.18	.16	.19	.33	.52	.69	.77	
Northwest	150 & over	.68	.28	.27	.25	.23	.22	.20	.19	.24	.41	.56	.67	.49	.44	.39	.36	.33	.30	.28	.26	.26	.30	.37	.44	Southwest
	100	.71	.31	.27	.24	.22	.21	.19	.18	.23	.40	.58	.70	.54	.49	.41	.35	.31	.28	.25	.23	.24	.30	.39	.48	
	30	.82	.33	.25	.20	.18	.15	.14	.13	.19	.41	.64	.80	.75	.53	.36	.28	.24	.19	.17	.15	.17	.30	.50	.66	
North and Shade	150 & over	.96	.96	.96	.96	.96	.96	.96	.96	.96	.96	.96	.96	.75	.75	.79	.83	.84	.86	.88	.88	.91	.92	.93	.93	South and Shade
	100	.98	.98	.98	.98	.98	.98	.98	.98	.98	.98	.98	.98	.81	.84	.86	.89	.91	.93	.93	.94	.94	.95	.95	.95	
	30	←1.00→												←1.00→												

Equation: Cooling Load, Btu/hr = [Peak solar heat gain, Btu/(hr) (sq ft), (*Table 6*)]
× [Window area, sq ft]
× [Shade factor, Haze factor, etc., (*Chapter 4*)]
× [Storage factor, (above Table at desired time)]

*Internal shading device is any type of shade located on the inside of the glass.

‡Bare glass-Any window with no inside shading device. Windows with shading devices on the outside or shaded by external projections are considered bare glass.

†These factors apply when maintaining a CONSTANT TEMPERATURE in the space during the operating period. Where the temperature is allowed to swing, additional storage will result during peak load periods. Refer to Table 13 for applicable storage factors.

§ Weight per sq ft of floor-

$$\text{Room on Bldg Exterior (One or more outside walls)} = \frac{\text{(Weight of Outside Walls, lb)} + \tfrac{1}{2}\text{ (Weight of Partitions, Floor and Ceiling, lb)}}{\text{Floor Area in Room, sq ft}}$$

$$\text{Room in Bldg Interior (No outside walls)} = \frac{\tfrac{1}{2}\text{ (Weight of Partitions, Floor and Ceiling, lb)}}{\text{Floor Area in Room, sq ft}}$$

$$\text{Basement Room (Floor on ground)} = \frac{\text{(Weight of Outside Walls, lb)} + \text{(Weight of Floor, lb)} + \tfrac{1}{2}\text{ (Weight of Partitions and Ceiling, lb)}}{\text{Floor Area in Room, sq ft}}$$

$$\text{Entire Building or Zone} = \frac{\text{(Weight of Outside Wall, Partitons, Floors, Ceilings, Structural Members and Supports,lb)}}{\text{Air Conditioned Floor Area, sq ft}}$$

With rug on floor-Weight of floor should be multiplied by 0.50 to compensate for insulating effect of rug.
Weights per sq ft of common types of construction are contained in *Tables 21 thru 33, pages 66 thru 76.*

TABLE 12-STORAGE LOAD FACTORS, HEAT GAIN-LIGHTS*

Lights On 10 Hours† with Equipment Operating 12, 16 and 24 Hours, Constant Space Temperature

	EQUIP. OPER-ATION Hours	WEIGHT (lb per sq ft of floor area)	NUMBER OF HOURS AFTER LIGHTS ARE TURNED ON																							
			0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23
Fluorescent Lights Exposed	24	150 & over	.37	.67	.71	.74	.76	.79	.81	.83	.84	.86	.87	.29	.26	.23	.20	.19	.17	.15	.14	.12	.11	.10	.09	.08
		100	.31	.67	.72	.76	.79	.81	.83	.85	.87	.88	.90	.30	.26	.22	.19	.16	.15	.13	.12	.10	.09	.08	.07	.06
		30	.25	.74	.83	.88	.91	.94	.96	.96	.98	.98	.99	.26	.17	.12	.08	.05	.04	.03	.02	.01	.01	.01	0	0
	16	150 & over	.60	.82	.83	.84	.84	.84	.85	.85	.86	.88	.90	.32	.28	.25	.23	.19								
		100	.46	.79	.84	.86	.87	.88	.88	.89	.89	.90	.90	.30	.26	.22	.19	.16								
		30	.29	.77	.85	.89	.92	.95	.96	.96	.98	.98	.99	.26	.17	.12	.08	.05								
	12	150 & over	.63	.90	.91	.93	.93	.94	.95	.95	.95	.96	.96	.37												
		100	.57	.89	.91	.92	.94	.94	.95	.95	.96	.96	.97	.36												
		30	.42	.86	.91	.93	.95	.97	.98	.98	.99	.99	.99	.26												
Fluorescent Lights Recessed in Susp. Ceiling or Exposed Incandescent Lights.	24	150 & over	.34	.55	.61	.65	.68	.71	.74	.77	.79	.81	.83	.39	.35	.31	.28	.25	.23	.20	.18	.16	.15	.14	.12	.11
		100	.24	.56	.63	.68	.72	.75	.78	.80	.82	.84	.86	.40	.34	.29	.25	.20	.18	.17	.15	.14	.12	.10	.09	.08
		30	.17	.65	.77	.84	.88	.92	.94	.95	.97	.98	.98	.35	.23	.16	.11	.07	.05	.04	.03	.02	.01	.01	0	0
	16	150 & over	.58	.75	.79	.80	.80	.81	.82	.83	.84	.86	.87	.39	.35	.31	.28	.25								
		100	.46	.73	.78	.82	.82	.82	.83	.84	.85	.87	.88	.40	.34	.29	.25	.20								
		30	.22	.69	.80	.86	.89	.93	.94	.95	.97	.98	.98	.35	.23	.16	.11	.07								
	12	150 & over	.69	.86	.89	.90	.91	.91	.92	.93	.94	.95	.95	.50												
		100	.58	.85	.88	.88	.90	.92	.93	.94	.94	.94	.95	.48												
		30	.40	.81	.88	.91	.93	.96	.97	.97	.98	.99	.99	.35												
Fluorescent or Incandescent Lights Recessed in Susp. Ceiling and Ceiling Plenum Return System.	24	150 & over	.23	.33	.41	.47	.52	.57	.61	.66	.69	.72	.74	.59	.52	.46	.42	.37	.34	.31	.27	.25	.23	.21	.18	.16
		100	.17	.33	.44	.52	.56	.61	.66	.69	.74	.77	.79	.60	.51	.37	.37	.32	.30	.27	.23	.20	.18	.16	.14	.12
		30	0	.48	.66	.76	.82	.87	.91	.93	.95	.97	.98	.52	.34	.16	.16	.11	.07	.05	.04	.02	.02	.01	0	0
	16	150 & over	.57	.64	.68	.72	.73	.73	.74	.74	.75	.76	.78	.59	.52	.42	.42	.37								
		100	.47	.60	.67	.72	.74	.77	.78	.79	.80	.81	.82	.60	.51	.37	.37	.32								
		30	.07	.53	.70	.78	.84	.88	.91	.93	.95	.97	.98	.52	.34	.16	.16	.11								
	12	150 & over	.75	.79	.83	.84	.86	.88	.89	.91	.91	.93	.93	.75												
		100	.68	.77	.81	.84	.86	.88	.89	.89	.92	.93	.93	.72												
		30	.34	.72	.82	.87	.89	.92	.95	.95	.97	.98	.98	.52												

†These factors apply when maintaining a CONSTANT TEMPERATURE in the space during the operating period. Where the temperature is allowed to swing, additional storage will result during peak load periods. Refer to Table 13 for applicable storage factors.

with lights operating the same number of hours as the time of equipment operation, use a load factor of 1.00.

†Lights On for Shorter or Longer Period than 10 Hours

Occasionally adjustments may be required to take account of lights operating less or more than the 10 hours on which the table is based. The following is the procedure to adjust the load factors:

A-WITH LIGHTS IN OPERATION FOR SHORTER PERIOD THAN 10 HOURS and the equipment operating 12, 16 or 24 hours at the time of the overall peak load, extrapolate load factors as follows:

1. Equipment operating for 24 hours:
   a. Use the storage load factors as listed up to the time the lights are turned off.
   b. Shift the load factors beyond the 10th hour (on the right of heavy line) to the left to the hour the lights are turned off. This leaves last few hours of equipment operation without designated load factors.
   c. Extrapolate the last few hours at thee same rate of reduction as the end hours in the table.
2. Equipment operating for 16 hours:
   a. Follow the procedure in Step 1, using the storage load factor values in 24-hour equipment operation table.
   b. Now construct a new set of load factors by adding the new values for the 16th hour to that denoted 0, 17th hour to the 1st hour, etc.
      b. The load factors for the hours succeeding the switching- off the lights are as in Steps 1b and 1c.
3. Equipment operating for 12 hours:

   Follow procedure in Step 2, except in Step 2b add values of 12th hour to that designated 0, 13th hour to the 1st hour, etc.

B-WITH LIGHTS IN OPERATION FOR LONGER PERIOD THAN 10 HOURS and the equipment operating 12, 16 or 24 hours at the time of the overall peak load, extrapolate load factors as follows:

1. Equipment operating for 24 hours:
   a. Use the load factors as listed through 10th hour and extrapolate beyond the 10th hour at the rate of the last 4 hours.
   b. Follow the same procedure as in Step 1b of "A" except shift load factors beyond 10th hour now to the right, dropping off the last few hours.
2. Equipment operating for 16 hours or 12 hours:
   a. Use the load factors in 24-hour equipment operation table as listed through 10th hour and extrapolate beyond the 10th hour at the rate of the last 4 hours.
   b. Follow the procedure in Step 1b of "A" except shift the load factors beyond 10th hour now to the right.
   c. For 16-hour equipment operation, follow the procedure in Steps 2b and 2c of "A".
   d. For 12-hor equipment operation, follow the procedure in Step 3 of "A".

Example

Adjust values for 24-hour equipment operation and derive new values for 16-hour equipment operation for fluorescent lights in operation 8 and 13 hours, and an enclosure of 150 lb/sq ft of floor.

EQUIP OPERATION Hours	WEIGHT§ (lb per sq ft of floor area)	NUMBER OF HOURS AFTER LIGHTS ARE TURNED ON																								LIGHTS ON Hours
		0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	
24	150	.37	.67	.71	.74	.76	.79	.81	.83	.84	.86	.87	.89	.90	.92	.29	.26	.23	.20	.19	.17	.15	.14	.12	.11	13
		.37	.67	.71	.74	.76	.79	.81	.83	.84	.29	.26	.23	.20	.19	.17	.15	.14	.12	.11	.10	.09	.08	.07	.06	8
		.37	.67	.71	.74	.76	.79	.81	.83	.84	.86	.87	.29	.26	.23	.20	.19	.17	.15	.14	.12	.11	.10	.09	.08	10
16	150	.60	.87	.90	.91	.91	.93	.93	.94	.94	.95	.95	.96	.96	.97	.29	.26									13
		.51	.79	.82	.84	.85	.87	.88	.89	.90	.29	.26	.23	.20	.19	.17	.15									8
		.60	.82	.83	.84	.84	.84	.85	.85	.86	.88	.90	.32	.28	.25	.23	.19									10

§ Weight per sq ft of floor-

$$\text{Room on Bldg Exterior (One or more outside walls)} = \frac{\text{(Weight of Outside Wall, lb)} + \tfrac{1}{2}\text{ (Weight of Partitions, Floor and Ceiling, lb)}}{\text{Floor Area in Room,sq ft}}$$

$$\text{Room in Bldg Interior (No outside walls)} = \frac{\tfrac{1}{2}\text{(Weight of Partitions, Floor and Ceiling, lb)}}{\text{Floor Area in Room,sq ft}}$$

$$\text{Basement Room (Floor on ground)} = \frac{\text{(Weight of Outside Wall, lb)} + \text{(Weight of Floor, lb)} + \tfrac{1}{2}\text{ (Weight of Partitions and Ceiling, lb)}}{\text{Floor Area in Room,sq ft}}$$

$$\text{Entire Building or Zone} = \frac{\text{(Weight of Outside Wall, Partitions, Floors, Ceilings, Structural Members and Supports, lb)}}{\text{Air Conditioned Floor Area, sq ft}}$$

With rug on floor-Weight of floor should be multiplied by 0.50 to compensate for insulating effect of rug.
Weights per sq ft of common types of construction are contained in *Tables 21 thru 33, pages 66 thru 76.*

SPACE TEMPERATURE SWING

In addition to the storage of radiant heat with a constant room temperature, heat is stored in the building structure when the space temperature is forced to swing.If the cooling capacity supplied to the space matches the cooling load, the temperature in thespace remains constant throughout the operating period. On the other hand, if the cooling capacity supplied to the space is lower than the actual cooling load at any point, the temperature in the space will rise. Ad the space temperature increases, less heat is convected from the surface and more radiant heat is stored in the structure. This process of storing additional heat is illustrated in *Fig. 11.*

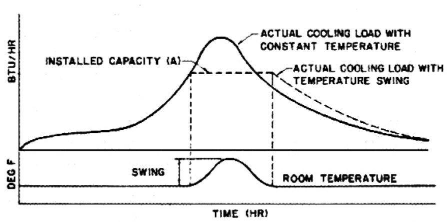


FIG. 11-ACTUAL COOLING LOAD WITH VARYING ROOM TEMPERATURE

The solid curve is the actual cooling load from the solar heat gain on a west exposure with a constant space temperature, 24-hour operation. Assume that the maximum cooling capacity available is represented by *A,* and that the capacity is controlled to maintain a constant temperature at partial load. When the actual cooling load exceeds the available cooling capacity, the temperature will swing as shown in the lower curve. The actual cooling load with temperature swing is shown by the dotted line. This operates in a similar manner with different periods of operation and with different types of construction.

NOTE: When a system is designed for a temperature swing, the maximum swing occurs only at the peak on design days, which are defined as those days when all loads simultaneously peak. Under normal operating conditions, the temperature remains constant or close to constant.

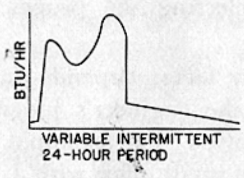
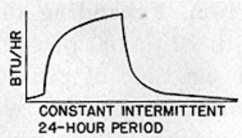
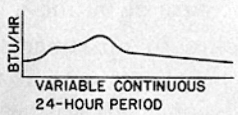
Basis of Table 13
-- Storage Factors, Space Temperature Swing

The storage factors in *Table 13* were computed using essentially the same procedure as *Tables 7 thru 12* with the exception that the equipment capacity

TABLE 13—STORAGE FACTORS, SPACE TEMPERATURE SWING

Btu/(hr) (deg F swing) (sq ft of floor area)

NOTE: This reduction is to be taken at the time of peak load only.

TYPE APPLICATION		WEIGHT (lb/sq ft floor area)	GLASS RATIO‡ (%)	HOURS OF OPERATION								
				24			16			12		
				Temperature Swing (F)								
Load Pattern	Bldg Type			1-2	3-4	5-6	1-2	3-4	5-6	1-2	3-4	5-6
	Office Bldg Periphery, Except North Side	150 and Over	75	1.90	1.80	1.65	1.80	1.70	1.55	1.60	1.50	1.40
			50	1.70	1.60	1.45	1.60	1.50	1.35	1.50	1.35	1.25
			25	1.50	1.40	—	1.40	1.30	—	1.30	1.20	—
		100	75	1.70	1.60	1.45	1.50	1.45	1.35	1.40	1.35	1.30
			50	1.50	1.40	1.30	1.35	1.30	1.20	1.30	1.25	1.10
			25	1.35	1.25	1.20	1.25	1.00	.90	1.20	.95	.70
		30	75	1.40	1.25	1.00	1.20	1.10	.95	1.00	.95	.88
			50	1.20	.95	.80	1.10	.90	.80	.90	.85	.80
			25	.90	.80	.70	.85	.75	.60	.80	.70	.55
	Interior Zones† Department Stores, Factories	150 and Over	—	1.60	1.55	1.50	1.50	1.45	—	1.35	—	—
		100	—	1.40	1.38	1.36	1.30	1.28	1.25	1.25	1.20	—
		30	—	.95	.92	.90	.90	.88	.85	.85	.80	—
	Apartment Houses, Hotels, Hospitals Residences	150 and Over	75	1.85	1.75	1.40	—	—	—	—	—	—
			50	1.65	1.50	—	—	—	—	—	—	—
			25	1.45	—	—	—	—	—	—	—	—
		100	75	1.55	1.45	1.40	—	—	—	—	—	—
			50	1.40	1.35	—	—	—	—	—	—	—
			25	1.30	—	—	—	—	—	—	—	—
		30	75	1.20	1.10	.95	—	—	—	—	—	—
			50	1.10	.90	.80	—	—	—	—	—	—
			25	.85	.70	—	—	—	—	—	—	—

Equation: Reduction in Peak Cooling Load, Btu/hr = (Floor Area, sq ft) × (Desired Temp Swing, *Table 4, page 20*) × (Storage Factor, above table)
*Weight per sq ft of floor may be obtained from equation on *page 30.*
†For 12-hour operation, use a 2 degree max temp swing.
‡Glass ratio is the percent of glass area to the total wall area.

available was limited and the swing in room temperature computed.

The magnitude of the storage effect is determined largely by the thermal capacity or heat holding capacity of the materials surrounding the space. It is limited by the amount of heat available for storage. Load patterns for different applications vary approximately as shown in the first column of *Table 13.* For instance, an office building has a rather large varying load with a high peak that occurs intermittently. An interior zone has an intermittent peak but the load pattern is relatively constant. A hospital, on the other hand, has a constant base load which is present for 24 hours with an additional intermittent load occurring during daylight hours. The thermal capacity of a material is the weight times the specific heat of the material. Since the specific heat of most construction material is approximately 0.20 Btu/(lb)(F), the thermal capacity is directly proportional to the weight of the material. Therefore, the data in the tables is based on weight of the materials surrounding the space, per square foot of floor area.

Use of Table 13 -- Storage Factors, Space Temperature Swing

Table 13 is used to determine the reduction in cooling load when the space temperature is forced to swing by reducing the equipment capacity below that required to maintain the temperature constant. This reduction is to be subtracted from the room sensible heat.

NOTE: This reduction is only taken at the time of *peak cooling load.*

Example 3 – Space Temperature Swing

Given:

The same room as in *Example 1, page 28.*

Find:

The actual cooling load at 4 p.m. from sun, lights, and people with 3 F temperature swing in the space.

Solution:

The peak sensible cooling load in this room from the sun, lights, and people (neglecting transmission infiltration, ventilation and other internal heat gain) is

5700+5190 = 10,890 Btu/hr. *(Examples 1 and 2.)*

NOTE: The peak cooling load in this room occurs at approximately 4 p.m. The solar and light loads are almost at their peak at 4 p.m. Although the transmission across the large glass window peaks at about 3 p.m., the peak infiltration and ventilation load also occurs at 3 p.m. and the relatively small transmission load across the wall peaks much later at about 12 midnight. The sum of these loads results in the peak cooling load occurring at about 4 p.m. in the spaces with this exposure.

The weight of the materials surrounding the room in *Example 1* is 97.4 lb/sq ft of floor area.

Reduction in cooling load for a 3 F swing *(Table 13)*

$= 20 \times 20 \times 1.4 \times 3 = 1680$ Btu/hr

Cooling load = 10,890 - 1680 = 9210 Btu/hr.

(For comparison purposes, the instantaneous heat gain from sun, lights, and people in this particular room is 14,610 Btu/hr.)

Since the normal thermostat setting is about 75 F or 76 F db, the design temperature (78 F = 75 F thermostat setting +3 F swinng) occurs only on design peak days at the time of peak load. Under partial load operation, the room temperature is between 75 F db and 78 F db, or at the thermostat setting (75 F), depending on the load.

PRECOOLING AS A MEANS OF INCREASING STORAGE

Precooling a space below the temperature normally desired *increases the storage of heat* at the time of peak load, only when the precooling temperature is maintained as the control point. This is because the potential temperature swing is increased, thus adding to the amount of heat stored at the time of peak load. Where the space is precooled to a lower temperature and the control point is reset upward to a comfortable condition when the occupants arrive, no additional storage occurs. In this situation, the cooling unit shuts off and there is no cooling during the period of warming up. When the cooling unit begins to supply cooling again, the cooling load is approximately up to the point it would have been without any precooling.

Precooling is very useful in reducing the cooling load in applications such as churches, supermarkets, theater, etc., where the precooled temperature can be maintained as the control point and the temperature swing increased to 8 F or 10 F.

DIVERSITY OF COOLING LOADS

Diversity of cooling load results from the probable non-occurrence of part of the cooling load on a design day. Diversity factors are applied to the refrigeration capacity in large air conditioning systems. These factors vary with location, type and size of the application, and are based entirely on the judgment of the engineer.

Generally, diversity factors can be applied to people and light loads in large multi-story office, hotel or apartment buildings. The possibility of having all of the people present in the building and all of the lights operating at the time of peak load are slight. Normally, in large office buildings, some people will be away from the office on other business. Also, the lighting arrangement will frequently be such that the lights in the vacant offices will not be on. In addition to lights being off because the people are not present, the normal maintenance procedure in large office buildings usually results in some lights being inoperative. Therefore, a diversity factor on the people and light loads should be applied for selecting the proper size refrigeration equipment.

The size of the diversity factor depends on the size of the building and the engineer's judgment of the circumstances involved. For example, the diversity factor on a single small office with 1 or 2 people is 1.0 or no reduction. Expanding this to one floor of a building with 50 to 100 people, 5% to 10% may be absent at the time of peak load, and expanding to a 20, 30 or 40-story building, 10% to 20% may be absent during the peak. A building with predominantly sales offices would have many people out in the normal course of business.

This same concept applies to apartments and hotels. Normally, very few people are present at the time the solar and transmission loads are peaking, and the lights are normally turned on only after sundown. Therefore, in apartments and hotels, the diversity factor can be much greater than with office buildings.

These reductions in cooling load are real and should be made where applicable. *Table 14* lists some typical diversity factors, based on judgment and experience.

TABLE 14-TYPICAL DIVERSITY FACTORS
FOR LARGE BUILDINGS
(Apply to Refrigeration Capacity)

TYPE OF APPLICATION	DIVERSITY FACTOR	
	People	Lights
Office	.75 to .90	.70 to .85
Apartment, Hotel	.40 to .60	.30 to .50
Department Store	.80 to .90	.90 to 1.0
Industrial*	.85 to .95	.80 to .90

Equation:

Cooling Load (for people and lights), Btu/hr
= (Heat Gain, Btu/hr, *Chapter 7)*
× (Storage Factor, *Table 12)* ×(Diversity Factor, above table)

*A diversity factor should also be applied to the machinery load. Refer to *Chapter 7.*

Use of Table 14
-- Typical Diversity Factors for Large Buildings

The diversity factors listed in *Table 14* are to be used as a guide in determining a diversity factor for any particular application. The final factor must necessarily be based on judgment of the effect of the many variables involved.

STRATIFICATION OF HEAT

There are generally two situations where heat is stratified and will reduce the cooling load on the air conditioning equipment:

1. Heat may be stratified in rooms with high ceilings where air is exhausted through the roof or ceiling.
2. Heat may be contained above suspended ceilings with recessed lighting and/or ceiling plenum return systems

The first situation generally applies to industrial applications, churches, auditoriums, and the like. The second situation applies to applications such as office buildings, hotels, and apartments. With both cases, the basic fact that hot air tends to rise makes it possible to stratify load such as convection from the roof, convection from lights, and convection from the upper part of the walls. The convective portion of the roof load is about 25% (the rest is radiation); the light load is about 50% with fluorescent (20^ with incandescent), and the wall transmission load about 40%.

In any room with a high ceiling, a large part of the convection load being released above the supply air stream will stratify at the ceiling or roof level. Some will be induced into the supply air stream. Normally, about 80% is stratified and 20% induced in the supply air. If air is exhausted through the ceiling or roof, this convection load released abovethe supply air may be subtracted from the air conditioning load. This results in a large reduction in load if the air is to be exhausted. It is not normally practical to exhaust more air than necessary, as it must be made up by bringing outdoor air through the apparatus. This usually results in a larger increase in load than the reduction realized by exhausting air.

Nominally, about a 10 F to 20 F rise in exhaust air temperature may be figured as load reduction if there is enough heat released by convection above the supply air stream.

Hot air stratifies at the ceiling event with no exhaust but rapidly builds up in temperature, and no reduction in load should be taken where air is not exhausted through the ceiling or roof.

With suspended ceilings, some of the convective heat from recessed lights flows into the plenum space. Also, the radiant heat within the room (sun, lights, people, etc.) striking the ceiling warms it up and causes heat to flow into the plenum space. These sources of heat increase the temperature of air in the plenum space which causes heat to flow into the underside of the floor structure above. When the ceiling plenum is used as a return air system, some of the return air flows through and over the light fixture, carrying more of the convective heat into the plenum space.

Containing heat within the ceiling plenum space tends to “flatten’ both the room and equipment load. The storage factors for estimating the load with the above conditions are contained in *Table 12.*

CHAPTER 4. SOLAR HEAT GAIN THRU GLASS

SOLAR HEAT – DIRECT AND DIFFUSE

The solar heat on the outer edge of the earth's atmosphere is about 445 Btu/(hr)(sq ft) on December 21 when the sun is closest to the earth, and about 415 Btu/(hr)(sq ft) on June 21 when it is farthest away. The amount of solar heat outside the earth's atmosphere varies between these limits throughout the year.

The solar heat reaching the earth's surface is reduced considerably below these figures because a large part of it is scattered, reflected back out into space, and absorbed by the atmosphere. The scattered radiation is termed *Diffuse or sky radiation,* and is more or less evenly distributed over the earth's surface because it is nothing more than a reflection from dust particles, water vapor and ozone in the atmosphere. The solar heat that comes directly through the atmosphere is termed *direct radiation.* The relationship between the total and the direct and diffuse radiation at any point on earth is dependent on the following two factors:

1. The distance traveled through the atmosphere to reach the point on the earth.
2. The amount of haze in the air.

As the distance traveled or the amount of haze increases, the diffuse radiation component increases but the direct component decreases. As either or both of these factors increase, the overall effect is to reduce the total quantity of heat reaching the earth's surface.

ORDINARY GLASS

Ordinary glass is specified as crystal glass of single thickness and single or double strength. The solar heat gain through ordinary glass depends on its location on the earth's surface (latitude), time of day, time of year, and facing direction of the window.The direct radiation component results in a heat gain to the conditioned space only when the window is in the direct rays of the sun, whereas the diffuse radiation component results in a heat gain, even when the window is not facing the sun.

Ordinary glass absorbs a small portion of the solar heat (5% to 6%) and reflects or transmits the rest. The amount reflected or transmitted depends on the angle of incidence. (The angle of incidence is the angle between the perpendicular to the window surface and the sun's rays, *Fig. 18, page 55.)* At low angles of incidence, about 89% or 87% is transmitted and 8% or 9% is reflected, as shown in *Fig. 12*. As the angle of incidence increases, more solar heat is reflected and less is transmitted, as shown in *Fig. 13*. The total solar heat gain to the conditioned space consists of the transmitted heat plus about 40% of the heat that is absorbed in the glass.

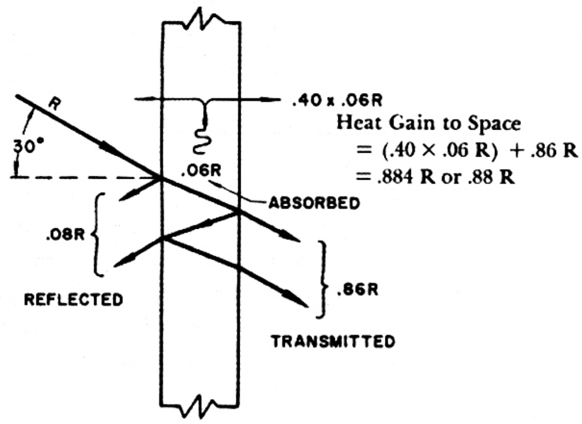


FIG. 12-REACTION ON SOLAR HEAT (R), ORDINARY GLASS, 30 ANGLE OF INCIDENCE

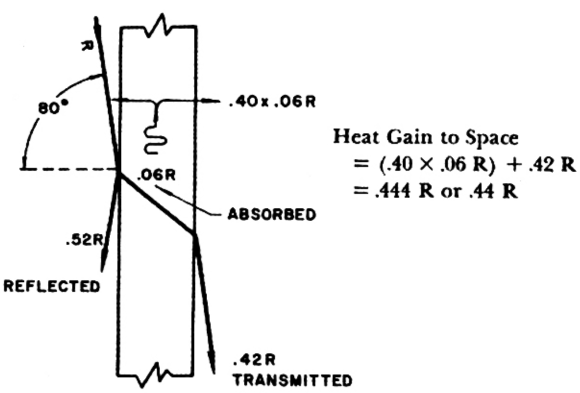


FIG. 13-REACTION ON SOLAR HEAT (R), ORDINARY GLASS, 80 ANGLE OF INCIDENCE

NOTE: The 40% of the absorbed solar heat going into the space is derived from the following reasoning:

1. The outdoor film coefficient is approximately 2.8 Btu/ (hr) (sq ft) (deg F) with a 5 mph wind velocity during the summer.

2. The inside film coefficient is approximately 1.8 Btu/ (hr) (sq ft) (deg F) because, in the average system design, air velocities across the glass are approximately 100-200 fpm.
3. If outdoor temperature is equal to room temperature, the glass
temperature is above both. Therefore absorbed heat

$$\text{flowing in} = \frac{1.8\text{x}100}{1.8+2.8} = 39.2\%\text{,or } 40\%$$

$$\text{Absorbed heat flowing out} = \frac{2.8\text{x}100}{1.8+2.8} = 60.8\%\text{, or } 60\%$$

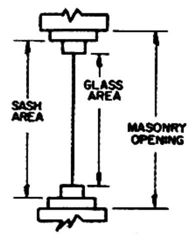
4. As the outdoor temperature rises, the glass temperature also irises, causing more of the absorbed heat to flow into the space. This can be accounted for by adding the transmission of heat across the glass (caused by temperature difference between inside and outdoors) to the constant 40% of the absorbed heat going inside.
5. This reasoning applies equally well when the outdoor temperature is below the room temperature.

Basis of Table 15
- Solar heat Gain thru Ordinary Glass

Table 15 provides data for 0°, 10°, 20°, 30°, 40°, and 50° latitudes, for each month of the year and for each hour of the day. This table includes the direct and diffuse radiation and that portion of the heat absorbed in the glass which gets into the space. It *does not* include the transmission of heat across the glass caused by a temperature difference between the outdoor and inside air. (See *Chapter 5* for "U" values.)

The data in *Table 15* is based on the following conditions:-

1. A glass area equal to 85% of the sash area.This is typical for wood sash windows. For metal sash windows, the glass area is assumed equal to 100% of the sash area because the conductivity of the metal sash is very high and the solar heat absorbed in the sash is transmitted almost instantaneously.



NOTE: The sash area equals approximately 85% of the masonry opening (or frame opening with frame walls) with wood sash windows, 90% of masonry opening with double hung metal sash windows, and 100% of masonry opening with casement windows.

FIG. 14
WINDOW AREAS

2. No haze in the air.
3. Sea level elevation.
4. A sea level dewpoint temperature of 66.8 F (95 F db, 75 F wb) which approximately corresponds to 4 centimeters of precipitable water vapor. Precipitable water vapor is all of the water vapor in a column of air from sea level to the outer edge of the atmosphere.

If these conditions do not apply, use the correction factors at the bottom of each page of *Table 15.*

Use of Table 15
- Solar Heat Gain thru Ordinary Glass

The bold face values in Table 15 indicate the maximum solar heat gain for the month for each exposure. The bold face values that are boxed indicate the yearly maximums for each exposure.

Table 15 is used to determine the solar heat gain thru ordinary glass at any time, in any space, zone or building.

To determine the actual cooling load *due to the solar heat gain,* refer to *Chapter 3, "Heat Storage, Diversity and Stratification."*

CAUTION – Where Estimation Multi-Exposure Rooms Or Buildings

If a haze factor is used on one exposure to determine the peak room or building load, the diffuse component listed for the other exposures must be divided by the haze factor to result in the actual room or building peak load. This is because the diffuse component increases with increasing haze, as explained on *page 41.*

Example 1 – Peak Solar Heat Gain (2 Exposures)

Since the time at which the peak solar load occurs in a space with 2 exposures is not always apparent, the solar heat gain is generally calculated at more than one time to determine its peak.

Given:

A room with equal glass areas on the West and South at 40° North latitude.

Find:

Peak solar heat gain.

Solution:

From *Table 15*

Solar heat gain-			
September 22	2:00	3:00	4:00 p.m.
West	99	1:39	149
South	110	81	44
Total	209	220	193
Solar heat gain-			
October 23	2:00	3:00	4:00 p.m.
West	88	122	117
South	137	104	59
Total	225	226	176
Solar heat gain-			
November 21	2:00	3:00	4:00 p.m.
West	74	100	91
South	139	104	59
Total	213	204	150

The peak solar heat gain to this room occurs at 3:00 p.m. on October 23. The peak room cooling load does not necessarily occur at the same time as the peak solar heat gain. because the peak transmission load, people land, etc., may occur at some other time.

Example 2 – Solar Gain Correction Factors

(Bottom Table 15)

The conditions on which *Table 15* is based do not apply to all locations, since many cities are above sea level, and many have different design dew points and some haze in their atmosphere.

Given:

A west exposure with steel casement windows
Location – Topeka, Kansas
Altitude – 991 ft
Design dewpoint – 69.8 F
39° North latitude

Find:

Peak solar heat gain

Solution:

By inspection of *Table 15* The boxed boldface values for peak solar heat gain, occurring at 4:00 p.m. on July 23

= 164 Btu/(hr) (sq ft)

Assume a somewhat hazy condition.

Altitude correction = 1.007 (bottom *Table 15)*

Dewpoint difference = 69.8-66.8 = 3 F

Dewpoint correction = 1 – (3/10×.07) = .979 (bottom *Table 15)*

Haze correction = 1 - .10 = .90 (bottom *Table 15)*

Steel sash correction = 1/.85 (bottom *Table 15)*

Solar heat gain at 4:00 p.m., July 23

= 164×1.007×.979×.90×1/.85

= 171 Btu/ (hr)(sq ft)

TABLE 15—SOLAR HEAT GAIN THRU ORDINARY GLASS

0° | Btu/(hr) (sq ft sash area) | 0°

0° NORTH LATITUDE		AM				SUN TIME						PM			0° SOUTH LATITUDE	
Time of Year	Exposure	6	7	8	9	10	11	Noon	1	2	3	4	5	6	Exposure	Time of Year
JUNE 21	North	0	45	65	74	78	80	82	80	78	74	65	45	0	South	DEC 22
	Northeast	0	119	156	154	133	95	53	20	14	13	11	6	0	Southeast	
	East	0	116	147	135	93	43	14	14	14	13	11	6	0	East	
	Southeast	0	37	42	27	15	14	14	14	14	13	11	6	0	Northeast	
	South	0	6	11	13	14	14	14	14	14	13	11	6	0	North	
	Southwest	0	6	11	13	14	14	14	14	15	27	42	37	0	Northwest	
	West	0	6	11	13	14	14	14	43	93	135	147	116	0	West	
	Northwest	0	6	11	13	14	20	53	95	133	154	156	119	0	Southwest	
	Horizontal	0	28	87	147	191	217	226	217	191	147	87	28	0	Horizontal	
JULY 23 & MAY 21	North	0	37	54	61	65	66	67	66	65	61	54	37	0	South	JAN 21 & NOV 21
	Northeast	0	118	153	150	124	86	43	16	14	13	11	6	0	Southeast	
	East	0	121	152	139	96	43	14	14	14	13	11	6	0	East	
	Southeast	0	46	52	36	18	14	14	14	14	13	11	6	0	Northeast	
	South	0	6	11	13	14	14	14	14	14	13	11	6	0	North	
	Southwest	0	6	11	13	14	14	14	14	18	36	52	46	0	Northwest	
	West	0	6	11	13	14	14	14	43	96	139	152	121	0	West	
	Northwest	0	6	11	13	14	16	43	86	124	150	153	118	0	Southwest	
	Horizontal	0	29	91	151	195	223	233	223	195	151	91	29	0	Horizontal	
AUG 24 & APR 20	North	0	17	28	31	33	34	34	34	33	31	28	17	0	South	FEB 20 & OCT 23
	Northeast	0	110	141	133	102	61	24	14	14	13	12	6	0	Southeast	
	East	0	129	163	148	103	46	14	14	14	13	12	6	0	East	
	Southeast	0	67	79	65	35	15	14	14	14	13	12	6	0	Northeast	
	South	0	6	12	13	14	14	14	14	14	13	12	6	0	North	
	Southwest	0	6	12	13	14	14	14	15	35	65	79	67	0	Northwest	
	West	0	6	12	13	14	14	14	46	103	148	163	129	0	West	
	Northwest	0	6	12	13	14	14	24	61	102	133	141	110	0	Southwest	
	Horizontal	0	31	97	150	206	234	245	234	206	150	97	31	0	Horizontal	
SEPT 22 & MAR 22	North	0	6	12	13	14	14	14	14	14	13	12	6	0	South	MAR 22 & SEPT 22
	Northeast	0	95	118	101	68	31	14	14	14	13	12	6	0	Southeast	
	East	0	134	167	151	107	47	14	14	14	13	12	6	0	East	
	Southeast	0	95	118	101	68	31	14	14	14	13	12	6	0	Northeast	
	South	0	6	12	13	14	14	14	14	14	13	12	6	0	North	
	Southwest	0	6	12	13	14	14	14	31	68	101	118	95	0	Northwest	
	West	0	6	12	13	14	14	14	47	107	151	167	134	0	West	
	Northwest	0	6	12	13	14	14	14	31	68	101	118	95	0	Southwest	
	Horizontal	0	32	100	163	210	240	250	240	210	163	100	32	0	Horizontal	
OCT 23 & FEB 20	North	0	6	12	13	14	14	14	14	14	13	12	6	0	South	APR 20 & AUG 24
	Northeast	0	67	79	65	35	15	14	14	14	13	12	6	0	Southeast	
	East	0	129	163	148	103	46	14	14	14	13	12	6	0	East	
	Southeast	0	110	141	133	102	61	24	14	14	13	12	6	0	Northeast	
	South	0	17	28	31	33	34	34	34	33	31	28	17	0	North	
	Southwest	0	6	12	13	14	14	24	61	102	133	141	110	0	Northwest	
	West	0	6	12	13	14	14	14	46	103	148	163	129	0	West	
	Northwest	0	6	12	13	14	14	14	15	35	65	79	67	0	Southwest	
	Horizontal	0	31	97	150	206	234	245	234	206	150	97	31	0	Horizontal	
NOV 21 & JAN 21	North	0	6	11	13	14	14	14	14	14	13	11	6	0	South	MAY 21 & JULY 23
	Northeast	0	46	52	36	18	14	14	14	14	13	11	6	0	Southeast	
	East	0	121	152	139	96	43	14	14	14	13	11	6	0	East	
	Southeast	0	118	153	150	124	86	43	16	14	13	11	6	0	Northeast	
	South	0	37	54	61	65	66	67	66	65	61	54	37	0	North	
	Southwest	0	6	11	13	14	16	43	86	124	150	153	118	0	Northwest	
	West	0	6	11	13	14	14	14	43	96	139	152	121	0	West	
	Northwest	0	6	11	13	14	14	14	14	18	36	52	46	0	Southwest	
	Horizontal	0	29	91	151	195	223	233	223	195	151	91	29	0	Horizontal	
DEC 22	North	0	6	11	13	14	14	14	14	14	13	11	6	0	South	JUNE 21
	Northeast	0	37	42	27	15	14	14	14	14	13	11	6	0	Southeast	
	East	0	116	147	135	93	43	14	14	14	13	11	6	0	East	
	Southeast	0	119	156	154	133	95	53	20	14	13	11	6	0	Northeast	
	South	0	45	65	74	78	80	82	80	78	74	65	45	0	North	
	Southwest	0	6	11	13	14	20	53	95	133	154	156	119	0	Northwest	
	West	0	6	11	13	14	14	14	43	93	135	147	116	0	West	
	Northwest	0	6	11	13	14	14	14	14	15	27	42	37	0	Southwest	
	Horizontal	0	28	87	147	191	217	226	217	191	147	87	28	0	Horizontal	

Solar Gain Correction	Steel Sash, or No Sash	Haze	Altitude	Dewpoint Decrease From 67 F	Dewpoint Increase From 67 F	South Lat. Dec. or Jan.
	× 1/.85 or 1.17	−15% (Max.)	+0.7% per 1000 Ft	+ 7% per 10 F	− 7% per 10 F	+ 7%

Bold Face Values — Monthly Maximums   Boxed Values — Yearly maximums

TABLE 15—SOLAR HEAT GAIN THRU ORDINARY GLASS (Contd)

10° Btu/(hr) (sq ft sash area) **10°**

10° NORTH LATITUDE		AM						SUN TIME				PM			10° SOUTH LATITUDE	
Time of Year	Exposure	6	7	8	9	10	11	Noon	1	2	3	4	5	6	Exposure	Time of Year
JUNE 21	North	19	44	50	45	44	43	41	43	44	45	50	44	2	South	DEC 22
	Northeast	55	131	153	140	106	65	28	14	14	13	11	8	2	Southeast	
	East	54	134	155	139	98	41	14	14	14	13	11	8	2	East	
	Southeast	18	49	55	43	25	14	14	14	14	13	11	8	2	Northeast	
	South	2	8	11	13	14	14	14	14	14	13	11	8	2	North	
	Southwest	2	8	8	13	14	14	14	14	25	43	55	49	18	Northwest	
	West	2	8	8	13	14	14	14	41	98	139	155	134	54	West	
	Northwest	2	8	8	13	14	18	28	65	106	140	153	131	55	Southwest	
	Horizontal	4	44	107	166	205	233	243	233	205	166	107	44	4	Horizontal	
JULY 23 & MAY 21	North	5	34	39	35	33	31	30	31	33	35	39	34	5	South	JAN 21 & NOV 21
	Northeast	42	127	148	133	109	56	22	14	14	13	11	7	1	Southeast	
	East	50	135	158	142	98	43	14	14	14	13	11	7	1	East	
	Southeast	26	57	66	56	32	14	14	14	14	13	11	7	1	Northeast	
	South	1	7	11	13	14	14	14	14	14	13	11	7	1	North	
	Southwest	1	7	11	13	14	14	14	14	32	56	66	57	26	Northwest	
	West	1	7	11	13	14	14	14	43	98	142	158	135	50	West	
	Northwest	1	7	11	13	14	14	22	56	109	133	148	127	42	Southwest	
	Horizontal	3	42	107	166	210	236	247	236	210	166	107	42	3	Horizontal	
AUG 24 & APR 20	North	1	15	16	15	15	14	14	14	15	15	16	15	1	South	FEB 20 & OCT 23
	Northeast	17	113	130	111	80	34	14	14	14	13	11	7	1	Southeast	
	East	25	138	163	149	104	46	14	14	14	13	11	7	1	East	
	Southeast	18	79	94	85	60	27	14	14	14	13	11	7	1	Northeast	
	South	1	7	11	13	14	14	14	14	14	13	11	7	1	North	
	Southwest	1	7	11	13	14	14	14	27	60	85	94	79	18	Northwest	
	West	1	7	11	13	14	14	14	46	80	149	163	138	25	West	
	Northwest	1	7	11	13	14	14	14	34	15	111	130	113	17	Southwest	
	Horizontal	2	38	105	167	213	242	250	242	213	167	105	38	2	Horizontal	
SEPT 22 & MAR 22	North	1	6	11	13	14	14	14	14	14	13	11	6	1	South	MAR 22 & SEPT 22
	Northeast	1	89	103	80	45	17	14	14	14	13	11	6	1	Southeast	
	East	1	130	164	151	106	47	14	14	14	13	11	6	1	East	
	Southeast	1	97	127	122	94	56	21	14	14	13	11	6	1	Northeast	
	South	1	6	13	19	24	27	28	27	24	19	13	6	1	North	
	Southwest	1	6	11	13	14	14	21	56	94	122	127	97	1	Northwest	
	West	1	6	11	13	14	14	14	47	106	151	164	130	1	West	
	Northwest	1	6	11	13	14	14	14	17	45	80	103	89	1	Southwest	
	Horizontal	1	31	97	160	207	235	247	235	207	160	97	31	1	Horizontal	
OCT 23 & FEB 20	North	0	5	10	13	14	14	14	14	14	13	10	5	0	South	APR 20 & AUG 24
	Northeast	0	58	66	44	28	14	14	14	14	13	10	5	0	Southeast	
	East	0	118	155	145	100	40	14	14	14	13	10	5	0	East	
	Southeast	0	103	147	149	123	81	46	18	14	13	10	5	0	Northeast	
	South	0	18	40	55	65	71	73	71	65	55	40	18	0	North	
	Southwest	0	5	10	13	14	18	46	81	123	149	147	103	0	Northwest	
	West	0	5	10	13	14	14	14	40	100	145	155	118	0	West	
	Northwest	0	5	10	13	14	14	14	14	28	44	66	58	0	Southwest	
	Horizontal	0	22	85	139	193	220	230	220	193	139	85	22	0	Horizontal	
NOV 21 & JAN 21	North	0	4	9	12	13	14	14	14	13	12	9	4	0	South	MAY 21 & JULY 23
	Northeast	0	27	37	17	13	14	14	14	13	12	9	4	0	Southeast	
	East	0	99	143	132	93	39	14	14	13	12	9	4	0	East	
	Southeast	0	99	153	161	146	109	70	31	17	12	9	4	0	Northeast	
	South	0	35	65	91	96	104	106	104	96	91	65	35	0	North	
	Southwest	0	4	9	12	17	31	70	109	146	161	153	99	0	Northwest	
	West	0	4	9	12	13	14	14	39	93	132	143	99	0	West	
	Northwest	0	4	9	12	13	14	14	14	13	17	37	27	0	Southwest	
	Horizontal	0	17	62	131	175	202	210	202	175	131	62	17	0	Horizontal	
DEC 22	North	0	4	9	12	13	14	14	14	13	12	9	4	0	South	JUNE 21
	Northeast	0	15	28	17	13	14	14	14	13	12	9	4	0	Southeast	
	East	0	86	137	130	91	42	14	14	13	12	9	4	0	East	
	Southeast	0	99	154	163	149	121	79	36	23	12	9	4	0	Northeast	
	South	0	50	74	94	109	116	120	116	109	94	74	50	0	North	
	Southwest	0	4	9	12	23	36	79	121	149	163	154	99	0	Northwest	
	West	0	4	9	12	13	14	14	42	91	130	137	86	0	West	
	Northwest	0	4	9	12	13	14	14	14	13	17	28	15	0	Southwest	
	Horizontal	0	14	66	120	167	193	202	193	167	120	66	14	0	Horizontal	

Solar Gain Correction	Steel Sash, or No Sash × 1/.85 or 1.17	Haze −15% (Max.)	Altitude +0.7% per 1000 Ft	Dewpoint Decrease From 67 F + 7% per 10 F	Dewpoint Increase From 67 F − 7% per 10 F	South Lat. Dec. or Jan. + 7%

Bold Face Values — Monthly Maximums Boxed Values — Yearly maximums

TABLE 15—SOLAR HEAT GAIN THRU ORDINARY GLASS (Contd)

20⁰ Btu/(hr) (sq ft sash area) **20⁰**

20° NORTH LATITUDE Time of Year	Exposure	AM 6	7	8	9	10	11	SUN TIME Noon	1	2	3	4	PM 5	6	20° SOUTH LATITUDE Exposure	Time of Year
JUNE 21	North	28	**41**	33	25	19	17	15	17	19	25	33	**41**	28	South	DEC 22
	Northeast	81	**154**	144	122	83	38	15	14	14	14	12	9	3	Southeast	
	East	81	148	**160**	143	96	41	14	14	14	14	12	9	3	East	
	Southeast	28	62	**73**	66	44	21	14	14	14	14	12	9	3	Northeast	
	South	3	9	12	14	14	14	**14**	14	14	14	12	9	3	North	
	Southwest	3	9	12	14	14	14	14	21	44	66	**73**	62	28	Northwest	
	West	3	9	12	14	14	14	14	41	96	143	**160**	148	81	West	
	Northwest	3	9	12	14	14	14	15	38	83	122	144	**154**	81	Southwest	
	Horizontal	11	60	121	176	216	232	**250**	232	216	176	121	60	11	Horizontal	
JULY 23 & MAY 21	North	20	**28**	23	17	15	14	14	14	15	17	23	**28**	20	South	JAN 21 & NOV 21
	Northeast	71	132	**138**	111	73	31	14	14	14	13	12	8	3	Southeast	
	East	75	148	**163**	145	99	46	14	14	14	13	12	8	3	East	
	Southeast	31	70	**85**	79	57	29	14	14	14	13	12	8	3	Northeast	
	South	3	8	12	13	14	14	**14**	14	14	13	12	8	3	North	
	Southwest	3	8	12	13	14	14	14	29	57	79	**85**	70	31	Northwest	
	West	3	8	12	13	14	14	14	46	99	145	**163**	148	75	West	
	Northwest	3	8	12	13	14	14	14	31	73	111	**138**	132	71	Southwest	
	Horizontal	8	55	118	175	216	240	**251**	240	216	175	118	55	8	Horizontal	
AUG 24 & APR 20	North	6	10	11	13	14	14	**14**	14	14	13	11	10	6	South	FEB 20 & OCT 23
	Northeast	45	111	**118**	89	50	18	14	14	14	13	11	7	2	Southeast	
	East	53	142	**165**	149	106	51	14	14	14	13	11	7	2	East	
	Southeast	29	89	**113**	108	98	55	20	14	14	13	11	7	2	Northeast	
	South	2	7	11	14	20	24	**26**	24	20	14	11	7	2	North	
	Southwest	2	7	11	13	14	14	20	55	98	108	**113**	89	29	Northwest	
	West	2	7	11	13	14	14	14	51	106	149	**165**	142	53	West	
	Northwest	2	7	11	13	14	14	14	18	50	89	**118**	111	45	Southwest	
	Horizontal	5	48	107	167	210	235	**247**	235	210	167	107	48	5	Horizontal	
SEPT 22 & MAR 22	North	0	6	11	13	14	14	**14**	14	14	13	11	6	0	South	MAR 22 & SEPT 22
	Northeast	0	83	**87**	59	22	14	14	14	14	13	11	6	0	Southeast	
	East	0	130	**163**	149	104	45	14	14	14	13	11	6	0	East	
	Southeast	0	99	136	**140**	120	84	41	15	14	13	11	6	0	Northeast	
	South	0	8	22	38	52	63	**65**	63	52	38	22	8	0	North	
	Southwest	0	6	11	13	14	15	41	84	120	**140**	136	99	0	Northwest	
	West	0	6	11	13	14	14	14	45	104	149	**163**	130	0	West	
	Northwest	0	6	11	13	14	14	14	14	22	59	**87**	83	0	Southwest	
	Horizontal	0	30	93	153	198	225	**233**	225	198	153	93	30	0	Horizontal	
OCT 23 & FEB 20	North	0	4	9	12	13	14	**14**	14	13	12	9	4	0	South	APR 20 & AUG 24
	Northeast	0	44	**52**	29	13	14	14	14	13	12	9	4	0	Southeast	
	East	0	99	**147**	141	100	49	14	14	13	12	9	4	0	East	
	Southeast	0	91	146	**160**	149	119	74	27	13	12	9	4	0	Northeast	
	South	0	21	50	76	93	106	**111**	106	93	76	50	21	0	North	
	Southwest	0	4	9	12	13	27	74	119	149	**160**	146	91	0	Northwest	
	West	0	4	9	12	13	14	14	49	100	141	**147**	99	0	West	
	Northwest	0	4	9	12	13	14	14	14	13	29	**52**	44	0	Southwest	
	Horizontal	0	18	68	127	171	196	**208**	196	171	127	68	18	0	Horizontal	
NOV 21 & JAN 21	North	0	3	8	11	13	13	**13**	13	13	11	8	3	0	South	MAY 21 & JULY 23
	Northeast	0	24	**26**	14	13	13	13	13	13	11	8	3	0	Southeast	
	East	0	71	**128**	127	91	43	13	13	13	11	8	3	0	East	
	Southeast	0	73	144	**164**	158	135	91	46	16	11	8	3	0	Northeast	
	South	0	28	69	100	123	136	**141**	136	123	100	69	28	0	North	
	Southwest	0	3	8	11	16	46	91	135	158	**164**	144	73	0	Northwest	
	West	0	3	8	11	12	13	13	43	91	127	**128**	71	0	West	
	Northwest	0	3	8	11	12	13	13	13	13	14	**26**	24	0	Southwest	
	Horizontal	0	5	48	101	146	172	**180**	172	146	101	48	5	0	Horizontal	
DEC 22	North	0	2	7	11	12	13	**13**	13	12	11	7	2	0	South	JUNE 21
	Northeast	0	14	**18**	12	12	13	13	13	12	11	7	2	0	Southeast	
	East	0	56	118	**121**	85	34	13	13	12	11	7	2	0	East	
	Southeast	0	59	139	**167**	159	134	97	60	20	11	7	2	0	Northeast	
	South	0	25	74	**111**	132	146	**149**	146	132	**111**	74	25	0	North	
	Southwest	0	2	7	11	20	60	97	134	159	**167**	139	59	0	Northwest	
	West	0	2	7	11	12	13	13	34	85	**121**	118	56	0	West	
	Northwest	0	2	7	11	12	13	13	13	12	12	**18**	14	0	Southwest	
	Horizontal	0	4	36	92	135	161	**170**	161	135	92	36	4	0	Horizontal	

Solar Gain Correction	Steel Sash, or No Sash	Haze	Altitude	Dewpoint Decrease From 67 F	Dewpoint Increase From 67 F	South Lat. Dec. or Jan.
	× 1/.85 or 1.17	−15% (Max.)	+0.7% per 1000 Ft	+ 7% per 10 F	− 7% per 10 F	+ 7%

Bold Face Values — Monthly Maximums Boxed Values — Yearly maximums

TABLE 15—SOLAR HEAT GAIN THRU ORDINARY GLASS (Contd)

30° Btu/(hr) (sq ft sash area) **30°**

30° NORTH LATITUDE		AM					SUN TIME						PM		30° SOUTH LATITUDE	
Time of Year	Exposure	6	7	8	9	10	11	Noon	1	2	3	4	5	6	Exposure	Time of Year
JUNE 21	North	**33**	29	18	14	14	14	14	14	14	14	18	29	**33**	South	DEC 22
	Northeast	105	**139**	130	97	55	19	14	14	14	14	12	10	5	Southeast	
	East	108	156	**161**	143	98	44	14	14	14	14	12	10	5	East	
	Southeast	42	75	**90**	**90**	73	44	17	14	14	14	12	10	5	Northeast	
	South	5	10	12	14	15	19	**21**	19	15	14	12	10	5	North	
	Southwest	5	10	12	14	14	14	17	44	73	**90**	**90**	75	42	Northwest	
	West	5	10	12	14	14	14	14	44	98	143	**161**	156	108	West	
	Northwest	5	10	12	14	14	14	14	19	55	97	130	**139**	105	Southwest	
	Horizontal	19	61	131	180	217	240	**250**	240	217	180	131	61	19	Horizontal	
JULY 23 & MAY 21	North	**22**	20	14	13	14	14	14	14	14	13	14	20	**22**	South	JAN 21 & NOV 21
	Northeast	93	**131**	123	89	46	16	14	14	14	13	12	9	4	Southeast	
	East	100	155	**164**	145	99	44	14	14	14	13	12	9	4	East	
	Southeast	42	82	**100**	**100**	83	53	22	14	14	13	12	9	4	Northeast	
	South	4	9	12	14	20	27	**30**	27	20	14	12	9	4	North	
	Southwest	4	9	12	13	14	14	14	53	83	**100**	**100**	82	42	Northwest	
	West	4	9	12	13	14	14	14	44	99	145	**164**	155	100	West	
	Northwest	4	9	12	13	14	14	14	16	46	89	123	**131**	93	Southwest	
	Horizontal	15	66	123	176	214	236	**246**	236	214	176	123	66	15	Horizontal	
AUG 24 & APR 20	North	6	8	11	13	13	14	**14**	14	13	13	11	8	6	South	FEB 20 & OCT 23
	Northeast	55	**108**	100	66	27	14	14	14	13	13	11	8	2	Southeast	
	East	66	147	**165**	148	102	46	14	14	13	13	11	8	2	East	
	Southeast	37	98	127	**129**	112	82	39	15	13	13	11	8	2	Northeast	
	South	2	8	13	27	47	58	**63**	58	47	27	13	8	2	North	
	Southwest	2	8	11	13	13	15	39	82	112	**129**	127	98	37	Northwest	
	West	2	8	11	13	13	14	14	46	102	148	**165**	147	66	West	
	Northwest	2	8	11	13	13	14	14	14	27	66	100	**108**	55	Southwest	
	Horizontal	6	47	107	161	200	225	**235**	225	200	161	107	47	6	Horizontal	
SEPT 22 & MAR 22	North	0	5	10	12	13	14	**14**	14	13	12	10	5	0	South	MAR 22 & SEPT 22
	Northeast	0	74	**90**	40	15	14	14	14	13	12	10	5	0	Southeast	
	East	0	124	**158**	144	103	48	14	14	13	12	10	5	0	East	
	Southeast	0	98	131	**152**	141	113	67	25	13	12	10	5	0	Northeast	
	South	0	9	18	60	82	98	**105**	98	82	60	18	9	0	North	
	Southwest	0	5	10	12	13	25	67	113	141	**152**	131	98	0	Northwest	
	West	0	5	10	12	13	14	14	48	103	144	**158**	124	0	West	
	Northwest	0	5	10	12	13	14	14	14	15	40	**90**	74	0	Southwest	
	Horizontal	0	25	81	135	179	202	**212**	202	179	135	81	25	0	Horizontal	
OCT 23 & FEB 20	North	0	3	8	11	12	13	**14**	13	12	11	8	3	0	South	APR 20 & AUG 24
	Northeast	0	33	**39**	18	12	13	14	13	12	11	8	3	0	Southeast	
	East	0	79	**135**	132	94	43	14	13	12	11	8	3	0	East	
	Southeast	0	73	142	**163**	159	136	92	47	15	11	8	3	0	Northeast	
	South	0	18	57	92	121	139	**145**	139	121	92	57	18	0	North	
	Southwest	0	3	8	11	15	47	92	136	159	**163**	142	73	0	Northwest	
	West	0	3	8	11	12	13	14	43	94	132	**135**	79	0	West	
	Northwest	0	3	8	11	12	13	14	13	12	18	**39**	33	0	Southwest	
	Horizontal	0	6	49	100	143	171	**179**	171	143	100	49	6	0	Horizontal	
NOV 21 & JAN 21	North	0	1	6	9	11	12	**12**	12	11	9	6	1	0	South	MAY 21 & JULY 23
	Northeast	0	8	**16**	9	11	12	12	12	11	9	6	1	0	Southeast	
	East	0	27	109	**116**	83	35	12	12	11	9	6	1	0	East	
	Southeast	0	28	127	161	**162**	143	104	64	23	9	6	1	0	Northeast	
	South	0	10	68	109	137	154	**159**	154	137	109	68	10	0	North	
	Southwest	0	1	6	9	23	64	104	143	**162**	161	127	28	0	Northwest	
	West	0	1	6	9	11	12	12	35	83	**116**	109	27	0	West	
	Northwest	0	1	6	9	11	12	12	12	11	9	**16**	8	0	Southwest	
	Horizontal	0	2	27	71	109	136	**145**	136	109	71	27	2	0	Horizontal	
DEC 22	North	0	0	4	9	11	12	**12**	12	11	9	4	0	0	South	JUNE 21
	Northeast	0	0	10	9	11	12	**12**	12	11	9	4	0	0	Southeast	
	East	0	0	92	**105**	80	32	12	12	11	9	4	0	0	East	
	Southeast	0	0	114	157	**162**	143	108	72	28	9	4	0	0	Northeast	
	South	0	0	64	113	142	159	**163**	159	142	113	64	0	0	North	
	Southwest	0	0	4	9	28	72	108	143	**162**	157	114	0	0	Northwest	
	West	0	0	4	9	11	12	12	32	80	**105**	92	0	0	West	
	Northwest	0	0	4	9	11	12	**12**	12	11	9	10	0	0	Southwest	
	Horizontal	0	0	19	60	97	122	**131**	122	97	60	19	0	0	Horizontal	

Solar Gain Correction	Steel Sash, or No Sash	Haze	Altitude	Dewpoint Decrease From 67 F	Dewpoint Increase From 67 F	South Lat. Dec. or Jan.
	× 1/.85 or 1.17	−15% (Max.)	+0.7% per 1000 Ft	+ 7% per 10 F	− 7% per 10 F	+ 7%

Bold Face Values — Monthly Maximums   Boxed Values — Yearly maximums

TABLE 15—SOLAR HEAT GAIN THRU ORDINARY GLASS (Contd)

40° Btu/(hr) (sq ft sash area) **40°**

40° NORTH LATITUDE		SUN TIME AM												PM	40° SOUTH LATITUDE	
Time of Year	Exposure	6	7	8	9	10	11	Noon	1	2	3	4	5	6	Exposure	Time of Year
JUNE 21	North	32	20	12	13	14	14	14	14	14	13	12	20	**32**	South	DEC 22
	Northeast	118	133	112	73	30	14	14	14	14	13	12	10	6	Southeast	
	East	126	161	**162**	142	95	44	14	14	14	13	12	10	6	East	
	Southeast	51	88	109	**111**	99	71	34	14	14	13	12	10	6	Northeast	
	South	6	10	12	19	35	44	**54**	44	35	19	12	10	6	North	
	Southwest	6	10	12	13	14	14	34	71	99	**111**	109	88	51	Northwest	
	West	6	10	12	13	14	14	14	44	95	142	**162**	161	126	West	
	Northwest	6	10	12	13	14	14	14	14	30	73	112	133	118	Southwest	
	Horizontal	31	82	134	179	210	232	237	232	210	179	134	82	31	Horizontal	
JULY 23 & MAY 21	North	**24**	14	12	13	14	14	14	14	14	13	12	14	**24**	South	JAN 21 & NOV 21
	Northeast	106	**127**	105	66	26	14	14	14	14	13	12	10	5	Southeast	
	East	118	161	164	144	98	43	14	14	14	13	12	10	5	East	
	Southeast	54	96	119	**125**	110	82	42	15	14	13	12	10	5	Northeast	
	South	5	10	13	26	44	63	**69**	63	44	26	13	10	5	North	
	Southwest	5	10	12	13	14	15	42	62	110	**125**	119	96	54	Northwest	
	West	5	10	12	13	14	14	14	43	98	144	164	161	118	West	
	Northwest	5	10	12	13	14	14	14	14	26	66	105	**127**	106	Southwest	
	Horizontal	24	73	126	171	203	225	**233**	225	203	171	126	73	24	Horizontal	
AUG 24 & APR 20	North	7	8	11	13	**14**	14	**14**	14	14	13	11	8	7	South	FEB 20 & OCT 23
	Northeast	68	**102**	82	46	16	14	14	14	14	13	11	8	3	Southeast	
	East	84	147	**162**	145	101	45	14	14	14	13	11	8	3	East	
	Southeast	48	105	138	**146**	139	107	66	25	14	13	11	8	3	Northeast	
	South	3	8	24	51	89	97	**102**	97	89	51	24	8	3	North	
	Southwest	3	8	11	13	14	25	66	107	139	**146**	138	105	48	Northwest	
	West	3	8	11	13	14	14	14	45	101	145	**162**	147	84	West	
	Northwest	3	8	11	13	14	14	14	14	16	46	82	**102**	68	Southwest	
	Horizontal	9	47	100	150	185	205	**214**	205	185	150	100	47	9	Horizontal	
SEPT 22 & MAR 22	North	0	5	9	12	13	13	**14**	13	13	12	9	5	0	South	MAR 22 & SEPT 22
	Northeast	0	51	**58**	26	13	13	14	13	13	12	9	5	0	Southeast	
	East	0	116	**149**	139	99	45	14	13	13	12	9	5	0	East	
	Southeast	0	95	144	162	157	133	90	41	14	12	9	5	0	Northeast	
	South	0	12	44	81	110	122	**140**	122	110	81	44	12	0	North	
	Southwest	0	5	9	12	14	41	90	133	157	**162**	144	95	0	Northwest	
	West	0	5	9	12	13	13	14	45	99	139	**149**	116	0	West	
	Northwest	0	5	9	12	13	13	14	13	13	26	**58**	51	0	Southwest	
	Horizontal	0	21	67	124	153	176	**183**	176	153	124	67	21	0	Horizontal	
OCT 23 & FEB 20	North	0	2	6	10	11	12	**12**	12	11	10	6	2	0	South	APR 20 & AUG 24
	Northeast	0	**35**	33	12	11	12	12	12	11	10	6	2	0	Southeast	
	East	0	85	117	**122**	88	39	12	12	11	10	6	2	0	East	
	Southeast	0	81	132	161	**163**	144	107	63	20	10	6	2	0	Northeast	
	South	0	21	59	104	137	154	**162**	154	137	104	59	21	0	North	
	Southwest	0	2	6	10	20	63	107	144	163	161	132	81	0	Northwest	
	West	0	2	6	10	11	12	12	39	88	**122**	117	85	0	West	
	Northwest	0	2	6	10	11	12	12	12	11	12	33	**35**	0	Southwest	
	Horizontal	0	8	29	64	101	123	**129**	123	101	64	29	8	0	Horizontal	
NOV 21 & JAN 21	North	0	0	3	7	9	10	**11**	10	9	7	3	0	0	South	MAY 21 & JULY 23
	Northeast	0	0	**12**	7	9	10	11	10	9	7	3	0	0	Southeast	
	East	0	0	91	**100**	74	33	11	10	9	7	3	0	0	East	
	Southeast	0	0	109	144	**156**	144	116	70	27	7	3	0	0	Northeast	
	South	0	0	59	104	139	158	166	158	139	104	59	0	0	North	
	Southwest	0	0	3	7	27	70	116	144	**156**	144	109	0	0	Northwest	
	West	0	0	3	7	9	10	11	33	74	**100**	91	0	0	West	
	Northwest	0	0	3	7	9	10	11	10	9	7	**12**	0	0	Southwest	
	Horizontal	0	0	16	43	73	92	**103**	92	73	43	16	0	0	Horizontal	
DEC 22	North	0	0	2	6	9	10	**10**	10	9	6	2	0	0	South	JUNE 21
	Northeast	0	0	**7**	6	9	10	**10**	10	9	6	2	0	0	Southeast	
	East	0	0	72	**86**	68	31	10	10	9	6	2	0	0	East	
	Southeast	0	0	88	134	**148**	142	115	73	30	7	2	0	0	Northeast	
	South	0	0	51	99	134	158	**165**	158	134	99	51	0	0	North	
	Southwest	0	0	2	7	30	73	115	142	**148**	134	88	0	0	Northwest	
	West	0	0	2	6	9	10	10	31	68	**86**	72	0	0	West	
	Northwest	0	0	2	6	9	10	**10**	10	9	6	7	0	0	Southwest	
	Horizontal	0	0	8	32	55	76	**85**	76	55	32	8	0	0	Horizontal	

Solar Gain Correction	Steel Sash, or No Sash	Haze	Altitude	Dewpoint Decrease From 67 F	Dewpoint Increase From 67 F	South Lat. Dec. or Jan.
	× 1/.85 or 1.17	−15% (Max.)	+0.7% per 1000 Ft	+ 7% per 10 F	− 7% per 10 F	+ 7%

Bold Face Values — Monthly Maximums Boxed Values — Yearly maximums

TABLE 15—SOLAR HEAT GAIN THRU ORDINARY GLASS (Contd)

50° Btu/(hr) (sq ft sash area) **50°**

50° NORTH LATITUDE		AM						SUN TIME					PM		50° SOUTH LATITUDE	
Time of Year	Exposure	6	7	8	9	10	11	Noon	1	2	3	4	5	6	Exposure	Time of Year
JUNE 21	North	**29**	12	12	13	14	14	14	14	14	13	12	12	**29**	South	DEC 22
	Northeast	**126**	125	94	50	16	14	14	14	14	13	12	10	8	Southeast	
	East	139	**164**	162	136	94	41	14	14	14	13	12	10	8	East	
	Southeast	64	102	126	**135**	124	98	61	23	14	13	12	10	8	Northeast	
	South	8	10	16	39	68	87	**93**	87	68	39	16	10	8	North	
	Southwest	8	10	12	13	14	23	61	98	124	**135**	126	102	64	Northwest	
	West	8	10	12	13	14	14	14	41	94	136	162	**164**	139	West	
	Northwest	8	10	12	13	14	14	14	14	16	50	94	125	**126**	Southwest	
	Horizontal	44	86	133	173	197	214	**220**	214	197	173	133	86	44	Horizontal	
JULY 23 & MAY 21	North	**21**	11	12	13	14	14	14	14	14	13	12	11	**21**	South	JAN 21 & NOV 21
	Northeast	114	**117**	87	44	15	14	14	14	14	13	12	10	6	Southeast	
	East	131	161	**163**	141	96	43	14	14	14	13	12	10	6	East	
	Southeast	65	107	134	**143**	136	109	70	26	14	13	12	10	6	Northeast	
	South	6	10	21	50	80	98	**106**	98	80	50	21	10	6	North	
	Southwest	6	10	12	13	14	26	70	109	136	**143**	134	107	65	Northwest	
	West	6	10	12	13	14	14	14	43	96	141	**163**	161	131	West	
	Northwest	6	10	12	13	14	14	14	14	15	44	87	**117**	114	Southwest	
	Horizontal	33	75	119	159	188	205	**211**	205	188	159	119	75	33	Horizontal	
AUG 24 & APR 20	North	8	8	10	12	13	14	**14**	14	13	12	10	8	8	South	FEB 20 & OCT 23
	Northeast	76	**94**	70	31	13	14	14	14	13	12	10	8	4	Southeast	
	East	94	145	**158**	141	98	45	14	14	13	12	10	8	4	East	
	Southeast	53	111	144	**157**	153	132	89	40	13	12	10	8	4	Northeast	
	South	4	9	36	73	105	130	**138**	130	105	73	36	9	4	North	
	Southwest	4	8	10	12	13	40	89	132	153	**157**	144	111	53	Northwest	
	West	4	8	10	12	13	14	14	45	98	141	**158**	145	94	West	
	Northwest	4	8	10	12	13	14	14	14	13	31	70	**94**	76	Southwest	
	Horizontal	13	46	89	131	160	179	**185**	179	160	131	89	46	13	Horizontal	
SEPT 22 & MAR 22	North	0	4	8	10	12	12	**12**	12	12	10	8	4	0	South	MAR 22 & SEPT 22
	Northeast	0	**58**	46	16	12	12	12	12	12	10	8	4	0	Southeast	
	East	0	102	**138**	130	93	43	12	12	12	10	8	4	0	East	
	Southeast	0	86	139	162	**163**	145	105	56	17	10	8	4	0	Northeast	
	South	0	11	51	93	131	150	**158**	150	131	93	51	11	0	North	
	Southwest	0	4	8	10	17	56	105	145	**163**	162	139	86	0	Northwest	
	West	0	4	8	10	12	12	12	43	93	130	**138**	102	0	West	
	Northwest	0	4	8	10	12	12	12	12	12	16	46	**58**	0	Southwest	
	Horizontal	0	15	49	88	118	140	**148**	140	118	88	49	15	0	Horizontal	
OCT 23 & FEB 20	North	0	0	4	7	9	10	**11**	10	9	7	4	0	0	South	APR 20 & AUG 24
	Northeast	0	**29**	20	7	9	10	11	10	9	7	4	0	0	Southeast	
	East	0	73	99	**105**	79	35	11	10	9	7	4	0	0	East	
	Southeast	0	69	111	145	**157**	144	115	69	24	7	4	0	0	Northeast	
	South	0	17	53	99	137	157	**167**	157	137	99	53	17	0	North	
	Southwest	0	0	4	7	24	69	115	144	**157**	145	111	69	0	Northwest	
	West	0	0	4	7	9	10	11	35	79	**105**	99	73	0	West	
	Northwest	0	0	4	7	9	10	11	10	9	7	20	**29**	0	Southwest	
	Horizontal	0	2	19	45	72	86	**94**	86	72	45	19	2	0	Horizontal	
NOV 21 & JAN 21	North	0	0	1	4	6	8	**9**	8	6	4	1	0	0	South	MAY 21 & JULY 23
	Northeast	0	0	5	4	6	8	9	8	6	4	1	0	0	Southeast	
	East	0	0	51	**64**	57	28	9	8	6	4	1	0	0	East	
	Southeast	0	0	62	95	**127**	**127**	107	67	21	4	1	0	0	Northeast	
	South	0	0	34	70	116	143	**153**	143	116	70	34	0	0	North	
	Southwest	0	0	1	4	21	67	107	**127**	**127**	95	62	0	0	Northwest	
	West	0	0	1	4	6	8	9	28	57	**64**	51	0	0	West	
	Northwest	0	0	1	4	6	8	9	8	6	4	5	0	0	Southwest	
	Horizontal	0	0	4	13	30	47	**53**	47	30	13	4	0	0	Horizontal	
DEC 22	North	0	0	0	3	5	6	**7**	6	5	3	0	0	0	South	JUNE 21
	Northeast	0	0	0	3	5	6	**7**	6	5	3	0	0	0	Southeast	
	East	0	0	0	27	**47**	23	7	6	5	3	0	0	0	East	
	Southeast	0	0	0	41	107	**116**	100	62	25	3	0	0	0	Northeast	
	South	0	0	0	31	99	131	**141**	131	99	31	0	0	0	North	
	Southwest	0	0	0	3	25	62	100	**116**	107	41	0	0	0	Northwest	
	West	0	0	0	3	5	6	7	23	**47**	27	0	0	0	West	
	Northwest	0	0	0	3	5	6	**7**	6	5	3	0	0	0	Southwest	
	Horizontal	0	0	0	5	19	33	**40**	33	19	5	0	0	0	Horizontal	

Solar Gain Correction	Steel Sash, or No Sash	Haze	Altitude	Dewpoint Decrease From 67 F	Dewpoint Increase From 67 F	South Lat. Dec. or Jan.
	× 1/.85 or 1.17	−15% (Max.)	+0.7% per 1000 Ft	+ 7% per 10 F	− 7% per 10 F	+ 7%

Bold Face Values — Monthly Maximums   Boxed Values — Yearly maximums

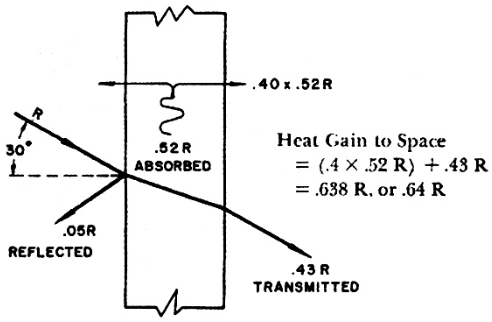
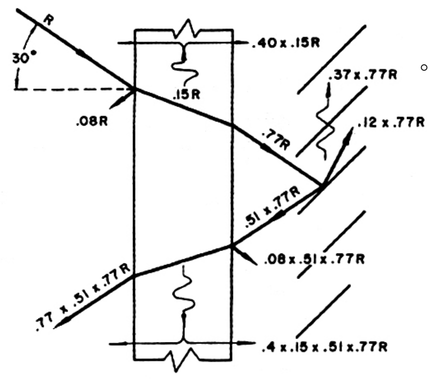


FIG. 15-REACTION ON SOLAR HEAT (R), 52% HEAT ABSORBING GLASS, 30 ANGLE OF INCIDENCE



Heat Gain to Space

$= (.40 \times .15\ R) + (.37 \times .77\ R) + (.12 \times .77\ R) + (.08 \times .51 \times .77\ R) + (.40 \times .15 \times .51 \times .77\ R)$

$= .492\ R$ or $.49\ R$

FIG. 16-REACTION ON SOLAR HEAT (R), ¼ -INCH PLATE GLASS, WHITE VENETIAN BLIND, 30 ANGLE OF INCIDENCE

ALL GLASS TYPES – WITH AND WITHOUT SHADING DEVICES

Glass, *other than ordinary glass,* absorbs more solar heat because it

1. May be thicker, or
2. May be specially treated to absorb solar heat (heat absorbing glass).

These special glass types reduce the transmitted solar heat but increase the amount of absorbed solar heat flowing into the space. Normally they reflect slightly less than ordinary glass because part of the reflection takes place on the inside surface. A portion of heat reflected from the inside surface is absorbed in passing back through the glass. The overall effect, however, is to reduce the solar heat gain to the conditioned space as shown in *Fig. 15.* (Refer to *Item 8, page 51,* for absorptivity, reflectivity and transmissibility of common types of glass at 30° angle of incidence.)

The solar heat gain factor through 52% heat absorbing glass as compared to ordinary glass is .64F/.88R = .728 or .73. This multiplier (.73) is used with *Table 15* to determine the solar heat gain thru 52% heat absorbing glass. Multipliers for various types of glass are listed in *Table 16.*

The effectiveness of a *shading device* depends on its ability to keep solar heat from the conditioned space. All shading devices reflect and absorb a major portion of the solar gain, leaving a small portion to be transmitted. The outdoor shading devices are much more effective than the inside devices because all of the reflected solar heat is kept out and the absorbed heat is dissipated to the outdoor air. Inside devices necessarily dissipate their absorbed heat within the conditioned space and must also reflect the solar heat back through the glass *(Fig. 16)* wherein some of it is absorbed. (Refer to *Item 8, page 51,* for absorptivity, reflectivity and transmissibility of common shading devices at 30° angle of incidence.)

The solar heat gain thru glass with an inside shading device may be expressed as follows:

$$Q = [.4a_g + t_g(a_{sd} + t_{sd} + r_g r_{sd} + .4a_g r_{sd})]\ \frac{R}{.88}$$

Where:

Q = solar heat gain to space, Btu/ (hr)(sq ft)
R = total solar intensity, Btu/(hr)(sq ft), (From *Table 15)*
a = solar absorptivity
t = solar transmissibility
r = solar reflectivity
g = glass
sd = shading device
.88 = conversion factor from *Fig. 12*

For drapes the above formula changes as follows, caused by the hot air space between glass and drapes:

$$Q = [.24a_g + t_g(.85a_{sd} + t_{sd} + r_g r_{sd} + .24a_g r_{sd})]\ \frac{R}{.88}$$

The transmission factor U for glass with 100% drape is 0.80 Btu/(hr) (sq ft) (F).

The solar heat gain factor thru the combination in *Fig. 16* as compared to ordinary glass is .49R/.88R = .557 or .56 (Refer to *Table 16* for ¼ -inch regular plate glass with a white venetian blind.)

NOTE: Actually the reaction on the solar heat reflected back through the glass from the blind is not always identical tot he first pass as assumed in this example. The first pass through the glass filters out most of solar radiation that is to be absorbed in the glass, and the second pass absorbs somewhat less. For simplicity, the reaction is assumed identical, since the quantities are normally small on the second pass.

Basis of Table 16

Over-all Factors for Solar Heat Gain thru Glass, With and Without Shading Devices

The factors in *Table 16* are based on:

1. An outdoor film coefficient of 2.8 Btu/(hr) (sq ft) (deg F) at 5 mph wind velocity.
2. An inside film coefficient of 1.8 Btu/(hr) (sq ft) (deg F), 100-200 fpm. This is not 1.47 as normally used, since the present practice in well designed systems is to sweep the window with a stream of air.
3. A 30° angle of incidence which is the angle at which most exposures peak. The 30° angle of incidence is approximately the balance point on reduction of solar heat coming through the atmosphere and the decreased transmissibility of glass. Above the 30° angle the transmissibility of glass decreases, and below the 30° angle the atmosphere absorbs or reflects more.
4. All shading devices fully drawn, except roller shades. Experience indicates that roller shads are *seldom fully drawn,* so the factors have been slightly increased.
5. Venetian blind slats horizontal at 45° and shading screen slats horizontal at 17°.
6. Outdoor canvas awnings ventilated at sides and top. (See *Table 16* footnote).
7. Since *Table 15* is based on the net solar heat gain thru ordinary glass, all calculated solar heat factors are divided by .88 *(Fig. 12)*.
8. The average absorptivity, reflectivity and transmissability for common glass and shading devices at a 30° angle of incidence along with shading factors appear in the table below.

Use of Table 16

- Over-all Factors for Solar Heat Gain thru Glass, With and Without Shading Devices

The factors in *Table 16* are multiplied by the values in *Table 15* to determine the solar heat gain thru different combinations of glass and shading devices. The correction factors listed under *Table 15* are to be used if applicable. Transmission due to temperature difference between the inside and outdoor air must be added to the solar heat gain to determine total gain thru glass.

Example 3 – Partially Drawn Shades

Occasionally it is necessary to estimate the cooling load in a building where the blinds are not to be fully drawn. The procedure is illustrated in the following example:

Given:

West exposure, 40° North latitude

Thermopane window with white venetian blind on inside, 3/4 drawn.

Find:

Peak solar heat gain.

Solution:

By inspection of *Table 15,* the boxed boldface values for peak solar heat gain, occurring at 4:00 p.m. on July 23

= 164 Btu/(hr) (sq ft)

TYPES OF GLASS OR SHADING DEVICES*	Absorptivity (a)	Reflectivity (r)	Transmissibility (t)	Solar Factor†
Ordinary Glass	.06	.08	.86	1.00
Regular Plate, 1/4 "	.15	.08	.77	.94
Glass, Heat Absorbing	by mfg.	.05	(1 - .05 – a)	-- --
Venetian Blind, Light Color	.37	.51	.12	.56 ‡
Medium Color	.58	.39	.03	.65 ‡
Dark Color	.72	.27	.01	.75 ‡
Fiberglass Cloth, Off White (5.72-61/58)	.05	.60	.35	.48 ‡
Cotton Cloth, Beige (6.18-91/36)	.26	.51	.23	.56 ‡
Fiberglass Cloth, Light Gray	.30	.47	.23	.59 ‡
Fiberglass Cloth, Tan (7.55-57/29)	.44	.42	.14	.64 ‡
Glass Cloth, White, Golden Stripes	.05	.41	.54	.65 ‡
Fiberglass Cloth, Dark Gray	.60	.29	.11	.75 ‡
Dacron Cloth, White (1.8-86/81)	.02	.28	.70	.76 ‡
Cotton Cloth, Dark Green, Vinyl Coated (similar to roller shade)	.82	.15	.00	.88 ‡
Cotton Cloth, Dark Green (6.06-91/36)	.02	.28	.70	.76 ‡

*Factors for various draperies are given for guidance only since the actual drapery material may be different in color and texture; figures in parentheses are ounces per sq yd, and yarn count warp/filling. Consult manufacturers for actual values.

†Compared to ordinary glass.

‡For a shading device in combination with ordinary glass.

Thermopane windows have no sash; therefore, sash area correction = 1/.85 (bottom *Table 15).*

In this example, ¾ of the window is covered with the venetian blind and ¼ is not; therefore, the solar heat gain factor equals ¾ of the overall factor + ¼ of the glass factor.

Factor for ¾ drawn = $(3/4 \times .52)+(1/4 \times .80)$ (*Table 16*)

$= .59$

Solar heat gain $= 164 \times \frac{.59}{.85}$

$= 114$ Btu/ (hr) (sq ft).

Example 4-Peak Solar Heat Gain thru Solex "R" Glass

Given:

West exposure, 40° North latitude

¼" Solex "R" glass in steel sash, double hung window

Find:

Peak solar heat gain.

Solution:

By inspection of Table 15 the boxed boldface value for peak solar heat gain, occurring at 4:00 p.m. on July 23

= 164 Btu/(hr) (sq ft).

Steel sash window correction = 1/.85 (bottom *Table 15*).

Solex "R" glass absorbs 50.9% of the solar heat (footnotes to *Table 16*) which places this glass in the 48% to 56% absorbing range.

From *Table 16*, the factor = .73.

Solar heat gain $= \frac{164 \text{x}.73}{.85}$ = 141 Btu/(hr) (sq ft)

TABLE 16-OVER-ALL FACTORS FOR SOLAR HEAT GAIN THRU GLASS

WITH AND WITHOUT SHADING DEVICES*

Apply Factors to Table 15

Outdoor wind velocity, 5 mph-Angle of incidence, 30 – Shading devices fully covering window

	GLASS FACTOR NO SHADE	INSIDE VENETIAL BLIND* 45° horiz. or vertical or ROLLER SHADE			OUTSIDE VENETIAN BLIND 45° horiz. slats		OUTSIDE SHADING SCREEN† 17° horiz. slats		OUTSIDE AWNING‡ vent. sides & top	
		Light Color	Medium Color	Dark Color	Light Color	Light on Outside Dark on Inside	Medium** Color	Dark§ Color	Light Color	Med. or Dark Color
ORDINARY GLASS	1.00	**.56**	.65	.75	.15	.13	.22	.15	.20	.25
REGULAR PLATE (1/4 inch)	.94	**.56**	.65	.74	.14	.12	.21	.14	.19	.24
HEAT ABSORBING GLASS††										
40 to 48% Absorbing	.80	**.56**	.62	.72	.12	.11	.18	.12	.16	.20
48 to 56% Absorbing	.73	**.53**	.59	.62	.11	.10	.16	.11	.15	.18
56 to 70% Absorbing	.62	**.51**	.54	.56	.10	.10	.14	.10	.12	.16
DOUBLE PANE										
Ordinary Glass	.90	**.54**	.61	.67	.14	.12	.20	.14	.18	.22
Regular Plate	.80	**.52**	.59	.65	.12	.11	.18	.12	.16	.20
48 to 56% Absorbing outside; Ordinary Glass inside.	.52	**.36**	.39	.43	.10	.10	.11	.10	.10	.13
48 to 56% Absorbing outside; Regular Plate inside.	.50	**.36**	.39	.43	.10	.10	.11	.10	.10	.12
TRIPLE PANE										
Ordinary Glass	.83	**.48**	.56	.64	.12	.11	.18	.12	.16	.20
Regular Plate	.69	**.47**	.52	.57	.10	.10	.15	.10	.14	.17
PAINTED GLASS										
Light Color	.28									
Medium Color	.39									
Dark Color	.50									
STAINED GLASS‡‡										
Amber Color	.70									
Dark Red	.56									
Dark Blue	.60									
Dark Green	.32									
Greyed Green	.46									
Light Opalescent	.43									
Dark Opalescent	.37									

Footnotes for *Table 16* appear on next page.

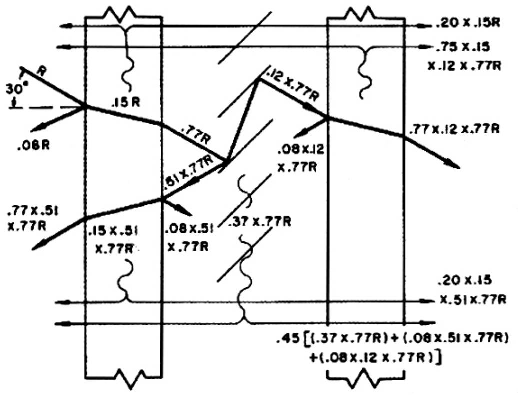


FIG. 17-REACTION ON SOLAR HEAT (R), ¼ -INCH PLATE GLASS, WHITE VENETIAN BLING, ¼ -INCH PLATE GLASS, 30 ANGLE OF INCIDENCE

APPROXIMATION OF FACTORS FOR COMBINATIONS NOT FOUND IN TABLE 16

Occasionally combinations of shading devices and types of glass may be encountered that are not covered in *Table 16.* These factors can be approximated (1) by using the solar heat gain flow diagrams in *Fig. 15 and 16,* (2) by applying the absorptivity, reflectivity and transmissibility of glass and shades listed in the table on *page 51,* or determined from manufacturer, and (3) by distributing heat absorbed within the dead air space and glass panes *(Fig. 17).*

Example 5-Approximation of Over-all Factor

Given:

A combination as in *Fig. 16* backed on the inside with another pane of $\frac{1}{4}$ -inch regular plate glass.

Find:

The over-all factor.

Solution:

Figure 17 shows the distribution of solar heat. The heat absorbed between the glass panes (dead air space) is divided 45% and 55% respectively between the in and out flow. The heat absorbed within the glass panes is divided 20% in and 80% out for the outer pane, and 75% in and 25% out for the inner pane. These divisions are based on reasoning partially stated in the notes under *Fig. 13,* which assume the outdoor film coefficient of 2.8 Btu/ (hr)(sq ft) (deg F), the indoor film coefficient of 1.8 Btu/(hr)(sq ft) (deg F), and the over-all thermal conductance of the air space of 1.37 Btu/ (hr)(sq ft)(deg F)

Heat gain to space *(Fig. 17)*

= (.75 × .15 × .12 × .77R) + (.77 × .12 × .77R)
+ .45 [(.37 × .77R) + (.08 × .51 × .77R)
+ (.08 × .12 × .77R)]
+ .20 [(.15R) + (.15 × .51 × .77R)]
= .2684R or .27R

Solar heat gain factor as compared to ordinary glass
= .27R/ .88F = .31

Equations: Solar Gain *Without Shades* = (Solar Data from Table 15) × (Glass Factor from table)

Solar Gain *With Shades* = (Solar Data from Table 15) × (Over-all Factor from table)

Solar Gain *With Shades Partially Drawn* = (Solar Data from Table 15) ×
[(Fraction Drawn × Over-all Factor) + (1 – Fraction Drawn) × (Glass Factor)]

Footnotes for Table 16:

*Shading devices fully drawn except roller shades. For fully drawn roller shades, multiply light colors by .73, medium colors by .95, and dark colors by 1.08.

†Factors for solar altitude angles of 40° or greater. At solar altitudes below 40°, some direct solar rays pass thru the slats. Use following multipliers:-

MULTIPLIERS FOR SOLAR ALTITUDES BELOW 40°

Approximate Sun Time, July 23			Solar Altitude Angle (deg)	Multiplier	
30° Lat.	40° Lat.	50° Lat.		Med. Color	Dark Color
6:00 a.m.	5:45 a.m.	5:30 a.m.	10	2.09	3.46
6:00 p.m.	6:15 p.m.	6:30 p.m.			
6:45 a.m.	6:40 a.m.	6:30 a.m.	20	1.59	2.66
5:15 p.m.	5:20 p.m.	5:30 p.m.			
7:30 a.m.	7:30 a.m.	7:30 a.m.	30	1.09	1.67
4:30 p.m.	4:30 p.m.	4:30 p.m.			

‡With outside canvas awnings tight against building on sides and top, multiply over-all factor by 1.4.

Commercial shade bronze. Metal slats 0.05 inches 0.05 inches wide, 17 per inch.

**Commercial shade, aluminum. Metal slats 0.057 inches wide, 17.5 per inch.

††Most heat absorbing glass used in comfort air conditioning is in the 40% to 56% range; industrial applications normally use 56% to 70%. The following table presents the absorption qualities of the most common glass types:-

SOLAR RADIATION ABSORBED BY HEAT ABSORBING GLASS

Trade Name or Descrip-tion	Manufacturer	Thick-ness (in.)	Color	Solar Radiation Absorbed (%)
Aklo	Blue Ridge Glass Corp.	1/8	Pale Blue-Green	56.6
Aklo	Blue Ridge Glass Corp.	1/4	Pale Blue-Green	69.7
Coolite	Mississippi Glass Co.	1/8	Light Blue	58.4
Coolite	Mississippi Glass Co.	1/4	Light Blue	70.4
L.O.F.	Libbey-Owens-Ford	1/4	Pale Blue-Green	48.2
Solex R	Pittsburgh Plate Glass Co.	1/4	Pale Green	50.9

‡‡ With multicolor windows, use the predominant color.

GLASS BLOCK

Glass block differs from sheet glass in that there is an appreciable absorption of solar heat and a fairly long time lag before the heat reaches the inside (about 3 hours). This is primarily caused by the thermal storage capacity of the glass block itself. The high absorption of heat increases the inside surface temperature of the sunlit glass block which may require lower room temperatures to maintain comfort conditions as explained in *Chapter 2.*

Shading devices on the outdoor side of glass block are almost as effective as with any other kind of glass since they keep the heat away from the glass. Shading devices on the inside are not effective in reducing the heat gain because most of the heat reflected is absorbed in the glass block.

Basis of Table 17
- Solar Heat Gain Factors for Glass block, With and Without Shading Devices

The factors in *Table 17* are the average of tests conducted by the ASHAE on several types of glass block.

Since glass block windows have no sash, the factors in *Table 17* have been increased to include the 1/.85 multiplier in *Table 15.*

Use of Table 17
- Solar Heat Gain Factors for Glass Block, With and Without Shading Devices

The factors in *Table 17* are used to determine the solar heat gain thru all types of glass block.

The transmission of heat caused by a difference between the inside and outdoor temperatures must also be figured, using the appropriate "U" value, *Chapter 5.*

Example 6-Peak Solar Heat Gain, Glass Block

Given:

West exposure, 40° North latitude

Glass block window

Find:

Peak solar heat gain

Solution:

By inspection of *Table 15,* the peak solar heat gain occurs on July 23.

Solar heat gain

At 4:00 p.m. = $(.39 \times 164) + (.21 \times 43) = 73$

At 5:00 p.m. = $(.39 \times 161) + (.21 \times 98) = 84$

At 6:00 p.m. = $(.39 \times 118) + (.21 \times 144) = 76$

Peak solar heat gain occurs at 5:00 p.m. on July 23.

TABLE 17-SOLAR HEAT GAIN FACTORS FOR GLASS BLOCK
WITH AND WITHOUT SHADING DEVICES*

Apply Factors to Table 15

EXPOSURE IN NORTH LATITUDES	MULTIPLYING FACTORS FOR GLASS BLOCK			EXPOSURE IN SOUTH LATITUDES
	Instantaneous Transmission Factor (B_i)	Absorption Transmission Factor (B_a)	Absorption Transmission Time Lag Hours	
Northeast	.27	.24	3.0	Southeast
East	.39	.21	3.0	East
Southeast	.35	.22	3.0	Northeast
South				North
Summer†	.27	.24	3.0	Summer†
Winter†	.39	.22	3.0	Winter†
Southwest	.35	.22	3.0	Northwest
West	.39	.21	3.0	West
Northwest	.27	.24	3.0	Southwest

*Factors include correction for no sash with glass block windows.

†Use the summer factors for all latitudes, North or South. Use the winter factor for intermediate seasons, 30° to 50° North or South latitude.

Equations:

Solar heat gain without shading devices

$= (B_i \times I_i) + (B_a \times I_a)$

Solar heat gain with outdoor shading devices

$= (B_i \times I_i + B_a \times I_a \times .25$

Solar heat gain with inside shading devices

$= (B_i \times I_i + B_a \times I_a) \times .90$

Where:

B_i = Instantaneous transmission factor from *Table 17.*

B_a = Absorption transmission factor from *Table 17.*

I_i = Solar heat gain value from Table 15 for the desired time and wall facing.

I_a = Solar heat gain value from Table 15 for 3 hours earlier than I_i and same wall facing.

SHADING FROM REVEALS, OVERHANGS, FINS AND ADJACENT BUILDINGS

All windows are shaded to a greater or lesser degree by the projections close to it and by buildings around it. This shading reduces the solar heat gain through these windows by keeping the direct rays of the sun off part of all of the window. The shaded portion has only the diffuse component striking it. Shading of windows is significant in monumental type buildings where the reveal may be large, even at the time of peak solar heat gain. *Chart 1,* this chapter, is presented to simplify the determination of the shading of windows by these projections.

Basis of Chart 1
- Shading from Reveals, Overhangs, Fins and Adjacent Buildings

The location of the sun is defined by the solar azimuth angle and the solar altitude angle as shown in *Fig. 18*. The solar azimuth angle is the angle in a horizontal plane between North and the vertical plane passing through the sun and the point on earth. The solar altitude angle is the angle in a vertical plane between the sun and a horizontal plane through a point on earth. The location of the sun with respect to the particular wall facing is defined by the wall solar azimuth angle and the solar altitude angle. The wall solar azimuth angle is the angle in the horizontal plane between the perpendicular to the wall and the vertical plane passing through the sun and the point on earth.

The shading of a window by a vertical projection alongside the window (see *Fig. 19*) is the tangent of the wall solar azimuth angle (B), times depth of the projection. The shading of a window by a horizontal projection above the window is the tangent of angle (X), a resultant of the combined effects of the altitude angle (A) and the wall solar azimuth angle (B), times the depth of the projection.

$$\text{Tan X} = \frac{\text{Tan A, solar altitude angle}}{\text{Cos B, wall solar azimuth angle}}$$

The upper part of *Chart 1* determines the tangent of the wall solar azimuth angle and the bottom part determines tan X.

Use of Chart 1
- Shading from Reveals, Overhangs, Fins and adjacent Buildings

The procedure to determine the top and side shading from *Chart 1* is.

1. Determine the solar azimuth and altitude angles from *Table 18.*

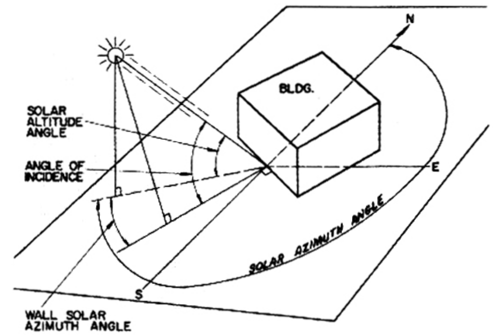


FIG. 18-SOLAR ANGLES

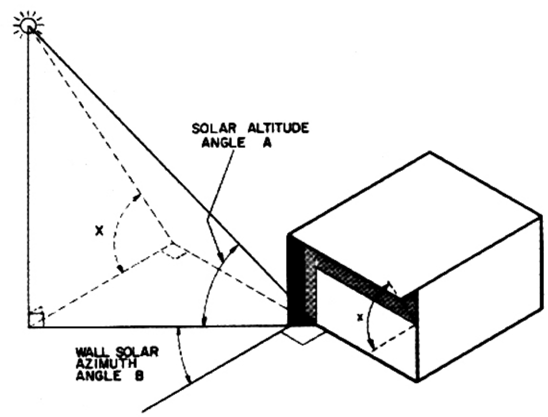


FIG. 19-SHADING BY WALL PROJECTIONS

2. Locate the solar azimuth angle on the scale in upper part of Chart 1.
3. Proceed horizontally to the exposure desired.
4. Drop vertically to “Shading from Side” scale.
5. Multiply the depth of the projection (plan view) by the “Shading from Side.”
6. Locate the solar altitude angle on the scale in lower part of Chart 1.
7. Move horizontally until the “Shading from Side” value (45 deg. lines) determined in Step 4 is intersected.
8. Drop vertically to “Shading from Top” from intersection.
9. Multiply the depth of the projection (elevation view) by the “Shading from Top.”

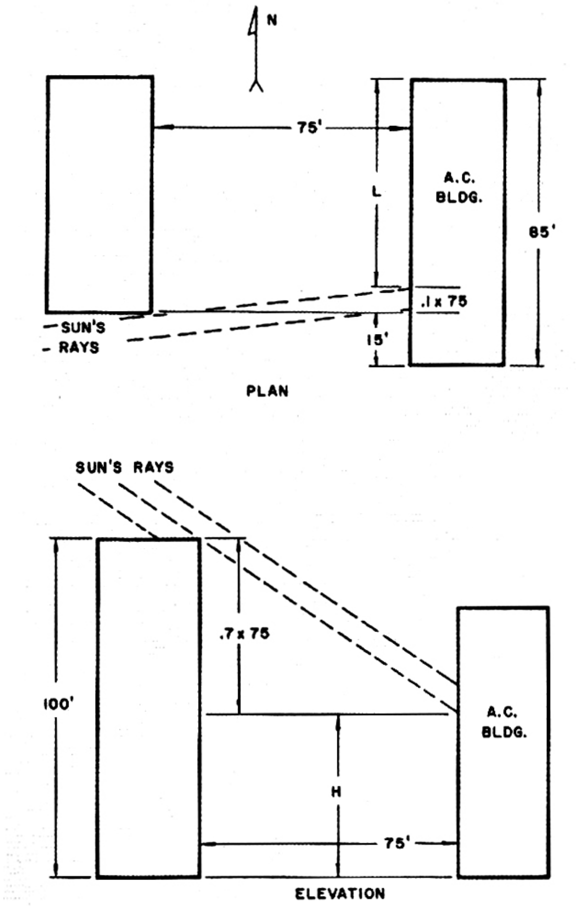


FIG. 20-SHADING OF BUILDING BY ADJACENT BUILDING

Example 7 – Shading of Building by Adjacent Building

Given:

Buildings located as shown in *Fig. 20.*

Find:

Shading at 4 p.m., July 23, of building to be air conditioned.

Solution:

It is recommended that the building plans and elevations be sketched to scale with approximate location of the sun, to enable the engineer to visualize the shading conditions.

From *Table 18,* solar azimuth angle = 267°

solar altitude angle = 35°

From *Chart 1,* shading from side = .1 ft/ft

shading from top = .7 ft/ft

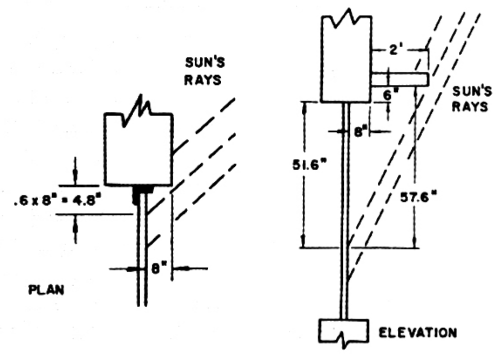


FIG. 21-SHADING OF REVEAL AND OVERHANG

Length of building in shade, L

= 85-15-(.1X75) = 62.5 ft

Height of building in shade, H = 100-(75X.7) = 47.5 ft

The air conditioned building is shaded to a height of 47.5 it and 62.5 ft along the face at 4:00 p.m. on July 23.

Example 8-Shading of Window by Reveals

Given:

A steel casement window on the west side with an 8-inch reveal.

Find:

Shading by the reveal at 2 p.m. on July 23, 40° North Latitude.

Solution:

From *Table 18,* solar azimuth-angle = 242°

solar altitude angle = 57°

From *Chart 1,* shading from side reveal = .6X8 = 4.8 in.

shading from top reveal = 1.8X8 = 14.4 in.

Example 9 – Shading of Window by Overhang and Reveal

Given:

The same window as in *Example 8* with a 2 ft overhang 6 inches above the window.

Find:

Shading by reveal and overhang a 2 p.m. on July 23, 40° North Latitude.

Solution:

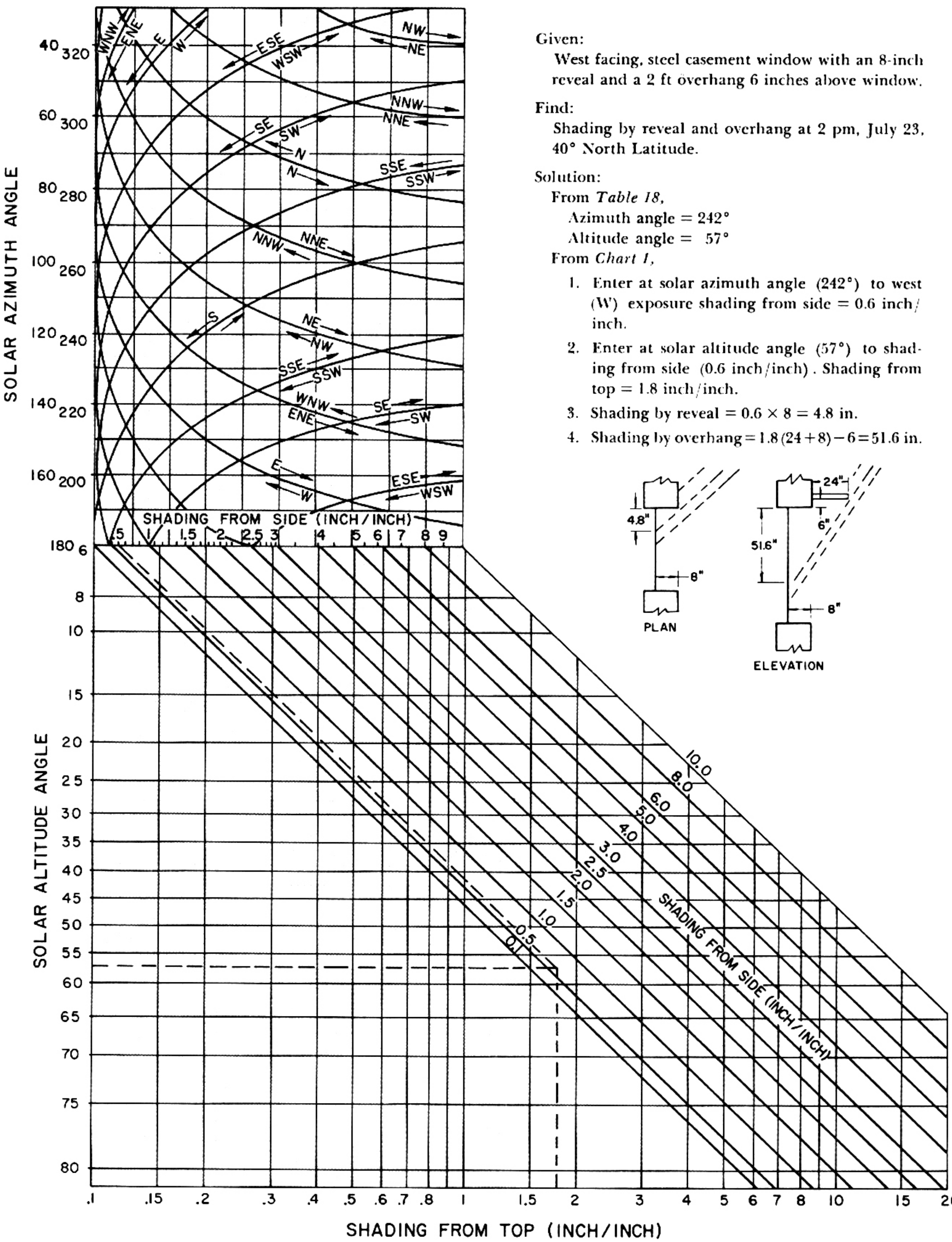
Refer to *Fig. 21.*

Shading from side reveal (same as *Example 8*) = 4.8 in.

Shading from overhang = 1.8X (24+8) = 57.6 in.

Since the overhang is 6 inches above the window, the portion of window shaded = 57.6 – 6.0 = 51.6 in.

CHART 1 — SHADING FROM REVEALS, OVERHANGS, FINS AND ADJACENT BUILDINGS



Given:

West facing, steel casement window with an 8-inch reveal and a 2 ft overhang 6 inches above window.

Find:

Shading by reveal and overhang at 2 pm, July 23, 40° North Latitude.

Solution:

From *Table 18*,

Azimuth angle = 242°

Altitude angle = 57°

From *Chart 1*,

1. Enter at solar azimuth angle (242°) to west (W) exposure shading from side = 0.6 inch/inch.
2. Enter at solar altitude angle (57°) to shading from side (0.6 inch/inch). Shading from top = 1.8 inch/inch.
3. Shading by reveal = 0.6 × 8 = 4.8 in.
4. Shading by overhang = 1.8 (24 + 8) − 6 = 51.6 in.

TABLE 18—SOLAR ALTITUDE AND AZIMUTH ANGLES

NORTH* LATITUDE	SUN TIME	Jan. 21		Feb. 20		Mar. 22		Apr. 20		May 21		June 21		July 23		Aug. 24		Sept. 22		Oct. 23		Nov. 21		Dec. 22		SUN TIME
		Alt	Az	Alt	Az	Alt	Az	Alt	Az	Alt	Az	Alt	Az	Alt	Az	Alt	Az	Alt	Az	Alt	Az	Alt	Az	Alt	Az	
	6 AM																									6 AM
	7	14	111	15	102	15	90	15	78	14	69	14	66	14	69	15	78	15	90	15	102	14	111	14	114	7
	8	28	113	30	103	30	89	30	77	28	67	27	63	28	67	30	77	30	89	30	103	28	113	27	117	8
	9	42	117	44	106	45	89	44	74	42	63	41	58	42	63	44	74	45	89	44	106	42	117	41	122	9
	10	54	126	58	112	60	89	58	68	54	54	53	49	54	54	58	68	60	89	58	112	54	126	53	131	10
	11	65	144	71	127	75	88	71	53	65	36	62	32	65	36	71	53	75	88	71	127	65	144	62	148	11
LAT 0°	12 N	70	180	79	180	90	0	79	0	70	0	67	0	70	0	79	0	90	0	79	180	70	180	67	180	12 N
	1 PM	65	216	71	233	75	272	71	307	65	324	62	328	65	324	71	307	75	272	71	233	65	216	62	212	1 PM
	2	54	234	58	248	60	271	58	292	54	306	53	311	54	306	58	292	60	271	58	248	54	234	53	229	2
	3	42	243	44	254	45	271	44	286	42	297	41	302	42	297	44	286	45	271	44	254	42	243	41	238	3
	4	28	247	30	257	30	271	30	283	28	293	27	297	28	293	30	283	30	271	30	257	28	247	27	243	4
	5	14	249	15	258	15	270	15	282	14	291	14	294	14	291	15	282	15	270	15	258	14	249	14	246	5
	6																									6
	6 AM					1	90	2	78	3	70	4	67	3	70	2	78	1	90							6 AM
	7	10	113	12	103	15	92	16	81	17	72	18	68	17	72	16	81	15	92	12	103	10	113	9	116	7
	8	24	117	27	108	30	95	31	83	32	72	32	68	32	72	31	83	30	95	27	108	24	117	23	121	8
	9	37	124	41	115	44	99	46	84	46	72	45	67	46	72	46	84	44	99	41	115	37	124	35	128	9
	10	48	136	54	125	59	106	61	84	60	67	58	61	60	67	61	84	59	106	54	125	48	136	46	139	10
	11	57	155	64	144	72	122	75	84	73	53	70	44	73	53	75	84	72	122	64	144	57	155	53	156	11
LAT 10°	12 N	60	180	69	180	80	180	89	0	80	0	77	0	80	0	89	0	80	180	69	180	60	180	57	180	12 N
	1 PM	57	205	64	216	72	238	75	276	73	307	70	316	73	307	75	276	72	238	64	216	57	205	53	204	1 PM
	2	48	224	54	235	59	254	61	276	60	293	58	299	60	293	61	276	59	254	54	235	48	224	46	221	2
	3	37	236	41	245	44	261	46	276	46	288	45	293	46	288	46	276	44	261	41	245	37	236	35	232	3
	4	24	243	27	252	30	265	31	277	32	288	32	292	32	288	31	277	30	265	27	252	24	243	23	239	4
	5	10	247	12	257	15	268	16	279	17	288	18	292	17	288	16	279	15	268	12	257	10	247	9	244	5
	6					1	270	2	282	3	290	4	293	3	290	2	282	1	270							6
	6 AM							4	79	7	71	8	68	7	71	4	79									6 AM
	7	6	114	10	106	14	95	18	84	20	75	21	72	20	75	18	84	14	95	10	106	6	114	5	117	7
	8	19	121	23	112	28	101	32	89	34	79	35	75	34	79	32	89	28	101	23	112	19	121	17	124	8
	9	30	130	36	121	42	108	46	94	48	82	48	77	48	82	46	94	42	108	36	121	30	130	28	133	9
	10	40	142	47	133	55	120	59	102	62	85	62	77	62	85	59	102	55	120	47	133	40	142	38	145	10
	11	47	158	55	152	66	143	72	117	75	88	76	74	75	88	72	117	66	143	55	152	47	158	44	163	11
LAT 20°	12 N	50	180	59	180	70	180	81	180	90	0	87	0	90	0	81	180	70	180	59	180	50	180	47	180	12 N
	1 PM	47	202	55	208	66	217	72	243	75	272	76	286	75	272	72	243	66	217	55	208	47	202	44	197	1 PM
	2	40	218	47	227	55	240	59	258	62	275	62	283	62	275	59	258	55	240	47	227	40	218	38	215	2
	3	30	230	36	239	42	252	46	266	48	278	48	283	48	278	46	266	42	252	36	239	30	230	28	227	3
	4	19	239	23	248	28	259	32	271	34	281	35	285	34	281	32	271	28	259	23	248	19	239	17	236	4
	5	6	246	10	254	14	265	18	276	20	285	21	288	20	285	18	276	14	265	10	254	6	246	5	243	5
	6							4	281	7	289	8	292	7	289	4	281									6
	6 AM							6	80	10	72	11	69	10	72	6	80									6 AM
	7	2	115	7	107	13	97	19	87	23	79	24	76	23	79	19	87	13	97	7	107	2	115			7
	8	14	124	19	116	26	106	31	95	35	86	37	82	35	86	31	95	26	106	19	116	14	124	11	126	8
	9	24	134	30	127	38	116	44	104	48	93	49	88	48	93	44	104	38	116	30	127	24	134	21	136	9
	10	32	146	40	141	49	130	56	117	61	103	62	96	61	103	56	117	49	130	40	141	32	146	29	149	10
	11	38	162	46	159	57	151	67	140	73	122	75	112	73	122	67	140	57	151	46	159	38	162	35	164	11
LAT 30°	12 N	40	180	49	180	60	180	71	180	80	180	83	180	80	180	71	180	60	180	49	180	40	180	37	180	12 N
	1 PM	38	198	46	201	57	209	67	220	73	238	75	248	73	238	67	220	57	209	46	201	38	198	35	196	1 PM
	2	32	214	40	219	49	230	56	243	61	257	62	264	61	257	56	243	49	230	40	219	32	214	29	211	2
	3	24	226	30	233	38	244	44	256	48	267	49	272	48	267	44	256	38	244	30	233	24	226	21	224	3
	4	14	236	19	244	26	254	31	265	35	274	37	278	35	274	31	265	26	254	19	244	14	236	11	234	4
	5	2	245	7	253	13	263	19	273	23	281	24	284	23	281	19	273	13	263	7	253	2	245			5
	6							6	280	10	288	11	291	10	288	6	280									6
	6 AM							7	81	13	74	15	72	13	74	7	81									6 AM
	7			5	110	12	99	19	91	24	83	26	80	24	83	19	91	12	99	5	110					7
	8	8	125	15	119	23	110	30	102	35	93	37	89	35	93	30	102	23	110	15	119	8	125	5	127	8
	9	17	136	24	131	33	122	41	113	47	104	49	100	47	104	41	113	33	122	24	131	17	136	14	138	9
	10	24	149	32	145	42	138	51	129	57	118	60	114	57	118	51	129	42	138	32	145	24	149	21	151	10
	11	28	164	37	162	48	157	58	151	66	143	69	138	66	143	58	151	48	157	37	162	28	164	25	165	11
LAT 40°	12 N	30	180	39	180	50	180	61	180	70	180	73	180	70	180	61	180	50	180	39	180	30	180	27	180	12 N
	1 PM	28	196	37	198	48	203	58	209	66	217	69	222	66	217	58	209	48	203	37	198	28	196	25	195	1 PM
	2	24	211	32	215	42	222	51	231	57	242	60	246	57	242	51	231	42	222	32	215	24	211	21	209	2
	3	17	224	24	229	33	238	41	247	47	256	49	260	47	256	41	247	33	238	24	229	17	224	14	222	3
	4	8	235	15	241	23	250	30	258	35	267	37	271	35	267	30	258	23	250	15	241	8	235	5	233	4
	5			5	250	12	261	19	269	24	277	26	280	24	277	19	269	12	261	5	250					5
	6							7	279	13	286	15	288	13	286	7	279									6
	6 AM							9	83	15	77	18	74	15	77	9	83									6 AM
	7					10	101	18	94	25	88	27	85	25	88	18	94	10	101							7
	8	3	125	10	121	19	114	28	106	34	100	37	97	34	100	28	106	19	114	10	121	3	125			8
	9	10	138	17	134	27	127	37	120	44	114	46	110	44	114	37	120	27	127	17	134	10	138	6	139	9
	10	15	151	23	148	34	143	44	137	52	131	55	128	52	131	44	137	34	143	23	148	15	151	12	152	10
	11	19	165	27	164	39	160	49	157	58	152	61	151	58	152	49	157	39	160	27	164	19	165	15	166	11
LAT 50°	12 N	20	180	29	180	40	180	51	180	60	180	63	180	60	180	51	180	40	180	29	180	20	180	17	180	12 N
	1 PM	19	195	27	196	39	200	49	203	58	208	61	209	58	208	49	203	39	200	27	196	19	195	15	194	1 PM
	2	15	209	23	212	34	217	44	223	52	229	55	232	52	229	44	223	34	217	23	212	15	209	12	208	2
	3	10	222	17	226	27	233	37	240	44	246	46	250	44	246	37	240	27	233	17	226	10	222	6	221	3
	4	3	235	10	239	19	246	28	254	34	260	37	263	34	260	28	254	19	246	10	239	3	235			4
	5					10	259	18	266	25	272	27	275	25	272	18	266	10	259							5
	6							9	277	15	283	18	286	15	283	9	277									6
SOUTH* LATITUDE	SUN TIME	July 23		Aug. 24		Sept. 22		Oct. 23		Nov. 21		Dec. 22		Jan. 21		Feb. 20		Mar. 22		Apr. 20		May 21		June 21		SUN TIME

*Use months indicated at top for North Latitudes; and use months at bottom for South Latitudes.

CHAPTER 5. HEAT AND WATER VAPOR FLOW THRU STRUCTURES

This chapter presents the methods and data for determining the sensible and latent heat gain or loss thru the outdoor structures of a building or thru a structure surrounding a space within the building. It also presents data for determining and preventing water vapor condensation on the enclosure surfaces of within the structure materials.

Heat flows from one point to another whenever a temperature difference exists between the two points; the direction of flow is always towards the lower temperature. Water vapor also flows form one point to another whenever a difference in vapor pressure exists between the two points; the direction of flow is towards the point of low vapor pressure. The rate at which the heat or water vapor will flow varies with the resistance to flow between the two points in the material. If the temperature and vapor pressure of the water vapor correspond to saturation conditions at any point, condensation occurs.

HEAT FLOW THRU BUILDING STRUCTURES

Heat gain thru the exterior construction (walls and roof) is normally calculated at the time of greatest heat flow. It is caused by solar heat being absorbed at the exterior surface and by the temperature difference between the outdoor and indoor air. Both heat sources are highly variable thruout any one day and, therefore, result in unsteady state heat flow thru the exterior construction. This unsteady state flow is difficult to evaluate for each individual situation; however, it can be handled best by means of an equivalent temperature difference across the structure.

The equivalent temperature difference is that temperature difference which results in the total heat flow thru the structure as caused by the variable solar radiation and outdoor temperature. The equivalent temperature difference across the structure must take into account the different types of construction and exposures, time of day, location of the building (latitude), and design conditions. The heat flow thru the structure may then be calculated, using the steady state heat flow equation with the equivalent temperature difference.

$$q = UA\Delta t_e$$

where q = heat flow, Btu/hr
U = transmission coefficient, Btu/(hr)(sq ft) (deg F temp diff)
A = area of surface, sq ft
Δt_e = equiv temp diff F

Heat loss thru the exterior construction (walls and roof) is normally calculated at the time of greatest heat flow. This occurs early in the morning after a few hours of very low outdoor temperatures. This approaches steady state heat flow conditions, and for all practical purposes may be assumed as such.

Heat flow thru the interior construction (floors, ceilings and partitions) is caused by a difference in temperature of the air on both sides of the structure. This temperature difference is essentially constant thru out the day and, therefore, the heat flow can be determined from the steady state heat flow equation, using the actual temperatures on either side.

EQUIVALENT TEMPERATURE DIFFERENCE-SUNLIT AND SHADED WALLS AND ROOFS

The process of transferring heat thru a wall under indicated unsteady state conditions may be visualized by picturing a 12-inch brick wall sliced into 12 one-inch sections. *Assume that temperatures in each slice are all equal at the beginning, and that the indoor and outdoor temperatures remain constant.*

When the sun shines on this wall, most of the solar heat is absorbed in the first slice, *Fig. 22.* This raises the temperature of the first slice above that of the outdoor air and the second slice, causing heat to flow to the outdoor air and also to the second slice, *Fig. 23*. The amount of heat flowing in either direction depends on the resistance to heat flow within the wall and thru the outdoor air film. The heat flow into the second slice, in turn, raises its temperature, causing heat to flow into the third slice, *Fig. 24.* This process of absorbing heat and passing some on to the next slice continues thru the wall to the last or 12th slice where the remaining heat is transferred to the inside by convection and radiation. For this particular wall, it takes approximately

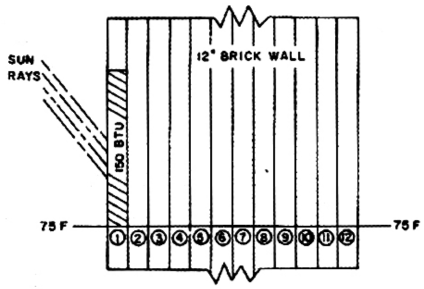


FIG. 22-SOLAR HEAT ABSORBED IN FIRST SLICE

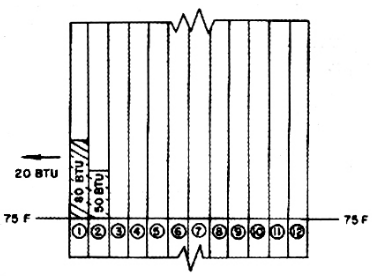


FIG. 23-BEHAVIOR OR ABSORBED SOLAR HEAT DURING SECOND TIME INTERVAL

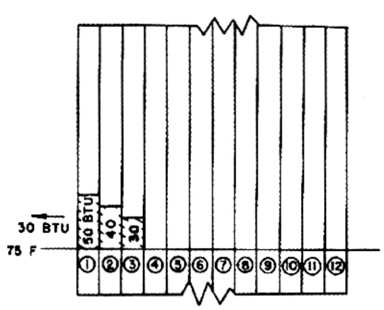


FIG. 24-BEHAVIOR OF ABSORBED SOLAR HEAT DURING THIRD TIME INTERVAL

7 hours for solar heat to pass thru the wall into the room. Because each slice must absorb some heat before passing it on, the magnitude of heat released to inside space would be reduced to about 10% of that absorbed in the slice exposed to the sun.

These diagrams do not account for possible changes in solar intensity or outdoor temperature.

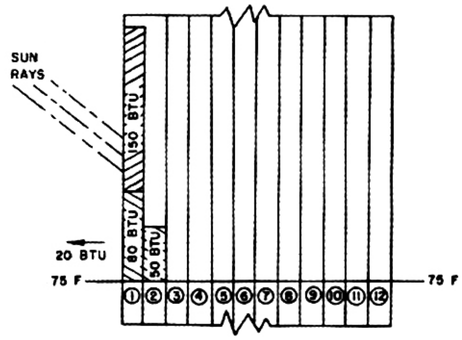


FIG. 25-BEHAVIOR OF ABSORBED SOLAR HEAT DURING SECOND TIME INTERVAL PLUS ADDITIONAL SOLAR HEAT ABSORBED DURING THIS INTERVAL

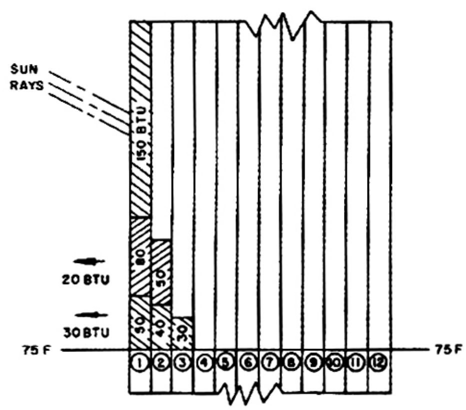


FIG. 26-BEHAVIOR OF ABSORBED SOLAR HEAT DURING THIRD TIME INTERVAL PLUS ADDITIONAL SOLAR HEAT ABSORBED DURING THIS INTERVAL

The solar heat absorbed at each time interval by the outdoor surface of the wall throughout the day goes thru this same process. *Figs. 25 and 26* show the total solar heat flow during the second and third time intervals.

A rise in outdoor temperature reduces the amount of absorbed heat going to the outdoors and more flows thru the wall.

This same process occurs with any type of wall construction to a greater or lesser degree, depending on the resistance to heat flow thru the wall and the thermal capacity of the wall.

NOTE: The thermal capacity of a wall or roof is the density of the material in the wall or roof, times the specific heat of the material, times the volume.

This progression of heat gain to the interior may occur over the full 24-hour period, and may result in a heat gain to the space during the night. If the equipment is operated less than 24 hours, i.e. either skipping the peak load requirement or as a routine procedure, the peak load requirement or as a routine procedure, the nighttime radiation to the sky and the lowering of the outdoor temperature may decrease the transmission gain and often may reverse it. Therefore, the heat gain estimate (sun and transmission thru the roof and outdoor walls), even with equipment operating less than 24 hours, may be evaluated by the use of the equivalent temperature data presented in *Tables 19 and 20.*

Basis of Tables 19 and 20
- Equivalent Temperature Difference for Sunlit and Shaded Walls and Roofs

Table 19 and 20 are analogue computer calculations using Schmidt's method based on the following conditions:

1. Solar heat in July at 40° North latitude.
2. Outdoor daily range of dry-bulb temperatures, 20 deg F.
3. Maximum outdoor temperature of 95 F db and a design indoor temperature of 80 F db, i.e. a design difference of 15 deg F.
4. Dark color walls and roofs with absorptivity of 0.90. For light color, absorptivity is 0.50; for medium color, 0.70.
5. Sun time.

The specific heat of most construction materials is approximately 0.20 Btu/(lb)(deg F); the thermal capacity of typical walls or roofs is proportional to the weight per sq ft; this permits easy interpolation.

Use of Tables 19 and 20
- Equivalent Temperature Difference for Sunlit and Shaded Wall and Roofs

The equivalent temperature differences in *Tables 19 and 20* are multiplied by the transmission coefficients listed in *Tables 21 thru 33* to determine the heat gain thru walls and roofs per sq ft of area during the summer. The total weight per sq ft of walls and roofs is obtained by adding the weights per sq ft of each component of a given structure. These weights and shown in italics and parentheses in *Tables 21 thru 33.*

Example 1 – Equivalent Temperature Difference, Roof

Given:

A flat roof exposed to the sun, with built-up roofing, $1\frac{1}{2}$ in. insulation, 3 in. wood deck and suspended acoustical tile ceiling.

Room design temperature = 80 F db

Outdoor design temperature = 95 F db

Daily range = 20 deg F

Find:

Equivalent temperature difference at 4 p.m. July.

Solution:

Wt/sq ft = 8 + 2 + 2 = 12 lb/sq ft *(Table 27, page 71)*

Equivalent temperature difference

= 43 deg F *(Table 20,* interpolated)

Example 2 – Daily Range and Design Temperature Difference Correction

At times the daily range may be more or less than 20 deg F; the difference between outdoor and room design temperatures may be more or less than 15 deg F. The corrections to be applied to the equivalent temperature difference for combinations of these two variables are listed in the notes following *Tables 19 and 20.*

Given:

The same roof as in *Example 1*

Room design temperature = 78 F db

Outdoor design temperature = 95 F db

Daily range = 26 deg F

Find:

Equivalent temperature difference under changed conditions

Solution:

Design temperature difference = 17 deg F

Daily range = 26 deg F

Correction to equivalent temperature difference

= -1 deg F *(Table 20A,* interpolated)

Equivalent temperature difference = 43 – 1 = 42 deg F

Example 3 – Other Months and Latitudes

Occasionally the heat gain thru a wall or roof must be known for months and latitudes other than those listed in *Note 3* following *Table 20.* This equivalent temperature difference is determined from the equation in *Note 3.* This equation adjusts the equivalent temperature difference for solar radiation only. Additional correction may have to be made for differences between outdoor and indoor design temperatures other than 15 deg F. Refer to *Tables 19 and 20, pages 62 and 63,* and to the correction *Table 20A.* Corrections for these differences must be made first; then the corrected equivalent temperature differences for both sun and shade must be applied in corrections for latitude.

Given:

12 in. common brick wall facing west, with no interior finish, located in New Orleans, 30° North latitude.

Find:

Equivalent temperature difference in November at 12 noon.

Find:

Equivalent temperature difference in November at 12 noon.

Solution:

The correction for design temperature difference is as follows:

Example 3, contd

Summer design dry-bulb for New Orleans

= 95 F db *(Table 1, page 11)*

Winter design dry-bulb for New Orleans

= 20 F db *(Table 1 page 11)*

Yearly range = 75 deg F

Correction in outdoor design temperature for November and a yearly range of 75 deg F

= -15 F *(Table 3, page 19)*

Outdoor design dry-bulb temperature in November at 3 p.m.

= 95 – 15 = 80 F

With and 80 F db room design, the outdoor to indoor difference is 80 – 80 = 0 deg F

Average daily range in New Orleans

= 13 deg F *(Table 1, page 11)*

The design difference of 0 deg F and a 13 deg F daily range results in a –11.5 deg F addition to the equivalent temperature difference, by interpolation in *Table 20A.*

Equivalent temperature differences for 12 in. brick wall in New Orleans at 12 noon in November:

Δt_{em} for west wall in sun

= 7 (*Table 19*)-11.5 = -4.5 deg F

TABLE 19-EQUIVALENT TEMPERATURE DIFFERENCE (DEG F)

FOR DARK COLORED†, SUNLIT AND SHADED WALLS*

Based on Dark Colored Walls; 95 F db Outdoor Design Temp; Constant 80F db Room Temp;

20 deg F Daily Range; 24-hour Operation; July and 40 N. Lat.†

EXPOSURE	WEIGHTS OF WALL ‡ (lb/sq ft)	SUN TIME																							
		AM							PM													AM				
		6	7	8	9	10	11	12	1	2	3	4	5	6	7	8	9	10	11	12	1	2	3	4	5
Northeast	20	5	15	22	23	24	19	14	13	12	13	14	14	14	12	10	8	6	4	2	0	-2	-3	-4	-2
	60	-1	-2	-2	5	24	22	20	15	10	11	12	13	14	13	12	11	10	8	6	4	2	1	0	-1
	100	4	3	4	4	4	10	16	15	14	12	10	11	12	12	12	11	10	9	8	7	6	6	5	5
	140	5	5	6	6	6	6	6	10	14	16	14	12	10	10	10	10	10	10	10	9	9	8	7	7
East	20	1	17	30	33	36	35	32	20	12	13	14	14	14	12	10	8	6	4	2	0	-1	-2	-3	-3
	60	-1	-1	0	21	30	31	31	19	14	13	12	13	14	13	12	11	10	8	5	4	3	1	1	0
	100	5	5	6	8	14	20	24	25	24	20	18	16	14	14	14	13	12	11	10	9	8	7	7	6
	140	11	10	10	9	8	9	10	15	18	19	18	17	16	14	12	13	14	14	14	13	13	12	12	12
Southeast	20	10	6	13	19	26	27	28	26	24	19	16	15	14	12	10	8	6	4	2	0	-1	-1	-2	-2
	60	1	1	0	13	20	24	28	26	25	21	18	15	14	13	12	11	10	8	6	5	4	3	3	2
	100	7	7	6	6	6	11	16	17	18	19	18	16	14	13	12	11	10	10	10	9	9	8	8	7
	140	9	8	8	8	8	7	6	11	14	15	16	18	16	15	14	13	12	12	12	11	11	10	10	9
South	20	-1	-2	-4	1	4	14	22	27	30	28	26	20	16	12	10	7	6	3	2	1	1	0	0	-1
	60	-1	-3	-4	-3	-2	7	12	20	24	25	26	23	20	15	12	10	8	6	4	2	1	1	0	-1
	100	4	4	2	2	2	3	4	8	12	15	16	18	18	15	14	11	10	9	8	8	7	6	6	5
	140	7	6	6	5	4	4	4	4	4	7	10	13	14	15	16	16	14	12	10	10	9	9	8	7
Southwest	20	-2	-4	-4	-2	0	4	6	19	26	34	40	41	42	30	24	12	6	4	2	1	1	0	-1	-1
	60	2	1	0	0	0	1	2	8	12	24	32	35	36	35	34	20	10	7	6	5	4	4	3	3
	100	7	5	6	5	4	5	6	7	8	12	14	19	22	23	24	23	22	15	10	10	9	9	8	7
	140	8	8	8	8	8	7	6	6	6	7	8	9	10	15	18	19	20	13	8	8	8	8	8	8
West	20	-2	-3	-4	-2	0	3	6	14	20	32	40	45	48	34	22	14	8	5	2	1	0	0	-1	-1
	60	2	1	0	0	0	2	4	7	10	19	26	34	40	41	36	28	16	10	6	5	4	3	3	2
	100	7	7	6	6	6	6	6	7	8	10	12	17	20	25	28	27	26	19	14	12	11	10	9	8
	140	12	11	10	9	8	8	8	9	10	10	10	11	12	14	16	21	22	23	22	20	18	16	15	13
Northwest	20	-3	-4	-4	-2	0	3	6	10	12	19	24	33	40	37	34	18	6	4	2	0	-1	-1	-2	-2
	60	-2	-3	-4	-3	-2	0	2	6	8	10	12	21	30	31	32	21	12	8	6	4	3	1	0	-1
	100	5	4	4	4	4	4	4	4	4	5	6	9	12	17	20	21	22	14	8	7	7	6	6	5
	140	8	7	6	6	6	6	6	6	6	6	6	7	8	9	10	14	18	19	20	16	7	11	10	9
North (Shade)	20	-3	-3	-4	-3	-2	1	4	8	10	12	14	13	12	10	8	6	4	2	0	0	13	-1	-2	-2
	60	-3	-3	-4	-3	-2	-1	0	3	6	8	10	11	12	12	12	10	8	6	4	2	-1	0	-1	-2
	100	1	1	0	0	0	0	0	1	2	3	4	5	5	5	8	7	6	5	4	3	1	2	2	1
	140	1	1	0	0	0	0	0	0	0	1	2	3	4	5	6	7	8	7	6	4	3	2	2	1
		6	7	8	9	10	11	12	1	2	3	4	5	6	7	8	9	10	11	12	1	2	3	4	5
		AM							PM													AM				
		SUN TIME																							

Equation: Heat Gain Thru Walls, Btu/hr = (Area, sq ft) × (equivalent temp diff) × (transmission coefficient U, *Tables 21 thru 25*)

*All values are for the both insulated and uninsulated walls.

†For other conditions, refer to corrections on *page 64.*

‡"Weight per sq ft" values for common types of construction are listed in *Tables 21 thru 25.*

For wall constructions less than 20 lb/sq ft, use listed values of 20 lb/sq ft.

Δt_{e8} for west wall in shade

$= 0$ (Table 19) – 11.5 = -11.5 deg F

No correction is needed for the time of day; this is accounted for in Table 19.

The correction for different solar intensity is

$$\Delta t_e = \Delta t_{es} + \frac{R_s}{R_m}(\Delta t_{em}\ \ \Delta t_{es}) = \frac{R_s}{R_m}\ \Delta t_{em} + (1 - \frac{R_s}{R_m})\ \Delta t_{es}$$

Wt/sq ft of wall = 120 lb/sq ft (Table 21)

Δt_{es} = - 11.5 deg F as corrected (Table 19 and 20A)

Δt_{em} = - 4.5 deg F as corrected (Table 19 and 20A)

R_s = 116 Btu/hr (Table 15, page 44)

R_m = 164 Btu/hr (Table 15, page 44)

$$\Delta t_e = -11.5 + \frac{116}{164}\,[-45\text{-}.5\text{-}(-11.5)]$$

= - 6.5 *deg* F (Novemder , 12 Noon)

TABLE 20-EQUIVALENT TEMPERATURE DIFFERENCE (DEG F)

FOR DARK COLORED†, SUNLIT AND SHADED ROOFS*

Based on 95 F db Outdoor Design Temp; Constant 80 F db Room Temp; 20 deg F Daily Range; 24-hour Operation; July and 40° N. Lat.†

CONDITION	WEIGHTS OF ROOF‡ (lb/sq ft)	SUN TIME																							
		AM							PM													AM				
		6	7	8	9	10	11	12	1	2	3	4	5	6	7	8	9	10	11	12	1	2	3	4	5
Exposed to Sun	10	-4	-6	-7	-5	-1	7	15	24	32	38	43	46	45	41	35	28	22	16	10	7	3	1	-1	-3
	20	0	-1	-2	-1	2	9	16	23	30	36	41	43	43	40	35	30	25	20	15	12	8	6	4	2
	40	4	3	2	3	6	10	16	23	28	33	38	40	41	39	35	32	28	24	20	17	13	11	9	6
	60	9	8	6	7	8	11	16	22	27	31	35	38	39	38	36	34	31	28	25	22	18	16	13	11
	80	13	12	11	11	12	13	16	22	26	28	32	35	37	37	35	34	34	32	30	27	23	20	18	14
Covered with Water	20	-5	-2	0	2	4	10	16	19	22	20	18	16	14	12	10	6	2	1	1	-1	-2	-3	-4	-5
	40	-3	-2	-1	-1	0	5	10	13	15	15	16	15	15	14	12	10	7	5	3	1	-1	-2	-3	-3
	60	-1	-2	-2	-2	-2	2	5	7	10	12	14	15	16	15	14	12	10	8	6	4	3	2	1	0
Sprayed	20	-4	-2	0	2	4	8	12	15	18	17	16	15	14	12	10	6	2	1	0	-1	-2	-2	-3	-3
	40	-2	-2	-1	-1	0	2	5	9	13	14	14	14	14	13	12	9	7	5	3	1	0	0	-1	-1
	60	-1	-2	-2	-2	-2	0	2	5	8	10	12	13	14	13	12	11	10	8	6	4	2	1	0	-1
Shaded	20	-5	-5	-4	-2	0	2	6	9	12	13	14	13	12	10	8	5	2	1	0	-1	-3	-4	-5	-5
	40	-5	-5	-4	-3	-2	0	2	5	8	10	12	13	12	11	10	8	6	4	2	0	-1	-3	-4	-5
	60	-3	-3	-2	-2	-2	-1	0	2	4	6	8	9	10	10	10	9	8	6	4	2	1	0	-1	-2
		6	7	8	9	10	11	12	1	2	3	4	5	6	7	8	9	10	11	12	1	2	3	4	5
		AM							PM													AM				
		SUN TIME																							

Equation: Heat Gain Thru Roofs, Btu/hr = (Area, sq ft) × (equivalent temp diff) × (transmission coefficient U, *Tables 27 or 28*)

*With attic ventilated and ceiling insulated roofs, reduce equivalent temp diff 25%

For peaked roofs, use the roof area projected on a horizontal plane.

†For other conditions, refer to corrections on *page 64.*

‡"Weight per sq ft" values for common types of construction are listed in *Tables 27 or 28*.

TABLE 20A-CORRECTIONS TO EQUIVALENT TEMPERATURES (DEG F)

OUTDOOR DESIGN FOR MONTH AT 3 P.M. MINUS ROOM TEMP (deg F)	DAILY RANGE (deg F)																
	8	10	12	14	16	18	20	22	24	26	28	30	32	34	36	38	40
-30	-39	-40	-41	-42	-43	-44	-45	-46	-47	-48	-49	-50	-51	-52	-53	-54	-55
-20	-29	-30	-31	-32	-33	-34	-35	-36	-37	-38	-39	-40	-41	-42	-43	-44	-45
-10	-19	-20	-21	-22	-23	-24	-25	-26	-27	-28	-29	-30	-31	-32	-33	-34	-35
0	- 9	-10	-11	-12	-13	-14	-15	-16	-17	-18	-19	-20	-21	-22	-23	-24	-25
5	- 4	- 5	- 6	- 7	- 8	- 9	-10	-11	-12	-13	-14	-15	-16	-17	-18	-19	-20
10	1	0	- 1	- 2	- 3	- 4	- 5	- 6	- 7	- 8	- 9	-10	-11	-12	-13	-14	-15
15	6	5	4	3	2	1	0	- 1	- 2	- 3	- 4	- 5	- 6	- 7	- 8	- 9	-10
20	11	10	9	8	7	6	5	4	3	2	1	0	- 1	- 2	- 3	- 4	- 5
25	16	15	14	13	12	11	10	9	8	7	6	5	4	3	2	1	0
30	21	20	19	18	17	16	15	14	13	12	11	10	9	8	7	6	5
35	26	25	24	23	22	21	20	19	18	17	16	15	14	13	12	11	10
40	31	30	29	28	27	26	25	24	23	22	21	20	19	18	17	16	15

Corrections to Equivalent Temperature Differences in Tables 19 & 20 for Conditions Other Than Basis of Table

1. Outdoor Design Temperature Minus Room Temperature *Greater or Less* Than 15 deg F db, and/or Daily Range *Greater or Less* Than 20 deg F db:

   Add the corrections listed in *Table 20A,* where the outdoor design temperature *(Table 1, page 10)* minus the room or indoor design temperature *(table 4, page 20)* is different from 15 deg F db, or the daily range is different from the 20 deg F db on which *Table 19 and 20* are based.

   This correction is to be applied to both equivalent temperature difference values, exposed to sun and shaded walls or roof.

2. Shaded walls

   For shaded walls on any exposure, use the values of equivalent temperature difference listed for north (shade), corrected if necessary as shown in Correction 1.

3. Latitudes other than 40° North and for other months with different solar intensities. *Tables 19 and 20* values are approximately correct for the east or west wall in any latitude during the hottest weather. In lower latitudes when the maximum solar altitude is 80° to 90° (the maximum occurs at noon), the temperature difference for either south or north wall is approximately the same as a north or shade wall. See *Table 18* for solar altitude angles.The temperature differential Δt_e for any wall facing or roof and for any latitude for any month is approximated as follows:

$$\Delta t_e = \Delta t_{es} + \frac{R_s}{R_m}(\Delta t_{em} \ \Delta t_{es}) = \frac{R_s}{R_m}\Delta t_{em} + (1 - \frac{R_s}{R_m})\Delta t_{es}$$

where

Δt_e = equivalent temperature difference for month and time of day desired.

Δt_{e8} = equivalent temperature difference for same wall or roof in shade at desired time of day, corrected if necessary for design conditions.

Δt_{em} = equivalent temperature difference for wall or foof exposed to the sun for desired time of day, corrected if necessary for design conditions.

R_8 = maximum solar heat gain in Btu/(hr) (sq ft) thru glass for wall facing or horizontal for roofs, for month and latitude desired, Table 15, page 44, or Table 6, page 29.

R_m = maximum solar heat gain in Btu/(hr)(sq ft) thru glass for wall facing or horizontal for roofs, for July at 40 North latitude, Table 15, page 44, or Table 6, page 29.

Example 3 illustrates the procedure.

4. Light or medium color wall or roof

   Light color wall or roof:

$$\Delta t_e = \Delta t_{es} + \frac{.50}{.90}(\Delta t_{em} - \Delta t_{es}) = .55\ \Delta t_{em} + .45\ \Delta t_{es}$$

   Medium color wall or roof:

$$\Delta t_e = \Delta t_{es} + \frac{.70}{.90}(\Delta t_{em} - \Delta t_{es}) = .78\ \Delta t_{em} + .22\ \Delta t_{es}$$

where:

Δt_e = equivalent temperature difference for month and time of day desired.

Δt_{e8} = equivalent temperature difference for same wall or roof in shade at desired time of day, corrected if necessary for design conditions.

Δt_{em} = equivalent temperature difference for wall or foof exposed to the sun for desired time of day, corrected if necessary for design conditions.

Note: Light color = white, cream, etc.
Medium color = light green, light blue, gray, etc.
Dark color = dark blue, dark red, dark brown, etc.

5. Other latitude, other month, light or medium color walls or roof.
The combined formulae are:
Light color walls or roof

$$\Delta t_e = .55 + \frac{R_s}{R_m}\ \Delta t_{em} + (1 - .55\ \frac{R_s}{R_m})\ \Delta t_{es}$$

Medium color walls or roof.

$$\Delta t_e = .78 + \frac{R_s}{R_m}\ \Delta t_{em} + (1 - .78\ \frac{R_s}{R_m})\ \Delta t_{es}$$

5. For South latitudes, use the following exposure values from *Table 19*:

South Latitude	Use Exposure Value
Northeast	Southeast
East	East
Southeast	Northeast
South	North (shade)
Southwest	Northwest
West	West
Northwest	Southwest
North (shade)	South

TRANSMISSION COEFFICIENT U

Transmission coefficient or *U* value is the rate at which heat is transferred thru a building structure in Btu/(hr)(sq ft)(deg F temp diff). The rate times the temperature difference is the heat flow thru the structure. The reciprocal of the *U* value for any wall is the total resistance of this wall to heat flow to the of heat. The total resistance of any wall to heat flow is the summation of the resistance in each component of the structure and the resistances of the outdoor and inside surface films. The transmission coefficients listed in *Tables 21 thru 33* have been calculated for the most common types of construction.

Basis of Tables 21 thru 33
- Transmission Coefficients U for Walls, Roofs, Partitions, Ceilings, Floors, Doors, and Windows

Table 21 thru 33 contain calculated *U* values based on the resistance listed in *Table 34, page 78.* The resistance of the outdoor surface film coefficient for summer and winter conditions and the inside surface film is listed in *Table 34*.

Note: The difference between summer and winter transmission coefficients for a typical wall is negligible. For example, with a transmission coefficient of 0.3 Btu/(hr)(sq ft) (F) for winter conditions, the coefficient for summer conditions will be:

1. Thermal resistance *R* (winter) of wall

$$= \frac{1}{U} = \frac{1}{0.3} = 3.33$$

2. Outdoor film thermal resistance (winter)
= 0.17 *(Table 34)*
3. Thermal resistance of wall without outdoor air film (winter = 3.33 – 0.17 = 3.16
4. Outdoor film thermal resistance (summer)
= 0.25 *(Table 34)*
5. Thermal resistance of wall with outdoor air film (summer) = 3.16 + 0.25 = 3.41
6. Transmission coefficient *U* of wall in summer

$$= \frac{1}{R} = \frac{1}{3.41} = 0.294$$

7. Difference between summer and winter transmission becomes greater with larger *U* values and less with smaller *U* values.

Use of Tables 21 thru 33
- Transmission Coefficients U for Walls, Roofs, Partitions, Ceilings, floors, Doors, and Windows

The transmission coefficients may be used for calculating the heat flow for both summer and winter conditions for the average application.

Example 4 – Transmission Coefficients

Given:

Masonry partition made of 8 in. hollow clay tile, both sides finished, metal lath plastered on furring with $\frac{3}{4}$ in. sand plaster.

Find:

Transmission coefficient

Solution:

Transmission coefficient *U*
= 0.18 Btu/(hr)(sq ft)(deg F), *Table 26, page 70*

Example 5 – Transmission Coefficient, Addition of Insulation

The transmission coefficients listed in *Tables 21 thru 30* do not include insulation (except for flat roofs, *Table 27, page 71).* Frequently, fibrous insulation or reflective insulation is included in the exterior building structure. The transmission coefficient for the typical constructions listed in *Table 21 thru 30,* with insulation, may be determined from *Table 31, page 75.*

Given:

Masonry wall consisting of 4 in. face brick, 8 in. concrete cinder block, metal lath plastered on furring with $\frac{3}{4}$ in. sand plaster and 3 in. of fibrous insulation in the stud space.

Find:

Transmission coefficient.

Solution:

Refer to *Tables 22 and 31.*

U value for wall without insulation
= 0.24 Btu/(hr)(sq ft)(deg F)

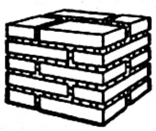
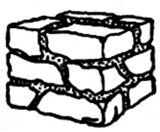
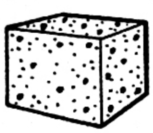
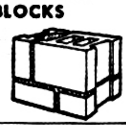
U value for wall with insulation
= 0.07 Btu/(hr)(sq ft)(deg F)

TABLE 21—TRANSMISSION COEFFICIENT U—MASONRY WALLS*

FOR SUMMER AND WINTER

Btu/(hr) (sq ft) (deg F temp diff)

All numbers in parentheses indicate weight per sq ft. Total weight per sq ft is sum of wall and finishes.

EXTERIOR FINISH		THICKNESS (inches) and WEIGHT (lb per sq ft)	INTERIOR FINISH: None	⅜" Gypsum Board (Plaster Board) (2)	⅝" Plaster on Wall: Sand Agg (6)	⅝" Plaster on Wall: Lt Wt Agg (3)	Metal Lath Plastered on Furring: ¾" Sand Plaster(7)	Metal Lath Plastered on Furring: ¾" Lt Wt Plaster(3)	⅜" Gypsum or Wood Lath Plastered on Furring: ½" Sand Plaster(7)	⅜" Gypsum or Wood Lath Plastered on Furring: ½" Lt Wt Plaster(2)	Insulating Board Plain or Plastered on Furring: ½" Board (2)	Insulating Board Plain or Plastered on Furring: 1" Board (4)
SOLID BRICK 	Face & Common	8 (87)	.48	.41	.45	.41	.31	.28	.29	.27	.22	.16
		12 (123)	.35	.31	.33	.30	.25	.23	.23	.22	.19	.14
		16 (173)	.27	.25	.26	.25	.21	.19	.20	.19	.16	.13
	Common Only	8 (80)	.41	.36	.39	.35	.28	.26	.26	.25	.21	.15
		12 (120)	.31	.28	.30	.27	.23	.22	.22	.21	.18	.14
		16 (160)	.25	.23	.24	.23	.19	.18	.18	.18	.16	.12
STONE 		8 (100)	.67	.55	.63	.53	.39	.34	.35	.32	.26	.18
		12 (150)	.55	.47	.52	.46	.34	.31	.31	.29	.24	.17
		16 (200)	.47	.41	.45	.40	.31	.28	.28	.27	.22	.16
		24 (300)	.36	.32	.35	.32	.26	.24	.24	.23	.19	.15
ADOBE-BLOCKS OR BRICK		8 (26)	.34	.30	.32	.30	.25	.23	.23	.22	18	.12
		12 (40)	.25	.23	.24	.23	.20	.18	.18	.18	.15	.14
POURED CONCRETE 	140 lb/cu ft	6 (70)	.75	.55	.69	.58	.41	.36	.37	.34	.27	.18
		8 (93)	.67	.49	.63	.53	.39	.34	.35	.32	.26	.17
		10 (117)	.61	.44	.57	.49	.36	.32	.33	.31	.25	.17
		12 (140)	.55	.40	.52	.45	.34	.31	.31	.29	.24	.16
	80 lb/cu ft	6 (40)	.31	.28	.30	.27	.23	.21	.22	.21	.18	.14
		8 (53)	.25	.23	.24	.23	.19	.18	.18	.18	.16	.12
		10 (66)	.21	.19	.20	.19	.17	.16	.15	.14	.14	.11
		12 (80)	.18	.17	.17	.15	.15	.14	.14	.14	.12	.10
	30 lb/cu ft	6 (15)	.13	.13	.13	.13	.12	.11	.11	.11	.13	.09
		8 (20)	.10	.10	.10	.10	.09	.09	.09	.09	.10	.07
		10 (25)	.08	.08	.08	.08	.08	.07	.08	.07	.08	.06
		12 (30)	.07	.07	.07	.07	.07	.07	.06	.06	.07	.06
HOLLOW CONCRETE BLOCKS 	Sand & Gravel Agg	8 (43)	.52	.44	.48	.43	.33	.29	.30	.28	.23	.17
		12 (63)	.47	.41	.45	.40	.31	.28	.28	.27	.22	.16
	Cinder Agg	8 (37)	.39	.35	.37	.34	.27	.25	.25	.24	.20	.15
		12 (53)	.36	.33	.35	.32	.26	.24	.23	.23	.19	.15
	Lt Wt Agg	8 (32)	.35	.32	.34	.31	.26	.23	.24	.22	.19	.15
		12 (43)	.32	.29	.31	.28	.24	.22	.22	.21	.18	.14
STUCCO ON HOLLOW CLAY TILE		8 (39)	.36	.32	.34	.32	.26	.24	.24	.23	.19	.15
		10 (44)	.32	.29	.31	.28	.23	.22	.22	.21	.18	.14
		12 (49)	.29	.27	.28	.26	.22	.20	.21	.20	.17	.13

1958 ASHAE Guide

Equations: Heat Gain, Btu/hr = (Area, sq ft) × (U value) × (equivalent temp diff, *Table 19*)
Heat Loss, Btu/hr = (Area, sq ft) × (U value) × (outdoor temp — inside temp)

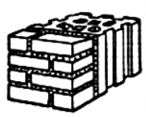
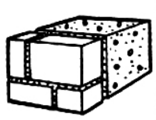
*For addition of insulation and air spaces to above walls, refer to *Table 31, page 75.*

TABLE 22—TRANSMISSION COEFFICIENT U—MASONRY VENEER WALLS*

FOR SUMMER AND WINTER

Btu/(hr) (sq ft) (deg F temp diff)

All numbers in parentheses indicate weight per sq ft. Total weight per sq ft is sum of wall and finishes.

EXTERIOR FINISH	BACKING	THICKNESS (inches) and WEIGHT (lb per sq ft)	INTERIOR FINISH: None	Gypsum Board (Plaster Board) (2)	5/8" Plaster on Wall		Metal Lath Plastered on Furring		3/8" Gypsum or Wood Lath Plastered on Furring		Insulating Board Plain or Plastered on Furring	
					Sand Agg (6)	Lt Wt Agg (3)	3/4" Sand Plaster(7)	3/4" Lt Wt Plaster(3)	½" Sand Plaster(7)	½" Lt Wt Plaster(2)	½" Board (2)	1" Board (4)
 4" Face Brick (43) —or— 4" Stone (50) —or— Precast Concrete (Sand Agg) 4" & 6" (39) (58)	Concrete Block (Cinder Agg)	4 (20)	.41	.37	.39	.35	.28	.26	.26	.25	.21	.16
		8 (37)	.33	.30	.32	.29	.24	.22	.23	.21	.18	.14
		12 (53)	.31	.29	.30	.28	.23	.21	.22	.21	.18	.14
	(Lt Wt Agg)	4 (17)	.35	.32	.34	.31	.25	.23	.24	.22	.19	.15
		8 (32)	.30	.28	.29	.27	.23	.21	.21	.20	.17	.14
		12 (43)	.28	.26	.27	.25	.21	.20	.20	.19	.17	.13
	(Sand & Gravel Agg)	4 (23)	.49	.44	.46	.41	.32	.29	.29	.27	.22	.17
		8 (43)	.41	.37	.39	.35	.28	.26	.26	.25	.21	.16
		12 (63)	.38	.35	.37	.33	.27	.25	.25	.24	.20	.15
	Hollow Clay Tile	4 (16)	.41	.37	.39	.35	.28	.26	.26	.25	.21	.16
		8 (30)	.31	.29	.30	.28	.23	.22	.22	.21	.18	.14
		12 (40)	.26	.25	.25	.24	.20	.19	.19	.18	.16	.13
	Concrete (Lt Wt Agg) 80 lb/cu ft	4 (26)	.35	.31	.34	.31	.25	.23	.24	.22	.19	.15
		6 (40)	.27	.25	.27	.25	.21	.20	.20	.19	.16	.13
		8 (54)	.22	.21	.22	.21	.18	.17	.17	.16	.14	.12
	(Sand & Gravel Agg)	4 (47)	.60	.53	.56	.49	.36	.32	.33	.31	.25	.18
		6 (70)	.55	.49	.52	.45	.34	.31	.32	.29	.24	.17
		8 (95)	.51	.45	.48	.42	.32	.29	.30	.28	.23	.17
	Common Brick	4 (40)	.49	.42	.46	.41	.32	.29	.29	.27	.22	.16
		8 (80)	.35	.31	.34	.31	.25	.23	.24	.22	.19	.15
 4" Common Brick (40) —or— Precast Concrete (Sand Agg) 8" & 10" (78) (98) —or— 4" Concrete Block (23) (Sand Agg) —or— 8" Stone (100)	Concrete Block (Cinder Agg)	4 (20)	.36	.33	.35	.32	.26	.24	.24	.23	.19	.15
		8 (37)	.29	.28	.29	.26	.22	.21	.21	.20	.17	.14
		12 (53)	.28	.26	.27	.25	.21	.20	.20	.19	.17	.13
	(Lt Wt Agg)	4 (17)	.32	.29	.30	.28	.23	.22	.22	.21	.18	.14
		8 (32)	.27	.26	.26	.25	.21	.20	.20	.19	.17	.13
		12 (43)	.25	.24	.25	.23	.20	.19	.19	.18	.16	.13
	(Sand & Gravel Agg)	4 (23)	.42	.38	.40	.36	.29	.26	.27	.25	.21	.16
		8 (43)	.36	.33	.35	.32	.26	.24	.24	.23	.19	.15
		12 (63)	.34	.32	.33	.30	.25	.23	.23	.22	.19	.15
	Hollow Clay Tile	4 (16)	.36	.33	.35	.32	.26	.24	.24	.23	.19	.15
		8 (30)	.28	.27	.28	.26	.22	.20	.20	.19	.17	.13
		12 (40)	.24	.23	.23	.22	.19	.18	.18	.17	.15	.12
	Concrete (Lt Wt Agg) 80 lb/cu ft	4 (26)	.32	.29	.30	.28	.23	.22	.22	.21	.18	.14
		6 (40)	.25	.23	.25	.23	.20	.18	.19	.18	.15	.13
		8 (54)	.21	.20	.20	.19	.17	.16	.16	.16	.14	.11
	(Sand & Gravel Agg)	4 (47)	.50	.45	.48	.42	.32	.29	.30	.28	.23	.17
		6 (70)	.47	.42	.44	.39	.31	.28	.29	.27	.22	.17
		8 (95)	.43	.40	.41	.37	.29	.27	.28	.26	.21	.16
	Common Brick	4 (40)	.42	.37	.40	.36	.29	.26	.27	.26	.21	.16
		8 (80)	.32	.29	.30	.28	.23	.22	.22	.21	.18	.14

1958 ASHAE Guide

Equations: Heat Gain, Btu/hr = (Area, sq ft) × (U value) × (equivalent temp diff, *Table 19*)

Heat Loss, Btu/hr = (Area, sq ft) × (U value) × (outdoor temp — inside temp)

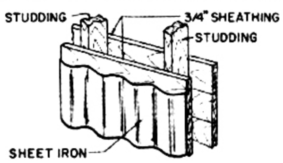
*For addition of insulation and air spaces to walls, refer to *Table 31, page 75.*

TABLE 23—TRANSMISSION COEFFICIENT U—LIGHT CONSTRUCTION, INDUSTRIAL WALLS*†

FOR SUMMER AND WINTER

Btu/(hr) (sq ft) (deg F temp diff)

All numbers in parentheses indicate weight per sq ft. Total weight per sq ft is sum of wall and finishes.



EXTERIOR FINISH	SHEATHING	WEIGHT (lb per sq ft)	INTERIOR FINISH: None	Flat Iron (1)	Insulating Board ½" (2)	Insulating Board 25/32" (3)	Wood ¾" (2)
⅜" Corrugated Transite	None	(1)	1.16	.55	.32	.26	.36
	½" Ins. Board	(2)	.34	.26	.19	.17	.21
	25/32" Ins. Board	(2)	.27	.21	.17	.15	.18
24 Gauge Corrugated Iron	None	(1)	1.40	.60	.33	.27	.38
	½" Ins. Board	(2)	.36	.27	.20	.17	.21
	25/32" Ins. Board	(2)	.28	.22	.17	.15	.18
	¾" Wood	(3)	.46	.33	.22	.19	.24
¾" Wood Siding	None	(2)	.58	.37	.25	.21	.27

1958 ASHAE Guide

Equations: Heat Gain, Btu/hr = (Area, sq ft) × (U value) × (equivalent temp diff, *Table 19*).
Heat Loss, Btu/hr = (Area, sq ft) × (U value) × (outdoor temp — inside temp).

*For addition of air spaces and insulation to walls, refer to *Table 31, page 75*.

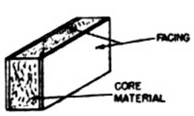
†Values apply when sealed with calking compound between sheets, and at ground and roof lines. When sheets are not sealed, increase U factors by 10%. These values may be used for roofs, heat flow up-winter; for heat flow down-summer, multiply above factors by 0.8.

TABLE 24—TRANSMISSION COEFFICIENT U—LIGHTWEIGHT, PREFABRICATED CURTAIN TYPE WALLS*

FOR SUMMER AND WINTER

Btu/(hr) (sq ft) (deg F temp diff)

All numbers in parentheses indicate weight per sq ft. Total weight per sq ft is sum of wall and finishes.



INSULATING CORE MATERIAL	DENSITY† (lb/cu ft)	METAL FACING (3) Core Thickness (in.) 1	2	3	4	METAL FACING WITH ¼" AIR SPACE (3) Core Thickness (in.) 1	2	3	4
Glass, Wood, Cotton Fibers	3	.21	.12	.08	.06	.19	.11	.08	.06
Paper Honeycomb	5	.39	.23	.17	.13	.32	.20	.15	.12
Paper Honeycomb with Perlite Fill, Foamglas	9	.29	.17	.12	.09	.25	.15	.11	.09
Fiberboard	15	.36	.21	.15	.12	.29	.19	.14	.11
Wood Shredded (Cemented in Preformed Slabs)	22	.31	.18	.13	.10	.25	.16	.12	.09
Expanded Vermiculite	7	.34	.20	.14	.11	.28	.18	.13	.10
Vermiculite or Perlite Concrete	20	.44	.27	.19	.15	.35	.23	.18	.14
	30	.51	.32	.24	.19	.39	.27	.21	.17
	40	.58	.38	.29	.23	.43	.31	.25	.20
	60	.69	.49	.38	.31	.49	.38	.31	.26

Equations: Heat Gain, Btu/hr = (Area, sq ft) × (U value) × (equivalent temp diff, *Table 19*).
Heat Loss, Btu/hr = (Area, sq ft) × (U value) × (outdoor temp — inside temp).

*For addition of insulation and air spaces to walls, refer to *Table 31, page 75*.

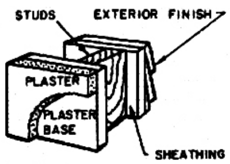
†Total weight per sq ft $= \dfrac{\text{core density} \times \text{core thickness}}{12} + 3$ lb/sq ft

TABLE 25—TRANSMISSION COEFFICIENT U—FRAME WALLS AND PARTITIONS*

FOR SUMMER AND WINTER

Btu/(hr) (sq ft) (deg F temp diff)

All numbers in parentheses indicate weight per sq ft. Total weight per sq ft is sum of component materials.



EXTERIOR FINISH	SHEATHING	INTERIOR FINISH: None	3/4" Wood Panel (2)	3/8" Gypsum Board (Plaster Board) (2)	Metal Lath Plastered: 3/4" Sand Plaster (7)	Metal Lath Plastered: 3/4" Lt Wt Plaster (3)	3/8" Gypsum or Wood Lath Plastered: 1/2" Sand Plaster (7)	3/8" Gypsum or Wood Lath Plastered: 1/2" Lt Wt Plaster (2)	Insulating Board Plain or Plastered: 1/2" Board (2)	Insulating Board Plain or Plastered: 1" Board (4)
1" Stucco (10)	None, Building Paper	.91	.33	.42	.45	.39	.40	.37	.29	.20
OR Asbestos	5/16" Plywood (1) or 1/2" Gyp (2)	.68	.30	.37	.40	.35	.36	.33	.26	.19
Cement Siding (1)	25/32" Wood & Bldg Paper (2)	.48	.25	.30	.31	.28	.29	.27	.22	.17
OR Asphalt	1/2" Insulating Board (2)	.42	.23	.27	.29	.26	.27	.25	.21	.16
Roll Siding (2)	25/32" Insulating Board (3)	.32	.20	.23	.24	.22	.22	.21	.18	.14
4" Face Brick	None, Building Paper	.73	.30	.37	.40	.35	.36	.33	.26	.19
Veneer (43) OR	5/16" Plywood (1) or 1/2" Gyp (2)	.57	.28	.33	.36	.32	.32	.30	.24	.18
3/8" Plywood (1)	25/32" Wood & Bldg Paper (2)	.42	.23	.27	.29	.26	.27	.25	.21	.16
OR Asphalt	1/2" Insulating Board (2)	.38	.22	.25	.27	.25	.25	.24	.20	.15
Siding (2)	25/32" Insulating Board (3)	.30	.19	.21	.22	.21	.21	.20	.17	.14
Wood Siding (3)	None, Building Paper	.57	.27	.33	.35	.31	.32	.30	.24	.18
OR	5/16" Plywood (1) or 1/2" Gyp (2)	.48	.25	.30	.31	.28	.29	.27	.22	.17
Wood Shingles (2)	25/32" Wood & Bldg Paper	.36	.22	.25	.26	.24	.24	.23	.19	.15
OR 3/4" Wood	1/2" Insulating Board (2)	.33	.20	.23	.24	.22	.23	.22	.18	.14
Panels (3)	25/32" Insulating Board (3)	.27	.18	.20	.21	.19	.19	.19	.16	.13
Wood Shingles	None, Building Paper	.43	.24	.28	.29	.27	.27	.25	.21	.16
Over 5/16" Insul	5/16" Plywood (1) or 1/2" Gyp (2)	.38	.22	.25	.27	.24	.25	.23	.19	.15
Backer Board (3)	25/32" Wood & Bldg Paper	.30	.19	.22	.23	.21	.21	.20	.17	.14
OR Asphalt	1/2" Insulating Board (2)	.28	.18	.20	.21	.20	.20	.19	.16	.13
Insulated Siding (4)	25/32" Insulating Board (3)	.23	.16	.18	.18	.17	.18	.17	.15	.12
Single Partition (Finish on one side only)			.43	.60	.67	.55	.57	.50	.36	.23
Double Partition (Finish on both sides)			.24	.34	.39	.31	.32	.28	.19	.12

1958 ASHAE Guide

Equations: Walls—Heat Gain, Btu/hr = (Area, sq ft) × (U value) × (equivalent temp diff, *Table 19*).
—Heat Loss, Btu/hr = (Area, sq ft) × (U value) × (outdoor temp—inside temp).

Partitions, unconditioned space adjacent—Heat Gain or Loss, Btu/hr = (Area sq ft) × (U value) × (outdoor temp—inside temp—5 F).

Partitions, kitchen or boiler room adjacent—Heat Gain, Btu/hr = (Area sq ft) × (U value) × (actual temp diff or outdoor temp—inside temp + 15 F to 25 F).

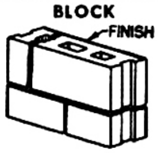
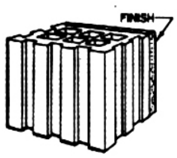
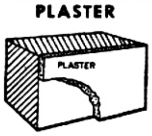
*For addition of insulation and air spaces to partitions, refer to *Table 31, page 75*.

TABLE 26—TRANSMISSION COEFFICIENT U—MASONRY PARTITIONS*

FOR SUMMER AND WINTER

Btu/(hr) (sq ft) (deg F temp diff)

All numbers in parentheses indicate weight per sq ft. Total weight per sq ft is sum of masonry unit and finish × 1 or 2 (finished one or both sides).

BACKING	THICKNESS (inches) and WEIGHT (per sq ft)	Both Sides Unfinished	No. of Sides Finished	FINISH: ⅜" Gypsum Board (Plaster Board) (2)	⅝" Plaster on Wall: Sand Agg (6)	⅝" Plaster on Wall: Lt Wt Agg (3)	Metal Lath Plastered on Furring: ¾" Sand Plaster(7)	Metal Lath Plastered on Furring: ¾" Lt Wt Plaster(3)	⅜" Gypsum or Wood Lath Plastered on Furring: ½" Sand Plaster(7)	⅜" Gypsum or Wood Lath Plastered on Furring: ½" Lt Wt Plaster(2)	Insulating Board Plain or Plastered on Furring: ½" Board(2)	Insulating Board Plain or Plastered on Furring: 1" Board(4)
HOLLOW CONCRETE BLOCK 	3 (17)	.45	One	.39	.43	.38	.30	.27	.28	.26	.21	.16
			Both	.35	.41	.33	.23	.20	.20	.18	.14	.10
	4 (20)	.40	One	.36	.39	.35	.28	.26	.26	.25	.20	.15
			Both	.32	.37	.31	.21	.19	.19	.18	.13	.11
	8 (37)	.32	One	.29	.31	.29	.24	.22	.22	.21	.18	.14
			Both	.27	.30	.26	.19	.17	.17	.16	.12	.09
Cinder Agg	12 (53)	.31	One	.28	.30	.27	.23	.21	.22	.21	.17	.14
			Both	.26	.29	.25	.18	.16	.17	.15	.12	.09
Lt Wt Agg	3 (15)	.38	One	.34	.36	.33	.27	.25	.25	.24	.20	.15
			Both	.31	.35	.30	.21	.18	.19	.17	.13	.09
	4 (17)	.35	One	.31	.34	.31	.25	.23	.24	.22	.19	.15
			Both	.29	.32	.27	.20	.17	.17	.16	.13	.09
	8 (32)	.30	One	.27	.29	.27	.22	.21	.21	.20	.17	.14
			Both	.25	.28	.24	.18	.16	.16	.15	.12	.09
	12 (43)	.28	One	.25	.27	.25	.21	.20	.20	.19	.16	.13
			Both	.23	.26	.23	.17	.15	.16	.15	.12	.08
Sand & Gravel Agg	8 (43)	.40	One	.36	.39	.35	.28	.26	.26	.25	.20	.15
			Both	.32	.37	.31	.21	.19	.19	.18	.13	.11
	12 (63)	.38	One	.34	.36	.33	.27	.25	.25	.24	.19	.15
			Both	.30	.35	.29	.21	.18	.19	.17	.13	.09
HOLLOW CLAY TILE 	3 (15)	.46	One	.40	.44	.39	.31	.28	.28	.27	.22	.16
			Both	.36	.42	.34	.23	.20	.20	.19	.14	.10
	4 (16)	.40	One	.36	.39	.35	.28	.26	.26	.25	.20	.15
			Both	.32	.37	.31	.21	.19	.19	.18	.13	.11
	6 (25)	.35	One	.31	.33	.31	.25	.23	.23	.22	.19	.15
			Both	.28	.32	.27	.20	.17	.18	.16	.13	.09
	8 (30)	.31	One	.28	.30	.28	.23	.22	.22	.21	.18	.14
			Both	.26	.29	.25	.18	.16	.17	.16	.12	.09
HOLLOW GYPSUM TILE	3 (9)	.37	One	.33	.35	.32	.26	.24	.24	.23	.19	.15
			Both	.30	.34	.29	.20	.18	.18	.13	.13	.09
	4 (13)	.33	One	.30	.32	.29	.24	.22	.23	.22	.18	.14
			Both	.27	.31	.26	.19	.17	.17	.16	.12	.09
SOLID GYPSUM PLASTER 	1½						.61 (13)	.43 (6)				
	2						.58 (18)	.38 (8)				
	2½						.55 (22)	.34 (9)				

1958 ASHAE Guide

Equations: Partitions, unconditioned space adjacent: Heat Gain or Loss, Btu/hr = (Area, sq ft) × (U value) × (outdoor temp—inside temp—5 F).

Partitions, kitchen or boiler room adjacent: Heat Gain or Loss, Btu/hr = (Area, sq ft) × (U value) × (actual temp diff or outdoor temp—inside temp + 15 F to 25 F).

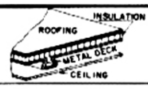
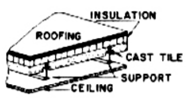
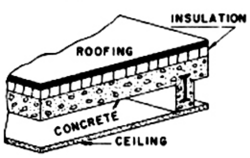
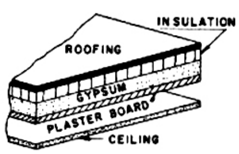
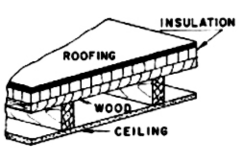
*For addition of insulation and air spaces to partitions, refer to *Table 31, page 75.*

TABLE 27—TRANSMISSION COEFFICIENT U—FLAT ROOFS COVERED WITH BUILT-UP ROOFING*

FOR HEAT FLOW DOWN—SUMMER. FOR HEAT FLOW UP—WINTER (See Equation at Bottom of Page).

Btu/(hr) (sq ft) (deg F temp diff)

All numbers in parentheses indicate weight per sq ft. Total weight per sq ft is sum of roof, finish and insulation.

TYPE OF DECK	THICKNESS OF DECK (inches) and WEIGHT (lb per sq ft)	CEILING†	INSULATION ON TOP OF DECK, INCHES						
			No Insulation	½ (1)	1 (1)	1½ (2)	2 (3)	2½ (3)	3 (4)
Flat Metal 	1 (5)	None or Plaster (6)	.67	.35	.23	.18	.15	.12	.10
		Suspended Plaster (5)	.32	.22	.17	.14	.12	.10	.09
		Suspended Acou Tile (2)	.23	.18	.14	.12	.11	.09	.08
Preformed Slabs—Wood Fiber and Cement Binder 	2 (4)	None or Plaster (6)	.20	.16	.13	.11	.10	.09	.08
		Suspended Plaster (5)	.15	.12	.11	.09	.08	.08	.07
		Suspended Acou Tile (2)	.13	.10	.09	.08	.08	.07	.06
	3 (7)	None or Plaster (6)	.14	.11	.10	.09	.08	.08	.07
		Suspended Plaster (5)	.12	.10	.09	.07	.07	.06	.05
		Suspended Acou Tile (2)	.10	.09	.08	.07	.07	.06	.05
Concrete (Sand & Gravel Agg)	4, 6, 8 (47),(70), (93)	None or Plaster (6)	.51	.30	.21	.16	.14	.12	.10
		Suspended Plaster (5)	.28	.20	.16	.13	.12	.10	.09
		Suspended Acou Tile(2)	.21	.16	.13	.11	.10	.09	.08
(Lt Wt Agg on Gypsum Board) 	2 (9)	None or Plaster (6)	.27	.20	.15	.13	.11	.10	.08
		Suspended Plaster (5)	.18	.14	.12	.10	.09	.09	.08
		Suspended Acou Tile (2)	.15	.12	.11	.09	.08	.08	.07
	3 (13)	None or Plaster (6)	.21	.16	.13	.11	.10	.09	.08
		Suspended Plaster (5)	.15	.12	.11	.09	.08	.08	.07
		Suspended Acou Tile (2)	.13	.11	.10	.08	.08	.07	.06
	4 (16)	None or Plaster (6)	.17	.14	.11	.10	.09	.08	.07
		Suspended Plaster (5)	.13	.11	.10	.08	.08	.07	.06
		Suspended Acou Tile(2)	.12	.10	.09	.07	.07	.06	.05
Gypsum Slab on ½" Gypsum Board 	2 (11)	None or Plaster (6)	.32	.22	.17	.14	.12	.10	.09
		Suspended Plaster (5)	.21	.17	.13	.11	.10	.09	.08
		Suspended Acou Tile (2)	.17	.13	.12	.10	.09	.08	.07
	3 (15)	None or Plaster (6)	.27	.19	.15	.13	.11	.10	.08
		Suspended Plaster (5)	.19	.15	.13	.11	.10	.09	.08
		Suspended Acou Tile (2)	.15	.12	.11	.09	.08	.08	.07
	4 (19)	None or Plaster (6)	.23	.17	.14	.12	.10	.09	.08
		Suspended Plaster (5)	.17	.13	.12	.10	.09	.08	.07
		Suspended Acou Tile (2)	.14	.12	.11	.09	.08	.08	.07
Wood 	1 (3)	None or Plaster (6)	.40	.26	.19	.15	.13	.11	.09
		Suspended Plaster (5)	.24	.18	.14	.12	.11	.09	.08
		Suspended Acou Tile (2)	.19	.15	.13	.11	.10	.08	.07
	2 (5)	None or Plaster (6)	.28	.20	.16	.13	.11	.10	.08
		Suspended Plaster (5)	.19	.15	.13	.11	.10	.09	.07
		Suspended Acou Tile (2)	.16	.13	.11	.10	.09	.08	.07
	3 (8)	None or Plaster (6)	.21	.16	.13	.11	.10	.09	.08
		Suspended Plaster (5)	.16	.13	.11	.09	.09	.08	.07
		Suspended Acou Tile (2)	.13	.11	.10	.09	.08	.07	.06

1958 ASHAE Guide

Equations: Summer—(Heat Flow Down) Heat Gain, Btu/hr = (Area, sq ft) × (U value) × (equivalent temp diff, *Table 20*).

Winter—(Heat Flow Up) Heat Loss, Btu/hr = (Area, sq ft) × (U value × 1.1) × (outdoor temp—inside temp).

*For addition of air spaces or insulation to roofs, refer to *Table 31, page 75.*

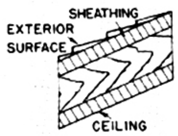
†For suspended ½" insulation board, plain (.6) or with ½" sand aggregate plaster (5). use values of suspended acou tile.

TABLE 28—TRANSMISSION COEFFICIENT U—PITCHED ROOFS*

FOR HEAT FLOW DOWN—SUMMER. FOR HEAT FLOW UP—WINTER (See Equation at Bottom of Page)

Btu/(hr) (sq ft projected area) (deg F temp diff)

All numbers in parentheses indicate weight per sq ft. Total weight per sq ft is sum of component materials.



PITCHED ROOFS		CEILING											
		None	3/4" Wood Panel (2)	3/8" Gypsum Board (Plaster Board) (2)	Metal Lath Plastered: 3/4" Sand Plaster (7)	Metal Lath Plastered: 3/4" Lt Wt Plaster (3)	3/8" Gypsum or Wood Lath Plastered: 1/2" Sand Plaster (5)	3/8" Gypsum or Wood Lath Plastered: 1/2" Lt Wt Plaster (2)	Insulating Board Plain or 1/2" Sand Agg Plastered: 1/2" Board (2)	Insulating Board Plain or 1/2" Sand Agg Plastered: 1" Board (4)	Acoustical Tile on Furring or 3/8" Gypsum: 1/2" Tile (2)	Acoustical Tile on Furring or 3/8" Gypsum: 3/4" Tile (3)
EXTERIOR SURFACE	**SHEATHING**											
Asphalt Shingles, (2)	Bldg paper on 5/16" plywood (2)	.51	.27	.30	.32	.29	.29	.28	.22	.17	.23	.21
	Bldg paper on 25/32" wood sheathing (3)	.30	.23	.26	.27	.25	.25	.24	.20	.16	.21	.19
Asbestos-Cement Shingles (3) or Asphalt Roll Roofing (1)	Bldg paper on 5/16" plywood (2)	.59	.28	.34	.37	.33	.33	.31	.25	.18	.25	.22
	Bldg paper on 25/32" wood sheathing (3)	.45	.25	.29	.31	.28	.28	.27	.22	.17	.22	.20
Slates (8) Tile (10) or Sheet Metal (1)	Bldg paper on 5/16" plywood (2)	.64	.29	.36	.38	.34	.35	.47	.26	.19	.26	.23
	Bldg paper on 25/32" wood sheathing (3)	.48	.25	.29	.31	.28	.28	.27	.22	.17	.23	.20
Wood Shingles (2)	Bldg paper on 1" x 4" strips (1)	.53	.26	.31	.33	.30	.30	.28	.23	.17	.24	.21
	Bldg paper on 5/16" plywood (2)	.41	.23	.27	.29	.26	.27	.25	.21	.16	.21	.19
	Bldg paper on 25/32" wood sheathing (3)	.34	.21	.24	.25	.23	.23	.22	.19	.15	.19	.17

1958 ASHAE Guide

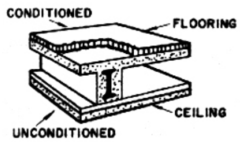
Equations: Summer (Heat Flow Down) Heat Gain, Btu/hr = (horizontal projected area, sq ft) × (U value) × (equivalent temp diff, *Table 20*).
Winter (Heat Flow Up) Heat Loss, Btu/hr = (horizontal projected area, sq ft) × (U value × 1.1) × (outdoor temp − inside temp).

*For addition of air spaces or insulation for above roofs, refer to *Table 31, page 75*.

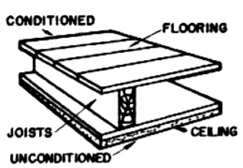
TABLE 29—TRANSMISSION COEFFICIENT U—CEILING AND FLOOR, (Heat Flow Up)

Based on Still Air Both Sides, Btu/(hr) (sq ft) (deg F temp diff)

All numbers in parentheses indicate weight per sq ft. Total weight per sq ft is sum of ceiling and floor.



FLOOR	CONCRETE SUBFLOOR	THICKNESS (inches) and WEIGHT (lb per sq ft)	MASONRY CEILING — Not Furred: None or ½" Sand Plaster (5)	Not Furred: ½" Lt Wt Plaster (3)	Suspended or Furred — Acoustical Tile Glued: ½" Tile (1)	Acoustical Tile Glued: ¾" Tile (1)	Metal Lath Plastered: ¾" Sand Plaster (7)	Metal Lath Plastered: ¾" Lt Wt Plaster (3)	⅜" Gypsum or Wood Lath Plastered: ½" Sand Plaster (5)	⅜" Gypsum or Wood Lath Plastered: ½" Lt Wt Plaster (2)	Insulating Board Plain or ½" Sand Agg Plastered: ½" Board (2)	Insulating Board Plain or ½" Sand Agg Plastered: 1" Board (4)	Acoustical Tile on Furring or ⅜" Gypsum: ½" Tile (1)	Acoustical Tile on Furring or ⅜" Gypsum: ¾" Tile (1)
None or ⅛" Linoleum or Floor Tile	Sand Agg	2 (19)	.70	.53	.38	.31	.43	.38	.44	.41	.26	.19	.28	.24
		4 (39)	.63	.49	.36	.30	.41	.36	.41	.38	.25	.18	.26	.23
		6 (59)	.57	.45	.34	.28	.38	.34	.39	.36	.24	.18	.25	.22
		8 (79)	.52	.42	.32	.27	.36	.32	.37	.34	.23	.17	.24	.21
		10 (99)	.48	.39	.31	.26	.34	.31	.35	.32	.23	.17	.23	.21
	Lt Wt Agg 80 lb/ft³	2 (15)	.48	.39	.31	.26	.34	.31	.35	.32	.23	.17	.23	.21
		4 (28)	.35	.30	.25	.22	.27	.25	.27	.26	.19	.15	.20	.18
		6 (41)	.27	.24	.21	.18	.22	.21	.22	.21	.17	.13	.17	.15
1 3/16" Wood Block on Slab	Sand Agg	2 (20)	.47	.39	.30	.26	.33	.30	.33	.40	.22	.17	.23	.20
		4 (40)	.44	.36	.29	.25	.31	.28	.32	.38	.22	.16	.22	.20
		6 (60)	.41	.34	.28	.24	.30	.27	.30	.36	.21	.16	.22	.19
		8 (80)	.38	.33	.26	.23	.28	.26	.29	.34	.20	.15	.21	.19
		10 (100)	.36	.31	.25	.22	.27	.25	.27	.32	.19	.15	.20	.18
	Lt Wt Agg 80 lb/ft³	2 (16)	.36	.31	.25	.22	.27	.25	.27	.32	.19	.15	.20	.18
		4 (29)	.28	.25	.21	.19	.22	.21	.23	.26	.17	.13	.17	.16
		6 (42)	.23	.21	.18	.16	.19	.18	.19	.21	.15	.12	.15	.14
Floor Tile or ⅛" Linoleum on ⅝" Plywood on 2" x 2" Sleepers	Sand Agg	2 (22)	.32	.28	.23	.21	.31	.28	.32	.30	.18	.14	.18	.17
		4 (42)	.31	.27	.23	.20	.30	.27	.30	.28	.18	.14	.18	.17
		6 (62)	.29	.26	.22	.19	.28	.26	.29	.27	.17	.14	.18	.16
		8 (82)	.28	.25	.21	.19	.27	.25	.27	.26	.17	.13	.17	.16
		10 (102)	.27	.24	.20	.18	.26	.24	.26	.25	.16	.13	.17	.15
	Lt Wt Agg 80 lb/ft³	2 (19)	.27	.24	.20	.18	.26	.24	.26	.25	.16	.13	.17	.15
		4 (31)	.22	.20	.17	.16	.22	.20	.22	.21	.14	.12	.15	.14
		6 (44)	.19	.17	.15	.14	.18	.17	.19	.18	.13	.11	.13	.12
¾" Hardwood on 25/32" Subfloor on 2" x 2" Sleepers	Sand Agg	2 (24)	.26	.23	.20	.18	.25	.23	.25	.24	.16	.13	.16	.15
		4 (44)	.25	.22	.19	.17	.24	.22	.24	.23	.16	.13	.16	.15
		6 (64)	.24	.21	.19	.17	.23	.21	.23	.22	.15	.12	.16	.14
		8 (84)	.23	.21	.18	.16	.22	.21	.22	.21	.15	.12	.15	.14
		10 (104)	.22	.20	.17	.16	.21	.20	.22	.21	.14	.12	.15	.14
	Lt Wt Agg 80 lb/ft³	2 (20)	.22	.20	.17	.16	.21	.20	.22	.21	.14	.12	.15	.14
		4 (33)	.19	.17	.15	.14	.18	.17	.18	.18	.13	.11	.13	.12
		6 (46)	.16	.15	.14	.13	.16	.15	.16	.16	.12	.099	.12	.11



FLOOR	SUBFLOOR	FRAME CONSTRUCTION CEILING — Not Furred: None	Suspended or Furred — Acoustical Tile Glued: ½" Tile (1)	Acoustical Tile Glued: ¾" Tile (1)	Metal Lath Plastered: ¾" Sand Plaster (7)	Metal Lath Plastered: ¾" Lt Wt Plaster (3)	⅜" Gypsum or Wood Lath Plastered: ½" Sand Plaster (5)	⅜" Gypsum or Wood Lath Plastered: ½" Lt Wt Plaster (2)	Insulating Board Plain or ½" Sand Agg Plastered: ½" Board (2)	Insulating Board Plain or ½" Sand Agg Plastered: 1" Board (4)	Acoustical Tile on Furring or ⅜" Gypsum: ½" Tile (1)	Acoustical Tile on Furring or ⅜" Gypsum: ¾" Tile (1)
None	None				.74	.59	.61	.54	.37	.24	.39	.31
	25/32" Wood (2)	.45	.30	.26	.31	.28	.29	.27	.22	.17	.23	.20
	2" Wood (5)	.27	.20	.18	.22	.20	.20	.19	.17	.14	.17	.15
½" Ceramic Tile on 1½" Cement	25/32" Wood (21)	.38	.21	.19	.28	.26	.26	.24	.20	.16	.21	.19
	2" Wood (24)	.24	.19	.17	.20	.19	.19	.18	.16	.13	.16	.15
¾" Hardwood Floor or Linoleum on ⅝" Plywood	25/32" Wood (5)	.33	.24	.21	.25	.23	.23	.22	.18	.15	.19	.17
	2" Wood (7)	.22	.17	.16	.18	.17	.17	.17	.15	.12	.15	.14
⅛" Linoleum on ¼" Hardboard on ⅜" Insulating Board	25/32" Wood (5)	.28	.21	.19	.22	.20	.21	.20	.17	.14	.18	.16
	2" Wood (8)	.20	.16	.15	.17	.16	.16	.16	.14	.12	.14	.13

1958 ASHAE Guide

Equations: Heat flow up, Unconditioned space below: Heat Gain, Btu/hr = (Area, sq ft) × (U value) × (outdoor temp − inside temp − 5 F).

Kitchen or boiler room below: Heat Gain, Btu/hr = (Area, sq ft) × (U value) × (actual temp diff, or outdoor temp − inside temp + 15 F to 25 F).

TABLE 31—TRANSMISSION COEFFICIENT U—WITH INSULATION & AIR SPACES

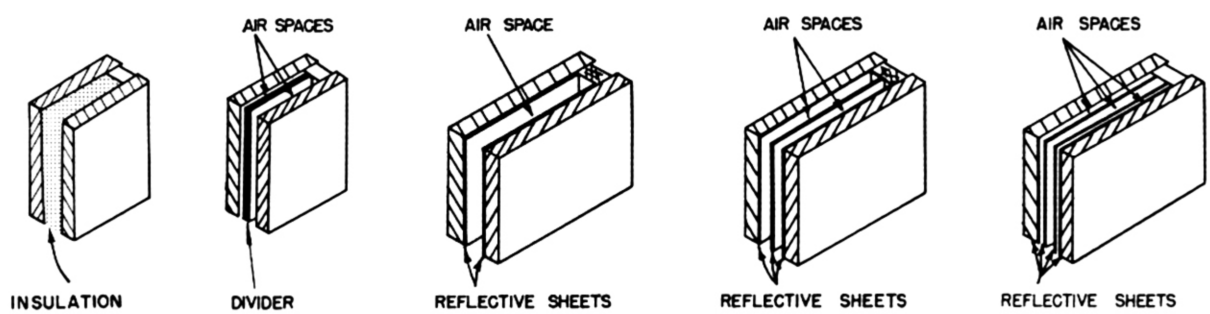
SUMMER AND WINTER

Btu/(hr) (sq ft) (deg F temp diff)

U Value Before Adding Insul. Wall, Ceiling, Roof Floor	Addition of Fibrous Insulation, Thickness (Inches)			Add'n of Air Space ¾" or more *	Addition of Reflective Sheets to Air Space (Aluminum Foil Average Emissivity = .05) — Direction of Heat Flow								
					Winter and Summer Horizontal			Summer Down			Winter Up		
	1	2	3		Added to one or both sides	One sheet in air space	Two sheets in air space	Added to one or both sides	One sheet in air space	Two sheets in air space	Added to one or both sides	One sheet in air space	Two sheets in air space
.60	.19	.11	.08	.38	.34	.18	.11	.12	.06	.05	.36	.20	.14
.58	.19	.11	.08	.37	.33	.18	.11	.12	.06	.05	.36	.20	.14
.56	.18	.11	.08	.36	.32	.18	.11	.11	.06	.05	.35	.20	.14
.54	.18	.11	.08	.36	.31	.17	.11	.11	.06	.05	.34	.19	.14
.52	.18	.11	.08	.35	.30	.17	.10	.11	.06	.05	.33	.19	.14
.50	.18	.11	.08	.34	.29	.17	.10	.11	.06	.05	.32	.19	.13
.48	.17	.11	.08	.33	.28	.16	.10	.11	.06	.04	.31	.18	.13
.46	.17	.10	.08	.32	.28	.16	.10	.11	.06	.04	.30	.18	.13
.44	.17	.10	.07	.31	.27	.16	.10	.11	.06	.04	.29	.18	.13
.42	.16	.10	.07	.30	.26	.15	.10	.11	.06	.04	.28	.17	.13
.40	.16	.10	.07	.29	.26	.15	.10	.10	.06	.04	.27	.17	.12
.38	.16	.10	.07	.28	.25	.15	.09	.10	.06	.04	.26	.17	.12
.36	.15	.10	.07	.27	.24	.14	.09	.10	.06	.04	.25	.16	.12
.34	.15	.10	.07	.26	.23	.14	.09	.10	.06	.04	.24	.16	.12
.32	.15	.10	.07	.25	.22	.13	.09	.10	.05	.04	.23	.15	.11
.30	.14	.09	.07	.23	.21	.13	.09	.10	.05	.04	.22	.15	.11
.28	.14	.09	.07	.22	.20	.13	.08	.09	.05	.04	.20	.14	.10
.26	.13	.09	.07	.21	.19	.12	.08	.09	.05	.04	.19	.13	.10
.24	.13	.09	.07	.20	.17	.12	.08	.09	.05	.04	.18	.13	.10
.22	.12	.08	.06	.18	.16	.11	.08	.08	.05	.04	.16	.12	.09
.20	.12	.08	.06	.17	.15	.10	.07	.08	.05	.04	.15	.11	.09
.18	.11	.08	.06	.15	.14	.10	.07	.08	.05	.04	.14	.11	.08
.16	.10	.07	.06	.14	.12	.09	.07	.07	.05	.04	.13	.10	.08
.14	.09	.07	.05	.12	.11	.08	.06	.07	.04	.04	.12	.09	.07
.12	.08	.06	.05	.11	.10	.08	.06	.06	.04	.03	.10	.08	.07
.10	.07	.06	.05	.09	.08	.07	.05	.06	.04	.03	.09	.07	.06

1958 ASHAE Guide

Insulation Added	Air Space Added	Reflective Sheets Added to One or Both Sides	Reflective Sheet in Air Space	Reflective Sheets in Air Space



*Checked for summer conditions for up, down and horizontal heat flow. Error from above values is less than 1%.

TABLE 32-TRANSMISSION COEFFICIENT U-FLAT ROOFS WITH ROOF-DECK INSULATION
SUMMER AND WINTER
Btu/(hr) (sq ft) (deg F temp diff)

U VALUE OF ROOF BEFORE ADDING ROOF DECK INSULATION	Addition of Roof-Deck Insulation Thickness (in.)					
	½	1	1½	2	2½	3
.60	.33	.22	.17	.14	.12	.10
.50	.29	.21	.16	.14	.12	.10
.40	.26	.19	.15	.13	.11	.09
.35	.24	.18	.14	.12	.10	.09
.30	.21	.16	.13	.12	.10	.09
.25	.19	.15	.12	.11	.09	.08
.20	.16	.13	.11	.10	.09	.08
.15	.12	.11	.09	.08	.08	.07
.10	.09	.08	.07	.07	.06	.05

TABLE 33-TRANSMISSION COEFFICIENT U-WINDOWS, SKYLIGHTS, DOORS & GLASS BLOCK WALLS
Btu/(hr) (sq ft) (deg F temp diff)

GLASS											
	Vertical Glass							Horizontal Glass			
	Single	Double			Triple			Single		Double (1/4")	
Air Space Thickness (in.)		¼	½	¾ -4	¼	½	¾ -4	Summer	Winter	Summer	Winter
Without Storm Windows	1.13	0.61	0.55	0.53	0.41	0.36	0.34	0.86	1.40	0.50	0.70
With Storm Windows	0.54							0.43	0.64		

DOORS		
Nominal Thickness of Wood (inches)	U Exposed Door	U With Storm Door
1	0.69	0.35
1¼	0.59	0.32
1½	0.52	0.30
1¾	0.51	0.30
2	0.46	0.28
2½	0.38	0.25
3	0.33	0.23
Glass (3/4" Herculite)	1.05	0.43

HOLLOW GLASS BLOCK WALLS	
Description*	U
5¾×5¾×37/8" Thick—Nominal Size 6×6×4 (14)	0.60
7¾×7¾×37/8" Thick–Nominal Size 8×8×4 (14)	0.56
11¾×11¾×37/8" Thick—Nominal Size 12×12×4 (16)	0.52
7¾×7¾×37/8" Thick with glass fiber screen dividing the cavity (14)	0.48
11¾×11¾×37/8" Thick with glass fiber screen dividing the cavity (16)	0.44

Equation: Heat Gain or Loss, Btu/hr = (Area, sq ft) × (U value) × (outdoor temp – inside temp)

*Italicized numbers in parentheses indicate weight in lb per sq ft.

CALCULATION OF TRANSMISSION COEFFICIENT U

For types of construction not listed in *Tables 21 thru 33,* calculate the *U* value as follows:

1. Determine the resistance of each component of a given structure and also the inside and outdoor air surface films from *Table 34*.
2. Add these resistances together, $R = r_1 + r_2 + r_3 + \ldots\ldots r_n$
3. Take the resistances, $U = \frac{1}{R}$

Basis of Table 34
- Thermal Resistance R, Building and Insulating Materials

Table 34 was extracted from the 1958 ASHAE Guide and the column "weight per sq ft" added.

Use of Table 34
- Thermal Resistance R, Building and Insulating Materials

The thermal resistances for building materials are listed in two columns. One column lists the thermal resistance per inch thickness, based on conductivity, while the other column lists the thermal resistance for a given thickness or construction, based on conductance.

Example 6 – Calculation of U Value

Given:

A wall as per *Fig. 27*

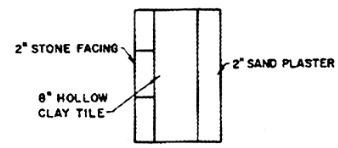


FIG. 27-OUTDOOR WALL

Find:

Transmission coefficient in summer.

Solution:

Refer to *Table 34*.

Construction	Resistance *R*
1. Outdoor air surface (7 ½ mph wind)	0.25
2. Stone facing, 2 in. (2 × .08)	0.16
3. Hollow clay tile, 8"	1.85
4. Sand aggregate plaster, 2 in. (2 × .20)	0.40
5. Inside air surface (still air)	0.68
Total Resistance	3.34

$$u = \frac{1}{R} = \frac{1}{3.34} = 0.30 \text{ Btu/(hr)(sq ft) (deg F)}$$

TABLE 34-THERMAL RESISTANCES R-BUILDING AND INSULATING MATERIALS

(deg F per Bu)/(hr) (sq ft)

MATERIAL	DESCRIPTION	THICK-NESS (in.)	DENSITY (lb per cu ft)	WEIGHT (lb per sq ft)	RESISTANCE R Per Inch Thickness 1/K	RESISTANCE R For Listed Thickness 1/C
	BUILDING MATERIALS					
BUILDING BOARD Boards, Panels, Sheathing, etc	Asbestos-Cement Board		120	-	0.25	-
	Asbestos-Cement Board	1/8	120	1.25	-	0.03
	Gypsum or Plaster Board	3/8	50	1.58	-	0.32
	Gypsum or Plaster Board	1/2	50	2.08	-	0.45
	Plywood		34	-	1.25	-
	Plywood	1/4	34	0.71	-	0.31
	Plywood	3/8	34	1.06	-	0.47
	Plywood	1/2	34	1.42	-	0.63
	Plywood or Wood Panels	3/4	34	2.13	-	0.94
	Wood Fiber Board, Laminated or Homogeneous		26	-	2.38	-
			31	-	2.00	-
	Wood Fiber, Hardboard Type		65	-	0.72	-
	Wood Fiber, Hardboard Type	1/4	65	1.35	-	0.18
	Wood, Fir or Pine Sheathing	25/32	32	2.08	-	0.98
	Wood, Fir or Pine	1 5/8	32	4.34	-	2.03
BUILDING PAPER	Vapor Permeable Felt		-	-	-	0.06
	Vapor Seal, 2 layers of Mopped 15 lb felt		-	-	-	0.12
	Vapor Seal, Plastic Film		-	-	-	Negl
WOODS	Maple, Oak, and Similar Hardwoods		45	-	0.91	-
	Fir, Pine, and Similar Softwoods		32	-	1.25	-
MASONRY UNITS	Brick, Common	4	120	40	-	.80
	Brick, Face	4	130	43	-	.44
	Clay Tile, Hollow:					
	1 Cell Deep	3	60	15	-	0.80
	1 Cell Deep	4	48	16	-	1.11
	2 Cells Deep	6	50	25	-	1.52
	2 Cells Deep	8	45	30	-	1.85
	2 Cells Deep	10	42	35	-	2.22
	3 Cells Deep	12	40	40	-	2.50
	Concrete Blocks, Three Oval Core Sand & Gravel Aggregate	3	76	19	-	0.40
		4	69	23	-	0.71
		6	64	32	-	0.91
		8	64	43	-	1.11
		12	63	63	-	1.28
	Cinder Aggregate	3	68	17	-	0.86
		4	60	20	-	1.11
		6	54	27	-	1.50
		8	56	37	-	1.72
		12	53	53	-	1.89
	Lightweight Aggregate (Expanded Shale, Clay, Slate or Slag; Pumice)	3	60	15	-	1.27
		4	52	17	-	1.50
		8	48	32	-	2.00
		12	43	43	-	2.27
	Gypsum Partition Tile:					
	3"×12"×30" solid	3	45	11	-	1.26
	3"×12"×30" 4-cell	3	35	9	-	1.35
	4"×12"×30" 3-cell	4	38	13	-	1.67
	Stone, Line or Sand		150	-	0.08	-

TABLE 34-THERMAL RESISTANCES R-BUILDING AND INSULATING MATERIALS (Contd)

(deg F per Bu)/(hr) (sq ft)

MATERIAL	DESCRIPTION	THICK-NESS (in.)	DENSITY (lb per cu ft)	WEIGHT (lb per sq ft)	RESISTANCE R Per Inch Thickness 1/K	RESISTANCE R For Listed Thickness 1/C
	BUILDING MATERIALS, (CONT.)					
MASONRY MATERIALS Concretes	Cement Mortar		116	-	0.20	-
	Gypsum-Fiber Concrete 87½ % gypsum, 12½ % wood chips		51	-	0.60	-
	Lightweight Aggregates		120	-	0.19	-
	Including Expanded		100	-	0.28	-
	Shale, Clay or Slate		80	-	0.40	-
	Expanded Slag; Cinders		60	-	0.59	-
	Pumice; Perlite; Vermiculite		40	-	0.86	-
	Also, Cellular Concretes		30	-	1.11	-
			20	-	1.43	-
	Sand & Gravel or Stone Aggregate (Oven Dried)		140	-	0.11	-
	Sand & Gravel or Stone Aggregate (Not Dried)		140	-	0.08	-
	Stucco		116	-	0.20	-
PLASTERING MATERIALS	Cement Plaster, Sand Aggregate		116	-	0.20	-
	Sand Aggregate	1/2	116	4.8	-	0.10
	Sand Aggregate	3/4	116	7.2	-	0.15
	Gypsum Plasten:					
	Lightweight Aggregate	1/2	45	1.88	-	0.32
	Lightweight Aggregate	5/8	45	2.34	-	0.39
	Lightweight Aggregate on Metal Lath	3/4	45	2.80	-	0.47
	Perlite Aggregate		45	-	0.67	-
	Sand Aggregate		105	-	0.18	-
	Sand Aggregate	1/2	105	4.4	-	0.09
	Sand Aggregate	5/8	105	5.5	-	0.11
	Sand Aggregate on Metal Lath	3/4	105	6.6	-	0.13
	Sand Aggregate on Wood Lath		105	-	-	0.40
	Vermiculite Aggregate		45	-	0.59	-
ROOFING	Asbestos-Cement Shingles		120	-	-	0.21
	Asphalt Roll Roofing		70	-	-	0.15
	Asphalt Shingles		70	-	-	0.44
	Built-up Roofing	3/8	70	2.2	-	0.33
	Slate	1/2	201	8.4	-	0.05
	Sheet Metal		-	-	Negl	-
	Wood Shingles		40	-	-	0.94
SIDING MATERIALS (On Flat Surface)	Shingles					
	Wood, 16", 7½ " exposure		-	-	-	0.87
	Wood, Double, 16", 12" exposure		-	-	-	1.19
	Wood, Plus Insul Backer Board, 5/16"		-	-	-	1.40
	Siding					
	Asbestos-Cement, ¼" lapped		-	-	-	0.21
	Asphalt Roll Siding		-	-	-	0.15
	Asphalt Insul Siding, ½" Board		-	-	-	1.45
	Wood, Drop, 1"×8"		-	-	-	0.79
	Wood, Bevel, ½"×8", lapped		-	-	-	0.81
	Wood, Bevel, ¾×"10", lapped		-	-	-	1.05
	Wood, Plywood, 3/8", lapped		-	-	-	0.59
	Structural Glass		-	-	-	0.10
FLOORING MATERIALS	Asphalt Tile	1/8	120	1.25	-	0.04
	Carpet and Fibrous Pad		-	-	-	2.08
	Carpet and Rubber Pad	1	-	-	-	1.23
	Ceramic Tile		-	-	-	0.08
	Cork Tile	1/8	25	-	2.22	-
	Cork Tile		25	0.26	-	0.28
	Felt, Flooring		-	-	-	0.06
	Floor Tile	1/8	-	-	-	0.05
	Linoleum	1/8	80	0.83	-	0.08
	Plywood Subfloor	5/8	34	1.77	-	0.78
	Rubber or Plastic Tile	1/8	110	1.15	-	0.02
	Terrazzo	1	140	11.7	-	0.08
	Wood Subfloor	25/32	32	2.08	-	0.98
	Wood, Hardwood Finish	3/4	45	2.81	-	0.68

TABLE 34-THERMAL RESISTANCES R-BUILDING AND INSULATING MATERIALS (Contd)

(deg F per Bu)/(hr) (sq ft)

MATERIAL	DESCRIPTION	THICK-NESS (in.)	DENSITY (lb per cu ft)	WEIGHT (lb per sq ft)	RESISTANCE R Per Inch Thickness $\frac{1}{K}$	RESISTANCE R For Listed Thickness $\frac{1}{C}$
INSULATING MATERIALS						
BLANKET AND BATT*	Cotton Fiber		0.8 - 2.0	-	3.85	-
	Mineral Wool, Fibrous Form Processed From Rock, Slag, or Glass		1.5 – 4.0	-	3.70	-
	Wood Fiber		3.2 – 3.6	-	4.00	-
	Wood Fiber, Milti-layer Stitched Expanded		1.5 – 2.0	-	3.70	-
BOARD AND SLABS	Glass Fiber		9.5	-	4.00	-
	Wood or Cane Fiber					
	Acoustical Tile	1/2	22.4	.93	-	1.19
	Acoustical Tile	3/4	22.4	1.4	-	1.78
	Interior Finish (Tile, Lath, Plank)		15.0	-	2.86	-
	Interior Finish (Tile, Lath, Plank)	1/2	15.0	0.62	-	1.43
	Roof Deck Slab					
	Sheathing (Impreg or Coated)		20.0	-	2.63	-
	Sheathing (Impreg or Coated)	1/2	20.0	0.83	-	1.32
	Sheathing (Impreg or Coated)	25/32	20.0	1.31	-	2.06
	Cellular Glass		9.0	-	2.50	-
	Cork Board (Without Added Binder)		6.5 – 8.0	-	3.70	-
	Hog Hair (With Asphalt Binder)		8.5	-	3.00	-
	Plastic (Foamed)		1.62	-	3.45	-
	Wood Shredded (Cemented in Preformed Slabs)		22.0	-	1.82	-
LOOSE FILL	Macerated Paper or Pulp Products		2.5 – 3.5	-	3.57	-
	Wood Fiber: Redwood, Hemlock, or Fir		2.0 – 3.5	-	3.33	-
	Mineral Wool (Glass, Slag, or Rock)		2.0 – 5.0	-	3.33	-
	Sawdust or Shavings		8.0 – 15.0	-	2.22	-
	Vermiculite (Expanded)		7.0	-	2.08	-
ROOF INSULATION	All Types					
	Preformed, for use above deck					
	Approximately	1/2	15.6	.7	-	1.39
	Approximately	1	15.6	1.3	-	2.78
	Approximately	1 1/2	15.6	1.9	-	4.17
	Approximately	2	15.6	2.6	-	5.26
	Approximately	2 1/2	15.6	3.2	-	6.67
	Approximately	3	15.6	3.9	-	8.33
AIR						
AIR SPACES	POSITION — HEAT FLOW					
	Horizontal — Up (Winter)	¾ -4	-	-	-	0.85
	Horizontal — Up (Summer)	¾ -4	-	-	-	0.78
	Horizontal — Down (Winter)	¾	-	-	-	1.02
	Horizontal — Down (Winter)	1 ½	-	-	-	1.15
	Horizontal — Down (Winter)	4	-	-	-	1.23
	Horizontal — Down (Winter)	8	-	-	-	1.25
	Horizontal — Down (Summer)	¾	-	-	-	0.85
	Horizontal — Down (Summer)	1½	-	-	-	0.93
	Horizontal — Down (Summer)	4	-	-	-	0.99
	Sloping 45° — Up (Winter)	¾ -4	-	-	-	0.90
	Sloping 45° — Down (Summer)	¾ -4	-	-	-	0.89
	Vertical — Horiz. (Winter)	¾ -4	-	-	-	0.97
	Vertical — Horiz. (Summer)	¾ -4	-	-	-	0.86
AIR FILM	POSITION — HEAT FLOW					
	Horizontal — Up		-	-	-	0.61
	Sloping 45° — Up		-	-	-	0.62
Still Air	Vertical — Horizontal		-	-	-	0.68
	Sloping 45° — Down		-	-	-	0.76
	Horizontal — Down		-	-	-	0.92
15 Mph Wind	Any Position (For Winter) — Any Direction		-	-	-	0.17
7½ Mph Wind	Any Position (For Summer) — Any Direction		-	-	-	0.25

*Includes paper backing and facing if and. In cases where the insulation froms a boundary (highjly refiective) of on air space, refer to *Table 31, page 75*

HEAT LOSS THRU BASEMENT WALLS AND FLOORS BELOW THE GROUND LEVEL

The loss through the floor is normally small and relatively constant year round because the ground temperature under the floor varies only a little throughout the year. The ground is a very good heat sink and can absorb or lose a large amount of heat without an appreciable change in temperature at about the 8 ft level. Above the 8 ft level, the ground temperature varies with the outdoor temperature, with the greatest variation at the surface and a decreasing variation down to the 8 ft depth. The heat loss thru a basement wall may be appreciable and it is difficult to calculate because the ground temperature varies with depth. *Tables 35 thru 37* have been empirically calculated to simplify the evaluation of heat loss thru basement walls and floors.

The heat loss thru a slab floor is large around the perimeter and small in the center. This is because the ground temperature around the perimeter varies with the outdoor temperature, whereas the ground temperature in the middle remains relatively constant, as with basement floors.

Basis of Tables 35 thru 37
- Heat Loss thru Masonry Floors and Walls in Ground

Tables 35 thru 37 are based on empirical data. The perimeter factors listed in *Table 36* were developed by calculating the heat transmitted for each foot of wall to an 8 ft depth. The ground was assumed to decrease the transmission coefficient, thus adding resistance between the wall and the outdoor air. The transmission coefficients were then added to arrive at the perimeter factors.

Use of Tables 35 thru 37
- Heat Loss thru Masonry Floors and Walls in Ground

The transmission coefficients listed in *Table 35* may be used for any thickness of uninsulated masonry floors where there is good contact between the floor and the ground.

The perimeter factors listed in *Table 36* are used for estimating heat loss thru basement walls and the outside strip of basement floors. This factor can be used only when the space is heated continuously. If there is only occasional heating, calculate the heat loss using the wall or floor transmission coefficients as listed in *Tables 21 thru 33* and the temperature difference between the basement and outdoor air or ground as listed in *Table 37.*

The heat loss in a basement is determined by adding the heat transferred thru the floor, the walls and the outside strip of the floor and the portion of the wall above the ground level.

Example 7- Heat Loss in a Basement

Given:

Basement-100'×40'×9'
Basement temp-65 F db, heated continuously
Outdoor temp-0° F db
Grade line-6 ft above basemen floor
Walls and floors-12 in. concrete (80 lb/cu ft)

Find:

Heat loss from basement

Solution:

1. Heat loss above ground
$H = UA_1(t_b - t_{oa})$
$= 0.18 \times (200+80) \times 3 \times (65\text{-}0) = 9828$ Btu/hr
2. Heat loss thru walls and outside strip of floor below ground.
$H = L_pQ\ (t_b - t_{oa})$
$= (200+80) \times 1.05 \times (65\text{-}0) =$ 19,100 Btu/hr
3. Heat loss thru floor
$H = UA_2\ (t_b - t_g)$
$= 0.05 \times (100 \times 40) \times (65\text{-}55) =$ 2000 Btu/hr
= 30,928 Btu/hr

Total Heat Loss

where U = Heat transmission coefficient of wall above ground (*Table 21*) and floor (*Table 35*) in Btu/(hr) (sq ft) (deg F)
A_1 = Area of wall above ground, sq ft
A_2 = Entire floor area, sq ft
L_p = Perimeter of wall, ft
Q = Perimeter factor (*Table 36*)
t_b = Basement dry-bulb temp, F
t_g = Ground temp, F, (*Table 37*)
t_{oa} = Outdoor design dry-bulb temp, F

TABLE 35-TRANSMISSION COEFFICIENT U-MASONRY FLOORS AND WALLS IN GROUND
(Use only in conjunction with *Table 36*)

Floor or Wall	Transmission Coefficient U Btu/(hr) (sq ft) (deg f)
*Basement Floor	.05
Portion of Wall exceeding 8 feet below ground level	.08

*Some additional floor loss is included in perimeter factor, see *Table 36.*

Equations:

Heat loss through floor, Btu/hr = (area of floor, sq ft) × (U value) × (basement-ground temp).

Heat loss through wall below 8 foot line, Btu/hr = (area of wall below 8 ft line, sq ft) × (U value) × (basement-ground temp).

NOTE: The factors in Tables 35 and 36 may be used for any thickness of uninsulated masonry wall or floor, but there must be a good contact (no air space which may connect to the outdoors) between the ground and the floor or wall. Where the ground is dry and sandy, or where there is cinder fill along wall or where the wall has a low heat transmission coefficient, the perimeter factor may be reduced slightly.

TABLE 36-PERIMETER FACTORS
FOR ESTIMATING HEAT LOSS THROUGH BASEMENT WALLS
AND OUTSIDE STRIP OF BASEMENT FLOOR
(Use only in conjunction with *Table 35*)

Distance of Floor From Ground Level	Perimeter Factor (q)
2 Feet above	.90
At ground level	.60
2 Feet below	.75
4 Feet below	.90
6 Feet below	1.05
8 Feet below	1.20

Equations:

Heat loss about perimeter, Btu/hr = (perimeter of wall, ft) × (perimeter factor) × (basement-outdoor temp).

TABLE 37-GROUND TEMPERATURES
FOR ESTIMATING HEAT LOSS THROUGH BASEMENT FLOORS

Outdoor Design Temp (F)	-30	-20	-10	0	+10	+20
Ground Temp (F)	40	45	50	55	60	65

TRANSMISSION COEFFICIENTS-PIPES IN WATER OR BRINE

Heat transmission coefficients for copper and steel pipes are listed in *Tables 38 and 39.* These coefficients may be useful in applications such as cold water or brine storage systems and ice skating rinks.

Basis of Tables 38 and 39
- Transmission coefficients, Pipes in Water or Brine

Table 38 is for ice coated pipes in water, based on a heat transfer film coefficient, inside the pipe, of 150 Btu/(hr)(sq ft internal pipe surface)(deg F).

Table 39 is for pipes in water or brine based on a heat transfer of 18 Btu/(hr)(sq ft external pipe surface) (deg F) in water, 14 Btu in brine. It is also based on a low rate of circulation on the outside of the pipe and 10 F to 15 F temperature difference between water or brine and refrigerant. High rates of circulation will increase the heat transfer rate. For special problems, consult heat transfer reference books.

TABLE 38- TRANSMISSION COEFFICIENT U-ICE COATED PIPES IN WATER
Btu/(hr) (lineal ft pipe) (deg F between 32 F db and refrig temp)

Copper Pipe Size (Inches O.D.)	Copper Pipe With Ice Thickness (Inches)				Steel Pipe Size Nominal (Inches)	Steel Pipe With Ice Thickness (Inches)				
	1/2	1	1 1/2	2		1/2	1	1 1/2	2	3
5/8	6.1	4.5	3.8	3.4	1/2	7.2	5.2	4.4	3.9	3.4
3/4	7.1	5.1	4.2	3.8	3/4	8.7	6.1	5.1	4.5	3.8
7/8	8.0	5.7	4.7	4.1	1	10.6	7.2	5.8	5.1	4.2
1 1/8	9.8	6.7	5.4	4.7	1 1/2	13.0	8.6	6.8	5.9	4.8

TABLE 39- TRANSMISSION COEFFICIENT U-PIPES IMMERSED IN WATER OR BRINE
Btu/(hr) (lineal ft pipe) (deg F between 32 F db and refrig temp)
Outside water film coefficient = 18 Btu/(hr) (sq ft) (deg F)
Outside brine film coefficient = 14 Btu/(hr) (sq ft) (deg F)
Water refrigerant temp = 10 F to 15 F

Copper Pipe Size (Inches O.D.)	Pipes in Water	Steel Pipe Nominal Size (Inches)	Pipes in Water	Pipes in Brine
1/2	2.4	1/2	4.0	3.1
5/8	2.9	3/4	5.0	3.9
3/4	3.5	1	6.2	4.8
1 1/8	5.3	1 1/4	7.8	6.1

WATER VAPOR FLOW THRU BUILDING STRUCTURES

Water vapor flows thru building structures, resulting in a latent load whenever a vapor pressure difference exists across a structure. The latent load from this source is usually insignificant in comfort applications and need be considered only in low or high dewpoint applications.

Water vapor flows from high to lower vapor pressure at a rate determined by the permeability of the structure. This process is quite similar to heat flow, except that there is transfer of mass with water vapor flow. As heat flow can be reduced by adding insulation, vapor flow can be reduced by vapor barriers. The vapor barrier may be paint (aluminum or asphalt), aluminum foil or galvanized iron. *It should always be placed on the side of a structure having the higher vapor pressure, to prevent the water vapor from flowing up to the barrier and condensing within the wall.*

Basis of Table 40
- Water Vapor Transmission thru Various Materials

The values for walls, floors, ceilings and partitions have been estimated from the source references listed in the bibliography. The resistance of a homogeneous material to water vapor transmission has been assumed to be directly proportional to the thickness, and it also has been assumed that there is no surface resistance to water vapor flow. The values for permeability of miscellaneous materials are based on test results.

NOTE: Some of the values for walls, roofs, etc., have been increased by a safety factor because conclusive data is not available.

Use of Table 40
- Water Vapor Transmission thru Various Materials

Table 40 is used to determine latent heat gain from water vapor transmission thru building structures in the high and low dewpoint applications where the air moisture content must be maintained.

Example 8 – Water Vapor Transmission

Given:

A 40 ft × 40 ft × 8 ft laboratory on second floor requiring inside design conditions of 40 F db, 50% rh, with the outdoor design conditions at 95 F db, 75 F wb. The outdoor wall is 12 inch brick with no windows. The partitions are metal lath and plaster on both sides of studs. Floor and ceiling are 4 inch concrete.

Find:

The latent heat gain from the water vapor transmission.

Solution:

Gr/lb at 95 F db, 75 F wb = 99 (psych chart)

Gr/lb at 40 F db, 50% rh = 18 (psych chart)

Moisture content difference = 81 gr/lb

Assume that the dewpoint in the areas surrounding the laboratory is uniform and equal to the outdoor dewpoint.

Latent heat gain:

$$\text{Outdoor wall} = \frac{40\text{x}8}{100}\text{x } 81 \text{ x } .04 \quad \text{(Table 40.)} = 10.4 \text{ Btu/hr}$$

$$\text{Floor and ceilings} = 2\text{x}\,\frac{40\text{x}40}{100}\text{ x } 81 \text{ x } .10 = 259 \text{ Btu/hr}$$

$$\text{Partitions} = 3\text{x}\,\frac{40\text{x}8}{100}\text{ x}81\text{x}1.0 = \underline{777 \text{ Btu/hr}}$$

Total Latent Heat Gain = 1046.4 Btu/hr

TABLE 40- WATER VAPOR TRANSMISSION THRU VARIOUS MATERIALS

DESCRIPTION OF MATERIAL OR CONSTRUCTION	PERMEANCE Btu/(hr) (100 sq ft) (gr/lb diff) latent heat		
	No Vapor Seal Unless Noted Under Description	With 2 Coats Vapor-seal Paint on Smooth Inside Surface*	With Aluminum Foil Mounted on One Side of Paper Cemented to Wall†
WALLS			
Brick -- 4 inches	.12	.075	.024
-- 8 inches	.06	.046	.020
-- 12 inches	.04	.033	.017
-- per inch of thickness	.49	- -	- -
Concrete -- 6 inches	.067	.050	.021
-- 12 inches	.034	.029	.016
-- per inch of thickness	.40	- -	- -
Frame -- with plaster interior finish	.79	.16	.029
-- same with asphalt coated insulating board lath	.42	.14	.028
Tile—hollow clay (face, glazed)--4 inches	.013	.012	.0091
--hollow clay (common))--4 inches	.24	.11	.025
--hollow clay, 4 inch face and 4 inch common	.012	.011	.0086
CEILINGS AND FLOORS			
Concrete--4 inches	.10	.067	.023
--8 inches	.051	.040	.019
Plaster on wood or metal lath on joist—no flooring	2.0	.18	.030
Plaster on wood or metal lath on joist—flooring	.50	.14	.028
Plaster on wood or metal lath on joists—double flooring	.40	.13	.028
PARTITIONS			
Insulating Board ½ inch on both sides of studding	4.0	.19	.030
Wood or metal lath and plaster on both sides of studding	1.0	.17	.029
ROOFS			
Concrete--2 inches, plus 3 layer felt roofing	.02	.018	
--6 inches, plus 3 layer felt roofing	.02	.018	
Shingles, sheathing, rafters--plus plaster on wood or metal lath	1.5	.18	
Wood –1 inch, plus 3 layer felt roofing	.02	.018	
--2 inches, plus 3 layer felt roofing	.02	.018	
MISCELLANEOUS			
Air Space, still air 3 5/8 inch	3.6		
1 inch	13.0		
Building Materials			
Masonite--1 thickness, 1/8 inch	1.1	.17	.027
--5 thicknesses	.32		
Plaster on wood lath	1.1		
--plus 2 coats aluminum paint	- -	.12	
Plaster on gypsum lath	1.95	- -	
--ditto plus primer and 2 coats lead and oil paint	- -	.13	
Plywood--1/4 inch Douglas fir (3 ply)	.63		
--ditto plus 2 coats asphalt paint	- -	.087	
--ditto plus 2 coats aluminum paint	- -	.13	
--1/2 inch Douglas fir (5 ply)	.27		
--ditto plus 2 coats asphalt paint	- -	.041	
--ditto plus 2 coats aluminum paint	- -	.12	
Wood--Pine .508 inch	.33		
--ditto plus 2 coats aluminum paint	- -	.046	
--spruce, .508 inch	.20		
Insulating Materials			
Corkboard, 1 inch thick	.63		
Interior finish insulating board, ½"	5.0 – 7.0		
--ditto plus 2 coats water emulsion paint	3.0 – 4.0		
--ditto plus 2 coats varnish base paint	.1 – 1.0		
--ditto plus 2 coats lead and oil paint	.17		
--ditto plus wall linoleum	.03 - .06		

TABLE 40- WATER VAPOR TRANSMISSION THRU VARIOUS MATERIALS (Contd)

DESCRIPTION OF MATERIAL OR CONSTRUCTION	PERMEANCE Btu/(hr) (100 sq ft) (gr/lb diff) latent heat		
	No Vapor Seal Unless Noted Under Description	With 2 Coats Vapor-seal Paint on Smooth Inside Surface*	With Aluminum Foil Mounted on One Side of Paper Cemented to Wall†
MISCELLANEOUS			
Insulating Materials, cont.			
Insulating board lath	4.6 – 8.2		
--ditto plus ½" plaster	1.5		
--ditto plus ½" plaster, sealer, and flat coat of paint	.16 - .31		
Insulating board sheathing, 25/32"	2.6 – 6.1		
--ditto plus asphalt coating both sides	.046 – 1.0		
Mineral wool (3 5/8 inches thick), unprotected	3.5		
Packaging materials			
Cellophane, moisture proof	.01 – 0.25		
Glassine (1 ply waxed or 3 ply plain)	.0015 - .006		
Kraft paper soaked with parafin wax, 4.5 lbs per 100 sq ft	1.4 – 3.1		
Pliofilm	.01 - .025		
Paint Films			
2 coats aluminum paint, estimated	.05 - .2		
2 coats asphalt paint, estimated	.05 - .1		
2 coats lead and oil paint, estimated	.1 - .6		
2 coats water emulsion, estimated	5.0 – 8.0		
Papers			
Duplex or asphalt laminae (untreated) 30-30, 3.1 lb per 100 sq ft	.15 - .27		
--ditto 30-60-30, 4.2 lb per 100 sq ft	.051 - .091		
Draft paper--1 sheet	8.1		
--2 sheets	5.1		
--aluminum foil on one side of sheet	.016		
--aluminum foil on both sides of sheet	.012		
Sheathing paper			
Asphalt impregnated and coated, 7 lb per 100 sq ft	.02 - .10		
Slaters felt, 6 lb per 100 sq ft, 50% saturated with tar	1.4		
Roofing Felt, saturated and coated with asphalt			
25 lb. per sq ft	.015		
50 lb. per sq ft	.011		
Tin sheet with 4 holes 1/16 diameter	.17		
Crack 12 inches long by 1/32 inches wide (approximated from above)	5.2		

*Painted surfaces: Two coats of a good vapor seal paint on a smooth surface give a fair vapor barrier. More surface treatment is required on a rough surface than on a smooth surface. Data indicates that either asphalt or aluminum paint are good for vapor seals.

†Aluminum Foil on Paper: This material should also be applied over a smooth surface and joints lapped and sealed with asphalt.

The vapor barrier should always be placed on the side of the wall having the higher vapor pressure if condensation of moisture in wall is possible.

Application: The heat gain due to water vapor transmission through walls may be neglected for the normal air conditioning or refrigeration job. This latent gain should be considered for air conditioning jobs where there is a great vapor pressure difference between the room and the outside, particularly when the dewpoint inside must be low. **Note that moisture gain due to infiltration usually is of much greater magnitude than moisture transmission through building structures.**

Conversion Factors: To convert above table values to: grain/(hr) (sq ft) (inch mercury vapor pressure difference), multiply by 9.8.
grain/(hr) (sq ft) (pounds per sq inch vapor pressure difference), multiply by 20.0

To convert Btu latent heat to grains, multiply by 7000/1060 = 6.6.

CONDENSATION OF WATER VAPOR

Whenever there is a difference of temperature and pressure of water vapor across a structure, conditions may develop that lead to a condensation of moisture. This condensation occurs at the point of saturation temperature and pressure.

As water vapor flows thru the structure, its temperature decreases and, if at any point it reaches the dewpoint or saturation temperature, condensation begins. As condensation occurs, the vapor pressure decreases, thereby lowering the dewpoint or saturation temperature until it corresponds to the actual temperature. The rate at which condensation occurs is determined by the rate at which heat is removed from the point of condensation. As the vapor continues to condense, latent heat of condensation is released, causing the dry-bulb temperature of the material to rise.

To illustrate this, assume a frame wall with wood sheathing and shingles on the outside, plasterboard on the inside and fibrous insulation between the two. Also, assume that the inside conditions are 75 F db and 50% rh and the outdoor conditions are 0° F db and 80% rh. Refer to *Fig. 28*.

The temperature and vapor pressure gradient decreases approximately as shown by the solid and dashed lines until condensation begins (saturation point). At this point, the latent heat of condensation decreases the rate of temperature drop thru the insulation. This is approximately indicated by the dotted line.

Another cause of concealed condensation may be evaporation of water from the ground or damp locations. This water vapor may condense on the underside of the floor joints (usually near the edges where it is coldest) or may flow up thru the outdoor side of the walls because of stack effect and/or vapor pressure differences.

Concealed condensation may cause wood, iron and brickwork to deteriorate and insulation to lose its insulating value. These effects may be corrected by the following methods:

1. *Provide vapor barriers on the high vapor pressure side.*
2. In winter, ventilate the building to reduce the vapor pressure within. No great volume of air change is necessary, and normal infiltration alone is frequently all that is required.
3. In winter, ventilate the structure cavities to remove vapor that has entered. Outdoor air thru vents shielded from entrance of rain and insects may be used.

Condensation may also form on the surface of a building structure. Visible condensation occurs when the surface of any material is colder than the dewpoint temperature of the surrounding air. In winter, the condensation may collect on cold closet walls and attic roofs and is commonly observed as frost on window panes. *Fig. 29* illustrates the condensation on a window with inside winter design conditions of 70 F db and 40% rh. Point A represents the room conditions; point B, the dewpoint temperature of the thin film of water vapor adjacent to the window surface; and point C, the point at which frost or ice appears on the window.

Once the temperature drops below the dewpoint, the vapor pressure at the window surface is also reduced, thereby establishing a gradient of vapor pressure from the room air to the window surface. This gradient operates, in conjunction with the convective action within Tive action within the room, to move water vapor continuously to the window surface to be condensed, as long as the concentration of the water vapor is maintained in a space.

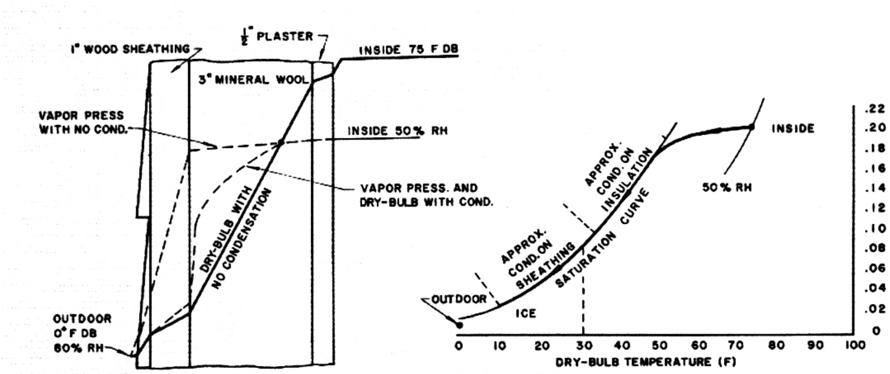


FIG. 28-CONDENSATION WITHIN FRAME WALL

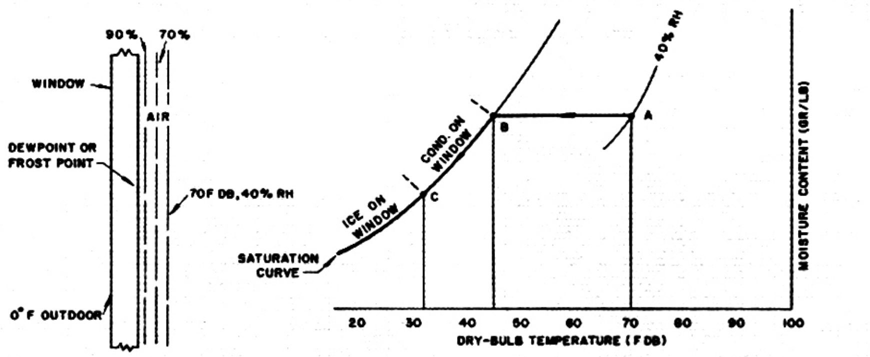


FIG. 29-CONDENSATION ON WINDOW SURFACE

Visible condensation is objectionable as it causes staining of surfaces, dripping on machinery and furnishings, and damage to materials in process of manufacture. Condensation of this type may be corrected by the following methods:

1. Increase the thermal resistance of walls, roofs and floors by adding insulation with vapor barriers to prevent condensation within the structures.
2. Increase the thermal resistance of glass by installing two or three panes with air space(s) between. In extreme cases, controlled heat, electric or other, may be applied between the glass of double glazed windows.
3. Maintain a room dewpoint lower than the lowest expected surface temperature in the room.
4. Decrease surface resistance by increasing the velocity of air passing over the surface. Decreasing the surface resistance increases the window surface temperature and brings it closer to the room dry-bulb temperature.

Basis of Chart 2

- Maximum Room RH; No Wall, Roof or Glass Condensation

Chart 2 has been calculated from the equation used to determine the maximum room dewpoint temperature that can exist with condensation.

$$t_{dp} = t_{rm} - \frac{U(t_{rm} - t_{oa})}{f_t}$$

where t_{dp} = dewpoint temp of room air, F db

t_{rm} = room temp, F

U = transmission coefficient, Btu/(hr)(sq ft)(deg F)

t_{oa} = outdoor temp, F

f_i = inside air film or surface conductance, Btu/(hr)(sq ft) (deg F)

Chart 2 is based upon a room dry-bulb temperature of 70 F db and an inside film conductance of 1.46 Btu/(hr)(sq ft) (deg F).

Use of Chart 2

- Maximum Room RH; No Wall, Roof or Glass Condensation

Chart 2 gives a rapid means of determining the maximum room relative humidity which can be maintained and yet avoid condensation with a 70 F db room.

Example 9-Moisture Condensation

Given:

12 in. stone wall with 5/8 in. sand aggregate plaster
Room temp – 70 F db
Outdoor temp - 0°F db

Find:

Maximum room rh without wall condensation.

Solution:

Transmission coefficient U = 0.52 Btu/(hr) (sq ft) (deg F)
(*Table 21, page 66*)

Maximum room rh = 40.05%, (*Chart 2*)

Corrections in room relative humidity for room temperatures other than 70 F db are listed in the table under Chart 2. Values other than those listed may be interpolated.

Example 10- Moisture Condensation

Given:

Same as *Example 9*, except room temp is 75 F db

Find:

Maximum room rh without wall condensation

Solution:

Transmission coefficient U = 0.52 Btu/(hr) (sq ft) (deg F)
(*Example 9*)

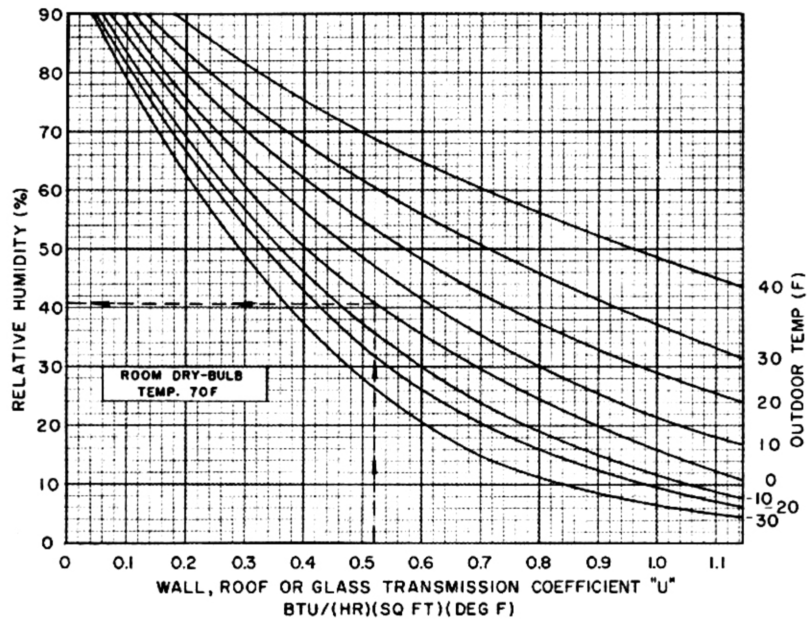
Maximum room rh for 70 F db room temp = 40.05%
(*Example 9*)

Rh correction for room temp of 75 F db with U factor of 0.52 = -1.57% (bottom *Chart 2*).

Maximum room rh = 40.05-1.57 = 38.48% or 38.5%

CHART 2—MAXIMUM ROOM RELATIVE HUMIDITY WITHOUT CONDENSATION

NO WALL, ROOF OR GLASS CONDENSATION



CORRECTION IN ROOM RH (%)

For Wall, Roof or Glass Transmission Coefficient U

Outdoor Temp (F db)	U = 1.1		U = .65		U = .35	
	Room Temp (F db)					
	60	80	60	80	60	80
-30	+1.0%	-1.0%	+1.5%	-2.0%	+2.5%	-2.0%
-20	+1.0	-1.5	+2.5	-2.5	+3.0	-2.0
-10	+2.0	-2.0	+3.5	-3.0	+3.0	-2.0
0	+3.5	-2.5	+4.0	-4.0	+3.5	-2.5
10	+5.0	-3.5	+5.0	-4.5	+4.0	-3.0
20	+7.0	-4.0	+6.5	-5.0	+4.5	-3.5
30	+9.0	-7.5	+8.5	-6.0	+5.0	-4.0
40	+12.0	-9.5	+9.5	-7.5	+6.0	-4.5

CHAPTER 6. INFILTRATION AND VENTILATION

The data in this chapter is based on ASHAE tests evaluating the infiltration and ventilation quantities of outdoor air. These outdoor air quantities normally have a different heat content than the air within the conditioned space and, therefore, impose a load on the air conditioning equipment.

In the case of infiltration, the load manifests itself directly within the conditioned space. The ventilation air, taken thru the conditioning apparatus, imposes a load both on the space thru apparatus bypass effect, and directly on the conditioning equipment.

The data in this chapter is based on ASHAE tests and years of practical experience.

INFILTRATION

Infiltration of air and particularly moisture into a conditioned space is frequently a source of sizable heat gain or loss. The quantity of infiltration air varies according to tightness of doors and windows, porosity of the building shell, height of the building, stairwells, elevators, direction and velocity of wind, and the amount of ventilation and exhaust air. Many of these cannot be accurately evaluated and must be based on the judgment of the estimator.

Generally, infiltration may be caused by wind velocity, or stack effort, or both:

1. *Wind Velocity*-The wind velocity builds up a pressure on the windward side of the building and a slight vacuum on the leeward side. The outdoor pressure build-up causes air to infiltrate thru crevices in the construction and cracks around the windows and doors. This, in turn, causes a slight build-up of pressure inside the building, resulting in an equal amount of exfiltration on the leeward side.
2. *Difference in Density or Stack Effect* – The variations in temperatures and humidities produce differences in density of air between inside and outside of the building. In tall buildings this density difference causes summer and winter infiltration and exfiltration as follows:

   Summer – Infiltration at the top and exfiltration at the bottom.

   Winter – Infiltration at the bottom and exfiltration at the top.

   This opposite direction flow balances at some neutral point near the mid-height of the building. Air flow thru the building openings increases proportionately between the neutral point and the top and the neutral point and bottom of the building. The infiltration from stack effect is greatly influenced by the height of the building and the presence of open stairways and elevators.

The combined infiltration from wind velocity and stack effect is proportional to the square root of the sum of the heads acting on it.

The increased air infiltration flow caused by stack effect is evaluated by converting the stack effect force to an equivalent wind velocity, and then calculating the flow from the wind velocity data in the tables.

In building over 100 ft tall, the equivalent wind velocity may be calculated from the following formula, assuming a temperature difference of 70 F db (winter) and a neutral point at the mid-height of the building:

$$V_e = \sqrt{V^2 - 1.75a} \quad \text{(for upper section of tall bldgs – winter)} \quad (1)$$

$$V_e = \sqrt{V^2 - 1.75b} \quad \text{(for lower section of tall bldgs – winter)} \quad (2)$$

where V_e = equivalent wind velocity, mph
V = wind velocity normally calculated for location, mph
a = distance window is above mid-height, ft
b = distance window is below mid-height, ft

NOTE: The total crackage is considered when calculating infiltration from stack effect.

INFILTRATION THRU WINDOWS AND DOORS, SUMMER

Infiltration during the summer is caused primarily by the wind velocity creating a pressure on the windward side. Stack effect is not normally a significant factor because the density difference is slight, (0.073 lb/cu ft at 75 F db, 50% rh and 0.070 lb/cu ft at 95 F db, 75 F wb). This small stack effect in tall buildings (over 100 ft) causes air to flow in the top and out the bottom. Therefore, the air infiltrating in the top of the building, because of the wind pressure, tends to flow down thru the building and out the doors on the street level, thereby offsetting some of the infiltration thru them.

In low buildings, air infiltrates thru open doors on the windward side unless sufficient outdoor air is introduced thru the air conditioning equipment to offset it; refer to "*Offsetting Infiltration with Outdoor Air*."

With *doors on opposite walls*, the infiltration can be considerable if the two are open at the same time.

Basis of Table 41
- Infiltration thru Windows and Doors, Summer

The data in *Tables 41a, b and c* is based on a wind velocity of 7.5 mph blowing directly at the window or door, and on observed crack widths around typical windows and doors. This data is derived from *Table 44* which lists infiltration thru cracks around windows and doors as established by ASHAE tests.

Table 41d shows values to be used for doors on opposite walls for various percentages of time that each door is open.

The data in *Table 41e* is based on actual tests of typical applications.

Use of Table 41
- Infiltration thru Windows and Doors, Summer

The data in Table 41 is used to determine the infiltration thru windows and doors on the windward side with the wind blowing directly at them. When the wind direction is oblique to the windows or doors, multiply the values in Tables 41a, b, c, d, by 0.60 and apply to total areas. For specific locations, adjust the values in Table 41 to the design wind velocity; refer to Table 1, page 10.

During the summer, infiltration is calculated for the windward side(s) only, because stack effect is small and, therefore, causes the infiltration air to flow in a downward direction in tall buildings (over 100 ft). Some of the air infiltrating thru the windows will exfiltrate thru the windows on the leeward side(s), while the remaining infiltration air flows out the doors, thus offsetting some of the infiltration thru the doors. To determine the net infiltration thru the doors, determine the infiltration thru the windows on the windward side, multiply this by .80, and subtract from the door infiltration. For low buildings the door infiltration on the windward side should be included in the estimate.

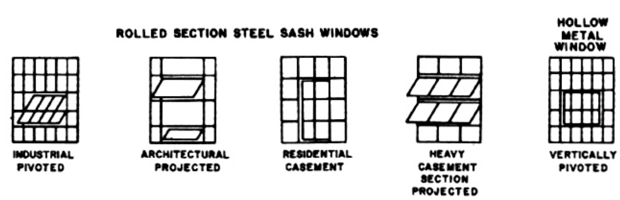
TABLE 41-INFILTRATION THRU WINDOWS AND DOORS-SUMMER*
7.5 mph Wind Velocity†

TABLE 41a-DOUBLE HUNG WINDOWS‡

DESCRIPTION	CFM PER SQ FT SASH AREA					
	Small-30X"72"			Large-54"X96"		
	No W-Strip	W-Strip	Storm Sash	No W-Strip	W-Strip	Storm Sash
Average Wood Sash	.43	.26	.22	.27	.17	.14
Poorly Fitted Wood Sash	1.20	.37	.60	.76	.24	.38
Metal Sash	.80	.35	.40	.51	.22	.25

TABLE 41b-CASEMENT TYPE WINDOWS ‡

DESCRIPTION	CFM PER SQ FT SASH AREA									
	Percent Openable Area									
	0%	25%	33%	40%	45%	50%	60%	66%	75%	100%
Rolled Section-Steel Sash										
Industrial Pivoted	.33	.72	-	.99	-	-	-	1.45	-	2.6
Architectural Projected	-	.39	-	-	-	.55	.74	-	-	-
Residential	-	-	.28	-	-	.49	-	-	-	6.3
Heavy Projected	-	-	-	-	.23	-	-	.32	.39	-
Hollow Metal-Vertically Pivoted	.27	.58	-	.82	-	-	-	1.2	-	2.2



REPRESENTATIVE TYPES OF WINDOWS
(VIEWED FROM OUTSIDE)

TABLE 41-INFILTRATION THRU WINDOWS AND DOORS-SUMMER* (Contd)

7.5 mph Wind Velocity†

Table 41c-DOORS ON ONE OR ADJACENT WALLS, CORNER ENTRANCES

DESCRIPTION	CFM PER SQ FT AREA**		CFM	
	No Use	Average Use	Standing Open	
			No Vestibute	Vestibule
Revolving Doors-Normal Operation	.8	5.2	-	-
Panels Open	-	-	1,200	900
Glass Door-3/4" Crack	4.5	10.0	700	500
Wood Door (3"X7")	1.0	6.5	700	500
Small Factory Door	.75	6.5	-	-
Garage & Shipping Room Door	2.0	4.5	-	-
Ramp Garage Door	2.0	6.75	-	-

TABLE 41d-SWINGING DOORS ON OPPOSITE WALLS

	% Time 2nd Door is Open	CFM PER PAIR OF DOORS				
		% Time 1st Door is Open				
		10	25	50	75	100
	10	100	250	500	750	1,000
	25	250	625	1250	1875	2,500
	50	500	1250	2500	3750	5,000
	75	750	1875	3750	5625	7,500
	100	1000	2500	5000	7500	10,000

TABLE 41e-DOORS

APPLICATION	CFM PER PERSON IN ROOM PER DOOR		
		36" Swinging Door	
	72" Revolving Door	No Vestibule	Vestibule
Bank	6.5	8.0	6.0
Barber Shop	4.0	5.0	3.8
Candy and Soda	5.5	7.0	5.3
Cigar Store	20.0	30.0	22.5
Department Store (Small)	6.5	8.0	6.0
Dress Shop	2.0	2.5	1.9
Drug Store	5.5	7.0	5.3
Hospital Room	-	3.5	2.6
Lunch Room	4.0	5.0	3.8
Men's Shop	2.7	3.7	2.8
Restaurant	2.0	2.5	1.9
Shoe Store	2.7	3.5	2.6

*All values in *Table 41* are based on the wind blowing directly at the window or door. When the wind direction is oblique to the window or door, multiply the above values by 0.60 and use the total window and door area on the windward side(s).

†Based on a wind velocity of 7.5 mph. For design wind velocities different from the base, multiply the above values by the ratio of velocities.

‡Includes frame leakage where applicable.

** Vestibules may decrease the infiltration as much as 30% when the door usage is light. When door usage is heavy, the vestibule is of little value for reducing infiltration.

Example 1-Infiltration in Tall Building, Summer

Given:

A 20-story building in New York City oriented true north. Building is 100 ft long and 100 ft wide with a floor-to-floor height of 12 ft. Wall area is 50% residential casement windows having 50% fixed sash. There are ten 7 ft × 3 ft swinging glass doors on the street level facing south.

Find:

Infiltration into the building thru doors and windows, disregarding outside air thru the equipment and the exhaust air quantity.

Solution:

The prevailing wind in New York City during the summer is south, 13 mph (*Table 1, page 10*).

Correction to Table 1 values for wind velocity
= 13/7.5 = 1.73
Glass area on south side
= 20×12×100×.50 = 12,000 sq ft
Infiltration thru windows
=12,000×.49×1.73 = 10,200 cfm (*Table 41b*)
Infiltration thru doors
=10×7×3×10×1.73 =3640 cfm (*Table 41c)*
Since this building is over 100 ft tall, net infiltration thru doors = 3640-(10,200×.80) = -4520 cfm.
Therefore, there is no infiltration thru the doors on the street level *on design days*, only exfiltration.

OFFSETTING INFILTRATION WITH OUTDOOR, AIR, SUMMER

Completely offsetting infiltration by the introduction of outdoor air thru the air conditioning apparatus is normally uneconomical except in buildings with few windows and doors. The outdoor air so introduced must develop a pressure equal to the wind velocity to offset infiltration. This pressure causes exfiltration thru the leeward walls at a rate equal to wind velocity. Therefore, in a four sided building with equal crack areas on each side and the wind blowing against one side, the amount of outdoor air introduced thru the apparatus must be a little more than three times the amount that infiltrates. Where the wind is blowing against two sides, the outdoor air must be a little more than equal to that which infiltrates.

Offsetting swinging door infiltration is not quite as difficult because air takes the path of least resistance, normally an open door. Most of the outdoor air introduced thru the apparatus flows out the door when it is opened. Also, in tall building the window infiltration tends to flow out the door.

The infiltration thru revolving doors is caused by displacement of the air in the door quadrants, is almost independent of wind velocity and, therefore, cannot be offset by outdoor air.

Basis of Table 42
- Offsetting Swinging Door Infiltration with Outdoor Air, Summer

Some of the outdoor air introduced thru the apparatus exfiltrates thru the cracks around the windows and in the construction on the leeward side. The outdoor air values have been increased by this amount for typical application as a result of experience.

Use of Table 42
- Offsetting Swinging Door Infiltration with Outdoor Air, Summer

Table 42 is used to determine the amount of outdoor air thru air conditioning apparatus required to offset infiltration thru swinging doors.

Example 2-Offsetting Swinging Door Infiltration

Given:

A restaurant with 3000 cfm outdoor air being introduced thru the air conditioning apparatus. Exhaust fans in the kitchen remove 2000 cfm. Two 7 ft × 3 ft glass swinging doors face the prevailing wind direction. At peak load conditions, there are 300 people in the restaurant.

Find:

The net infiltration thru the outside doors.

Solution:

Infiltration thru doors = 300×2.5=750 cfm (*Table 41e*)
Net outdoor air = 3000-2000=1000 cfm
Only 975 cfm of outdoor air is required to offset 750 cfm of door infiltration (*Table 42*).
Therefore, there will be no net infiltration thru the outside doors unless there are windows on the leeward side. If there are window in the building, calculate as outlined in *Example 1*.

TABLE 42-OFFSETTING SWINGING DOOR INFILTRATION WITH OUTDOOR AIR-SUMMER

Net Outdoor Air* (cfm)	Door Infiltration (cfm)	Net Outdoor Air* (cfm)	Door Infiltration (cfm)
140	100	1370	1100
270	200	1480	1200
410	300	1560	1300
530	400	1670	1400
660	500	1760	1500
790	600	1890	1600
920	700	2070	1800
1030	800	2250	2000
1150	900	2450	2200
1260	1000	2650	2400

*Net outdoor air is equal to the outdoor air quantity introduced thru the apparatus minus the exhaust air quantity.

INFILTRATION THRU WINDOWS AND DOORS, WINTER

Infiltration thru windows and doors during the winter is caused by the wind velocity and also stack effect. The temperature differences during the winter are considerably greater than in summer and, therefore, the density difference is greater; at 75 F db and 30% rh, density is .0738; at 0° F db, 40% rh, density is .0865. Stack effect causes air to flow in at the bottom and out at the top, and in many cases requires spot heating at the doors on the street level to maintain conditions. In applications where there is considerable infiltration on the street level, much of the infiltration thru the windows in the upper levels will be offset.

Basis of Table 43
- Infiltration thru Windows and Doors, Winter

The data in *Table 43* is based on a wind velocity of 15 mph blowing directly at the window or door and on observed crack widths around typical windows and doors. The infiltration thru these cracks is calculated from *Table 44* which is based on ASHAE tests.

Use of Table 43
- Infiltration thru Windows and Doors, Winter

Table 43 is used to determine the infiltration of air thru windows and doors on the windward side during the winter. The stack effect in tall buildings increases the infiltration thru the doors and windows on the lower levels and decreases it on the upper levels. Therefore, whenever the door infiltration is increased, the infiltration thru the upper levels must be decreased by 80% of the net increase in door infiltration. The infiltration from stack effect on the leeward sides of the building is determined by using the difference between the equivalent velocity (V_e) and the actual velocity (V) as outlined in Example 3. The data in Table 43 is based on the wind blowing directly at the windows and doors. When the wind direction is oblique to the windows and doors, multiply the values by 0.60 and use the total window and door area on the windward sides.

Example 3-Infiltration in Tall Buildings, Winter

Given:

The building described in *Example 1*.

Find:

The infiltration thru the doors and windows.

Solution:

The prevailing wind in New York City during the winter is NW at 16.8 mph (*Table 1, page 10*)

Correction on *Table 43* for wind velocity is 16.8/15= 1.12. Since the wind is coming from the Northwest, the crackage on the north and west sides will allow infiltration but the wind is only 60% effective. Correction for wind direction is .6.

Since this building is over 100 ft tall, stack effect causes infiltration on all sides at the lower levels and exfiltration at the upper levels. The total infiltration on the windward sides remains the same because the increase at the bottom is exactly equal to the decrease at the top. (For a floor-by-floor analysis, use equivalent wind velocity formulas.) Infiltration thru windows on the windward sides of the lower levels

$= 12{,}000 \times 2 \times 1.12 \times .6 \times .98 = 15{,}810$ cfm.

The total infiltration thru the windows on the leeward sides of the building is equal to the difference between the equivalent velocity at the first floor and the design velocity at the midpoint of the building.

$$V_e = \sqrt{V^2 + 1.75b}$$

$$= \sqrt{(16.8)^2 + (1.75 \times \frac{240}{2})} = 22.2 \text{ mph}$$

$$V_e - V = 22.2 - 16.8 = 5.4 \text{ mph}$$

Total infiltration thru windows in lower half of building (upper half is exfiltration) on leeward side

$= 12{,}000 \times 2 \times {}^{1}/_{2} \times (5.4/15) \times {}^{1}/_{2} \times .98$

= 2160 cfm (*Table 43*)

NOTE: This is the total infiltration thru the windows on the leeward side. A floor-by-floor analysis should be made to balance the system to maintain proper conditions on each floor.

(on leeward side)

$= 10 \times 7 \times 3 \times (5.4/15) \times 30$

= 2310 cfm (Table 43c, average use, 1 and 2 story building).

Example 4-Offsetting Infiltration with Outdoor Air

Any outdoor air mechanically introduced into the building offsets some of the infiltration. In *Example 3* all of the outdoor air is effective in reducing the window infiltration. Infiltration is reduced on two windward sides, and the air introduced thru the apparatus exfiltrates thru the other two sides.

Given:

The building described in *Example 1* with .25 cfm/sq ft supplied thru the apparatus and 40,000 cfm being exhausted from the building.

Find:

The net infiltration into this building.

Solution:

Net outdoor air = $(.25 \times 10{,}000 \times 20) - 40{,}000 = 10{,}000$ cfm

Net infiltration thru windows

= 15,800+2160-10,000 = 7970 cfm

Net infiltration thru doors = 2310 cfm (*Example 3*)

Net infiltration into building = 7970+2310 = 10,280 cfm

TABLE 43-INFILTRATION THRU WINDOWS AND DOORS-WINTER*

15 mph Wind Velocity†

TABLE 43a-DOUBLE HUNG WINDOWS ON WINDOW SIDE‡

DESCRIPTION	CFM PER SQ FT SASH AREA					
	Small-30×"72"			Large-54"×96"		
	No W-Strip	W-Strip	Storm Sash	No W-Strip	W-Strip	Storm Sash
Average Wood Sash	.85	.52	.42	.53	.33	.26
Poorly Fitted Wood Sash	2.4	.74	1.2	1.52	.47	.74
Metal Sash	1.60	.69	.80	1.01	.44	.50

NOTE: W-Strip denotes weatherstrip.

TABLE 43b-CASEMENT TYPE WINDOWS ON WINDWARD SIDE‡

DESCRIPTION	CFM PER SQ FT SASH AREA									
	Percent Openable Area									
	0%	25%	33%	40%	45%	50%	60%	66%	75%	100%
Rolled Section-Steel Sash										
Industrial Pivoted	.65	1.44	-	1.98	-	-	-	2.9	-	5.2
Architectural Projected	-	.78	-	-	-	1.1	1.48	-	-	-
Residential	-	-	.56	-	-	.98	-	-	-	1.26
Heavy Projected	-	-	-	-	.45	-	-	.63	.78	-
Hollow Metal-Vertically Pivoted	.54	1.19	-	1.64	-	-	-	2.4	-	4.3

TABLE 43c-DOORS ON ONE OR ADJACENT WINDWARD SIDES‡

DESCRIPTION	CFM PER SQ FT AREA**				
	Infrequent Use	Average Use			
		1 & 2 Story Bldg.	Tall Building (ft)		
			50	100	200
Revolving Door	1.6	10.5	12.6	14.2	17.3
Glass Door-(3/16" Crack)	9.0	30.0	36.0	40.5	49.5
Wood Door 3'×7'	2.0	13.0	15.5	17.5	21.5
Small Factory Door	1.5	13.0			
Garage & Shipping Room Door	4.0	9.0			
Ramp Garage Door	4.0	13.5			

*All values in *Table 43* are based on the wind blowing directly at the window or door. When the prevailing wind direction is oblique to the window or door, multiply the above values by 0.60 and use the total window and door area on the windward side(s).

†Based on a wind velocity of 15 mph. For design wind velocities different from the base, multiply the table values by the ratio of velocities.

‡Stack effect in tall buildings may also cause infiltration on the leeward side. To evaluate this, determine the equivalent velocity (V_e) and subtract the design velocity (V). The equivalent velocity is:

$$V_e = \sqrt{V^2 - 1.75a} \text{ (upper section)}$$

$$V_e = \sqrt{V^2 + 1.75b} \text{ (lower section)}$$

Where a and b are the distances above and below the mid-height of the building, respectively, in ft.

Multiply the table values by the ratio (V_e-V)/15 for doors and one half of the windows on the leeward side of the building. (Use values under "1 and 2 Story Bldgs" for doors on leeward side of tall buildings.)

**Doors on opposite sides increase the above values 25%. Vestibules may decrease the infiltration as much as 30% when door usage is light. If door usage is heavy, the vestibule is of little value in reducing infiltration. Heat added to the vestibule will help maintain room temperature near the door.

INFILTRATION-CRACK METHOD (Summer or Winter)

The crack method of evaluating infiltration is more accurate than the area methods. It is difficult to establish the exact crack dimensions but, in certain close tolerance applications, it may be necessary to evaluate the load accurately. The crack method is applicable both summer and winter.

Basis of Table 44
- Infiltration thru Windows and Doors, Crack Method

The data on windows in *Table 44* are based on ASHAE tests. These test results have been reduced 20% because, as infiltration occurs on one side, a certain amount of pressure builds up in the building, thereby reducing the infiltration. The data on glass and factory doors has been calculated from observed typical crack widths.

Use of Table 44
- Infiltration thru Windows and Doors, Crack Method

Table 44 is used to determine the infiltration thru the doors and windows listed. This table does not take into account winter stack effect which must be evaluated separately, using the equivalent wind velocity formulas previously presented.

Example 5-Infiltration thru Windows, Crack Method

Given:

A 4 ft × 7 ft residential casement window facing south.

Find:

The infiltration thru this window:

Solution:

Assume the crack widths are measured as follows:

Window frame-none, well sealed

Window openable area-1/32 in. crack; length, 20 ft

Assume the wind velocity is 30 mph due south.

Infiltration thru window =20 × 2.1 = 42 cfm (*Table 44*)

TABLE 44-INFILTRATION THRU WINDOWS AND DOORS-CRACK METHOD-SUMMER-WINTER*

TABLE 44a-DOUBLE HUNG WINDOWS-UNLOCKED ON WINDWARD SIDE

TYPE OF DOUBLE HUNG WINDOW	CFM PER LINEAR FOOT OF CRACK											
	Wind Velocity-Mph											
	5		10		15		20		25		30	
	No W-Strip	W-Strip	No W-Strip	W-Strip	No W-Strip	W-Strip	No W-Strip	W-Strip	No W-Strip	W-Strip	No W-Strip	W-Strip
Wood Sash												
Average Window	.12	.07	.35	.22	.65	.40	.98	.60	1.33	.82	1.73	1.05
Poorly Fitted Window	.45	.10	1.15	.32	1.85	.57	2.60	.85	3.30	1.18	4.20	1.53
Poorly Fitted-with Storm Sash	.23	.05	.57	.16	.93	.29	1.30	.43	1.60	.59	2.10	.76
Metal Sash	.33	.10	.78	.32	1.23	.53	1.73	.77	2.3	1.00	2.8	1.27

TABLE 44b-CASEMENT TYPE WINDOWS ON WINDWARD SIDE

TYPE OF DOUBLE HUNG WINDOW		CFM PER LINEAR FOOT OF CRACK					
		Wind velocity-Mph					
		5	10	15	20	25	30
Rolled Section-Steel Sash							
Industrial Pivoted	1/16” crack	.87	1.80	2.9	4.1	5.1	6.2
Architectural Projected	1/32” crack	.25	.60	1.03	1.43	1.86	2.3
Architectural Projected	3/64” crack	.33	.87	1.47	1.93	2.5	3.0
Residential casement	1/64” crack	.10	.30	.55	.78	1.00	1.23
Residential Casement	1/32” crack	.23	.53	.87	1.27	1.67	2.10
Heavy Casement Section Projected	1/64” crack	.05	.17	.30	.43	.58	.80
Heavy Casement Section Projected	1/32” crack	.13	.40	.63	.90	1.20	1.53
Hollow Metal-Vertically Pivoted		.50	1.46	2.40	3.10	3.70	4.00

*Infiltration caused by stack effect must be calculated separately during the winter.

†No allowance has been made for usage. See Table 43 for infiltration due to usage.

TABLE 44-INFILTRATION THRU WINDOWS AND DOORS-CRACK METHOD-SUMMER-WINTER*
(Contd)

TABLE 44c-DOORS†ON WINDWARD SIDE

TYPE OF DOOR		CFM PER LINEAR FOOT OF CRACK					
		Wind Velocity-mph					
		5	10	15	20	25	30
Glass Door-Herculite							
Good Installation	1/16” crack	3.2	6.4	9.6	13.0	16.0	19.0
Average Installation	1/32” crack	4.8	10.0	14.0	20.0	24.0	29.0
Poor Installation	3/64” crack	6.4	13.0	19.0	26.0	26.0	38.0
Ordinary Wood or Metal							
Well Fitted-W-Strip		.45	.60	.90	1.3	1.7	2.1
Well Fitted-No W-Strip		.90	1.2	1.8	2.6	3.3	4.2
Poorly Fitted-No W-Strip		.90	2.3	3.7	5.2	6.6	8.4
Factory Door 1/8” crack		3.2	6.4	9.6	13.0	16.0	19.0

VENTILATION

VENTILATION STANDARDS

The introduction of outdoor air for ventilation of conditioned spaces is necessary to dilute the odors given off by people, smoking and other internal air contaminants.

The amount of ventilation required varies primarily with the total number of people, the ceiling height and the number of people smoking. People give off body odors which require a minimum of 5 cfm per person for satisfactory dilution. Seven and one half cfm per person is recommended. This is based on a population density of 50 to 75 sq ft per person and a typical ceiling height of 8 ft. With greater population densities, the ventilation quantity should be increased. When people smoke, the additional odors given off by cigarettes or cigars require a minimum of 15 to 25 cfm per person. In special gathering rooms with heavy smoking, 30 to 50 cfm per person is recommended.

Basis of Table 45
- Ventilation Standards

The data in *Table 45* is based on test observation of the clean outdoor air required to maintain satisfactory odor levels with people smoking and not smoking. These test results were then extrapolated for typical concentrations of people, both smoking and not smoking, for the applications listed.

Use of Table 45
- Ventilation Standards

Table 45 is used to determine the minimum and recommended ventilation air quantity for the listed applications. In applications where the minimum values are used and the minimum cfm per person and cfm per sq ft of floor area are listed, use the larger minimum quantity. Where the crowd density is greater than normal or where better than satisfactory conditions are desired, use the recommended values.

SCHEDULED VENTILATION

In comfort applications, where local codes permit, it is possible to reduce the capacity requirements of the installed equipment by reducing the ventilation air quantity at the time of peak load. This quantity can be reduced at the time of peak to, in effect, minimize the outdoor air load. At times other than peak load, the calculated outdoor air quantity is used. Scheduled ventilation is recommended *only for installations operating more than 12 hours or 3 hours longer than occupancy*, to allow some time for flushing out the building when no odors are being generated. It has been found, by tests, that few complaints of stuffiness are encountered when the outdoor air quantity is reduced for short periods of time, provided the flushing period is available. It is recommended that the outdoor air quantity be reduced to no less than 40% of the recommended quantity as listed in *Table 45*.

The procedure for estimating and controlling scheduled ventilation is as follows:

1. In estimating the cooling load, reduce the air quantity at design conditions to a minimum of 40% of the recommended air quantity.
2. Use a dry-bulb thermostat following the cooling and dehumidifying apparatus to control the leaving dewpoint such that:
   a. With the dewpoint at design, the damper motor closes the outdoor air damper to 40% of the design ventilation air quantity.
   b. As the dewpoint decreases below design, the outdoor air damper opens to the design setting.
3. Another method which could be used is a thermostat located in the leaving chilled water from the refrigeration machine.

Example 6-Ventilation Air Quantity, Office Space

Given:

A 5000 sq ft office with a ceiling height of 8 ft and 50 people. Approximately 40% of the people smoke.

Find:

The ventilation air quantity.

Solution:

The population density is typical, 100 sq ft per person, but the number of smokers is considerable.

Recommended ventilation = 50 × 15 = 750 cfm (*Table 45*)

Minimum ventilation = 50 × 10 = 500 cfm (*Table 45*)

500 cfm will more than likely not maintain satisfactory conditions within the space because the number of smokers is considerable. Therefore, 750 cfm should be used in this application.

NOTE: Many applications have exhaust fans. This means that the outdoor air quantity must at least equal the exhausted air; otherwise the infiltration rate will increase. *Tables 46 and 47* list the approximate capacities of typical exhaust fans. The data in these tables were obtained from published ratings of several manufacturers of exhaust fans.

TABLE 45-VENTILATION STANDARDS

APPLICATION	SMOKING	CFM PER PERSON		CFM PER SQ FT OF FLOOR
		Recommended	Minimum*	Minimum*
Apartment – Average	Some	20	15	-
Apartment – De Luxe	Some	30	25	.33
Banking Space	Occasional	10	7½	-
Barber Shops	Considerable	15	10	-
Beauty Parlors	Occasional	10	7½	-
Broker's Board Rooms	Very Heavy	50	30	-
Cocktail Bars	Heavy	30	25	-
Corridors (Supply or Exhaust)	-	-	-	.25
Department Stores	None	7½	5	.05
Directors Rooms	Extreme	50	30	-
Drug Stores†	Considerable	10	7½	-
Factories‡§	None	10	7½	.10
Five and Ten Cent Stores	None	7½	5	-
Funeral Parlors	None	10	7½	-
Garage‡	-	-	-	1.0
Hospitals – Operating Rooms‡**	None	-	-	2.0
Hospitals – Private Rooms	None	30	25	.33
Hospitals – Wards	None	20	15	-
Hotel Roms	Heavy	30	25	.33
Kitchen – Restaurant†	-	-	-	4.0
Kitchen – Residence	-	-	-	2.0
Laboratories†	Some	20	15	-
Meeting Rooms	Very Heavy	50	30	1.25
Office – General	Some	15	10	-
Office – Private	None	25	15	.25
Office – Private	Considerable	30	25	.25
Restaurant – Cafeteria†	Considerable	12	10	-
Restaurant – Dining Room†	Considerable	15	12	-
School Rooms‡	None	-	-	-
Shop Retail	None	10	7½	-
Theater‡	None	7½	5	-
Theater	Some	15	10	-
Toilets‡ (Exhaust)	-	-	-	2.0

*When minimum is used, use the larger.

‡See local codes which may govern.

†May be governed by exhaust..

§Use these values unless governed by other sources of contamination or by local codes.

**All outdoor air is recommended to overcome explosion hazard of anesthetics.

TABLE 46-CENTRIFUGAL FAN CAPACITIES

Inlet Diameter (in.)	Capacity* (cfm)	Motor Horsepower Range	Outlet Velocity Range (fpm)
4	50-250	1/70-1/20	800-2000
6	100-550	1/20-1/6	500-2500
8	300-1000	1/20-1/2	850-2900
10	600-2800	1/5-2	950-4300
12†	800-1600	1/8-1/2	1000-2000
15†	1200-2500	1/4-1	1000-2000
18†	1700-3600	1/4-1 1/4	1000-2000
21†	2300-5000	1/3-1 1/2	1000-2000

*These typical air capacities were obtained from published rating of several manufacturers of nationally known exhaust fans, single width, single inlet. Range of static pressures $^1/_4$ to 1 $^1/_4$ inches. Fans with inlet diameter 10 inches and smaller are direct connected.

†The capacity of these fans has been arbitrarily taken at 1000 fpm minimum and 2000 fpm maximum outlet velocity. For these fans the usual selection probably is approximately 1500 fpm outlet velocity for ventilation.

TABLE 47-PROPELLER FAN CAPACITIES-FREE DELIVERY

Fan Diameter (in.)	Speed (rpm)	Capacity* (cfm)
8	1500	500
12	1140	825
12	1725	1100
16	855	1000
16	1140	1500
18	850	1800
18	1140	2350
20	850	2400
20	1140	2750
20	1620	3300

*The capacities of fans of various manufacturers may vary ±10% from the values given above.

CHAPTER 7. INTERNAL AND SYSTEM HEAT GAIN

INTERNAL HEAT GAIN

Internal heat gain is the sensible and latent heat released within the air conditioned space by the occupants, lights, appliances, machines, pipes, etc. This chapter outlines the procedures for determining the instantaneous *heat gain* from these sources. A portion of the heat gain from internal sources is radiant heat which is partially absorbed in the building structure, thereby reducing the instantaneous heat gain. *Chapter 3, "Heat Storage, Diversity and Stratification*," contains the data and methods for estimating the actual cooling load from the heat sources referred to in the following text.

PEOPLE

Heat is generated within the human body by oxidation, commonly called metabolic rate. The metabolic rate varies with the individual and with his activity level. The normal body processes are performed most efficiently at a deep tissue temperature of about 98.6 F; this temperature may vary only thru a narrow range. However, the human body is capable of maintaining this temperature, thru a wide ambient temperature range, by conserving or dissipating the heat generated within itself.

This heat is carried to the surface of the body by the blood stream and is dissipated by:

1. Radiation from the body surface to the surrounding surfaces.
2. Convection from the body surface and the respiratory tract to the surrounding air.
3. Evaporation of moisture from the body surface and in the respiratory tract to the surrounding air.

The amount of heat dissipated by radiation and convection is determined by the difference in temperature between the body surface and its surroundings. The body surface temperature is regulated by the quantity of blood being pumped to the surface; the more blood, the higher the surface temperature up to a limit of about 96 F. The heat dissipated by evaporation is determined by the difference in vapor pressure between the body and the air.

Basis of Table 48
- Heat Gain from People

Table 48 is based on the metabolic rate of an average adult male, weighing 150 pounds, at different levels of activity, and generally for occupancies longer than 3 hours. These have been adjusted for typical compositions of mixed groups of males and females for the listed applications. The metabolic rate of women is about 85% of that for a male, and for children about 75%.

The heat gain for restaurant applications has been increased 30 Btu/hr sensible and 30 Btu/hr latent heat per person to include the food served.

The data in *Table 48* as noted are for continuous occupancy. The excess heat and moisture brought in by people, where short time occupancy is occurring (under 15 minutes), may increase the heat gain from people by as much as 10%.

Use of Table 48
- Heat Gain from People

To establish the proper heat gain, the room design temperature and the activity level of the occupants must be known.

Example 1-Bowling Alley

Given:

A 10 lane bowing alley, 50 people, with a room design dry-bulb temperature of 75 F. Estimate one person per alley bowling, 20 of the remainder seated, and 20 standing.

Find:

Sensible heat gain = (10×525)+(20×240)+(20×280)
= 15,650 Btu/hr

Latent heat gain = (10×925)+(20×160)+(20×270)
= 17,850 Btu/hr

LIGHTS

Lights generate sensible heat by the conversion of the electrical power input into light and heat. The heat is dissipated by radiation to the surrounding surfaces, by conduction into the adjacent materials and by convection to the surrounding air. The radiant portion of the light load is partially stored, and the convection portion may be stratified as described on page 39. Refer to *Table 12, page 35*, to determine the actual cooling load.

Incandescent lights convert approximately 10% of the power input into light with the rest being generated as heat within the bulb and dissipated by radiation, convection and conduction. About 80% of the power input is dissipated by radiation and only about 10% by convection and conduction, *Fig. 30*.

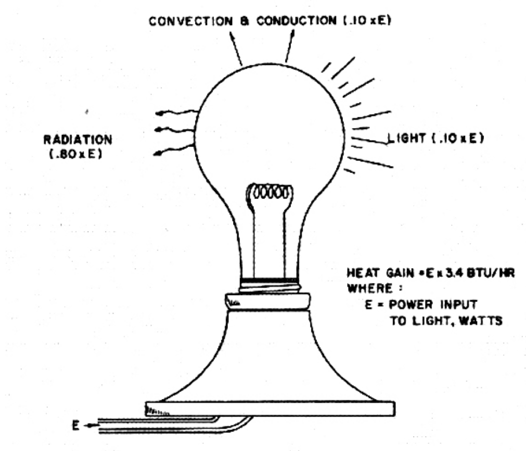


FIG. 30-CONVERSION OF ELECTRIC POWER TO HEAT AND LIGHT WITH INCANDESCENT LIGHTS, APPROXIMATE

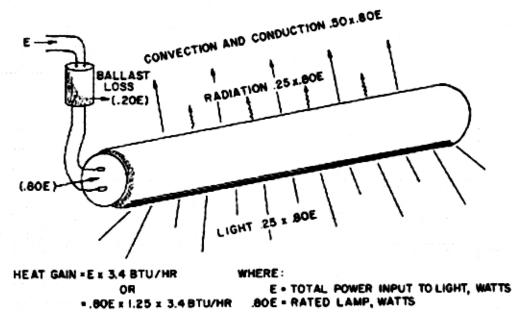


FIG. 31-CONVERSION OF ELECTRIC POWER TO HEAT AND LIGHT WITH FLUORESCENT LIGHTS, APPROXIMATE

Fluorescent lights convert about 25% of the power input into light, with about 25% being dissipated by radiation to the surrounding surfaces. The other 50% is dissipated by conduction and convection. In addition to this, approximately 25% more heat is generated as heat in the ballast of the fluorescent lamp, *Fig. 31*.

Table 49 indicates the basis for arriving at the gross heat gain from fluorescent or incandescent lights.

TABLE 48-HEAT GAIN FROM PEOPLE

DEGREE OF ACTIVITY	TYPICAL APPLICATION	Metabolic Rate (Adult Male) Btu/hr	Average Adjusted Metabolic Rate* Btu/hr	ROOM DRY-BULB TEMPERATURE									
				82 F Btu/hr		80 F Btu/hr		78 F Btu/hr		75 F But/hr		70 F Btu/hr	
				Sensible	Latent	Sensible	Latent	Sensible	Latent	Sensible	Latent	Sensible	Latent
Seated at rest	Theater, Grade School	390	350	175	175	195	155	210	140	230	120	260	90
Seated, very light work	High School	450	400	180	220	195	205	215	185	240	160	275	125
Office worker	Offices, Hotels, Apts., College	475	450	180	270	200	250	215	235	245	205	285	165
Standing, walking slowly	Dept., Retail, or Variety Store	550											
Walking, seated	Drug Store	550	500	180	320	200	300	220	280	255	245	290	210
Standing, walking slowly	Bank	550											
Sedentary work	Restaurant†	500	550	190	360	220	330	240	310	280	270	320	230
Light bench work	Factory, light work	800	750	190	560	220	530	245	505	295	455	365	385
Moderate dancing	Dance Hall	900	850	220	630	245	605	275	575	325	525	400	450
Walking, 3 mph	Factory, fairly heavy work	1000	1000	270	730	300	700	330	670	380	620	460	540
Heavy work	Bowling Alley‡ Factory	1500	1450	450	1000	465	985	485	965	525	925	605	845

*Adjusted Metabolic Rate is the metabolic rate to be applied to a mixed group of people with a typical percent composition based on the following factors:

Metabolic rate, adult female=Metabolic rate, adult male×0.85

Metabolic rate, children =Metabolic rate, adult male×0.75

†Restaurant-Values for this application include 60 Bu per hr for food per Individual (30 Btu sensible and 30 Btu latent heat per hr).

‡Bowling-Assume one person per alley actually bowling and all others sitting, metabolic rate 400 Btu per hr; or standing, 550 Btu per hr.

TABLE 49 – HEAT GAIN FROM LIGHT

TYPE	HEAT GAIN* Btu/hr
Fluorescent	Total Light Watts×1.25†×3.4
Incandescent	Total Light Watts×3.4

*Refer to Tables 12 and 13, pages 35-37 to determine actual cooling load.

†Fluorescent light wattage is multiplied by 1.25 to include heat gain in ballast.

APPLIANCES

Most appliances contribute both sensible and latent heat to a space. Electric appliances contribute latent heat, only by virtue of the function they perform, that is, drying, cooking, etc, whereas gas burning appliances contribute additional moisture as a product of combustion. A properly designed hood with a positive exhaust system removes a considerable amount of the generated heat and moisture from most types of appliances.

Basis of Tables 50 thru 52
- Heat Gain from Restaurant Appliances and Miscellaneous Appliances

The data in these tables have been determined from manufacturers data, the American Gas Association data, Directory of Approved Gas Appliances and actual tests by Carrier Corporation.

TABLE 50-HEAT GAIN FROM RESTAURANT APPLIANCES

NOT HOODED*-ELECTRIC

APPLIANCE	OVERALL DIMENSIONS Less Legs and Handles (In.)	TYPE OF CON-TROL	MISCELLANEOUS DATA	MFR MAX RATING Btu/hr	MAIN-TAIN-ING RATE Btu/hr	RECOM HEAT GAIN FOR AVG USE		
						Sensible Heat Btu/hr	Latent Heat Btu/hr	Total Heat Btu/hr
Coffee Brewer-1/2 gal		Man.		2240	306	900	220	1120
Warmer-1/2 gal		Man.		306	306	230	90	320
4 Coffee Brewing Units with 41/2 gal Tank	20×30×26 H	Auto.	Water heater—2000 watts Brewers—2960 watts	16900		4800	1200	6000
Coffee Urn--3 gal	15 Dia×34H	Man.	Black finish	11900	3000	2600	1700	4300
--3 gal	12×23 oval ×21H	Auto.	Nickel plated	15300	2600	2200	1500	3700
--5 gal	18 Dia ×37H	Auto.	Nickel plated	17000	3600	3400	2300	5700
Doughnut Machine	22×22×57H	Auto.	Exhaust system to outdoors-1/2 hp motor	16000		5000		5000
Egg Boiler	10×13×25H	Man.	Med. ht. –550 watts Low ht—275 watts	3740		1200	800	2000
Food Warmer with Plate Warmer, per sq ft top surface		Auto.	Insulated, separate heating unit for each pot. Plate warmer in base	1350	500	350	350	700
Food Warmer without Plate Warmer, per sq ft top surface		Auto.	Ditto, without plate warmer	1020	400	200	350	550
Fry Kettle--111/2 lb fat	12 Dia×14H	Auto.		8840	1100	1600	2400	4000
Fry Kettle—25 lb fal	16×18×12H	Auto.	Frying area 12"×14"	23800	2000	3800	5700	9500
Griddle, Frying	18×18×8H	Auto.	Frying top 18"×14"	8000	2800	3100	1700	4800
Grille, Meat	14×14×10H	Auto.	Cooking area 10"×12"	10200	1900	3900	2100	6000
Grille, Sandwich	13×14×10H	Auto.	Grill area 12"×12"	5600	1900	2700	700	3400
Roll Warmer	26×17×13H	Auto.	One drawer	1500	400	1100	100	1200
Toaster, Continuous	15×15×28H	Auto.	2 Slices wide-- 360 slices/hr	7500	5000	5100	1300	6400
Toaster, Continuous	20×15×28H	Auto.	4 Slices wide-- 720 slices/hr	10200	6000	6100	2600	8700
Toaster, Pop-Up	6×11×9H	Auto.	2 Slices	4150	1000	2450	450	2900
Waffle Iron	12×13×10H	Auto.	One waffle 7" dia	2480	600	1100	750	1850
Waffle Iron for Ice Cream Sandwich	14×13×10H	Auto.	12 Cakes, each 2 1/2"×3 3/4"	7500	1500	3100	2100	5200

*If properly designed positive exhaust hood is used, multiply recommended value by .50.

Use of Tables 50 thru 52
- Heat Gain from Restaurant Appliances and
- Miscellaneous Appliances

The *Maintaining Rate* is the heat generated when the appliance is being maintained at operating temperature but not being used.

The *Recommended for Average Use* values are those which the appliance generates under normal use. These appliances seldom operate at maximum capacity during peak load since they are normally warmed up prior to the peak.

The values in Tables 50 thru 52 are for unhooded appliances. If the appliance has a properly designed positive exhaust hood, reduce the sensible and the latent heat gains by 50%. A hood, to be effective, should extend beyond the appliance approximately 4 inches per foot of height between the appliance and the face of the hood. The lower edge should not be higher than 4 feet above the appliance and the average face velocity across the hood should not be less than 70 fpm.

TABLE 51-HEAT GAIN FROM RESTAURANT APPLIANCES
NOT HOODED*--GAS BURNING AND STEAM HEATED

APPLIANCE	OVERALL DIMENSIONS Less Legs and Handles (In.)	TYPE OF CON-TROL	MISCELLANEOUS DATA	MFR MAX RATING Btu/hr	MAIN-TAIN-ING RATE Btu/hr	RECOM HEAT GAIN FOR AVG USE Sensible Heat Btu/hr	RECOM HEAT GAIN FOR AVG USE Latent Heat Btu/hr	RECOM HEAT GAIN FOR AVG USE Total Heat Btu/hr
GAS BURNING								
Coffee Brewer-1/2 gal Warmer-1/2 gal		Man. Man.	Combination brewer and warmer	3400 500	 500	1350 400	350 100	1700 500
Coffee Brewing Units with Tank	19X30X26 H		4 Brewers and 4½ gal tank			7200	1800	9000
Coffee Urn--3 gal	15" DiaX34H	Auto.	Black finish	3200	3900	2900	2900	5800
Coffee Urn --3 gal	12X23 oval X21H	Auto.	Nickel plated		3400	2500	2500	5000
Coffee Urn --5 gal	18 Dia X37H	Auto.	Nickel plated		4700	3900	3900	7800
Food Warmer, Values per sq ft top surface		Man.	Water bath type	2000	900	850	450	1300
Fry Kettle—15 lb fat	12X20X18H	Auto.	Frying area 10X10	14250	3000	4200	2800	7000
Fry Kettle—28 lb fal	15X35X11H		Frying area 11X16	24000	4500	7200	4800	12000
Grill—Broil-O-Grill Top Burner Bottom Burner	22X14X17H (1.4 sq ft) grill surface)	Man.	Insulated 22,000 Btu/hr 15,000 Btu/hr	37000		14400	3600	18000
Stoves, Short Order-- Open Top. Values per sq ft top surface		Man.	Ring type burners 12000 to 22000 Btu/ea	14000		4200	4200	8400
Stoves, Short Order-- Closed Top. Values per sq ft top surface		Man.	Ring type burners 10000 to 12000 Btu/ea	11000		3300	3300	6600
Toaster, Continuous	15X15X28H	Auto.	2 Slices wide-- 360 slices/hr	12000	10000	7700	3300	11000
STEAM HEATED								
Coffee Urn--3 gal --3 gal --5 gal	15 DiaX34H 12X23 ovalX21H 18 DiaX37H	Auto. Auto. Auto.	Black finish Nickel plated Nickel plated			2900 2400 3400	1900 1600 2300	4800 4000 5700
Coffee Urn--3 gal --3 gal --5 gal	15 DiaX34H 12X23 ovalX21H 18 DiaX37H	Man. Man. Man.	Black finish Nickel plated Nickel plated			3100 2600 3700	3100 2600 3700	6200 5200 7400
Food Warmer, per sq ft top surface		Auto.				400	500	900
Food Warmer, per sq ft top surface		Man.				450	1150	1500

*If properly designed positive exhaust hood is used, multiply recommended value by. 50.

TABLE 52-HEAT GAIN FROM MISCELLANEOUS APPLIANCES
NOT HOODED*

APPLIANCE	TYPE OF CONTROL	MISCELLANEOUS DATA	MFR MAX RATING Btu/hr	RECOM HEAT GAIN FOR AVG USE Sensible Heat Btu/hr	Latent Heat Btu/hr	Total Heat Btu/hr
GAS BURNING						
Hair Dryer, Blower Type 15 amps, 115 volts AC	Man.	Fan 165 watts, (low 915 watts, high 1580 watts)	5,370	2,300	400	2,700
Hair Dryer, helmet type, 6.5 amps, 115 volts AC	Man.	Fan 80 watts, (low 300 watts, high 710 watts)	2,400	1,870	330	2,200
Permanent Wave Machine	Man.	60 heaters at 25 watts each, 36 in normal use	5,100	850	150	1,000
Pressurized Instrument Washer and Sterilizer		11"X11"X22"		12,000	23,460	35,460
Neon Sign, per Linear ft tube		1/2" outside dia		30		30
		3/8" outside dia		60		60
Solution and/or Blanket Warmer		18"X30"X72"		1,200	3,000	4,200
		18"X24"X72"		1,050	2,400	3,450
Sterilizer	Auto.	16"X24"		9,600	8,700	18,300
Dressing	Auto.	20"X36"		23,300	24,000	47,300
Sterilizer, Rectangular Bulk	Auto.	24"X24"X36"		34,800	21,000	55,800
	Auto.	24"X24"X48"		41,700	27,000	68,700
	Auto.	24"X36"X48"		56,200	36,000	92,200
	Auto.	24"X36"X60"		68,500	45,000	113,500
	Auto.	36"X42"X84"		161,700	97,500	259,200
	Auto.	42"X48"X96"		184,000	140,000	324,000
	Auto.	48"X54"X96"		210,000	180,000	390,000
Sterilizer, Water	Auto.	10 gallon		4,100	16,500	20,600
	Auto.	15 gallon		6,100	24,600	30,700
Sterilizer, Instrument	Auto.	6"X8"X17"		2,700	2,400	5,100
	Auto.	9"X10"X20"		5,100	3,900	9,000
	Auto.	10"X12"X22"		8,100	5,900	14,000
	Auto.	10"X12"X36"		10,200	9,400	19,600
	Auto.	12"X16"X24"		9,200	8,600	17,800
Sterilizer, Utensil	Auto.	16"X16"X24"		10,600	20,400	31,000
	Auto.	20"X20"X24"		12,300	25,600	37,900
Sterilizer, Hot Air	Auto.	Model 120 Amer Sterilizer Co		2,000	4,200	6,200
	Auto.	Model 100 Amer Sterilizer Co		1,200	2,100	3,300
Water Still		5 gal/hour		1,700	2,700	4,400
X-ray Machines, for making pictures		Physicians and Dentists office		None	None	None
X-ray Machines, for therapy		Heat load may be appreciable-- write mfg for data				
GAS BURNING						
Burner, Laboratory small bunsen	Man.	7/16 dia barrel with manufactured gas	1,800	960	240	1,200
small bunsen	Man.	7/16 dia with nat gas	3,000	1,680	420	2,100
fishtail burner	Man.	7/16 dia with not gas	3,500	1,960	490	2,450
fishtail burner	Man.	7/16 dia bar with not gas	5,500	3,080	770	3,850
large bunsen	Man.	1 ½ dia mouth, adj orifice	6,000	3,350	850	4,200
Cigar Lighter	Man.	Continuous flame type	2,500	900	100	1,000
Hair Dryer System 5 helmets	Auto.	Consists of heater & fan which blows hot air thru duct system to helmets	33,000	15,000	4,000	19,000
10 helmets	Auto.			21,000	6,000	27,000

*If properly designed positive exhaust hood is used, multiply recommended value by. 50.

Example 2-Restaurant

Given:

A restaurant with the following electric appliances with a properly designed positive exhaust hood on each:

1. Two 5-gallon coffee urns, both used in the morning, only one used either in the afternoon or evening.
2. One 20 sq ft food warner without plate warmer.
3. Two 24 × 20 × 10 inch frying griddles.
4. One 4-slice pop-up toaster, used only in the morning.
5. Two 25 lb deep fat, fry kettles.

Find:

Heat gain from these appliances during the afternoon and evening meal.

Solution:

Use *Table 50*.	Sensible	Latent
1. Coffee Urn-only one in use:		
Sensible heat gain = 3400 × .50 =	1700	
Latent heat gain = 2300 × .50 =		1150
2. Food Warmer:		
Sensible heat gain =20 × 200 × .50=	2000	
Latent heat gain =20 × 350 × .50=		3500
3. Frying Griddles:		
Sensible heat gain =2×5300×.50 =	5300	
Latent heat gain =2×2900×.50 =		2900
4. Toaster—not in use		
5. Fry Kettles:		
Sensible heat gain =2 × 3800 × .50 =	3800	
Latent heat gain =2 × 5700 × .50 =		5700
Total sensible heat gain =	12,800	
Total latent heat gain =		13,250

ELECTRIC MOTORS

Electric motors contribute sensible heat to a space by converting the electrical power input to heat. Some of this power input is dissipated as heat in the motor frame and can be evaluated as

input × (1 - motor eff).

The rest of the power input (brake horsepower or motor output) is dissipated by the driven machine and in the drive mechanism. The driven machine utilizes this motor output to do work which may or may not result in a heat gain to the space.

Motors driving fans and pumps: The power input increases the pressure and velocity of the fluid and the temperature of the fluid.

The increased energy level in the fluid is degenerated in pressure drop throughout the system and appears as a heat gain to the fluid at the point where pressure drop occurs. This heat gain does not appear as a temperature rise because, as the pressure reduces, the fluid expands. The fluid expansion is a cooling process which exactly offsets the heat generated by friction. The heat of compression required to increase the energy level is generated at the fan or pump *and* is a heat gain at this point.

If the fluid is conveyed outside of the air conditioned space, only the inefficiency of the motor driving fan or pump should be included in room sensible heat gain.

If the temperature of the fluid is maintained by a separate source, these heat gains to the fluid heat of compression are a load on this separate source only.

The heat gain or loss from the system should be calculated separately ("*System Heat Gain," p. 110*).

Motors driving process machinery (lathe, punch press, etc.): The total power input to the machine is dissipated as heat at the machine. If the product is removed from the conditioned space at a higher temperature than it came in, some of the heat input into the machine is removed and should not be considered a heat gain to the conditioned space. The heat added to a product is determined by multiplying the number of pounds of material handled per hour by the specific heat and temperature rise.

Basis of Table 53
- Heat Gain from Electric Motors

Table 53 is based on average efficiencies of squirrel cage induction open type integral horsepower and fractional horsepower motors. Power supply for fractional horsepower motors is 110 or 220 volts, 60 cycle, single phase; for integral horsepower motors, 208, 220, or 440 volts, 60 cycle, 2 or 3 phase general purpose and constant speed, 1160 or 1750 rpm. This table may also be applied with reasonable accuracy to 50 cycle, single phase a-c, 50 and 60 cycle enclosed and fractional horsepower polyphase motors.

Use of Table 53
- Heat Gain from Electric Motors

The data in *Table 53* includes the heat gain from electric motors and their driven machines when both the motor and the driven machine are in the conditioned space, or when only the driven machine is in the conditioned space, or when only the motor is in the conditioned space.

Caution: The power input to electric motors does not necessarily equal the rated horsepower divided by the motor efficiency. Frequently these motors may be operating under a continuous overload, or may be operating at less than rated capacity. It is always advisable to measure the power input wherever possible. This is especially important in estimates for industrial installations where the motor-machine load is normally a major portion of the cooling load.

When reading are obtained directly in watts and when both motors and driven machines are in the air conditioned space, the heat gain is equal to the number of watts times the factor 3.4 Btu/(watt)(hr).

When the machine is in the conditioned space and the motor outside, multiply the watts by the motor efficiency and by the factor 3.4 to determine heat gain to the space.

When the machine is outside the conditioned space, multiply the watts by one minus the motor efficiency and by the factor 3.4.

Although the results are less accurate, it may be expedient to obtain power input measurements using a clamp-on ammeter and voltmeter. These instruments permit instantaneous readings only. They afford means for determining the load factor but the usage factor must be obtained by a careful investigation of the operating conditions.

TABLE 53-HEAT GAIN FROM ELECTRIC MOTORS

CONTINUOUS OPERATION*

NAMEPLATE† OR BRAKE HORSEPOWER	FULL LOAD MOTOR EFFICIENCY PERCENT	LOCATION OF EQUIPMENT WITH RESPECT TO CONDITIONED SPACE OR AIR STREAM‡		
		Motor In- Driven Machine in $\frac{HP \times 2545}{\% Eff}$	Motor Out- Driven Machine in HP×2545	Motor In- Driven Machine out $\frac{HP \times 2545\ (1-\% Eff)}{\% Eff}$
		Btu per Hour		
1/20	40	320	130	190
1/12	49	430	210	220
1/8	55	580	320	260
1/6	60	710	430	280
1/4	64	1,000	640	360
1/3	66	1,290	850	440
1/2	70	1,820	1,280	540
3/4	72	2,680	1,930	750
1	79	3,220	2,540	680
1 1/2	80	4,770	3,820	950
2	80	6,380	5,100	1,280
3	81	9,450	7,650	1,800
5	82	15,600	12,800	2,800
7 1/2	85	22,500	19,100	3,400
10	85	30,000	25,500	4,500
15	86	44,500	38,200	6,300
20	87	58,500	51,000	7,500
25	88	72,400	63,600	8,800
30	89	85,800	76,400	9,400
40	89	115,000	102,000	13,000
50	89	143,000	127,000	16,000
60	89	172,000	153,000	19,000
75	90	212,000	191,000	21,000
100	90	284,000	255,000	29,000
125	90	354,000	318,000	36,000
150	91	420,000	382,000	38,000
200	91	560,000	510,000	50,000
250	91	700,000	636,000	64,000

*For intermittent operation, an appropriate usage factor should be used, preferably measured.

†If motors are overloaded and amount of overloading is unknown, multiply the above heat gain factors by the following maximum service factors:

Maximum Service Factors

Horsepower	1/20 - 1/8	1/6 - 1/3	1/2 - 3/4	1	1 1/2 - 2	3 - 250
AC Open Type	1.4	1.35	1.25	1.25	1.20	1.15
DC Open Type	--	--	--	1.15	1.15	1.15

No overload is allowable with enclosed motors

‡For a *fan or pump* in air conditioned space, exhausting air and pumping fluid to outside of space, use values in last column.

The following is a conversion table which can be used to determine load factors from measurements:

TO FIND →	HP OUTPUT	KILOWATTS INPUT
Direct Current	$\frac{I \times E \times eff}{746}$	$\frac{I \times E}{1,000}$
1 Phase	$\frac{I \times E \times pf \times eff}{746}$	$\frac{I \times E \times pf}{1,000}$
3 or 4 Wire 3 Phase	$\frac{I \times E \times pf \times eff \times 1.73}{746}$	$\frac{I \times E \times pf \times 1.73}{1,000}$
4 Wire 2 Phase	$\frac{I \times E \times pf \times eff \times 2}{746}$	$\frac{I \times E \times 2 \times pf}{1,000}$

Where I = amperes eff = efficiency

E = volts pf = power factor

NOTE: For 2 phase, 3 wire circuit, common conductor current is 1.41 times that in either of the other two conductors.

Example 3-Electric Motor Heat Gain in a Factory

(Motor Bhp Established by a Survey)

Given:

1. Forty-five 10 hp motors operated at 80% rated capacity, driving various types of machines located within air conditioned space (lathes, screw machines, etc.).
   Five 10 hp motors operated at 80% rated capacity, driving screw machines, each handling 5000 lbs of bronze per hr. Both the final product and the shaving from the screw machines are removed from the space on conveyor belts. Rise in bronze temperature is 30 F; sp ht is .01 Btu/(lb) (F).
2. Ten 5 hp motors (5 bhp) driving fans, exhausting air to the outdoors.
3. Three 20 hp motors (20 bhp) driving process water pumps, water discarded outdoors.

Find:

Total heat gain from motors.

Solution:

Use *Table 53*.	Sensible Heat Gain Btu/hr
1. Machines-Heat gain to space	
= 45 × 30,000 × .80=	1,080,000
Heat gain from screw machines	
= 5 × 30,000 × .80 = 120,000 Btu/hr	
Heat removed from space from screw machine work	
= 5000 × 5 × 30 × .01 = 7,500 Btu/hr	
Net heat gain from screw machines to space	
= 120,000 – 7500 =	112,500
2. Fan exhausting air to the outdoors:	
Heat gain to space = 10 × 2800 =	28,000
3. Process water pumped to outside air conditioned space	
Heat gain to space =3 × 7500	22,500
Total heat gain from motors on machines, fans, and pumps =	1,243,000

NOTE: If the process water were to be recirculated and cooled in the circuit from an outside source, the heat gain to the water

3 × (58,500 - 7500) = 153,000 Btu/hr

would become a load on this outside source.

PIPING, TANKS AND EVAPORATION OF WATER FROM A FREE SURFACE

Hot pipes and tanks add sensible heat to a space by convection and radiation. Conversely, cold pipes remove sensible heat. All open tanks containing hot water contribute not only sensible heat but also latent heat due to evaporation.

In industrial plants, furnaces or dryers are often encountered. These contribute sensible heat to the space by convection and radiation from the outside surfaces, and frequently dryers also contribute sensible and latent heat from the drying process.

Basis of Tables 54 thru 58

- Heat Gain from Piping, Tanks and Evaporation of Water

Table 54 is based on nominal flow in the pipe and a convection heat flow from a horizontal pipe of--

$$1.016 \times \left(\frac{1}{Dia}\right)^{.2} \times \left(\frac{1}{T_1}\right)^{.181}$$

× (temp diff between hot water or steam and room).

The radiation from horizontal pipes is expressed by-

$$17.23 \times 10^{-10} \times \text{emissivity} \times (T_1{}^4 - T_2{}^4)$$

where T_1 = room surface temp, deg R

T_2 = pipe surface temp, deg R

Tables 55 and 56 are based on the same equation and an insulation resistance of approximately 2.5 per inch of thickness for 85% magnesia and 2.9 per inch of thickness with moulded type.

Caution: *Table 55 and 56* do not include an allowance for fittings. A safety factor of 10% should be added for pipe runs having numerous fittings.

Table 57 is based on an emissivity of 0.9 for painted metal and painted or bare wood and concrete. The emissivity of chrome, bright nickel plate, stainless steel, or galvanized iron is 0.4. The resistance (r) of wood is 0.833 per inch and of concrete 0.08 per inch. The metal surface temperature has been assumed equal to the water temperature.

NOTE: The heat gain from furnaces and ovens can be estimated from *Table 57*, using the outside temperature of furnace and oven.

Table 58 is based on the following formula for still air: Heat of evaporation = 95 (vapor pressure differential between water and air), where vapor pressure is expressed in inches of mercury, and the room conditions are 75 F db and 50% rh.

Use of Tables 54 thru 58

- Heat Gain from Piping, Tanks and Evaporation of Water

Example 4-Heat Gain from Hot Water Pipe and Storage Tank

Given:

Room conditions – 75 F db, 50% rh

50 ft of 10-inch uninsulated hot water (125 F) pipe.

The hot water is stored in a 10 ft wide x 20 ft long x 10 ft high, painted metal tank with the top open to the atmosphere. The tank is supported on open steel framework.

Find:

Sensible and latent heat gain

Solution:

Use *Tables 54, 57 and 58*	Btu/hr
Piping-Sensible heat gain = 50 × 50 × 4.76 =	11,900
Tank - Sensible heat gain, sides	
= (20 × 10 × 2) + (10 × 10 × 2)	
× 50 × 1.8 =	54,000
- Sensible heat gain, bottom	
= (20×10) ×50×1.5=	15,000
Total sensible heat gain =	80,900
Total latent heat gain, top =(20 × 10) × 330 =	66,000

STEAM

When steam is escaping into the conditioned space, the room sensible heat gain is only that heat represented by the difference in heat content of steam at the steam temperature and at the room drybulb temperature (lb/hr × temp dift × .45). The latent heat gain is equal to the pounds per hour escaping times 1050 Btu/lb.

MOISTURE ABSORPTION

When moisture (regain) is absorbed by hygroscopic materials, sensible heat is added to the space. The heat so gained is equal to the latent heat of vaporization which is approximately 1050 Btu/lb times the pounds of water absorbed. This sensible heat is an addition to room sensible heat, and a deduction from room latent heat if the hygroscopic materials is removed from the conditioned space.

LATENT HEAT GAIN - CREDIT TO ROOM SENSIBLE HEAT

Some forms of latent heat gain reduce room sensible heat. Moisture evaporating at the room wet-bulb temperature (not heated or cooled from external source) utilizes room sensible heat for heat of evaporation. This form of latent heat gain should be deducted from room sensible heat and added to room latent heat. This does not change the total room heat gain, but may have considerable effect on the sensible heat factor.

When the evaporation of moisture derives its heat from another source such as steam or electric heating coils, only the latent heat gain to the room is figured; room sensible heat is not reduced. The power input to the steam or electric coils balances the heat of evaporation except for the initial warmup of the water.

TABLE 54-HEAT TRANSMISSION COEFFICIENTS FOR BARE STEEL PIPES

Btu/(hr) (linear ft) (deg F diff between pipe and surrounding air)

NOMINAL PIPE SIZE (in.)	HOT WATER				STEAM		
	120 F	150 F	180 F	210 F	5 psig 227 F	50 psig 300 F	100 psig 338 F
	TEMPERATURE DIFFERENCE*						
	50 F	80 F	110 F	140 F	157 F	230 F	268 F
1/2	0.46	0.50	0.55	0.58	0.61	0.71	0.76
3/4	0.56	0.61	0.67	0.72	0.75	0.87	0.93
1	0.68	0.74	0.82	0.88	0.92	1.07	1.15
1 1/4	0.85	0.92	1.01	1.09	1.14	1.32	1.43
1 1/2	0.96	1.04	1.15	1.23	1.29	1.49	1.63
2	1.18	1.28	1.41	1.51	1.58	1.84	1.99
2 1/2	1.40	1.53	1.68	1.80	1.88	2.19	2.36
3	1.68	1.83	2.01	2.15	2.26	2.63	2.84
3 1/2	1.90	2.06	2.22	2.43	2.55	2.97	3.22
4	2.12	2.30	2.53	2.72	2.85	3.32	3.59
5	2.58	2.80	3.08	3.30	3.47	4.05	4.39
6	3.04	3.29	3.63	3.89	4.07	4.77	5.16
8	3.88	4.22	4.64	4.96	5.21	6.10	6.61
10	4.76	5.18	5.68	6.09	6.41	7.49	8.12
12	5.59	6.07	6.67	7.15	7.50	8.80	9.53

*At 70 F db room temperature

TABLE 55-HEAT TRANSMISSION COEFFICIENTS FOR INSULATED PIPES*

Btu/(hr) (linear ft) (deg F diff between pipe and room)

IRON PIPE SIZE (In.)	85 PERCENT MAGNESIA INSULATION†		
	1 In. Thick	1½ In. Thick	2 In. Thick
1/2	0.16	0.14	0.12
3/4	0.18	0.15	0.13
1	0.20	0.17	0.15
1 1/4	0.24	0.20	0.17
1 1/2	0.26	0.21	0.18
2	0.30	0.24	0.21
2 1/2	0.35	0.27	0.24
3	0.40	0.32	0.27
3 1/2	0.45	0.35	0.30
4	0.49	0.38	0.32
5	0.59	0.45	0.38
6	0.68	0.52	0.43
8	0.85	0.65	0.53
10	1.04	0.78	0.64
12	1.22	0.90	0.73

* No allowance for fittings. This table applies only to straight runs of pipe. When numerous fittings exists, a suitable safely factor must be included. This added heat gain at the fittings may be as much as 10%. Generally this table can be used without adding this safety factor.

†Other insulation. If other types of insulation are used, multiply the above values by the factors shown in the following table:

MATERIAL	PIPE COVERING FACTORS
Corrugated Asbestos (Air Cell)	
4 Ply per inch	1.36
6 Ply per inch	1.23
8 Ply per inch	1.19
Laminated Asbestos (Sponge Felt)	0.98
Mineral Wool	1.00
Diatomaceous Silica (Super-X)	1.36
Brown Asbestos Fiber (Wool Felt)	0.88

TABLE 56-HEAT TRANSMISSION COEFFICIENTS FOR INSULATED COLD PIPES*

MOULDED TYPE†

Btu/(hr) (linear ft) (deg F diff between pipe and room)

IRON PIPE SIZE (in.)	ICE WATER		BRINE		HEAVY BRINE	
	Actual Thickness of Insulation (In.)	Coefficient	Actual Thickness of Insulation (In.)	Coefficient	Actual Thickness of Insulation (In.)	Coefficient
1/2	1.5	0.11	2.0	0.10	2.8	0.09
3/4	1.6	0.12	2.0	0.11	2.9	0.09
1	1.6	0.14	2.0	0.12	3.0	0.10
1 1/4	1.6	0.16	2.4	0.13	3.1	0.11
1 1/2	1.5	0.17	2.5	0.13	3.2	0.12
2	1.5	0.20	2.5	0.15	3.3	0.13
2 1/2	1.5	0.23	2.6	0.17	3.3	0.15
3	1.5	0.27	2.7	0.19	3.4	0.16
3 1/2	1.5	0.29	2.9	0.19	3.5	0.18
4	1.7	0.30	2.9	0.21	3.7	0.18
5	1.7	0.35	3.0	0.24	3.9	0.20
6	1.7	0.40	3.0	0.26	4.0	0.23
8	1.9	0.46	3.0	0.32	4.0	0.26
10	1.9	0.56	3.0	0.38	4.0	0.31
12	1.9	0.65	3.0	0.44	4.0	0.36

*No allowance for fittings. This table applies only to straight runs of pipe. When numerous fittings exist, a suitable safety factor must be included. This added heat gain at the fitting may be as much as 10%. Generally this table can be used without adding this safety factor.

†Insulation material. Values in this table are based on a material having a conductivity k=0.30. However, a 15% safety factor was added to this k value to compensate for seams and imperfect workmanship. The table applies to either cork covering (k=0.29), or mineral wool board (k = 0.32). The thickness given above is for molded mineral wool board which is usually some 5 to 10% greater than molded cork board.

TABLE 57-HEAT TRANSMISSION COEFFICIENTS FOR UNINSULATED TANKS

SENSIBLE HEAT GAIN*

Btu/(hr) (sq ft) (deg F diff between liquid and room)

CONSTRUCTION	METAL								WOOD 2½ in. Thick				CONCRETE 6 in. Thick			
	Painted				Bright (Nickel)				Painted or Bare				Painted or Bare			
	Temp Diff				Temp Diff				Temp Diff				Temp Diff			
	50 F	100 F	150 F	200 F	50 F	100 F	150 F	200 F	50 F	100 F	150 F	200 F	50 F	100 F	150 F	200 F
Vertical (Sides)	1.8	2.0	2.3	2.6	1.3	1.7	1.6	1.7	.37	.37	.37	.37	.91	.93	.96	.97
Top	2.1	2.4	2.7	2.9	1.6	1.4	1.9	2.1	.38	.38	.38	.38	.99	1.0	1.0	1.1
Bottom	1.5	1.7	2.0	2.2	0.97	1.1	1.3	1.4	.35	.36	.36	.36	.83	.86	.88	.90

*To estimate latent heat load if water is being evaporated, see *Table 58*

TABLE 58-EVAPORATION FROM A FREE WATER SURFACE-LATENT HEAT GAIN

STILL AIR, ROOM AT 75 F db, 50% RH

WATER TEMP	75 F	100 F	125 F	150 F	175 F	200 F
Btu/(hr) (sq ft)	42	140	330	680	1260	2190

SYSTEM HEAT GAIN

The system heat gain is considered as the heat added to or lost by the system components, such as the ducts, piping, air conditioning fan, and pump, etc. This heat gain must be estimated and included in the load estimate but can be accurately evaluated only after the system has been designed.

SUPPLY AIR DUCT HEAT GAIN

The supply duct normally has 50 F db to 60 F db air flowing through it. The duct may pass through an unconditioned space having a temperature of, say, 90 F db and up. This results in a heat gain to the duct before it reaches the space to be conditioned. This, in effect, reduces the cooling capacity of the conditioned air. To compensate for it, the cooling capacity of the air quantity must be increased. It is recommended that long runs of ducts in unconditioned spaces be insulated to minimize heat gain.

Basis or Chart 3

- Percent Room Sensible Heat to be Added for Heat Gain to Supply Duct

Chart 3 is based on a difference of 30 F db between supply air and unconditioned space, a supply duct velocity of 1800 fpm in a quare duct, still air on the outside of the duct and a supply air rise of 17 F db.

Correction factors for different room temperatures, duct velocities and temperature differences are included below *Chart 3*. Values are plotted for use with uninsulated, furred and insulated ducts.

Use of Chart 3

-- Percent Room Sensible Heat to be Added for Heat Gain to Supply Duct

To use this chart, evaluate the length of duct running thru the unconditioned space, the temperature of unconditioned space, the duct velocity, the supply air temperature, and room sensible heat subtotal.

Example 5- Heat Gain to Supply Duct

Given:

20 ft of uninsulated duct in unconditioned space at 100 F db

Duct velocity – 2000 fpm

Supply air temperature – 60 F db

Room sensible heat gain – 100,000 Btu/hr

Find:

Percent addition to room sensible heat

Solution:

The supply air to unconditioned space temperature difference = 100 – 60 = 40 F db

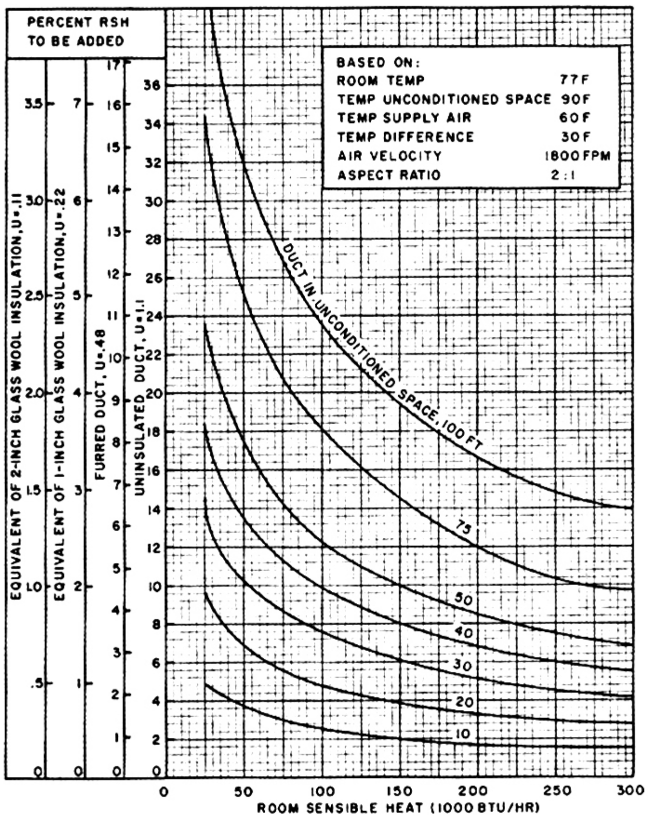
From *Chart 3*, percent addition = 4.5%

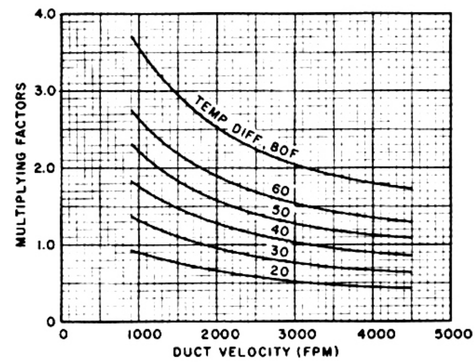
Correction for 40 F db temperature difference and 2000 fpm duct velocity = 1.26

Actual percent addition = 4.5 × 1.26 = 5.7%

CHART 3- HEAT GAIN TO SUPPLY DUCT

Percent of Room Sensible Heat





MULTIPLYING FACTORS FOR OTHER ROOM TEMPERATURES

Room Temp	Multiplying Factor
75	1.10
76	1.06
77	1.00
78	0.97
79	0.94
80	0.92

$$Q = UPl \times \frac{2.165 \times AV}{(2.165 \times AV) + UPl} (t_3 - t_1)$$

where:

- Q = duct heat gain (Btu/hr)
- U = duct heat transmission factor (Btu/hr-sq ft-F)
- P = rectangular duct perimeter (ft)
- l = duct length (ft)
- A = duct area (sq ft)
- V = duct velocity (fpm)
- t_1 = temperature of supply air entering duct (F)
- t_3 = temperature of surrounding air (F)

Based on formulas in ASHRAE Guide 1963, p. 184, 185.

SUPPLY AIR DUCT LEAKAGE LOSS

Air leakage from the supply duct may be a serious loss of cooling effect, except when it leaks into the conditioned space. This loss of cooling effect must be added to the room sensible and latent heat load.

Experience indicates that the average air leakage from the entire length of *low velocity* supply ducts, whether large or small systems, averages around 10% of the supply air quantity. Smaller leakage per foot of length for larger perimeter ducts appears to be counterbalanced by the longer length of run. Individual workmanship is the greatest variable, and duct leakages from 5% to 30% have been found. The following is a guide to the evaluation of duct leakages under various conditions:

1. Bare ducts within conditioned space-usually not necessary to figure leakage.
2. Furred or insulated ducts within conditioned space-a matter of judgment, depending on whether the leakage air actually gets into the room.

TABLE 59- HEAT GAIN FROM AIR CONDITIONING FAN HORSEPOWER, DRAW-THRU SYSTEM‡‡

	FAN TOTAL PRESSURE† (In. of Water)	CENTRAL STATION SYSTEMS‡					APPLIED OR UNITARY SYSTEM**				
		Temp Diff Room to Supply Air					Temp Diff Room to Supply Air				
		10 F	15 F	20 F	25 F	30 F	10 F	15 F	20 F	25 F	30 F
		PERCENT OF ROOM SENSIBLE HEAT*									
Fan Motor Not in Conditioned Space or Air Stream	0.50	1.2	0.8	0.6	0.5	0.4	2.2	1.5	1.1	0.9	0.7
	0.75	1.9	1.3	1.0	0.8	0.6	3.5	2.4	1.8	1.4	1.2
	1.00	2.7	1.8	1.4	1.1	0.9	4.8	3.2	2.4	1.9	1.6
	1.25	3.9	2.6	1.9	1.6	1.3	6.5	4.3	3.2	2.6	2.2
	1.50	4.6	3.1	2.3	1.9	1.6	7.8	5.2	3.9	3.1	2.6
	1.75	5.4	3.6	2.7	2.2	1.8	9.1	6.1	4.6	3.6	3.0
	2.00	6.2	4.1	3.1	2.5	2.1	10.4	6.9	5.2	4.2	3.5
	3.00	10.4	6.9	5.2	4.2	3.5	16.7	11.2	8.4	6.7	5.6
	4.00	15.3	10.2	7.7	6.1	5.1					
	5.00	19.2	12.8	9.6	7.7	6.4					
	6.00	24.4	16.3	12.2	9.9	8.2					
	8.00	38.0	25.4	19.0	15.2	12.7					
Fan Motor†† in Conditioned Space or Air Stream	0.50	1.6	1.1	0.8	0.6	0.5	2.7	1.8	1.4	1.1	0.9
	0.75	2.6	1.8	1.3	1.1	0.9	4.2	2.8	2.1	1.7	1.4
	1.00	3.6	2.4	1.8	1.5	1.2	5.8	3.8	2.9	2.3	1.9
	1.25	5.0	3.4	2.5	2.0	1.7	7.6	5.1	3.8	3.1	2.6
	1.50	6.0	4.0	3.0	2.4	2.0	9.2	6.1	4.6	3.7	3.1
	1.75	7.0	4.7	3.5	2.8	2.4	10.7	7.2	5.4	4.3	3.6
	2.00	8.0	5.4	4.0	3.2	2.7	12.2	8.2	6.1	4.9	4.1
	3.00	13.2	8.8	6.6	5.3	4.4	19.5	13.1	9.8	7.8	6.5
	4.00	19.0	12.7	9.5	7.6	6.4					
	5.00	23.8	15.9	11.9	9.5	8.0					
	6.00	30.0	20.0	15.0	12.0	10.0					
	8.00	45.5	30.3	22.8	18.2	15.2					

*Excludes from heat gain, typical values for bearing losses, etc. which are dissipated in apparatus room.

†Fan Total Pressure equals fan static pressure plus velocity pressure at fan discharge. Below 1200 fpm the fan total pressure is approximately equal to the fan static. Above 1200 fpm the total pressure should be figured.

‡70% fan efficiency assumed.

**50% fan efficiency assumed.

††80% motor and drive efficiency assumed.

‡‡For draw-thru systems, this heat is an addition to the supply air heat gain and is added to the room sensible heat. For blow-thru systems this fan heat is added to the grand total heat; use the RSH times the percent listed and add to the GTH.

3. All ducts outside the conditioned space- assume 10% leakage. This leakage is a total loss and the full amount must be included. When only part of the supply duct is outside the conditioned space, include that fraction of 10% as the leakage. (Fraction is ratio of length outside of conditioned space to total length of supply duct.)

High velocity systems usually limit leakage to 1%.

HEAT GAIN FROM AIR CONDITIONING FAN HORSEPOWER

The inefficiency of the air conditioning equipment fan and the heat of compression adds heat to the system as described under "*Electric Motors*." In the case of draw-through systems, this heat is an addition to the supply air heat gain and should be added to the room sensible heat. With blow-through systems (fan blowing air through the coil, etc.) the fan heat added is a load on the dehumidifier and, therefore, should be added to the grand total heat *(see "Percent Addition to Grand Total Heat")*.

Basis of Table 59

— Heat Gain from Air Conditioning Fan Horsepower

The air conditioning fan adds heat to the system in the following manner:

1. Immediate temperature rise in the air due to the inefficiency of the fan.
2. Energy gain in the air as a pressure and/or velocity rise.

3. With the motor and drive in the air stream or conditioned space, the heat generated by the inefficiency of the motor and drive is also an immediate heat gain.

The fan efficiencies are about 70% for central station type fans and about 50% for packaged equipment fans.

Use of Table 59
-- Heat Gain from Air Conditioning Fan Horsepower

The approximate system pressure loss and dehumidified air rise (room minus supply air temperature) differential must be estimated from the system characteristics and type of application. These should be checked from the final system design.

The normal comfort application has a dehumidified air rise of between 15 F db and 25 F db and the fan total pressure depends on the amount of ductwork involved, the number of fittings (elbows, etc.) in the ductwork and the type of air distribution system used. Normally, the fan total pressure can be approximated as follows:

1. No ductwork (packaged equipment) – 0.5 to 1.00 inches of water.
2. Moderate amount of ductwork, low velocity systems - 0.75 to 1.50 inches of water.
3. Considerable ductwork, low velocity system- 1.25 to 2.00 inches of water.
4. Moderate amount of ductwork, high pressure system - 2.00 to 4.00 inches of water.
5. Considerable ductwork, high pressure system – 3.00 to 6.00 inches of water.

Example 6- Heat Gain from Air Conditioning Fan Horsepower

Given:

Same data as *Example 5*

80 ft of supply duct in conditioned space

Find:

Percent addition to room sensible heat.

Solution:

Assume 1.50 inches of water, fan total pressure, and 20 F db dehumidifer rise. Refer to *Table 59*.

Heat gain from fan horsepower = 2.3%

SAFETY FACTOR AND PERCENT ADDITIONS TO ROOM SENSIBLE AND LATENT HEAT

A safety factor to be added to the room sensible heat sub-total should be considered as strictly a factor of probable error in the survey or estimate, and should usually be between 0% and 5%.

The total room sensible heat is the sub-total plus percentage additions to allow for (1) supply duct heat gain, (2) supply duct leakage losses, (3) fan horsepower and (4) safety factor, as explained in the preceding paragraph.

Example 7-Percent Addition to Room Sensible Heat

Given:

Same data as *Examples 5 and 6*

Find:

Percent addition to room sensible heat gain sub-total

Solution:

Supply duct heat gain	=	5.7%
Supply duct leakage (20 ft duct of total 100 ft)	=	2.0%
Fan horsepower	=	2.3%
Safety factor	=	0.0%
Total percent addition to RSH	=	10.0%

The percent additions to room latent heat for supply duct leakage loss and safety factor should be the same as the corresponding percent additions to room sensible heat.

RETURN AIR DUCT HEAT AND LEAKAGE GAIN

The evaluation of heat and leakage effects on return air ducts is made in the same manner as for supply air ducts, except that the process is reversed; there is inward gain of hot moist air instead of loss of cooling effect.

Chart 3 can be used to approximate heat gain to the return duct system in terms of percent of RSH, using the following procedure:

1. Using RSH and the length of return air duct, use *Chart 3* to establish the percent heat gain.
2. Use the multiplying factor from table below *Chart 3* to adjust the percent heat gain for actual temperature difference between the air surrounding the return air duct and the air inside the duct, and also for the actual velocity.
3. Multiply the resulting percentage of heat gain by the ratio of RSH to GTH.
4. Apply the resulting heat gain percentage to GTH.

To determine the return air duct leakage, apply the following reasoning:

1. Bare duct within conditioned space – no in-leakage.
2. Furred duct within conditioned space or furred space used for return air – a matter of judgment, depending on whether the furred space may connect to unconditioned space.

TABLE 60-HEAT GAIN FROM DEHUMIDIFIER PUMP HORSEPOWER

PUMP HEAD (ft)	SMALL PUMPS* 0-100 GPM					LARGE PUMPS† 100 GPM AND LARGER				
	CHILLED WATER TEMP RISE					CHILLED WATER TEMP RISE				
	5 F	7 F	10 F	12 F	15 F	5 F	7 F	10 F	12 F	15 F
	PERCENT OF GRAND TOTAL HEAT									
35	2.0	1.5	1.0	1.0	0.5	1.5	1.0	0.5	0.5	0.5
70	3.5	2.5	2.0	1.5	1.0	2.5	2.0	1.5	1.0	1.0
100	5.0	4.0	2.5	2.0	1.5	4.0	3.0	2.0	1.5	1.0

*Efficiency 50% †Efficiency 70%

3. Ducts outside conditioned space – assume up to 3% inleakage, depending on the length of duct. If there is only a short connection between conditioned space and apparatus, inleakage may be disregarded. If there is a long run of duct, then apply judgment as to the amount of inleakage.

HEAT GAIN FROM DEHUMIDIFIER PUMP HORSEPOWER

With dehumidifier systems, the horsepower required to pump the water adds heat to the system as outlined under "Electric Motors". This heat will be an addition to the grand total heat.

Basis of Table 60

-- Heat Gain from Dehumidifier Pump Horsepower

Table 60 is based on pump efficiencies of 50% for small pumps and 70% for large pumps. Small pumps are considered to have a capacity of less than 100 gallons; large pumps, more than 100 gallons.

Use of Table 60

-- Heat Gain from Dehumidifier Pump Horsepower

The chilled water temperature rise in the dehumidifier and the pump head must be approximated to use *Table 60.*

1. Large systems with considerable piping and fitting may require up to 100 ft pump head; normally, 70 ft head is the average.
2. The normal water temperature rise in the dehumidifier is between 7 F and 12 F. Applications using large amounts of water have a lower rise; those using small amounts of water have a higher rise.

PERCENT ADDITION TO GRAND TOTAL HEAT

The percent additions to the grand total heat to compensate for various external losses consist of heat and leakage gain to return air ducts, heat gain from the dehumidifier pump horsepower, and the heat gain to the dehumidifier and piping system.

These heat gains can be estimated as follows:

1. Heat and leakage gain to return air ducts, see above.
2. Heat gain from dehumidifier pump horsepower, *Table 60*.
3. Dehumidifier and piping losses:
   a. Very little external piping - 1% of GTH.
   b. Average external piping - 2% of GTH.
   c. Extensive external piping - 4% of GTH.
4. Blow-through fan system-add percent room sensible heat from *Table 59* to GTH.
5. Dehumidifier in conditioned apparatus room-reduce the above percentages by one half.

CHAPTER 8. APPLIED PSYCHROMETRICS

The preceding chapters contain the practical data to properly evaluate the heating and cooling loads. They also recommend outdoor air quantities for ventilation purposes in areas where state, city or local codes do not exist.

This chapter describes practical psychrometrics as applied to apparatus selection. It is divided into three parts:

1. *Description of terms, processes and factors*-as encountered in normal air conditioning applications.
2. *Air conditioning apparatus*-factors affecting common processes and the effect of these factors on selection of air conditioning equipment.
3. *Psychrometrics of partial load control* – the effect of partial load on equipment selection and on the common processes.

To help recognize terms, factors and processes described in this chapter, a brief definition of psychrometrics is offered at this point, along with an illustration and definition of terms appearing on a standard psychrometric chart *(Fig. 32)*.

Dry-bulb Temperature — The temperature of air as registered by an ordinary thermometer.

Wet-bulb Temperature — The temperature registered by a thermometer whose bulb is covered by a wetted wick and exposed to a current of rapidly moving air.

Dewpoint Temperature — The temperature at which condensation of moisture begins when the air is cooled.

Relative Humidity — Ratio of the actual water vapor pressure of the air to the saturated water vapor pressure of the air at the same temperature.

Specific Humidity or Moisture Content — The weight of water vapor in grains or pounds of moisture per pound of dry air.

Enthalpy — A thermal property indicating the quantity of heat in the air above an arbitrary datum, in Btu per pound of dry air. The datum for dry air is 0°F and, for the moisture content, 32 F water.

Enthalpy Deviation — Enthalpy indicated above, for any given condition, is the enthalpy of saturation. It should be corrected by the enthalpy deviation due to the air not being in the saturated state. Enthalpy deviation is in Btu per pound of dry air. Enthalpy deviation is applied where extreme accuracy is required; however, on normal air conditioning estimates it is omitted.

Specific Volume — The cubic feet of the mixture per pound of dry air.

Sensible Heat Factor — The ratio of sensible to total heat.

Alignment Circle — Located at 80 F db and 50% rh and used in conjunction with the sensible heat factor to plot the various air conditioning process lines.

Pounds of Dry Air — The basis for all psychrometric calculations. Remains constant during all psychrometric processes.

The dry-bulb, wet-bulb, and dewpoint temperatures and the relative humidity are so related that, if two properties are known, all other properties shown may then be determined. When air is saturated, dry-bulb, wet-bulb, and dewpoint temperatures are all equal.

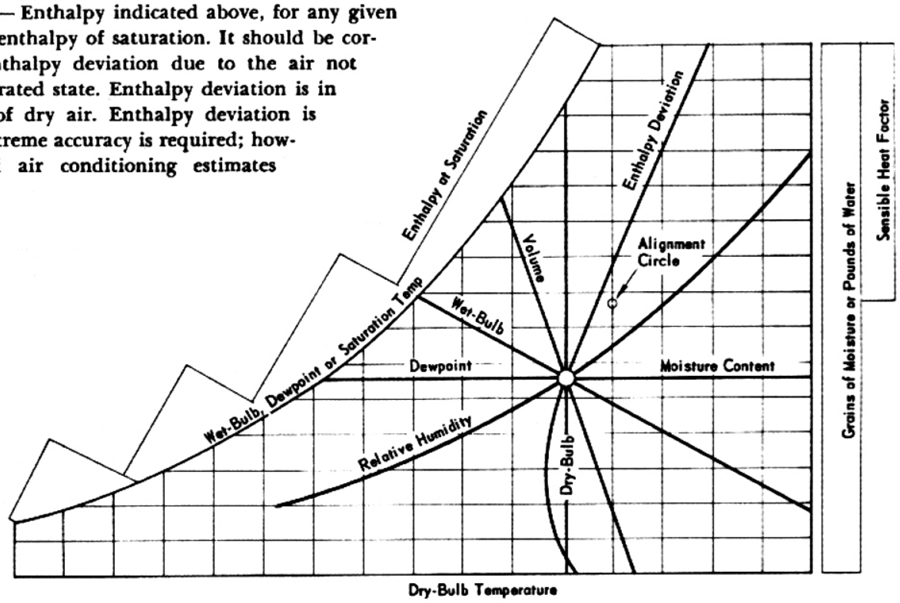


FIG. 32 – SKELETON PSYCHROMETRIC CHART

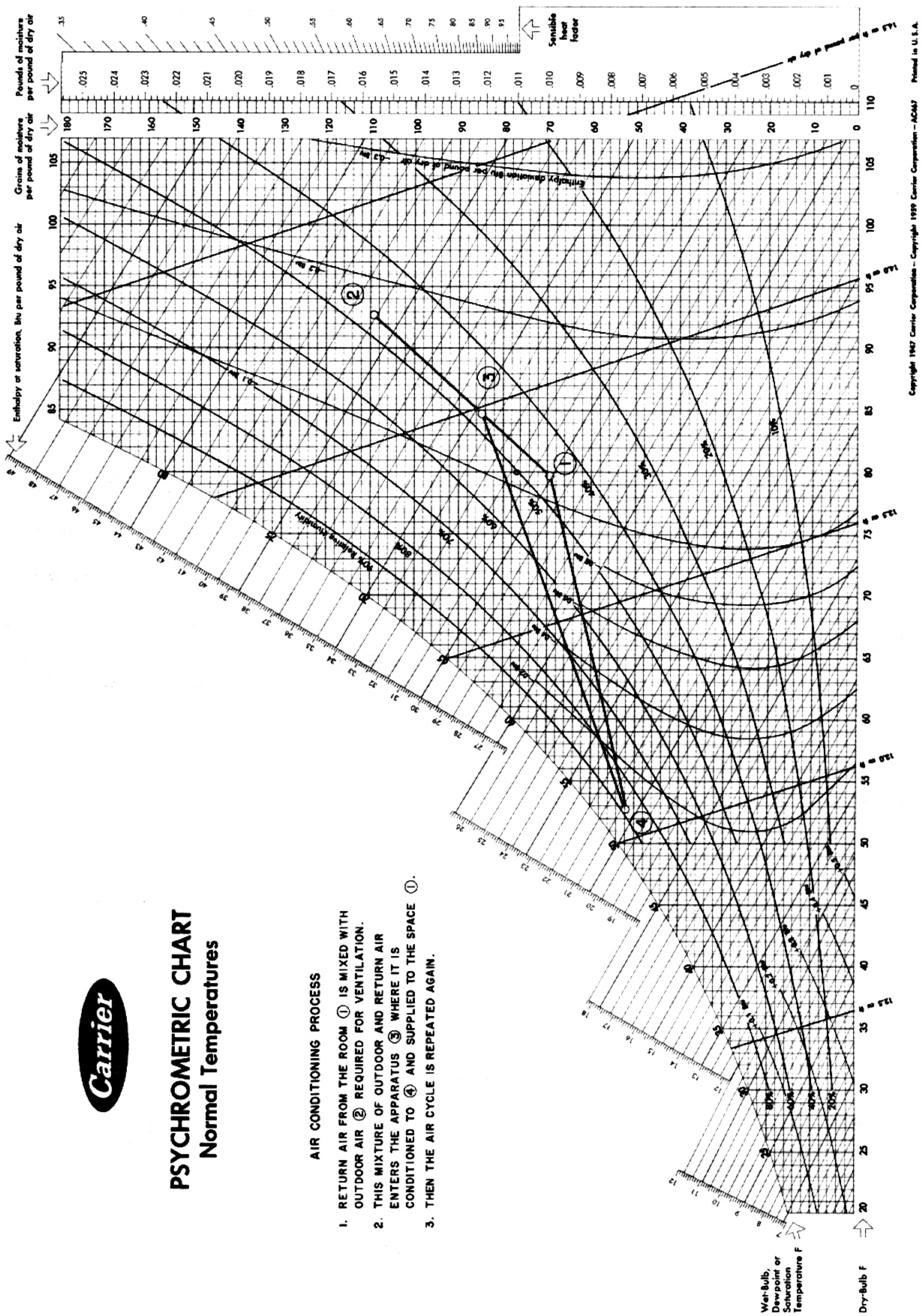


FIG. 33-TYPICAL AIR CONDITIONING PROCESS TRACED ON A STANDARD PSYCHROMETRIC CHART

DEFINITION

Psychrometrics is the science involving thermodynamic properties of moist air and the effect of atmospheric moisture on materials and human comfort. As it applies to this chapter, the definition must be broadened to include the method of controlling the thermal properties of moist air.

AIR CONDITIONING PROCESSES

Fig. 33 shows a typical air conditioning process traced on a psychrometric chart. Outdoor air (*2*)* is mixed with return air from the room (*1*) and enters the apparatus (*3*). Air flows through the conditioning apparatus (*3-4*) and is supplied to the space (*4*). The air supplied to the space moves along line (*4-1*) as it picks up the room loads, and the cycle is repeated. Normally most of the air supplied to the space by the air conditioning system is returned to the conditioning apparatus. There, it is mixed with outdoor air required for ventilation. The mixture then passes thru the apparatus where heat and moisture are added or removed, as required, to maintain the desired conditions.

The selection of proper equipment to accomplish this conditioning and to control the thermodynamic properties of the air depends upon a variety of elements. However, only those which affect the psychrometric properties of air will be discussed in this chapter. These elements are: room sensible heat factor (RSHF)†, grand sensible heat factor (GSHF), effective surface temperature (t_{es}), bypass factor (BF), and effective sensible heat factor (ESHF).

DESCRIPTION OF TERMS, PROCESSES AND FACTORS

SENSIBLE HEAT FACTOR

The thermal properties of air can be separated into latent and sensible heat. The term *sensible heat factor* is the ratio of sensible to total heat, where total heat is the sum of sensible and latent heat. This ratio may be expressed as:

$$SHF = \frac{SH}{SH + LH} = \frac{SH}{TH}$$

where: SHF = sensible heat factor
SH = sensible heat
LH = latent heat
TH = total heat

ROOM SENSIBLE HEAT FACTOR (RSHF)

The *room sensible heat factor* is the ratio of room sensible heat to the summation of room sensible and room latent heat. This ratio is expressed in the following formula:

$$RSHF = \frac{RSH}{RSH + RLH} = \frac{RSH}{RTH}$$

The supply air to a conditioned space must have the capacity to offset simultaneously both the room sensible and room latent heat loads. The room and the supply air conditions to the space may be plotted on the standard psychrometric chart and these points connected with a straight line (*1-2*), *Fig. 34*. This line represents the psychrometric process of the supply air within the conditioned space and is called the room sensible heat factor line.

The slope of the RSHF line illustrates the ratio of sensible to latent loads within the space and is illustrated in *Fig. 34* by Δh_s (sensible heat) and Δh_l (latent heat). Thus, if adequate air is supplied to offset these room loads, the room requirements will be satisfied, provided both the dry-and wet-bulb temperatures of the supply air fall on this line.

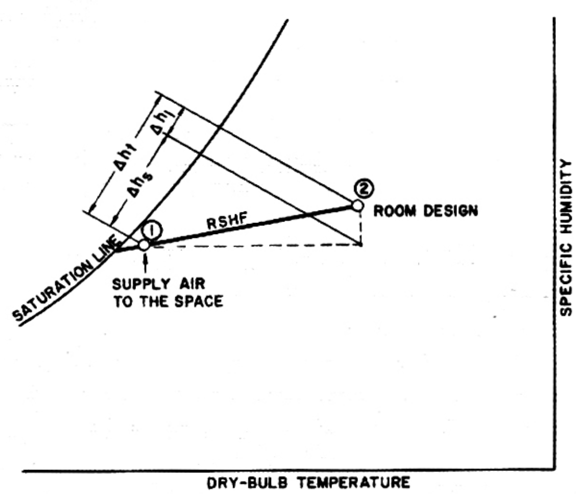


FIG. 34-RSHF LINE PLOTTED BETWEEN ROOM AND SUPPLY AIR CONDITIONS

*One italic number in parentheses represents a point, and two italic numbers in parentheses represent a line, plotted on the accompanying psychrometric chart examples.

†Refer to page 149 for a description of all abbreviations and symbols used in this chapter.

The room sensible heat factor line can also be drawn on the psychrometric chart without knowing the condition of supply air. The following procedure illustrates how to plot this line, using the calculated RSHF, the room design conditions, the sensible heat factor scale in the upper right hand corner of the psychrometric chart, and the alignment circle at 80 F dry-bulb and 50% relative humidity:

1. Draw a base line thru the alignment circle and the calculated RSHF shown on the sensible heat factor scale in the upper right corner of psychrometric chart *(1-2), Fig. 35*.
2. Draw the actual room sensible heat factor line thru the room design conditions parallel to the base line in *Step 1 (3-4), Fig. 35.* As shown, this line may be drawn to the saturation line on the psychrometric chart.

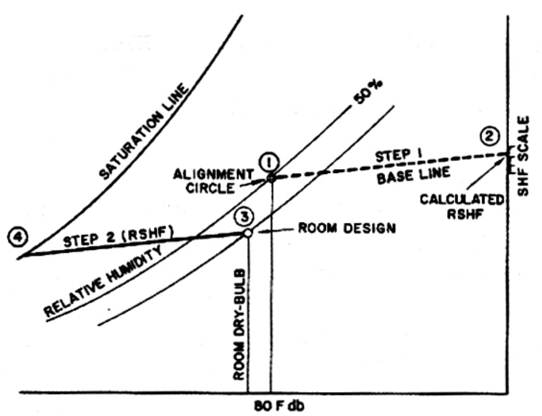


FIG. 35-RSHF LINE PLOTTED ON SKELETON PSYCHROMETRIC CHART

GRAND SENSIBLE HEAT FACTOR (GSHF)

The *grand sensible heat factor* is the ratio of the total sensible heat to the grand total heat load that the conditioning apparatus must handle, including the outdoor air heat loads. This ratio is determined from the following equation:

$$\text{GSHF} = \frac{\text{TSH}}{\text{TLH} + \text{TSH}} = \frac{\text{TSH}}{\text{GTH}}$$

Air passing thru the conditioning apparatus increases or decreases in temperature and/or moisture content. The amount of rise or fall is determined by the total sensible and latent heat loads that the conditioning apparatus must handle. The condition of the air entering the apparatus (mixture condition of outdoor and return room air) and the condition of the air leaving the apparatus may be plotted on the psychrometric chart and connected by a straight line (1-2), *Fig. 36*. This line represents the psychrometric process of the air as it passes through the conditioning apparatus, and is referred to as the grand sensible heat factor line.

The slope of the GSHF line represents the ratio of sensible and latent heat that the apparatus must handle. This is illustrated in *Fig. 36* by Δh_s (sensible heat) and Δh_1 (latent heat).

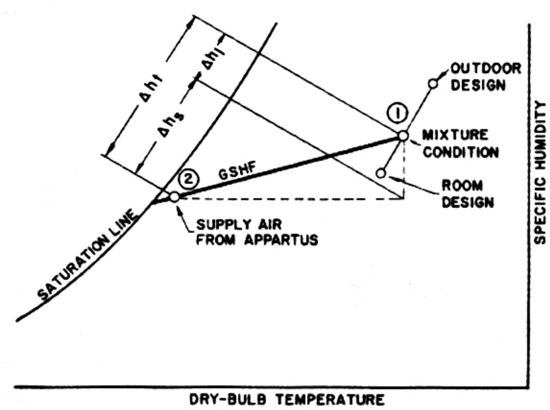


FIG. 36-GSHF LINE PLOTTED BETWEEN MIXTURE CONDITIONS TO APPARATUS AND LEAVING CONDITION FROM APPARATUS

The grand sensible heat factor line can be plotted on the psychrometric chart without knowing the condition of supply air, in much the same manner as the RSHF line. *Fig. 37, Step 1 (1-2)* and *Step 2 (3-4)* show the procedure, using the calculated GSHF, the mixture condition of air to the apparatus, the sensible heat factor scale, and the alignment circle on the psychrometric chart. The resulting GSHF line is plotted thru the mixture conditions of the air to the apparatus.

REQUIRED AIR QUANTITY

The air quantity required to offset simultaneously the room sensible and latent loads and the air quantity required thru the apparatus to handle the total sensible and latent loads may be calculated, using the conditions on their respective RSHF and GSHF lines. For a particular application, when both the RSHF and GSHF ratio lines are plotted on the psychrometric chart, the intersection of the two lines *(1) Fig. 38*, represents the condition of the supply air to the space. It is also the condition of the air leaving the apparatus.

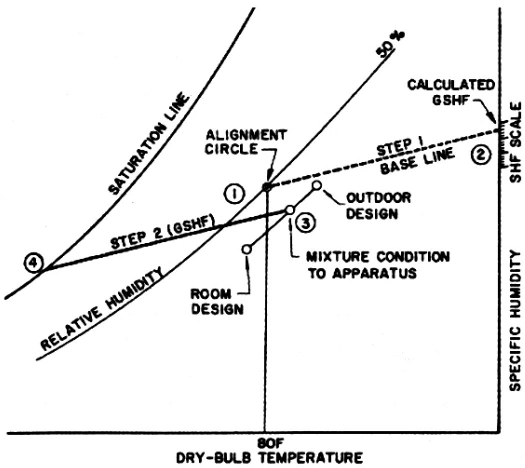


FIG. 37 – GSHF LINE PLOTTED ON SKELETON PSYCHROMETRIC CHART

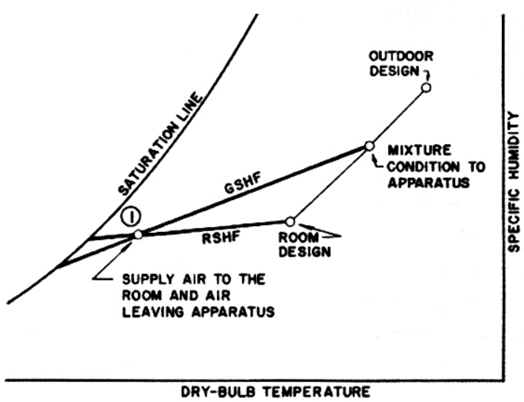


FIG. 38 – RSHF AND GSHF LINES PLOTTED ON SKELETON PSYCHROMETRIC CHART

This neglects fan and duct heat gain, duct leakage losses, etc. In actual practice, these heat gains and losses are taken into account in estimating the cooling load. *Chapter 7* gives the necessary data for evaluating these supplementary loads. Therefore, the temperature of the air leaving the apparatus is not necessarily equal to the temperature of the air supplied to the space as indicated in *Fig. 38*.

Fig. 39 illustrates what actually happens when these supplementary loads are considered in plotting the RSHF and GSHF lines.

Point (*1*) is the condition of air leaving the apparatus and point (*2*) is the condition of supply air to the space. Line (*1-2*) represents the temperature rise of the air stream resulting from fan horsepower and heat gain to the duct.

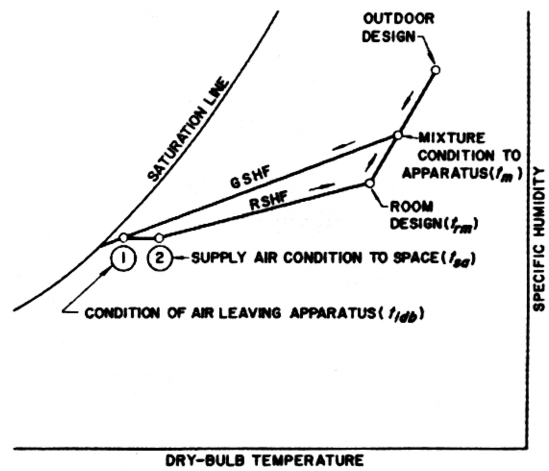


FIG. 39-RSHF AND GSHF LINES PLOTTED WITH SUPPLEMENTARY LOAD LINE

The air quantity required to satisfy the room load may be calculated from the following equation:

$$cfm_{sa} = \frac{\text{RSH}}{1.08(t_{rm} - t_{sa})}$$

The air quantity required thru the conditioning apparatus to satisfy the total air conditioning load (including the supplementary loads) is calculated from the following equation:

$$cfm_{da} = \frac{\text{TSH}}{1.08(t_m - t_{ldb})}$$

The required air quantity supplied to the space is equal to the air quantity required thru the apparatus, neglecting leakage losses. The above equation contains the term t_m which is the mixture condition of air entering the apparatus. With the exception of an all outdoor air application, the term t_m can only be determined by trial and error.

One possible procedure to determine the mixture temperature and the air quantities is outlined below. This procedure illustrates one method of apparatus selection and is presented to show how cumbersome and time consuming it may be.

1. Assume a rise *(t_{rm}-t_{sa})* in the supply air to the space, and calculate the supply air quantity *(cfm_{sa})* to the space.
2. Use this air quantity to calculate the mixture condition of the air *(t_m)* to the space, *(Equation 1, page 150)*.
3. Substitute this supply air quantity and mixture condition of the air in the formula for air quantity thru the apparatus (cfm_{da}) and determine the leaving condition of the air from the conditioning apparatus *(t_{ldb})*.
4. The rise between the leaving condition from the apparatus and supply air condition to the space (t_{sa}-t_{1db}) must be able to handle the supplementary loads (duct heat gain and fan heat). These temperatures (t_{1db}, t_{sa}) may be plotted on their respective GSHF and RSHF lines (*Fig. 39*) to determine if these conditions can handle the supplementary loads. If they cannot, a new rise in supply air is assumed and the trial-and-error procedure repeated.

In a normal, well designed, tight system this difference in supply air temperature and the condition of the air leaving the apparatus (t_{sa}-t_{1db}) is usually not more than a few degrees. To simplify the discussion on the interrelationship of RSHF and GSHF, the supplementary loads have been neglected in the various discussions, formulas and problems in the remainder of this chapter. It can not be over-emphasized, however, that these supplementary loads must be recognized when estimating the cooling and heating loads. These loads are taken into account on the air conditioning load estimate in *Chapter 1*, and are evaluated in *Chapter 7*.

The RSHF ratio will be constant (at full load) under a specified set of conditions; however, the GSHF ratio may increase or decrease as the outdoor air quantity and mixture conditions are varied for design purposes. As the GSHF ratio changes, the supply air condition to the space varies along the RSHF line (*Fig. 38*).

The difference in temperature between the room and the air supply to the room determines the air quantity required to satisfy the room sensible and room latent loads. As this temperature difference increases (supplying colder air, since the room conditions are fixed), the required air quantity to the space decreases. This temperature difference can increase up to a limit where the RSHF line crosses the saturation line on the psychrometric chart, *Fig. 38*; assuming, of course, that the available conditioning equipment is able to take the air to 100% saturation. Since this is impossible, the condition of the air normally falls on the RSHF line close to the saturation line. How close to the saturation line depends on the physical operating characteristics and the efficiency of the conditioning equipment.

In determining the required air quantity, when neglecting the supplementary loads, the supply air temperature is assumed to equal the condition of the air leaving the apparatus (t_{sa}-t_{1db}). This is illustrated in Fig. 38. The calculation for the required air quantity still remains a trial-and-error procedure, since the mixture temperature of the air (t_m) entering the apparatus is dependent on the required air quantity. The same procedure previously described for determining the air quantity is used. Assume a supply air rise and calculate the supply air quantity and the mixture temperature to the conditioning apparatus. Substitute the supply air quantity and mixture temperature in the equation for determining the air quantity thru the apparatus, and calculate the leaving condition of the air from the apparatus. This temperature must equal the supply air temperature; if it does not, a new supply air rise is assumed and the procedure repeated.

Determining the required air quantity by either method previously described is a tedious process, since it involves a trial-and-error procedure, plotting the RSHF and GSHF ratios on a psychrometric chart, and in actual practice accounting for the supplementary loads in determining the supply air, mixture and leaving air temperatures.

This procedure has been simplified, however, by relating all the conditioning loads to the physical performance of the conditioning equipment, and then including this equipment performance in the actual calculation of the load.

This relationship is generally recognized as a psychrometric correlation of loads to equipment performance. The correlation is accomplished by calculating the "effective surface temperature," "bypass factor" and "effective sensible heat factor." These alone will permit the simplified calculation of supply air quantity.

EFFECTIVE SURFACE TEMPERATURE (t_{es})

The surface temperature of the conditioning equipment varies throughout the surface of the apparatus as the air comes in contact with it. However, the effective surface temperature can be considered to be the uniform surface temperature which would produce the same leaving air conditions as the non-uniform surface temperature that actually occurs when the apparatus is in operation. This is more clearly understood by illustrating the heat transfer effect between the air and the cooling (or heating) medium. *Fig. 40* illustrates this process and

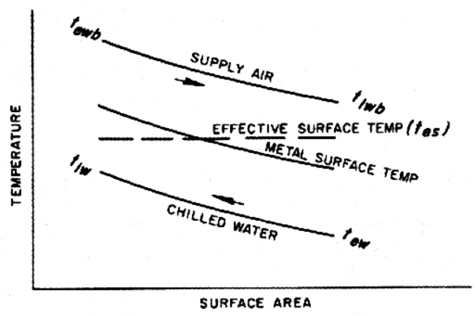


FIG. 40- RELATIONSHIP OF EFFECTIVE SURFACE TEMP TO SUPPLY AIR AND CHILLED WATER

is applicable to a chilled water cooling medium with the supply air counterflow in relation to the chilled water.

The relationship shown in *Fig. 40* may also be illustrated for heating, direct expansion cooling and for air flowing parallel to the cooling or heating medium. The direction, slope and position of the lines change, but the theory is identical.

Since conditioning the air thru the apparatus reduces to the basic principle of heat transfer between the heating or cooling media of the conditioning apparatus and the air thru that apparatus, there must be a common reference point. This point is the effective surface temperature of the apparatus. The two heat transfers are relatively independent of each other, but are quantitatively equal when referred to the effective surface temperature.

Therefore, to obtain the most economical apparatus selection, the effective surface temperature is used in calculating the required air quantity and in selecting the apparatus.

For applications involving cooling and dehumidification, the effective surface temperature is at the point where the GSHF line crosses the saturation line on the psychrometric chart (*Fig. 36*). As such, this effective surface temperature is considered to be the dewpoint of the apparatus, and hence the term apparatus dewpoint (adp) has come into common usage for cooling and dehumidifying processes.

Since cooling and dehumidification is one of the most common applications for central station apparatus, the "Air Conditioning Load Estimate" form, *Fig. 44*, is designed around the term apparatus dewpoint (adp). The term is used exclusively in this chapter when referring to cooling and dehumidifying applications. The psychrometrics of air can be applied equally well to other types of heat transfer applications such as sensible heating, evaporative cooling, sensible cooling, etc., but for these applications the effective surface temperature will not necessarily fall on the saturation line.

BYPASS FACTOR (BF)

Bypass factor is a function of the physical and operating characteristics of the conditioning apparatus and, as such, represents that portion of the air which is considered to pass through the conditioning apparatus completely unaltered.

The physical and operating characteristics affecting the bypass factor are as follows:

1. A decreasing amount of available apparatus heat transfer surface results in an increase in bypass factor, i.e. less rows of coil, less coil surface area, wider spacing of coil tubes.
2. A decrease in the velocity of air through the conditioning apparatus results in a decrease in bypass factor, i.e. more time for the air to contact the heat transfer surface.

Decreasing or increasing the amount of heat transfer surface has a greater effect on bypass factor than varying the velocity of air through the apparatus.

There is a psychrometric relationship of bypass factor to GSHF and RSHF. Under specified room, outdoor design conditions and quantity of outdoor air, RSHF and GSHF are fixed. The position of RSHF is also fixed, but the relative position of GSHF may vary as the supply air quantity and supply air condition change.

To properly maintain room design conditions, the air must be supplied to the space at some point along the RSHF line. Therefore, as the bypass factor varies, the relative position of GSHF to RSHF changes, as shown by the dotted lines in *Fig. 41*. As the position of GSHF changes, the entering and leaving air conditions at the apparatus, the required air quantity, bypass factor and the apparatus dewpoint also change.

The effect of varying the bypass factor on the conditioning equipment is as follows:

1. Smaller bypass factor—
   a. Higher adp—DX equipment selected for higher refrigerant temperature and chilled water equipment would be selected for less or higher temperature chilled water. Possibly smaller refrigeration machine.
   b. Less air-smaller fan and fan motor.
   c. More heat transfer surface—more rows of coil or more coil surface available.
   d. Smaller piping if less chilled water is used.
2. Larger bypass factor--
   a. Lower adp—Lower refrigerant temperature to select DX equipment, and more water or lower temperature for chilled water equipment. Possibly larger refrigeration machine.

b. More air-larger fan and fan motor.
c. Less heat transfer surface—less rows of coil or less coil surface available.
d. Larger piping if more chilled water is used.

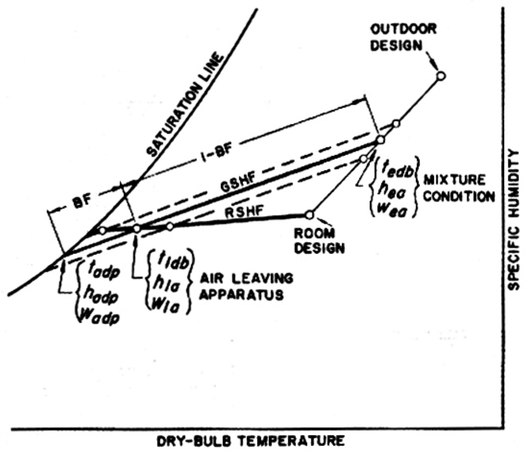


FIG. 41-RSHF AND GSHF LINES PLOTTED ON SKELETON PSYCHROMETRIC CHART

It is, therefore, an economic balance of first cost and operating cost in selecting the proper bypass factor for a particular application. *Table 62, page 127*, lists suggested bypass factors for various applications and is a guide for the engineer to proper bypass factor selection for use in load calculations.

Tables have also been prepared to illustrate the various configurations of heat transfer surfaces and the resulting bypass factor for different air velocities. *Table 61, page 127*, lists bypass factors for various coil surfaces. Spray washer equipment is normally rated in terms of saturation efficiency which is the complement of bypass factor (1-BF). *Table 63, page 136*, is a guide to representative saturation efficiencies for various spray arrangements.

As previously indicated, the entering and leaving air conditions at the conditioning apparatus and the apparatus dewpoint are related psychrometrically to the bypass factor. Although it is recognized that bypass factor is not a true straight line function, it can be accurately evaluated mathematically from the following equations:

$$BF = \frac{t_{ldb} - t_{adp}}{t_{edb} - t_{adp}} = \frac{h_{la} - h_{adp}}{h_{ea} - h_{adp}} = \frac{W_{la} - W_{adp}}{W_{ea} - W_{adp}}$$

and

$$1\text{-}BF = \frac{t_{edb} - t_{ldb}}{t_{edb} - t_{adp}} = \frac{h_{ea} - h_{la}}{h_{ea} - h_{adp}} = \frac{W_{ea} - W_{la}}{W_{ea} - W_{adp}}$$

NOTE: The quantity (1-BF) is frequently called contact factor and is considered to be that portion of the air leaving the apparatus at the adp.

EFFECTIVE SENSIBLE HEAT FACTOR (ESHF)

To relate bypass factor and apparatus dewpoint to the load calculation, the *effective sensible heat factor* term was developed. ESHF is interwoven with BF and adp, and thus greatly simplifies the calculation of air quantity and apparatus selection.

The effective sensible heat factor is the ratio of effective room sensible heat to the effective room sensible and latent heats. Effective room sensible heat is composed of room sensible heat (see RSHF) plus that portion of the outdoor air sensible load which is considered as being bypassed, unaltered, thru the conditioning apparatus. The effective room latent heat is composed of the room latent heat (see RSHF) plus that portion of the outdoor air latent heat load which is considered as being bypassed, unaltered, thru the conditioning apparatus. This ratio is expressed in the following formula:

$$ESHF = \frac{ERSH}{ERSH + ERLH} = \frac{ERSH}{ERTH}$$

The bypassed outdoor air loads that are included in the calculation of ESHF are, in effect, loads imposed on the conditioned space in exactly the same manner as the infiltration load. The infiltration load comes thru the doors and windows; the bypassed outdoor air load is supplied to the space thru the air distribution system.

Plotting RSHF and GSHF on the psychrometric chart defines the adp and BF as explained previously. Drawing a straight line between the adp and room design conditions (1-2), *Fig. 42* represents the ESHF ratio. The interrelationship of RSHF and GSHF to BF, adp and ESHF is graphically illustrated in *Fig. 42*.

The effective sensible heat factor line may also be drawn on the psychrometric chart without initially knowing the adp. The procedure is identical to the one described for RSHF on *page 118*. The calculated ESHF, however, is plotted thru the room design conditions to the saturation line *(1-2), Fig. 43*, thus indicating the adp.

Tables have been prepared to simplify the method of determining adp from ESHF. Adp can be obtained by entering *Table 65* at room design conditions and at the calculated ESHF. It is not necessary to plot ESHF on a psychrometric chart.

AIR QUANTITY USING ESHF, ADP AND BF

A simplified approach for determining the required air quantity is to use the psychrometric correlation of effective sensible heat factor, apparatus dewpoint and bypass factor. Previously in this chapter, the interrelationship of ESHF, BF and adp was shown with GSHF and RSHF. These two factors need not be calculated to determine the required air quantity, since the use of ESHF, BF and adp results in the same air quantity.

The formula for calculating air quantity, using BF and t_{adp}, is:

$$cfm_{da} = \frac{\text{ERSH}}{1.08\,(t_{rm} - t_{adp})\,(1 - \text{BF})}$$

This air quantity simultaneously offsets the room sensible and room latent loads, and also handles the total sensible and latent loads for which the conditioning apparatus is designed, including the outdoor air loads and the supplementary loads.

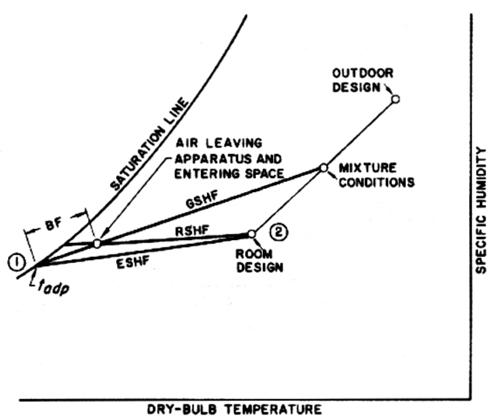
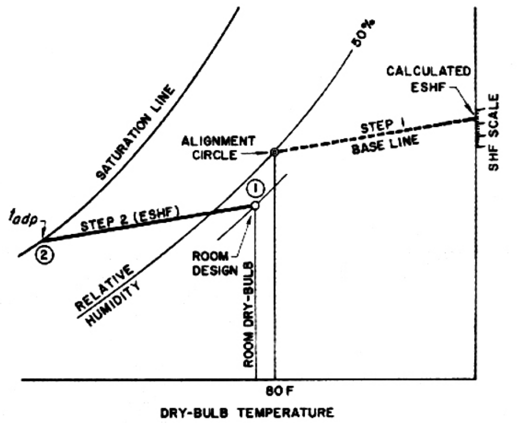


FIG. 42- RSHF, GSHF AND ESHF LINES PLOTTED ON SKELETON PSYCHROMETRIC CHART



PSYCHROMETRIC CHART

AIR CONDITIONING LOAD ESTIMATE FORM

The "Air Conditioning Load Estimate" form is designed for cooling and dehumidifying applications, and may be used for psychrometric calculations. Normally, only ESHF, BF and adp are required to determine air quantity and to select the apparatus. But for those instances when it is desirable to know RSHF and GSHF, this form is designed so that these factors may also be calculated. *Fig. 44*, in conjunction with the following items, explains how each factor is calculated. (The circled numbers correspond to numbers in *Fig. 44*).

1. $\text{RSHF} = \dfrac{\text{RSH}}{\text{RSH} + \text{RLH}} = \dfrac{1}{1 + 2}$

2. $\text{GSHF} = \dfrac{\text{TSH}}{\text{GTH}} = \dfrac{3 + 4}{5}$

3. $\text{ESHF} = \dfrac{\text{ERSH}}{\text{ERSH} + \text{ERLH}} = \dfrac{\text{ERSH}}{\text{ERTH}}$

$$8 = \frac{3}{3 + 6} = \frac{3}{7}$$

4. Adp located where ESHF crosses the saturation line, or from *Table 65*, ESHF 8 and room conditions 9 give adp 10
5. BF 11 used in the outdoor air calculations is obtained from the equipment performance table or charts. Typical bypass factors for different surfaces and for various applications are given on *page 127*. These are to guide the engineer and may be used in the outdoor air calculation when the actual equipment performance tables are not readily available.

SHEET ____
PREPARED BY ____ OFFICE ____
NAME OF JOB ____
LOCATION ____

Carrier

DATE ____
PROP NO. ____ JOB NO. ____
APPROVED ____

SPACE USED FOR ____
SIZE ____ × ____ = ____ SQ FT × ____ = ____ CU FT

ITEM	AREA OR QUANTITY	SUN GAIN OR TEMP. DIFF.	FACTOR	BTU/HOUR
SOLAR GAIN—GLASS				
GLASS	SQ FT ×		×	
GLASS	SQ FT ×		×	
GLASS	SQ FT ×		×	
GLASS	SQ FT ×		×	
SKYLIGHT	SQ FT ×		×	
SOLAR & TRANS. GAIN—WALLS & ROOF				
WALL	SQ FT ×		×	
WALL	SQ FT ×		×	
WALL	SQ FT ×		×	
WALL	SQ FT ×		×	
ROOF—SUN	SQ FT ×		×	
ROOF—SHADED	SQ FT ×		×	
TRANS. GAIN—EXCEPT WALLS & ROOF				
ALL GLASS	SQ FT ×		×	
PARTITION	SQ FT ×		×	
CEILING	SQ FT ×		×	
FLOOR	SQ FT ×		×	
INFILTRATION	CFM ×		× 1.08	
INTERNAL HEAT				
PEOPLE		PEOPLE ×		
POWER		HP OR KW ×		
LIGHTS		WATTS × 3.4 ×		
APPLIANCES, ETC.		×		
ADDITIONAL HEAT GAINS		×		
			SUB TOTAL	
STORAGE	SQ FT ×		× (−)	
			SUB TOTAL	
SAFETY FACTOR		%		
			ROOM SENSIBLE HEAT ■	1
SUPPLY DUCT HEAT GAIN % + SUPPLY DUCT LEAK. LOSS % + FAN H. P. %				
OUTDOOR AIR	CFM ×	F ×	11 BF × 1.08	
			EFFECTIVE ROOM SENSIBLE HEAT ■	3
LATENT HEAT				
INFILTRATION	CFM ×		GR/LB × 0.68	
PEOPLE		PEOPLE ×		
STEAM		LB/HR × 1050		
APPLIANCES, ETC.				
ADDITIONAL HEAT GAINS				
VAPOR TRANS.	SQ FT × 1/100 ×	GR/LB ×		
			SUB TOTAL	
SAFETY FACTOR		%		
			ROOM LATENT HEAT	2
SUPPLY DUCT LEAKAGE LOSS			%	
OUTDOOR AIR	CFM ×	GR/LB ×	11 BF × 0.68	
			EFFECTIVE ROOM LATENT HEAT	6 7
			EFFECTIVE ROOM TOTAL HEAT ■	4
OUTDOOR AIR HEAT				
SENSIBLE:	CFM ×	F × (1 − 11 BF) × 1.08		
LATENT:	CFM ×	GR/LB × (1 − 11 BF) × 0.68		
RETURN DUCT HEAT GAIN % + RETURN DUCT LEAK. GAIN % + HP PUMP % + DEHUM. & PIPE LOSS %			SUB TOTAL	
			GRAND TOTAL HEAT ■	5

16 9

ESTIMATE FOR	LOCAL TIME SUN TIME	PEAK LOAD	LOCAL TIME SUN TIME
HOURS OF OPERATION			

CONDITIONS	DB	WB	% RH	DP	GR/LB
OUTDOOR (OA)					
ROOM (RM)					
DIFFERENCE		X X X	X X X	X X X	

OUTDOOR AIR

VENTILATION: ____ PEOPLE × ____ CFM/PERSON = ____
____ SQ FT × ____ CFM/SQ FT = ____
CFM VENTILATION ■

INFILTRATION: SWINGING/REVOLVING DOORS ____ PEOPLE × ____ CFM/PERSON = ____
OPEN DOORS ____ DOORS × ____ CFM/DOOR = ____
EXHAUST FAN ____
CRACK ____ FEET × ____ CFM/FT = ____
CFM INFILTRATION ■

15 CFM OUTDOOR AIR THRU APPARATUS ■ ____ CFM_{OA}

APPARATUS DEWPOINT

8 ESHF: EFFECTIVE SENS HEAT FACTOR = $\dfrac{(3)\ \text{EFFECTIVE ROOM SENS. HEAT}}{(7)\ \text{EFFECTIVE ROOM TOTAL HEAT}}$ = ____

10 ADP: INDICATED ADP = ____ F SELECTED ADP = ____ F

DEHUMIDIFIED AIR QUANTITY

12 TEMP. RISE: $(1 - \underline{(11)}\ BF) \times (T_{RM}\ \underline{(9)}\ F - T_{ADP}\ \underline{(10)}\ F) =$ ____ F

13 DEHUM. CFM: $\dfrac{(3)\ \text{EFFECTIVE ROOM SENS. HEAT}}{1.08 \times (12)\ F\ \text{TEMP. RISE}} =$ ____ CFM_{DA}

OUTLET TEMP. DIFF.: $\dfrac{(1)\ \text{ROOM SENS. HEAT}}{1.08 \times (13)\ CFM_{DA}} =$ ____ F (RM—OUTLET AIR)*

SUPPLY AIR QUANTITY

14 SUPPLY CFM: $\dfrac{(1)\ \text{ROOM SENS. HEAT}}{1.08 \times\ F\ \text{DESIRED DIFF}} =$ ____ CFM_{SA}

BYPASS CFM: (14) CFM_{SA} − (13) CFM_{DA} = ____ CFM_{BA}

RESULTING ENT & LVG CONDITIONS AT APPARATUS

17 EDB: $T_{RM}\ \underline{(9)}\ F + \dfrac{(15)\ CFM_{OA}}{(13)\ \text{OR}\ (14)\ CFM^{\dagger}} \times (T_{OA}\ \underline{(16)}\ F - T_{RM}\ \underline{(9)}\ F) = T_{EDB}$ ____ F

18 LDB: $T_{ADP}\ \underline{(10)}\ F + \underline{(11)}\ BF \times (T_{EDB}\ \underline{(17)}\ F - T_{ADP}\ \underline{(10)}\ F) = T_{LDB}$ ____ F

FROM PSYCH. CHART: T_{EWB} ____ F, T_{LWB} ____ F

NOTES

*IF THIS ΔT IS TOO HIGH, DETERMINE SUPPLY CFM FOR DESIRED DIFFERENCE BY SUPPLY AIR QUANTITY FORMULA.

†WHEN BYPASSING A MIXTURE OF OUTDOOR AND RETURN AIR, USE SUPPLY CFM. WHEN BYPASSING RETURN AIR ONLY, USE DEHUMIDIFIED CFM.

Form E 20

NOTE: The circled numbers are explained on the previous page under "Air Conditioning Load Estimate" form.

FIG. 44 AIR CONDITIONING LOAD ESTIMATE

6. $$cfm_{da} = \frac{\text{ERSH}}{1.08\ (t_{rm} - t_{adp})\ (1 - \text{BF})}$$

$$13 = \frac{3}{1.08\ (9 - 10)\ (1 - 11)}$$

Once the dehumidified air quantity is calculated, the conditioning apparatus may be selected. The usual procedure is to use the grand total heat 5, dehumidified air quantity 13 , and the apparatus dewpoint 10, to select apparatus.

Since guides are available, the bypass factor of the apparatus selected is usually in close agreement with the originally assumed bypass factor. If, because of some peculiarity in loading in a particular application, there is a wide divergence in bypass factor, that portion of the load estimate form involving bypass factor should be adjusted accordingly.

7. Outlet temperature difference – *Fig. 44* shows a calculation for determining the temperature difference between room design dry-bulb and the supply air dry-bulb to the room. Frequently a maximum temperature difference is established for the application involved. If the outlet temperature difference calculation is larger than desired, the total air quantity in the system is increased by bypassing air around the conditioning apparatus. This temperature difference calculation is:

$$\text{Outlet temp diff} = \frac{\text{RSH}}{1.08 \times cfm_{da}}$$

$$= \frac{1}{1.08 \times 13}$$

8. Total air quantity when outlet temperature difference is greater than desired- The calculation for the total supply air quantity for a desired temperature difference (between room and outlet) is:

$$cfm_{sa} = \frac{\text{RSH}}{1.08 \times \Delta t} = \frac{1}{1.08 \times \Delta t}$$

The amount of air that must be bypassed around the conditioning apparatus to maintain this desired temperature difference (Δt) is the difference between cfm_{sa} and cfm_{da}.

9. Entering and leaving conditions at the apparatus—Often it is desired to specify the selected conditioning apparatus in terms of entering and leaving air conditions at the apparatus. Once the apparatus has been selected from ESHF, adp, BF and GTH, the entering and leaving air conditions are easily determined. The calculations for the entering and leaving dry-bulb temperatures at the apparatus are illustrated in *Fig. 44*.

The entering dry-bulb calculation contains the term "cfm†"*. This air quantity "cfm†" depends on whether a mixture of outdoor and return air or return air only is bypassed around the conditioning apparatus.

The total supply air quantity cfm_{sa} 14 is used for "cfm†" when bypassing a mixture of outdoor and return air. *Fig. 45* is a schematicsketch of a system bypassing a mixture of outdoor and return air.

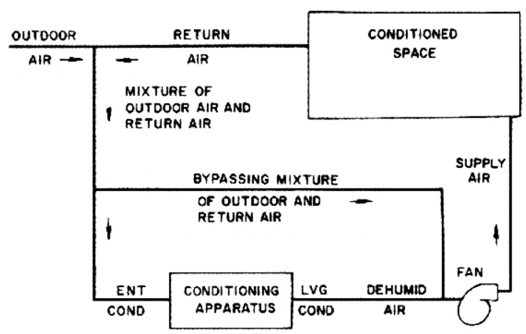


FIG. 45- BYPASSING MIXTURE OF OUTDOOR AND RETURN AIR

When bypassing a mixture of return air only or when there is no need for a bypass around the apparatus, use the cfm_{da} 13 for the value of "cfm†" *Fig. 46* is a schematic sketch of a system bypassing room return air only.

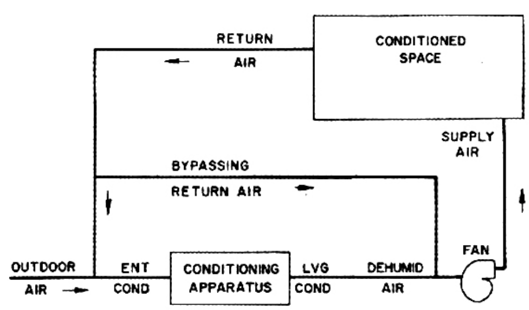


FIG. 46 – BYPASSING RETURN AIR ONLY OR NO FIXED BYPASS

*"cfm†" is a symbol appearing in the equation next to 17 in *Fig. 44*

The entering and leaving wet-bulb temperatures at the apparatus are determined on the standard psychrometric chart, once the entering and leaving dry-bulb temperatures are calculated. The procedure for determining the wet-bulb temperatures at the apparatus is illustrated in *Fig. 47* and described in the following items:

a. Draw a straight line connecting room design conditions and outdoor design conditions.
b. The point at which entering dry-bulb crosses the line plotted in *Step a* defines the entering conditions to the apparatus. The entering wet-bulb is read on the psychrometric chart.
c. Draw a straight line from the adp ⑩ to the entering mixture conditions at the apparatus (*Step b*.) (This line defines the GSHF line of the apparatus.)
d. The point at which the leaving dry-bulb crosses the line drawn in *Step c* defines the leaving conditions of the apparatus. Read the leaving wet-bulb from the apparatus at this point. (This point defines the intersection of the RSHF and GSHF as described previously.)

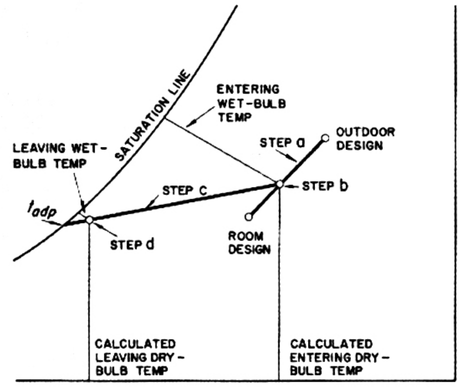


FIG. 47- ENTERING AND LEAVING CONDITIONS AT APPARATUS

AIR CONDITIONING APPARATUS

The following section describes the characteristic psychrometric performance of air conditioning equipment.

Coils, sprays and sorbent dehumidifiers are the three basic types of heat transfer equipment required for air conditioning applications. These components may be used singly or in combination to control the psychrometric properties of the air passing thru them.

The selection of this equipment is normally determined by the requirements of the specific application. The components must be selected and integrated to result in a practical system; that is, one having the most economical owning and operating cost.

An economical system requires the optimum combination of air conditioning components. It also requires an air distribution system that provides good air distribution within the conditioned space, using a practical rise between supply air and room air temperatures.

Since the only known items are the load in the space and the conditions to be maintained within the space, the selection of the various components is based on thes items. Normally, performance requirements aestablished and then equipment is selected to meet the requirement

COIL CHARACTERISTICS

In the operation of coils, air is drawn or forcedover a series of tubes thru which chilled water, brine, volatile refrigerant, hot water or steam is flowing. As the air passes over the surface of the coil, it is cooled, cooled and dehumidified, or heated, depending upon the temperature of the media flowing thru the tubes. the media in turn is heated or cooled in the process.

The amount of coil surface not only affects the heat transfer but also the bypass factor of the coil. The bypass factor, as previously explained, is the measure of air side performance. Consequently, it is a function of the type and amount of coil surface and the time available for contact as the air passes thru the coil. *Table 61* gives approximate bypass factors for various finned coil surfaces and air velocities.

TABLE 61- TYPICAL BYPASS FACTORS
(For Finned Coils)

DEPTH OF COILS	WITHOUT SPRAYS		WITH SPRAYS	
	8 fins/in.	14 fins/in.	8 fins/in.	14 fins/in.
	Velocity (fpm)			
(rows)	300-700	300-700	300-600	300-600
2	.42-.55	.22-.38		
3	.27-.40	.10-.23		
4	.15-.28	.05-.14	.12-.22	.04-.10
5	.10-.22	.03-.14	.08-.16	.02-.06
6	.06-.15	.01-.05	.05-.11	.01-.03
8	.02-.08	.00-.02	.02-.06	.00-.02

These bypass factors apply to coils with 5/8 in. O.D. tubes and spaced on approximately 11/4 in. centers. The values are approximate. Bypass factors for coils with plate fins, or for combinations other than those shown, should be obtained from the coil manufacturer.

Table 61 contains bypass factors for a wide range of coils. This range is offered to provide sufficient latitude in selecting coils for the most economical system. *Table 62* lists some of the more common applications with representative coil bypass factors. This table is intended only as a guide for the design engineer.

TABLE 62- TYPICAL BYPASS FACTORS
(For Various Applications)

COIL BYPASS FACTOR	TYPE OF APPLICATION	EXAMPLE
0.30- to 0.50	A small total load or a load that is somewhat larger with a low sensible heat factor (high latent load).	Residence
0.20 to 0.30	Typical comfort application with a relatively small total load or a low sensible heat factor with a somewhat larger load.	Residence, Small Retail Shop, Factory
0.10 to 0.20	Typical comfort application.	Dept. Store, Bank, Factory
0.05 to 0.10	Application with high internal sensible loads or requiring a large amount of outdoor air for ventilation.	Dept. Store, Restaurant, Factory
0 to 0.10	All outdoor air applications.	Hospital Operating Room, Factory

COIL PROCESSES

Coils are capable of heating or cooling air at a constant moisture content, or simultaneously cooling and dehumidifying the air. They are used to control dry-bulb temperature and maximum relative humidity at peak load conditions. Since coils alone cannot *raise* the moisture content of the air, a water spray on the coil surface must be added if humidification is required. If this spray water is recirculated, it will not materially affect the

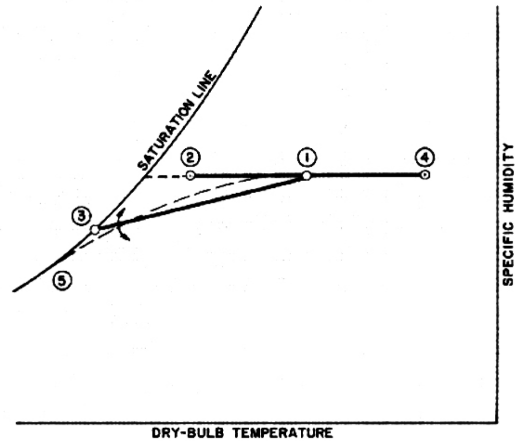


FIG. 48- COIL PROCESSES

psychrometric process when the air is being cooled and dehumidified.

Fig. 48 illustrates the various processes that can be accomplished by using coils.

Sensible Cooling

The first process, illustrated by line (1-2), represents a sensible cooling application in which the heat is removed from the air at a constant moisture content.

Cooling and Dehumidification

Line (*1-3*) represents a cooling and dehumidification process in which there is a simultaneous removal of heat and moisture from the air.

For practical considerations, line (*1-3*) has been plotted as a straight line. It is, in effect, a line that starts at point (*1*) and curves toward the saturation line below point (*3*). This is indicated by line (*1-5*).

Sensible Heating

Sensible heating is illustrated by line (*1-4*); heat is added to the air at constant moisture content.

COIL PROCESS EXAMPLES

To better understand these processes and their variations, a description of each with illustrated examples is presented in the following: (Refer to *page 149* for definition of symbols and abbreviations.)

Cooling and Dehumidification

Cooling and dehumidification is the simultaneous removal of the heat and moisture from the air, line (*1-3*), *Fig. 48*. Cooling and dehumidification occurs when the ESHF and GSHF are less than 1.0. The ESHF for these applications can vary from 0.95, where the load is predominantly sensible, to 0.45 where the load is predominantly latent.

The air conditioning load estimate form illustrated in *Fig. 44* presents the procedure that is used to determine the ESHF, dehumidified air quantity, and entering and leaving air conditions at the apparatus. *Example 1* illustrates the psychrometrics involved in establishing these values.

Example 1- Cooling and Dehumidification

Given:

- Application –5¢ & 10¢ Store
- Location –Bloomfield, N. J.
- Summer design –95 F db, 75 F wb
- Inside design –75 F db, 50% rh
- RSH –200,000 Btu/hr
- RLH –50,000 But/hr
- Ventilation –2,000 cfm_{oa}

Find:

1. Outdoor air load (OATH)
2. Grand total heat (GTH)
3. Effective sensible heat factor (ESHF)
4. Apparatus dewpoint temperature (t_{adp})
5. Dehumidified air quantity (cfm_{da})
6. Entering and leaving conditions at the apparatus

$(t_{edb}, t_{ewb}, t_{ldb}, t_{lwb})$

Solution:

1. OASH = 1.08×2000×(95-75) = 43,200 Btu/hr (14)
   OALH = .68×2000× (99-65) = 46,200 Btu/hr (15)
   OATH = 43,200+46,200 = 89,400 Btu/hr (17)

2. TSH = 200,000+43,200 = 243,200 Btu/hr (7)
   TLH = 50,000+46,200 = 96,200 Btu/hr (8)
   GTH = 243,200+96,200 = 339,400 Btu/hr (9)

3. Assume a bypass factor of 0.15 from *Table 62*.

$$ESHF = \frac{200{,}000 + (.15)(43{,}200)}{200{,}000 + (.15)(43{,}200) + 50{,}000 + (.15)(46{,}200)}$$

$$= .785$$

4. Determine the apparatus dewpoint from the room design conditions and the ESHF, by either plotting on the psychrometric chart or using *Table 65*. *Fig. 49* illustrates the ESHF plotted on the psychrometric chart.

$$t_{adp} = 50 \text{ F}$$

5. $$cfm_{da} = \frac{200{,}000 + (.15)(43{,}200)}{1.08\,(75 - 50)\,(1 - .15)} \qquad (36)$$

NOTE: Numbers in parentheses at right edge of column refer to equations beginning on *page 150*.

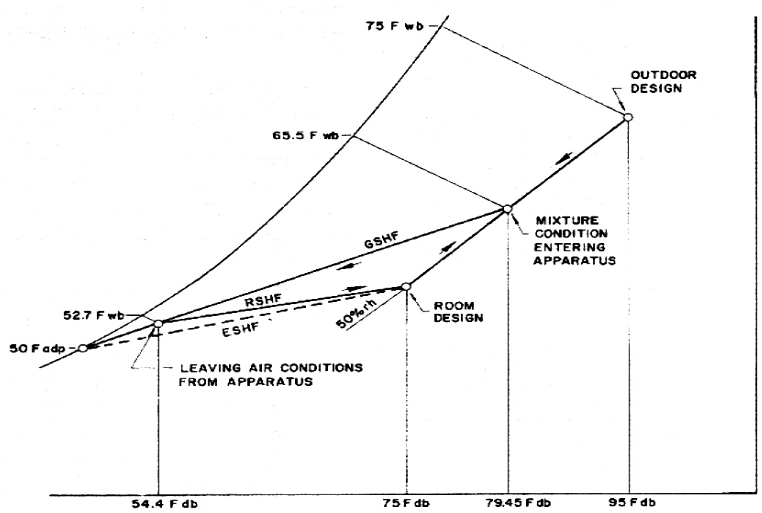


FIG. 49- COOLING AND DEHUMIDIFICATION

6. Assume for this example that the apparatus selected for 9,000 cfm, 50 F adp, and GTH = 339,400, has a bypass factor that is equal, or nearly equal, to the assumed BF = 0.15. Also, assume that it is not necessary to physically bypass air around the apparatus.

$$t_{edb} = \frac{(2000 \times 95) + (7000 \times 75)}{9000} = 79.45 \text{ db} \qquad (31)$$

Read t_{ewb} where the t_{edb} crosses the straight line plotted between the outdoor and room design conditions on the psychrometric chart, *Fig. 49.*
t_{ewb} = 65.5 F wb
t_{ldb} = 50+.15(79.45-50) = 54.4 F db (32)
Determine that t_{lwb} by drawing a straight line between the adp and the entering conditions at the apparatus. (This is the GSHF line.) Where t_{ldb} intersects this line, read t_{lwb}.
t_{lwb} = 52.7 F wb

Cooling and Dehumidification – High Latent Load Application

On some applications a special situation exists if the ESHF and GSHF lines do not intersect the saturation line when plotted on the psychrometric chart of if they do the adp is absurdly low. This may occur where the latent load is high with respect to the total loads (dance halls, etc.). In such applications, an appropriate apparatus dewpoint is selected and the air is reheated to the RSHF line. Occasionally, altering the room design conditions eliminates the need for reheat, or reduces the quantity of reheat required. Similarly, the utilization of a large air side surface (low bypass factor) coil may eliminate the need for reheat or reduce the required reheat.

Once the ventilation air requirement is determined, and if the supply air quantity is not fixed, the best approach to determining the apparatus dewpoint is to assume a maximum allowable temperature difference between the supply air and the room. Then, calculate the supply air conditions to the space. The supply air conditions to the space must fall on the RSHF line to properly offset the sensible and latent loads in the space.

There are four criteria which should be examined, to aid in establishing the supply air requirements to the space. These are:

1. Air movement in the space.
2. Maximum temperature difference between the supply air and the room.
3. The selected adp should provide an economical refrigeration machine selection.
4. In some cases, the ventilation air quantity required may result in an all outdoor air application.

Example 2 is a laboratory application with a high latent load. In this example the ESHF intersects the saturation line, but the resulting adp is too low.

Example 2- Cooling and Dehumidification – High Latent Load

Given:

Application – Laboratory
Location – Bangor, Maine
Summer design – 90 F db, 73 F wb
Inside design – 75 F db, 50% rh
RSH – 120,000 Btu/hr
RLH – 65,000 Btu/hr
Ventilation – 2,500 cfm_{oa}
Temp. diff. between room and supply air, 20 F maximum

Find:

1. Outdoor air load (OATH)
2. Effective sensible heat factor (ESHF)
3. Apparatus dewpoint (t_{adp})
4. Reheat required
5. Supply air quantity (cfm_{sa})
6. Entering conditions to coil ($t_{edb,}$ $t_{ewb,}$ W_{ea})
7. Leaving conditions from coil ($t_{ldb,}$ t_{lwb})
8. Supply air condition to the space ($t_{sa,}$ W_{sa})
9. Grand total heat (GTH)

Solution:

1. OASH = 1.08×2500×(90-75) = 40,500 Btu/hr (14)
   OALH = .68×2500× (95-65) = 51,000 Btu/hr (15)
   OATH = 40,500+51,000 = 91,500 Btu/hr (17)
2. Assume a bypass factor of 0.05 because of high latent load.

$$\text{ESHF} = \frac{120{,}000 + .05\,(40{,}500)}{120{,}000 + .05\,(40{,}500) + 65{,}000 + (.05)\ (51{,}000)}$$

$$= .645 \qquad (26)$$

When plotted on the psychrometric chart, this ESHF (.645) intersects the saturation vurve at 35 F. With such a low adp an appropriate apparatus dewpoint should be selected and the air reheated to the RSHF line.

3. Refer to Table 65. For inside design conditions of 75 F db, 50% rh, an ESHF of .74 results in an adp of 48 F which is a reasonable minimum figure.
4. Determine amount of reheat (Btu/hr) required to produce an ESHF of .74.
   ESHF (.74) =

$$\frac{120{,}000 + .05\,(40{,}500) + \text{reheat}}{120{,}000 + .05\,(40{,}500) + \text{reheat} + 65{,}000 + (.05)\ 51{,}000}$$

$$.74 = \frac{122{,}025 + \text{reheat}}{189{,}575 + \text{reheat}} \qquad (25)$$

reheat = 70,230 Btu/hr

5. Determine dehumidifier air quantity (cfm_{da})

$$cfm_{da} = \frac{\text{ERSH}}{1.08 \times (1 - \text{BF})\ (t_{rm} - t_{adp})} \qquad (36)$$

$$= \frac{122{,}025 + 70{,}230}{1.08\ (1 - .05)\ (75 - 48)} = 6940 \text{ cfm}$$

cfm_{da} is also cfm_{sa} when no air is to be physically bypassed around the cooling coil.

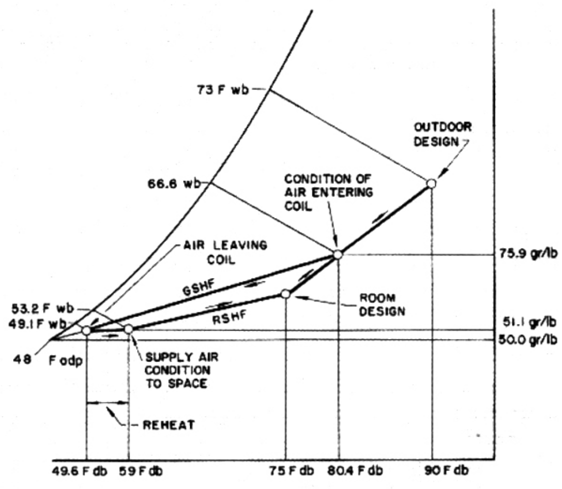


FIG. 50- COOLING AND DEHUMIDIFICATION WITH HIGH LATENT LOAD

6. $$t_{edb} = \frac{(2500 \times 90) + (4440 \times 75)}{6940}$$

$= 80.4$ (31)

Read t_{ewb} where the t_{edb} crosses the straight line plotted between the outdoor air and room design conditions on the psychrometric chart, *Fig. 50*.

$t_{ewb} = 66.6$ F

The moisture content at the entering conditions to the coil is real from the psychrometric chart.

$W_{ea} = 75.9$ gr/lb

Determine leaving conditions of air from cooling coil.

$t_{ldb} = t_{adp} + \text{BF}\,(t_{edb} - t_{adp})$ (32)

$= 48 + .05\,(80.4 - 48)$

$= 49.6$

$h_{sa} = h_{adp} + \text{BF}\,(h_{ea} - h_{adp})$ (34)

$= 19.21 + .05\,(31.3 - 19.21)$

$= 19.82$

$t_{lwb} = 49.1$ F

8. Determine supply air temperature to space

$$t_{sa} = t_{rm} - \frac{\text{RSH}}{1.08\,(cfm_{sa})} \quad (35)$$

$$= 75 - \frac{(120{,}000)}{1.08\,(6940)}$$

$= 59$ F

$W_{sa} = 51.1$ gr/lb

Temp. diff between room and supply air

$= t_{rm} - t_{sa} = 75 - 59 = 16$ F

Which is less than 20 F

9. GTH = 4.45×6940(31.3-19.82) = 354,500 Btu/hr (24)

Cooling and Dehumidification –Using All Outdoor Air

In some applications it may be necessary to supply all outdoor air; for example, a hospital operating room, or an area that requires large quantities of ventilation air. For such applications, the ventilation or code requirements may be equal to, or more than, the air quantity required to handle the room loads.

Items 1 thru 5 explain the procedure for determining the dehumidified air requirements using the "Air Conditioning Load Estimate" form when all outdoor air is required.

1. Calculate the various loads and determine the apparatus dewpoint and dehumidified air quantity.
2. If the dehumidified air quantity is *equal* to the outdoor air requirements, the solution is self-evident.
3. If the dehumidified air quantity is *less* than the outdoor air requirements, a coil with a larger bypass factor should be investigated when the difference in air quantities is small. If a large difference exists, however, reheat is required. This situation sometimes occurs when the application requires large exhaust air quantities.
4. If the dehumidified air quantity is *greater* than the outdoor air requirements, substitute cfm_{da} for cfm_{oa} in the outdoor air load calculations.
5. Use the recalculated outdoor air loads to determine a new apparatus dewpoints and dehumidified air quantity. This new dehumidified air quantity should check reasonably close to the cfm_{da} in *Item 1.*

A special situation may arise when the condition explained in *Item 4* occurs. This happens when the ESHF, as plotted on the psychrometric chart, does not intersect the saturation line. This situation is handled in a manner similar to that previously described under *"Cooling and Dehumidification –High Latent Load Application."*

Example 3 illustrates an application where codes specify that all outdoor air be supplied to the space.

Example 3 – Cooling and Dehumidification-
All Outdoor Air

Given:

Application – Laboratory
Location – Wheeling, West Virginia
Summer design – 95 F db, 75 F wb
Inside design – 75 F db, 55% rh
RSH – 50,000 Btu/hr
RLH – 11,000 But/hr
Ventilation – 1600 cfm_{oa}
All outdoor air to be supplied to space.

Find:

1. Outdoor air load (OATH)
2. Effective sensible heat factor (ESHF)
3. Apparatus dewpoint (t_{adp})
4. Dehumidified air quantity (cfm_{da})
5. Recalculated outdoor air load (OATH)
6. Recalculated effective sensible heat factor (ESHF)

NOTE : Number in parentheses at right edge of column refer to equations beginning on *page 150*

7. Final apparatus dewpoint temperature (t_{adp})
8. Recalculated dehumidified air quantity (cfm_{da})

Solution:

1. OASH = 1.08×1600×(95-75) = 34,600 Btu/hr (14)
   OALH = .68×1600× (98.5-71) = 30,000 Btu/hr (15)
   OATH = 34,500+30,000 = 64,600 Btu/hr (17)
2. Assume a bypass factor of 0.05 from *Tables 61 and 62*.

$$ESHF = \frac{50,000 + (.05)(34,600)}{50,000 + (.05)(34,600) + 11,000 + (.05)(30,000)}$$

$$= .81 \quad (26)$$

3. Table 65 shows that, at the given room design conditions and effective sensible heat factor, t_{adp} = 54.5 F.
4. $$cfm_{da} = \frac{50,000 + (.05)(34,600)}{1.08(1 - .05)(75 - 54.5)} = 2450 \text{ cfm} \quad (36)$$

   Since 2450 cfm is larger than the ventilation requirements, and by code all OA is required, the O.A loads, the adp, and the dehumidified air quantity must be recalculated using 2450 cfm as the OA requirements.
5. Recalculating outdoor air load
   OASH = 1.08×2450× (95-75) = 53,000 Btu/hr (14)
   OALH = .68×2450× (98.5-71) = 46,000 Btu/hr (15)
   OATH = 53,000+46,000 = 99,000 Btu/hr (17)
6. $$ESHF = \frac{50,000 + (.05)(53,000)}{(50,000) + (.05)(53,000) + 11,000 + (.05)(46,000)}$$

   $$= .80 \quad (26)$$
7. t_{adp} = 54 F
8. $$cfm_{da} = \frac{50,000 + (.05)(53,000)}{1.08(1 - .05)(75 - 54)} = 2500 \text{ cfm} \quad (36)$$

   This checks reasonably close to the value in Step 4, and recalculation is not necessary.

Cooling With Humidification

Cooling with humidification may be required at partial load operation to make up a deficiency in the room latent load. It may also be used at design conditions for industrial applications having relatively high sensible loads and high room relative humidity requirements. Without humidification, excessively high supply air quantities may be required. This not only creates air distribution problems but also is often economically unsound. Excessive supply air quantity requirements can be avoided by introducing moisture into the space to convert sensible heat to latent heat. This is sometimes referred to as a "split system." The moisture is introduced into the space by using steam or electric humidifiers or auxiliary sprays.

NOTE : Numbers in parentheses at right edge of column refer to equations beginning on *page 150*.

When humidification is performed in the space, the room sensible load is decreased by an amount equal to the latent heat added, since the process is merely an interchange of heat. The humidifier motor adds sensible heat to the room but the amount is negligible and is usually ignored.

Where humidification is required at design to reduce the air quantity, then a credit to the room sensible heat should be taken in the amount of the latent heat from the added moisture. No credit to the room sensible load is taken when humidification is used to make up a deficiency in the room latent load during partial load operation.

When the humidifiers and sprays are used to reduce the required air quantity, the latent load introduced into the space is added to the room latent load.

When the humidifier or sprays are operated only to make up the room deficiency, the latent load introduced into the room by the humidifier or auxiliary sprays in the space is not added to the room latent load.

The introduction of this moisture into the space to reduce the required air quantity decreases the RSHF, ESHF and the apparatus dewpoint. This method of reducing the required air quantity is normally advantageous when designing for high room relative humidities.

The method of determining the amount of moisture necessary to reduce the required air quantity results in a trial-and-error procedure. The method is outlined in the following steps:

1. Assume an amount of moisture to be added and determine the latent heat available from this moisture. *Table 64* gives the maximum moisture that may be added to a space without causing condensation on supply air ducts and equipment.
2. Deduct this assumed latent heat from the original effective room sensible heat and use the difference in the following equation for ERSH to determine t_{adp}.

   $$t_{adp} = t_{rm} - \frac{ERSH}{1.08 \times (1 - BF)\, cfm_{da}}$$

   Cfm_{da} is the reduced air quantity permissible in the air distribution system.
3. The ESHF is obtained from a psychrometric chart or *Table 65*, using the apparatus dewpoint (from *Step 2*) and room design conditions.
4. The new effective room latent load is determined from the following equation:

   $$ERLH = ERSH \times \frac{1 - ESHF}{ESHF}$$

   The ERSH is from *Step 2* and ESHF is from *Step 3.*
5. Deduct the original ERLH (before adding sprays or

humidifier in the space) from the new effective room latent heat in *Step 4.* The result is equal to the latent heat from the added moisture, and must check with the value assumed in *Step 1.* If it does not check, assume another value and repeat the procedure.

Example 4 illustrates the procedure for investigating an application where humidification is accomplished within the space to reduce the air quantity.

Example 4- Cooling With Humidification in the Space

Given:

Application – A high humidity chamber

Location – St. Louis, Missouri

Summer design –95 F db, 70% F wb

Inside design – 70 F db, 70% rh

RSH – 160,000 Btu/hr

RLH – 10,000 Btu/hr

RSHF - .94

Ventilation – 4000 cfm_{oa}

Find:

A. When space humidification is not used:
1. Outdoor air load (OATH)
2. Grand total heat (GTH)
3. Effective sensible heat factor (ESHF)
4. Apparatus dewpoint (t_{adp})
5. Dehumidified air quantity (cfm_{da})
6. Dehumidified air quantity (cfm_{da}) (t_{edb}, t_{ewb}, t_{ldb}, t_{lwb})

B. When humidification is used in the space:
1. Determine maximum air quantity and assume an amount of moisture added to the space and latent heat from this moisture.
2. New effective room sensible heat (ERSH)
3. New apparatus dewpoint (t_{adp})
4. New effective sensible heat factor (ESHF)
5. New effective room latent heat (ERLH)
6. Check calculated latent heat from the moisture added with amount assumed in Item 1.
7. Theoretical conditions of the air entering the evaporative humidifier before humidification.
8. Entering and leaving conditions at the apparatus (t_{edb}, t_{ewb}, t_{ldb}, t_{lwb})

Solution:

A. When space humidification isnot used:

1. OASH = 1.08×4000× (95-70) = 108,000 Btu/hr (14)
   OALH = .68×4000× (117-77) = 109,000 Btu/hr (15)
   OATH = 108,000+109,000 = 217,000 Btu/hr (17)
2. GTH = 160,000+10,000+108,000+109,000 (9)
   = 387,000 Btu/hr
3. Assume a bypass factor of 0.05 from *Tables 61 and 62*.

$$ESHF = \frac{160{,}000 + (.05)(108{,}000)}{160{,}000 + 10{,}000 + (.05)(108{,}000) + (.05)(109{,}000)}$$

$= .92$ (26)

4. Plot the ESHF on a psychrometric chart and read the adp (dotted line in *Fig. 51*).
   t_{adp} = 59.5 F

5. $cfm_{da} = \dfrac{160{,}000 + (.05)(108{,}000)}{1.08\,(1 - .05)\,(70 - 59.5)} = 15{,}4000$ cfm (36)

6. $t_{edb} = \dfrac{(400 \times 95) + (11{,}400 \times 70)}{15{,}400} = 76.7$ F db (31)

Read t_{ewb} where the t_{edb} crosses the straight line plotted between the outdoor and room design conditions on the psychrometric chart (*Fig. 51*).
t_{ewb} = 67.9 F wb
T_{ldb} = 59.5+.05 (76.7-59.5) = 60.4 F db (32)
Determine the t_{lwb} by drawing a straight line between the adp and the entering conditions to the apparatus (the GSHF line). Where t_{ldb} intersects this line, read the t_{lwb} (*Fig. 51*).
t_{lwb} = 60 F wb

B. When humidification is used in the space:

1. Assume, for the purpose of illustration in this problem, that the maximum air quantity permitted in the air distribution system is 10,000 cfm. Assume 5 grains of moisture per pound of dry air is to be added to convert sensible to latent heat. The latent heat is calculated by multiplying the air quantity times the moisture added times the factor .68.
2. NEW ERSH = Original ERSH – latent heat of added moisture
   = [160,000+(.05×108,000]-34,000
   = 131,400 Btu/hr
3. $t_{adp} = 70 - \dfrac{131{,}400}{1.08\,(1 - .05)\,(10{,}000)} = 57.2$ F (36)
4. ESHF is read from the psychrometric chart as .73 (dotted line in *Fig. 52*).
5. NEW ERLH $= \text{New ERSH} \times \dfrac{1 - ESHF}{ESHF}$
   $= 131{,}400 \times \dfrac{1 - .73}{.73}$
   = 48,600 Btu/hr
6. Check for latent heat of added moisture.
   Latent heat of added moisture
   = New ERLH – Original ERLH
   = 48,600 [[10,000+(.05×109,000]
   = 33,200 Btu/hr
   This checks reasonably close with the assumed value in *Step 1* (34,000 Btu/hr).

NOTE : Numbers in parentheses at right edge of column refer to equations beginning on *page 150*.

7. Psychrometrically, it can be assumed that the atomized water from the spray heads in the space absorbs part of the room sensible heat and turns into water vapor at the final room wet-bulb temperature. The theoretical dry-bulb of the air entering the spray is at the intersection of the room design wet-bulb line and the moisture of the air entering the sprays. This moisture content is determined by subtracting the moisture added by the room sprays from the room design moisture content.
   Moisture content of air entering humidifier
   = 77-5 = 72 gr/lb.
   The theoretical dry-bulb is determined from the psychrometric chart as 73.3 db, illustrated on Fig. 52.

8. $$t_{edb} = \frac{(4000 \times 95) + (6000 \times 70)}{10{,}000} = 80 \text{ F db} \qquad (31)$$

   Read t_{ewb} where the t_{edb} crosses the straight line plotted between the outdoor and room design conditions on the psychrometric chart (*Fig. 52*).

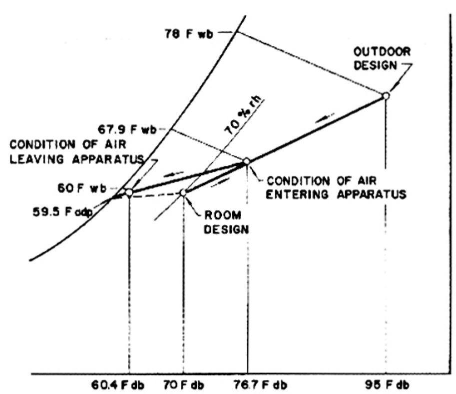


FIG. 51- COOLING AND DEHUMIDIFICATION ADDING NO MOISTURE TO THE SPACE

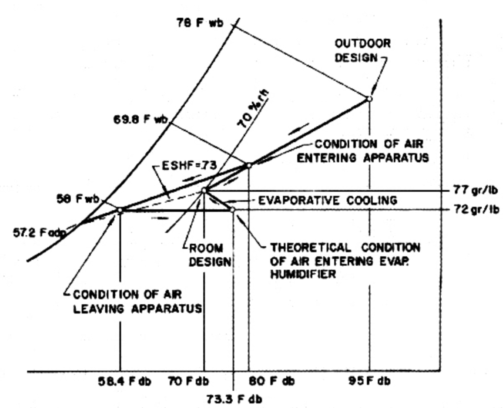


FIG. 52- COOLING AND DEHUMIDIFICATION ADDING MOISTURE INTO THE SPACE

t_{ewb} = 69.8 F wb

$$t_{ldb} = 57.2 + (.05)(80 - 57.2) = 58.4 \text{ F db} \qquad (32)$$

Determine t_{lwb} by drawing a straight line between the adp and the entering conditions to the apparatus (GSHF line). Where t_{ldb} intersects this line, read the
t_{lwb} *(Fig. 52).*
t_{lwb} = 58 F wb

The straight line connecting the leaving conditions at the apparatus with the theoretical condition of the air entering the evaporative humidifier represents the theoretical process line of the air. This theoretical condition of the air entering the humidifier represents what the room conditions are if the humidifier is not operating. The slope of this theoretical process line is the same as RSHF (.94).

The heavy lines on *Fig. 52* illustrate the theoretical air cycle as air passes through the conditioning apparatus to the evaporative humidifier, then to the room, and finally back to the apparatus where the return air is mixed with the ventilation air. Actually, if a straight line were drawn from the leaving conditions of the apparatus (58.4 F db, 58 F wb) to the room design conditions, this line would be the RSHF line and would be the process line for the supply air as it picks up the sensible and latent loads in the space (including the latent heat added by the sprays).

The following two methods of laying out the system are recommended when the humidifier is to be used for both partial load control and reducing the air quantity.

1. Use two humidifiers; one to operate continuously, adding the moisture to reduce the air quantity, and the other to operate intermittently to control the humidity. The humidifier used for partial load is sized for the effective room latent load, not including that produced by the other humidifier. If the winter requirements for moisture addition are larger than summer requirements, then the humidifier is selected for these conditions. This method of using two humidifiers gives the best control.
2. Use one humidifier of sufficient capacity to handle the effective room latent heat plus the calculated amount of latent heat from the added moisture required to reduce the air quantity. In *Part B*, Step 5, the humidifier would be sized for a latent load of 48,600 Btu/hr.

Sensible Cooling

A sensible cooling process is one that removes heat from the air at a constant moisture content, line (1-2, Fig. 48. Sensible cooling occurs when either of the following conditions exist:

1. The GSHF as calculated or plotted on the psychrometric chart is 1.0.
2. The ESHF calculated on the air conditioning load estimate form is equal to 1.0.

In a sensible cooling application, the GSHF equals 1.0. The ESHF and the RSHF may equal 1.0. When only the RSHF equals. 1.0, however, it does not necessarily indicate a sensible cooling process because latent load, introduced by outdoor air can give a GSHF less than 1.0.

The apparatus dewpoint is referred to as the effective surface temperature (t_{es}) in sensible cooling applications. The effective surface temperature must be equal to, or higher than, the dewpoint temperature of the entering air. In most instances, the t_{es} does not lie on the saturated line and, therefore, will not be the dewpoint of the apparatus. However, the calculations for ESHF, t_{adp} and cfm_{da} may still be performed on the term t_{es} for t_{adp}. The use of the term cfm_{da} in a sensible cooling application should not be construed to indicate that dehumidification is occurring. It is used in the "Air Conditioning Load Estimate" form and in *Example 5* to determine the air quantity required thru the apparatus to offset the conditioning loads.

The leaving air conditions from the coil are dictated by the room design conditions, the load and the required air quantity. The effective surface temperature may be found by using equation 36.

Example 5 illustrates the method of determining the apparatus dewpoint or the effective surface temperature for a sensible cooling application.

Example 5- Sensible Cooling

Given:

Location – Bakersfield, California
Summer design – 105 F db, 70 F wb
Inside design – 75 F db, 50% maximum rh
RSH – 200,000 Btu/hr
RLH – 50,000 Btu/hr
Ventilation – 13,000 cfm_{oa}

Find:

1. Outdoor air load (OATH)
2. Grand total heat (GTH)
3. Grand sensible heat factor (GSHF)
4. Effective sensible heat factor (ESHF)
5. Apparatus dewpoint (t_{adp}) or the effective surface temp. (t_{es})
6. Dehumidified air quantity (cfm_{da})
7. Entering and leaving conditions at the apparatus (t_{edb}, t_{ewb}, t_{ldb}, t_{lwb})

Solution:

1. OASH = 1.08×(105-75)×(13,000 = 420,000 Btu/hr (14)
   OALH = .68×(54-64)×13,000 = -88,500 Btu/hr (15)
   The latent load is negative and a greater absolute value than the room latent load. Therefore, the inside design conditions must be adjusted unless there is a means to humidify the air.
   Room latent heat = 50,000 Btu/hr
   $$\text{Room moisture content} = 54 + \frac{50{,}000}{.68 \times 13{,}000} = 59.65 \text{ grains}$$
   Adjusted inside design –75 F db, 59.65 grains
   OALH = .68× (54-59.65) ×13,000 = -50,000 Btu/hr (15)
   OATH = 420,000+(-50,000)= 370,000 Btu/hr (17)
2. TSH = 200,000+420,000 = 620,000 Btu/hr (7)
   TLH = 50,000+(-50,000) = 0 (8)
   GTH = 620,000+0 = 620,000 Btu/hr (9)
3. $$\text{GSHF} = \frac{620{,}000}{620{,}000} = 1 \qquad (27)$$
   This is a sensible cooling application since GSHF=1
4. Assume a bypass factor of 0.05 from *tables 61 and 62*.
   $$\text{ESHF} = \frac{200{,}000 + (.05)\,420{,}000}{200{,}000 + (.05)\,420{,}000 + 50{,}000 + (.05)\,(-50{,}000)} = .823 \qquad (26)$$
5. Plot the ESHF to the saturation line on the psychrometric chart. The apparatus dewpoint is read as t_{adp} = 48.8 F, *fig. 53*. (36)
6. $$cfm_{da} = \frac{200{,}000 + (.05)\,420{,}000}{1.08 \times (75 - 48.8)\,(1 - .05)} = \frac{221{,}000}{26.9} = 8{,}230 \text{ CFM} \qquad (36)$$
   Since the dehumidified air quantity is less than the outdoor ventilation requirements, substitute the cfm_{oa} for cfm_{da}. This results in a new effective surface temperature which does not lie on the saturated line.
   $$t_{es} = 75 - \frac{200{,}000 + (.05)\,420{,}000}{1.08 \times (1 - .05) \times 13{,}000} = 58.4 \text{ F} \qquad (36)$$
   This temperature, t_{es}, falls on the GSHF line.
7. This is an all outdoor air application since the cfm_{da} is less than the ventilation requirements therefore:
   $t_{edb} = t_{oa}$ = 105F
   t_{ewb} = 70F
   Calculate the t_{sa} which equals the t_{ldb} by subsituting t_{es} for t_{adb} in equation (28).
   $t_{ldb} = t_{sa}$ = 105-(1-.05) (105-58.4) = 60.7 F (28)
   Determine the t_{lwb} by drawing a straight line between the t_{es} and the entering conditions at the apparatus. (This is the GSHF line.) Where t_{ldb} intersects this line, read t_{lwb}. t_{lwb}=54.6 F

In Example 5, the assumed .05 bypass factor is used to determine t_{es} and dehumidified air quantity. Since the dehumidified air quantity is less than the

NOTE: Numbers in parentheses at right edge of column refer to equations beginning on *page 150*.

ventilation air requirement, the .05 bypass factor is used again to determine a new t_{es}, substituting the ventilation air requirement for the dehumidified air quantity. The new t_{es} is 58.4 F.

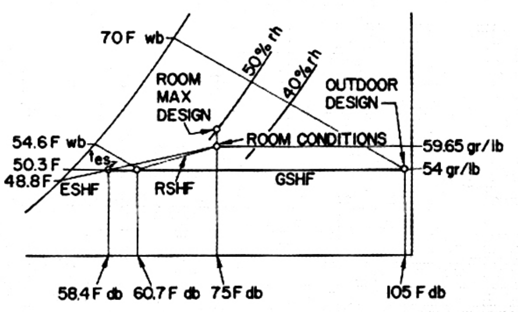


FIG. 53- SENSIBLE COOLING

If a coil with a higher bypass factor is substituted in *Example 5*, a lower t_{es} results. Under these conditions, it becomes a question of economic balance when determining which coil selection and which refrigerant temperature is the best for the application. For instance, the maximum possible coil bypass factor that can be used is .19. This still results in a t_{es} above 50.3 F and at the same time maintains a dehumidified air cfm of 13,000 which equals the ventilation requirements.

SPRAY CHARACTERSTICS

In the operation of spray type equipment, air is drawn or forced thru a chamber where water is sprayed thru nozzles into the air stream. The spray nozzles may be arranged within the chamber to spray the water counter to air flow, parallel to air flow, or in a pattern that is a combination of these two. Generally, the counter-flow sprays are the most efficient; parallel flow sprays are the most efficient; parallel flow sprays are the least efficient; and when both are employed, the efficiency falls somewhere in between these extremes.

SATURATION EFFICIENCY

In a spray chamber, air is brought into contact with a dense spray of water. The air approaches the state of complete saturation. The degree of saturation is termed saturation efficiency (sometimes called contact or performance factor). Saturation efficiency is, therefore, a easure of the spray chamber efficiency. It can be considered to represent that portion of the air passing thru the spray chamber which contacts the spray water surface. This contacted air is considered to be leaving the spray chamber at the effective surface temperature of the spray water. This effective surface temperature is the temperature at complete saturation of the air.

Though not a straight line function, the effect of saturation efficiency on the leaving air conditions from a spray chamber may be determined with a sufficient degree of accuracy from the following equation:

$$\text{Sat Eft} = \frac{t_{edb} - t_{ldb}}{t_{edb} - t_{es}} = \frac{W_{ea} - W_{la}}{W_{ea} - W_{es}} = \frac{h_{ea} - h_{la}}{h_{ea} - h_{es}}$$

The saturation efficiency is the complement of bypass factor, and with spray equipment the bypass factor is used in the calculation of the cooling load. Bypass factor, therefore, represents that portion of the air passing thru the spray equipment which is considered to be leaving the spray chamber completely unaltered from its entering condition.

This efficiency of the sprays in the spray chamber is dependent on the spray surface available and on the time available for the air to contact the spray water surface. The available surface is determined by the water particle size in the spray mist (pressure at the spray nozzle and the nozzle size), the quantity of water sprayed, number of banks of nozzles, and the number of nozzles in each bank. The time available for contact depends on the velocity of the air thru the chamber, the length of the effective spray chamber, and the direction of the sprays relative to the air flow. As the available surface decreases or as the time available surface decreases or as the time available for contact decreases, the saturation efficiency of the spray chamber decreases. *Table 63* illustrates the relative efficiency of different spray chamber arrangements.

The relationship of the spray water temperatures to the air temperatures is essential in understanding the psychrometrics of the various spray processes. It can be assumed that the leaving water temperature from a spray chamber, after it has contacted the air, is equal to the leaving air wet-bulb temperature. The leaving water temperature will not usually vary more than a degree from the leaving air wet-bulb temperature. Then the entering water quantity and the heat required to be added or removed from the air.

Table 63 illustrates the relative efficiency of different spray chamber arrangements.

TABLE 63- TYPICAL SATURATION EFFICIENCY*
For Spray Chambers

NO. OF BANKS	DIREC-TION OF WATER SPRAY	¼" NOZZLE (25 psig Nozzle Pressure 3 gpm/sq ft†)		1/8" NOZZLE (30 psig Nozzle Pressure 2.5 gpm/sq ft†)	
		Velocity‡ (fpm)			
		300	700	300	700
1	Parallel	70%	50%	80%	60%
	Counter	75%	65%	82%	70%
2	Parallel	90%	85%	92%	87%
	Opposing	98%	92%	98%	93%
	Counter	99%	93%	99%	94%

*Saturation efficiency = 1-BF

†Gpm/sq ft of chamber face area

‡Velocities above 700 fpm and below 300 fpm normally do not permit eliminators to adequately remove moisture from the air. Reference to manufacturer' data is suggested for limiting velocity and performance.

SPRAY PROCESSES

Sprays are capable of cooling and dehumidifying, sensible cooling, cooling and humidifying, and heating and humidifying. Sensible cooling may be accomplished only when the entering air dewpoint is the same as the effective surface temperature of the spray water.

The various spray processes are represented on the psychrometric chart in *Fig. 54*. All process lines must go toward the saturation line, in order to be at or near saturation.

Adiabatic Saturation or Evaporative Cooling

Line *(1-2)* represents the evaporative cooling process. This process occurs when air passes thru a spray chamber where heat has not been added to or removed from the spray water. (This does not include heat gain from the water pump and thru the apparatus casing.) When plotted on the psychrometric chart, this line approximately follows up the line of the wet-bulb temperature of the air entering the spray chamber. The spray water temperature remains essentially constant at this wet-bulb temperature.

Cooling and Humidification –With Chilled Spray Water

If the spray water receives limited cooling before it is sprayed into the air stream, the slope of the process line will move down from the evaporative cooling line. This process is represented by line *(1-3)*. Limited cooling causes the leaving air to be lower in dry-and wet-bulb temperatures, but higher in moisture content, than the air entering the spray chamber.

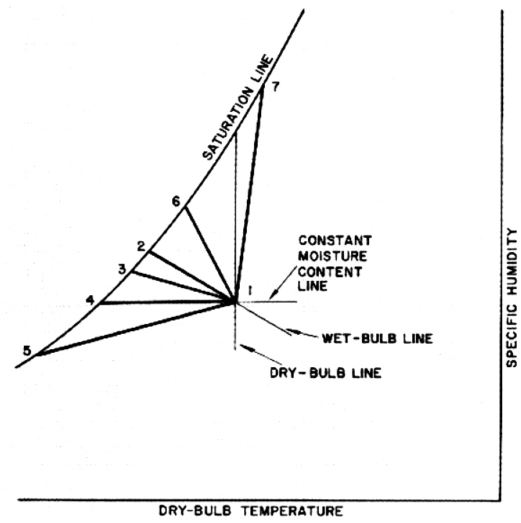


FIG. 54- SPRAY PROCESSES

Sensible Cooling

If the spray water is cooled further, sensible cooling occurs. This process is represented by line *(1-4)*. Sensible cooling occurs only when the entering air dewpoint is equal to the effective surface temperature of the spray water; this condition is rare. In a sensible cooling process, the air leaving the spray chamber is lower in dry-and wet-bulb temperatures but equal in moisture content to the entering air.

Cooling and Dehumidification

If the spray water is cooled still further, cooling and dehumidification takes place. This is illustrated by line *(1-5)*. The leaving air is lower in dry-and wet-bulb temperatures and in moisture content than the air entering the spray chamber.

Cooling and Humidification – With Heated Spray Water

When the spray water is heated to a limited degree before it is sprayed into the air stream, the slope of the process line rises to a point above the evaporative cooling line. This is illustrated by line *(1-6)*. Note that the leaving air is lower in dry-bulb temperature, but higher in wet-bulb temperature and moisture content, than the air entering the spray chamber.

Heating and Humidification

If the spray water is sufficiently heated, a heating and humidification process results. This is represented by line *(1-7)*. In this process the dry-bulb temperature, wet-bulb temperature, and moisture content of the leaving air is greater than that of the entering air.

SPRAY PROCESS EXAMPLES

The following descriptions and examples provide a better understanding of the various psychrometric processes involved in spray washer equipment.

Cooling and Dehumidification

When a spray chamber is to used for cooling and dehumidification, the procedure for estimating the load and selecting the equipment is identical to the procedure described on *page 128* for coils. The "Air Conditioning Load Estimate" form is used to evaluate the load; bypass factor is determined by subtracting the selected saturation efficiency from one. Spray chamber dehumidifiers may not be rated in terms of apparatus dewpoint but in terms of entering and leaving wet-bulb temperatures at the apparatus. The apparatus dewpoint must still be determined, however, to evaluate properly the entering and leaving wet-bulb temperatures and the dehumidified air quantity.

Although originally prepared to exemplify the operation of a coil, *Example 1, page 128*, is also typical of the cooling and dehumidifying process using sprays.

Cooling and Dehumidification –Using All Outdoor Air

When a spray chamber is to be used for cooling and dehumidifying with all outdoor air, the procedure for determining adp, entering and leaving conditions at the chamber, ESHF and cfm_{da} is identical to the procedure for determining these items for coils using all outdoor air. Therefore, the description on *page 130* and *Example 3* may be used to analyze this type of application.

Evaporative Cooling

An evaporative cooling application is the simultaneous removal of sensible heat and the addition of moisture to the air, line *(1-2), Fig. 54*. The spray water temperature remains essentially constant at the wet-bulb temperature of the air. This is a process in which heat is not added to or removed from the spray water. (Heat gain from the water pump and heat gain thru the apparatus casing are not included.)

Evaporative cooling is commonly used for those applications where the relative humidity is to be controlled but where no control is required for the room dry-bulb temperature, except to hold it above a predetermined minimum. When the dry-bulb temperature is to be maintained during the winter or intermediate season, heat must be available to the system. This is usually accomplished by adding a reheat coil. When relative humidity is to be maintained in addition to room dry-bulb during the winter or intermediate season, a combination of preheat and reheat coils, or a reheat coil and spray water heating, is required. The latter method changes the process from evaporative cooling to one of the humidification processes illustrated by lines *(1-6)* or *(1-7)* in *Fig. 54.*

Evaporative cooling may be used in industrial applications where the humidity alone is critical, and also in dry climates where evaporative cooling gives some measure of relief by removing sensible heat.

Example 6 illustrates an industrial application designed to maintain the space relative humidity only

Example 6-Evaporative Cooling

Given:

An industrial application

Location – Columbia, South Carolina

Summer design – 95 F db, 75 F wb

Inside design – 55% rh

RSH – 2,100,000 Btu/hr

RSHF – 1.0

Use all outdoor air at design load conditions

Find:

1. Room dry-bulb temperature at design (t_{rm})
2. Supply air quantity (cfm_{sa})

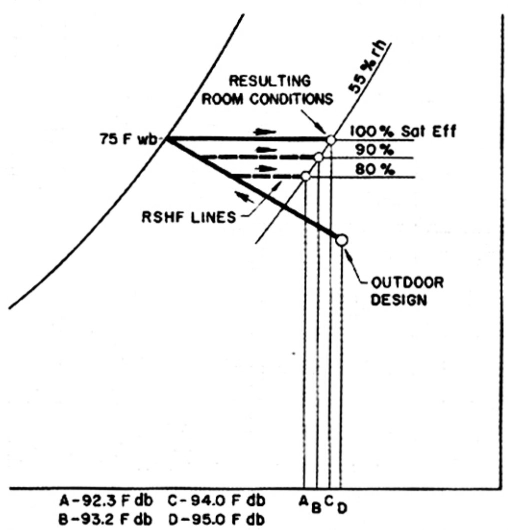


FIG. 55- EVAPORATIVE COOLING, WITH VARYING SATURATION EFFICIENCY

Solution:

1. Determine the room dry-bulb temperature by compromising between the spray saturation efficiency, the acceptable room dry-bulb temperature, and the supply air quantity. To evaluate these items, use the following equation to determine the leaving conditions from the spray for various saturation efficiencies:

   $t_{ldb} = t_{edb} - \text{(Sat Eff)}\ (t_{edb} - t_{ewb})$*

   The room dry-bulb temperature in the following table results from various spray saturation efficiencies and is determined by plotting the RSHF thru the various leaving conditions, to the design relative humidity, *Fig. 55*. Note that the supply air temperature rise decreases more rapidly than the room dry-bulb temperature. Correspondingly, as the supply air temperature rise decreases, the supply air temperature rise decreases, the supply air quantity increases in the same proportion.

SAT EFF (%)	DRY-BULE TEMP LEAVING SPRAYS (t_{ldb})	SUPPLY AIR TEMP RISE (Δt)	ROOM DRY-BULB TEMP AT 55% RH (t_{rm})
100	75	19	94
95	76	17.6	93.6
90	77	16.2	93.2
85	78	14.7	92.7
80	79	13.3	92.3

2. Calculate the supply air quantity for the various temperature rises from the following equation:

$$cfm_{sa} = \frac{RSH}{1.08\ (t_{rm} - t_{ldb})}$$

SUPPLY AIR TEMP RISE (t_{rm}-t_{ldb})	SUPPLY AIR QUANTITY (cfm_{sa})
19	102,400
17.6	110,600
16.2	120,000
14.7	132,300
13.3	146,200

The spray chamber and supply air quantity should then be selected to result in the best owning and operating costs. The selection is based primarily on economic considerations.

Evaporative Cooling Used With A Split System

There are occasions when using straight evaporative cooling results in excessive air quantity requirements and an unsatisfactory air distribution system. This situation usually arises in applications that are to be maintained at higher relative humidities (70% or more). To use straight evaporative cooling with the large air quantity, or to use a split system with the auxiliary sprays in the space, becomes a problem of economics which should be analyzed for each particular application.

When a split system is used, supplemental spray heads are usually added to the straight evaporative cooling system. These spray heads atomize water and add supplementary moisture directly to the room. This added moisture is evaporated at the final room wet-bulb temperature, and the room sensible heat is reduced by the amount of heat required to evaporate the sprayed water.

Table 64 gives the recommended maximum moisture to be added, based on a 65 F db room temperature or over, without causing condensation on the ductwork.

*This equation is applicable only to evaporative cooling applications where the entering air wet-bulb temperature, the leaving air wet-bulb temperature, and the entering and leaving water temperature to the sprays are all equal.

TABLE 64- MAXIMUM RECOMMENDED MOISTURE ADDED TO SUPPLY AIR

Without Causing Condensation on Ducts†

ROOM DESIGN RH	MOISTURE Gr/Cu Ft Dry Air	ROOM DESIGN RH	MOISTURE Gr/Cu Ft Dry Air
85	1.25	65	1.50
80	1.30	60	1.60
75	1.35	55	1.70
70	1.40	50	1.80

†These are arbitrary limits which have been established by a combination of theory and field experience. These limits apply where the room dry-bulb temperature is 65 F db or over.

As a rule of thumb, the air is reduced in temperature approximately 8.3 F for every grain of moisture per cubic foot added. This value is often used as a check on the final room temperature as read from the psychrometric chart.

Example 7 illustrates an evaporative cooling application with supplemental spray heads used in the space.

Example 7 –Evaporative Cooling-With Auxiliary Sprays

Given:

An industrial application

Location – Columbia, South Carolina

Summer design – 95 F db, 75 F wb

Inside design – 70% rh

RSH – 2,100,000 Btu/hr

RSHF – 1.0

Moisture added by auxiliary spray heads – 19 gr/lb (13.9 cu ft/lb×1.4 gr/cu ft)

Use all outdoor air thru a spray chamber with 90% saturation efficiency.

Find:

1. Leaving conditions from spray chamber (t_{ldb}, t_{lwb})
2. Room dry-bulb temperature (t_{rm})
3. Supply air quantity (cfm_{sa}) with auxiliary sprays
4. Supply air quantity (cfm_{sa}) without auxiliary sprays

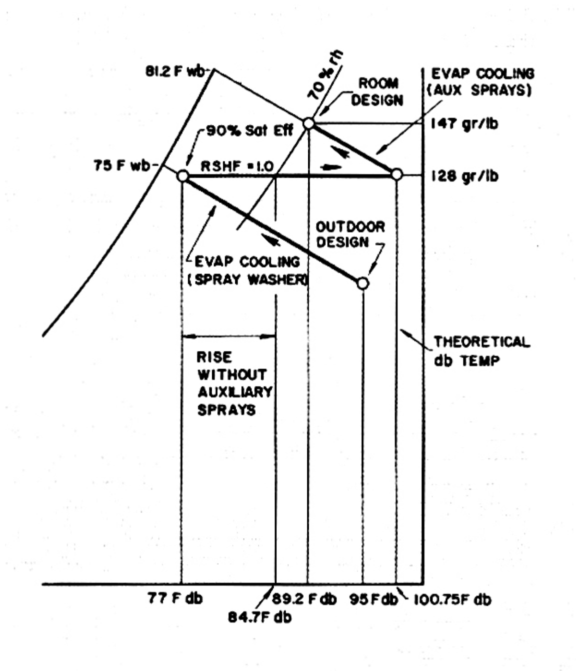


FIG. 56- EVAPORATIVE COOLING, WITH AUXILIARY SPRAYS WITHIN THE SPACE

Solution:

1. $t_{ldb} = t_{edb}$-(Sat Eff) (t_{edb}-t_{ewb})
   = 95-.90 (95-75) = 77 F db
   t_{lwb} = is the same as the t_{ewb} in an evaporative cooling process, *Fig. 56*.
2. Room dry-bulb temperature is evaluated by determining the moisture content of the space.
   $W_{rm} = W_{sa}$+19=128+19=147 gr/lb
   The 19 gr/lb is the moisture added to the space by the auxiliary spray heads.
   The t_{rm} is the point on the psychrometric chart where the W_{rm} intersects the 70% design relative humidity line, *Fig. 56*.
   t_{rm} = 89.2 F db
3. Psychrometrically, it can be assumed that the atomized water from the spray heads absorbs part of the room sensible heat and turns into water vapor at the final room wet-bulb temperature. The intersection of this wet-bulb temperature with the moisture content of the air leaving the evaporative cooler is the theoretical dry-bulb equivalent temperature if the auxiliary sprays were not operating. The difference between this theoretical dry-bulb equivalent temperature and the temperature of the spray chamber, t_{ldb}, is used to determine the supply air quantity.
   t_{ldb} (from spray chamber) = 77 F.
   The theoretical dry-bulb temp is 100.75 F, *Fig. 56*.
   Temp rise = 23.75 F db

   $$cfm_{sa} = \frac{\text{RSH}}{1.08 \text{ x temp rise}} = \frac{2{,}100{,}000}{1.08 \text{ x } 23.75} = 82{,}000 \text{ cfm}$$

4. If no auxiliary sprays were to be used, the room design dry-bulb would be where the RSHF line intersects the room design relative humidity. From *Fig. 56*, the room dry-bulb is read
   t_{rm} = 84.7 F db
   The supply air quantity required to maintain the room design relative humidity is determined from the following equation:

   $$cfm_{sa} = \frac{\text{RSH}}{1.08\ (t_{rm} - t_{ldb})} = \frac{2{,}100{,}000}{1.08\ (84.7 - 77)}$$

   $$= 253{,}000 \text{ cfm}$$

   This air quantity is over three time the air quantity required when auxiliary sprays are used in the space. However, it should be noted that, by reducing the air quantity, the room dry-bulb temperature increased from 84.7 F to 89.2 F.

Heating and Humidification –With Sprays

A heating and humidifying application is one in which heat and moisture are simultaneously added to the air, line *(1-7)*, *Fig. 54*. This may be required during the intermediate and winter seasons or during partial loads where both the dry-bulb temperature and relative humidity are to be maintained.

Heating and humidification may be accomplished by either of the following methods:

1. Add heat to the spray water before it is sprayed into the air stream.
2. Preheat the air with a steam or hot water coil and then evaporatively cool it in the spray chamber.

Spray water is heated, by a steam to water interchanger or by direct injection of steam into the water system. Since the supply air quantity and the spray water quantity have been determined from the summer design conditions, the only other requirement is to determine the amount of heat to be added to the spray water or to the preheater.

For applications requiring humidification, the room latent load is usually not calculated and the room sensible heat factor is assumed to be 1.0.

Example 8 illustrates the psychrometric calculations for a heating and humidifying application when the spray water is heated. It should be noted that this type of application occurs only when the quantity of outdoor air required is large in relation to the total air quantity.

Example 8- Heating and Humidification-
With Heated Spray Water

Given:

An industrial application
Location – Richmond, Virginia
Winter design – 15 F db
Inside design – 72 F db, 35% rh
Ventilation – 50,000 cfm_{oa} (see explanation above)
Supply air – 85,000 cfm_{sa}
Design room heat loss – 2,500,000 Btu/hr
Spray saturation efficiency – 95%
RSHF (winter conditions) – 95%
Make-up water – 65 F

Find:

1. Supply air conditions to the space (t_{sa})
2. Entering and leaving spray water temperature (t_{ew}, t_{lw})
3. Heat added to spray water to select water heater.

Solution:

1. $$t_{sa} = \frac{\text{design room heat loss}}{1.08 \times cfm_{sa}} + t_{rm}$$

$$= \frac{2{,}500{,}000}{1.08 \times 85{,}000} + 72 = 99.2 \text{ Fdb}$$

To determine the wet-bulb temperature, plot the RSHF line on the psychrometric chart and read the wet-bulb at the point where t_{sa} crosses this line *(Fig. 57)*. Supply air wet-bulb to the space = 65.8 F wb.

2. To determine the entering and leaving spray water temperature, calculate the entering and leaving air conditions at the spray chamber:

$$t_{edb} = \frac{(15 \times 50{,}000) + (72 \times 35{,}000)}{85{,}000} = 38.5 \text{ F db} \qquad (31)$$

To determine wet-bulb temperature of the air entering the spray chamber, plot the mixture line of outdoor and return room air on the psychrometric chart, and read the wet-bulb temperature where t_{edb} crosses the mixture line, *Fig. 54.*

t_{ewb}= 32.4 F wb

The air leaving the spray chamber must have the same moisture content as the air in the room.

W_{rm}= W_{la} = 41 gr/lb

Since the spray chamber has a saturation efficiency of 95%, the moisture content of completely saturated air is calculated as follows:

$$W_{sat} = \frac{W_{la} - W_{ea}}{\text{Sat Eff}} + W_{ea}$$

$$= \frac{41 - 17}{.95} + 17 = 42.3 \text{ gr/lb}$$

The heating and humidification process line is plotted on the psychrometric chart between the moisture content of saturated air (42.3 gr/lb) and the entering conditions to the spray chamber (38.5 F db and 32.4 F wb), *Fig. 57.*

The leaving conditions are read from the psychrometric chart where the room moisture content line (41 gr/lb) intersects the heating and humidification process line, *Fig. 57.*

t_{lwb} = 43.6 F db
t_{lwb} = 43.4 F wb

The temperature of the leaving spray water is approximately equal to the wet-bulb temperature of the air leaving the spray chamber.

t_{lw} = 43.4 F

NOTE: Numbers in parentheses at right edge of column refer to equations beginning on *page 150*.

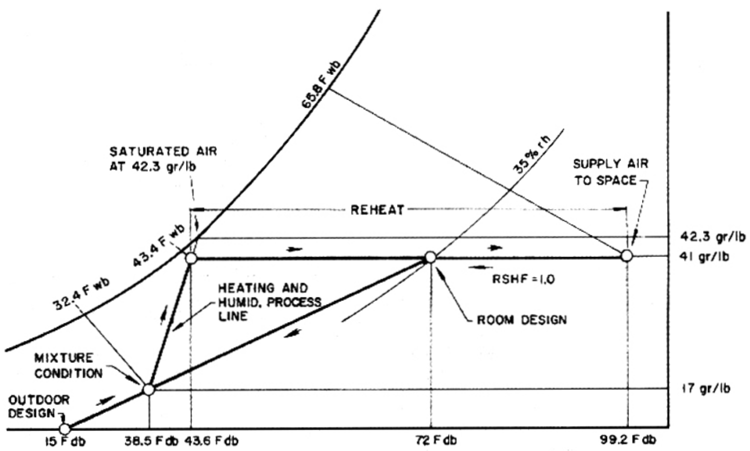


FIG. 57- HEATING AND HUMIDIFICATION, WITH HEATING SPRAY WATER

The temperature of the entering spray water is dependent on the water quantity and the heat to be added or removed from the air. In this type of application, the water quantity is usually dictated by the cooling load design requirements. Assume, for illustration purposes, that this spray washer is selected for 110 gpm for cooling.

The heat added to the air as it passes through the washer

$= cfm_{sa} \times 4.45 \times (h_{la}-h_{ea})$

$= 85{,}000 \times 4.45 \times (16.85-12)$

$= 1{,}830{,}000$ Btu/hr

The entering water temperature is determined from the following equation:

$$t_{ew} = t_{lw} + \frac{\text{heat added to air}}{500 \times \text{gpm}}$$

$$= 43.4 + \frac{1{,}830{,}000}{500 \times 110}$$

$= 76.8$ F

3. The heat added to the spray water (for selecting spray water heater) is equal to the heat added to the air plus the heat added to the make-up water. The amount of make-up water is equal to the amount of moisture evaporated into the air and is determined from the following equation:

$$\text{Make-up water} = \frac{cfm_{sa}\,(W_{la} - W_{ea})}{7000 \times 12.7 \times 8.34}$$

where:

w_{ea}, W_{la} = moisture content of the air entering and leaving the spray washer in grains per pound of dry air

7000 = grains of moisture per pound of dry air

12.7 = volume of the mixture in cubic feet per pound of dry air, determined from psychrometric chart

8.34 = water in pounds per gallon

$$\text{Make-up water} = \frac{85{,}000\,(41 - 17)}{7000 \times 12.7 \times 8.34} = 2.8 \text{ gpm}$$

The heat added to the make-up spray water is determind from the following equation:

Heat added to make-up water

$= \text{gpm} \times 500\,(t_{ew}\text{-make-up water temp})$

$= 2.8 \times 500\,(76.8-65)$

$= 16{,}200$ Btu/hr

To select a water heater, the total amount of heat added to the spray water is determined by totaling the heat added to the air and the heat added to the make-up spray water.

Heat added to spray water

$= 1{,}830{,}000+16{,}200$

$= 1{,}846{,}200$ Btu/hr

It the make-up water was at a higher temperature than the required entering water temperature to the sprays, then a credit to the heat added to the spray water may be taken.

In this example a reheat coil is required to heat the air leaving the spray chamber, at 43.6 F db and at a constant moisture content of 41 gr/lb, to the required supply air temperature of 99.2 F db.

The requirements of the application illustrated in *Example 8* can also be met by preheating the outdoor air and mixing it with the return air from the space. This mixture must then be evaporatively cooled to the room dewpoint (or room moisture content). And finally, the air leaving the spray chamber must be reheated to the required supply air temperature.

SORBENT DEHUMIDIFIERS

Sorbent dehumidifiers contain liquid absorbent or solid adsorbent which are either sprayed directly into, or located in, the path of the air stream. The liquid absorbent changes either physically or chemically, or both, during the sorption process. The solid adsorbent does not change during the sorption process.

As moist air comes in contact with either the liquid absorbent or solid adsorbent, moisture is removed from the air by the difference in vapor pressure between the air stream and the sorbent. As this moisture condenses,

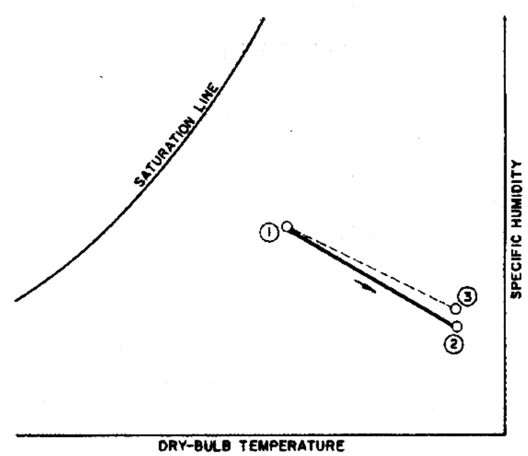


FIG. 58- SORBENT DEHUMIDIFICATION PROCESSES

latent heat of condensation is liberated, causing a rise in the temperature of the air stream and the sorbent material. This process occurs at a wet-bulb temperature that is approximately constant. However, instead of adding moisture to the air as in an evaporative cooling process, the reverse occurs. Heat is added to the air and moisture is removed from the air stream; thus it is a dehumidification and heating process as illustrated in *Fig. 58*. = Line *(1-2)* is the theoretical process and the dotted line *(1-3)* can vary, depending on the type of sorbent used.

PSYCHROMETRICS OF PARTIAL LOAD CONTROL

The apparatus required to maintain proper space conditions is normally selected for peak load operation. Actually, peak load occurs but a few times each year and operation is predominantly at partial load conditions. Partial load may be caused by a reduction in sensible or latent loads in the space, or in the outdoor air load. It may also be caused by a reduction in these loads in any combination.

PARTIAL LOAD ANALYSIS

Since the system operates at partial load most of the time and must maintain conditions commensurate with job requirements, partial load analysis is at least as important as the selection of equipment. Partial load analysis should include a study of resultant room conditions at minimum total load. Usually this will be sufficient. Certain applications, however, should be evaluated at minimum latent load with design sensible load, or minimum sensible load and full latent load. Realistic minimum and maximum loads should be assumed for the particular application so that, psychrometrically, the resulting room conditions are properly analyzed.

The six most common methods, used singly or in combination, of controlling space conditions for cooling applications at partial load are the following:

1. Reheat the supply air.
2. Bypass the heat transfer equipment.
3. Control the volume of the supply air.
4. Use on-off control of the air handing equipment.
5. Use on-off control of the refrigeration machine.
6. Control the refrigeration capacity.

The type of control selected for a specific application depends on the nature of the loads, the conditions to be maintained within the space, and available plant facilities.

REHEAT CONTROL

Reheat control maintains the dry-bulb temperature within the space by replacing any decrease in the sensible loads by an artificial load. As the internal latent load and/or the outdoor latent load decreases, the space relative humidity decreases. If humidity is to be maintained, rehumidifying is required in addition to reheat. This was described previously under *"Spray Process, Heating and Humidifying."*

Figure 59 illustrates the psychrometrics of reheat control. The solid lines represent the process at design load, and the broken lines indicate the resulting process at partial load. The RSHF value, plotted from room design conditions to point *(2),* must be calculated for the minimum practical room sensible load. The room thermostat then controls the temperature of the air leaving the reheat coil along line *(1-2).* This type of control is applicable for any RSHF ratio that intersects line *(1-2).*

If the internal latent loads decrease, the resulting room conditions are at point *(3),* and the new RSHF process line is along line *(2-3).* However, if humidity is to be maintained within the space, the reduced latent load is compensated by humidifying, thus returning to the design room conditions.

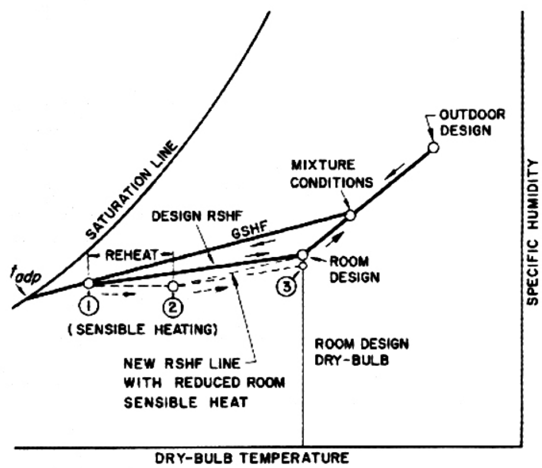


FIG. 59- PSYCHROMETRICS OF REHEAT CONTROL

BYPASS CONTROL

Bypass control maintains the dry-bulb temperature within the space by modulating the amount of air to be cooled, thus varying the supply air temperature to the space. *Fig. 60* illustrates one method of bypass control when bypassing return air only.

Bypass control may also be accomplished by bypassing a mixture of outdoor and return air around the heat transfer equipment. This method of control is inferior to bypassing return air only since it introduces raw unconditioned air into the space, thus allowing an increase in room relative humidity.

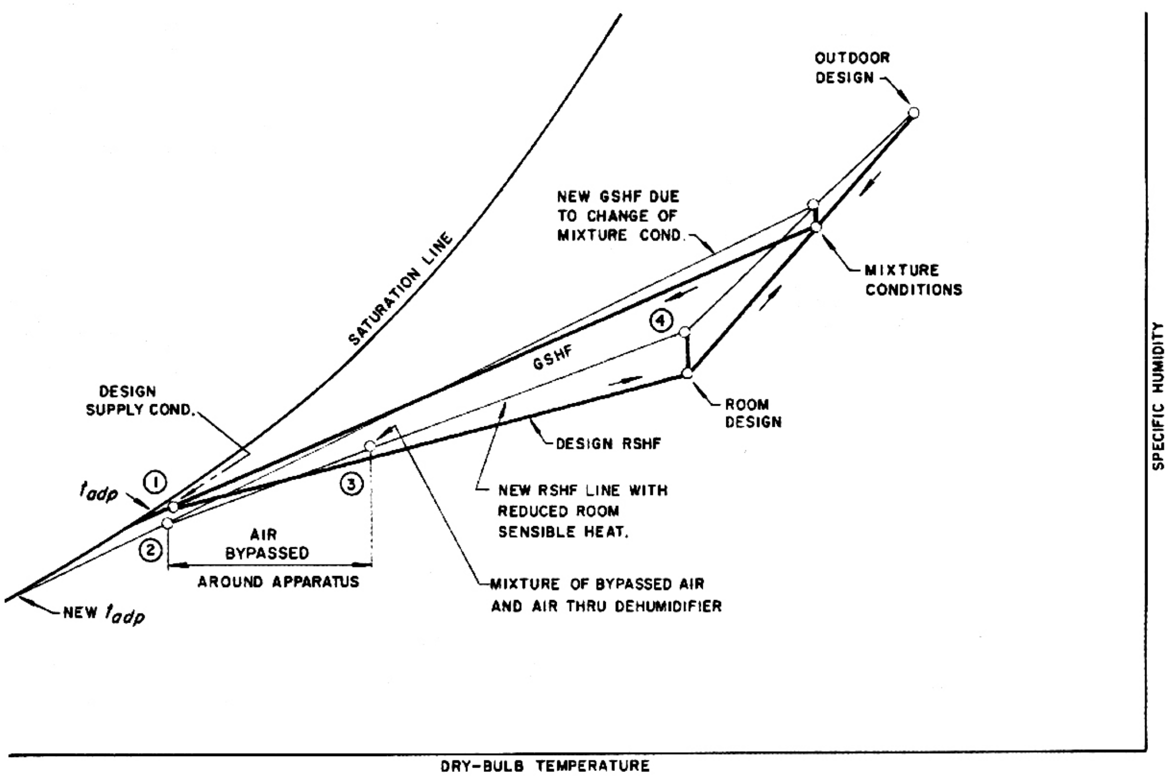


FIG. 60- PSYCHROMETRICS OF BYPASS CONTROL WITH RETURN AIR ONLY

A reduction in room sensible load causes the bypass control to reduce the amount of air thru the dehumidifier. This reduced air quantity results in equipment operation at a lower apparatus dewpoint. Also, the air leaves the dehumidifier at a lower temperature so that there is a tendency to adjust for a decrease in sensible load that is proportionately greater than the decrease in latent load.

Bypass control maintains the room dry-bulb temperature but does not prevent the relative humidity from rising above design. With bypass control, therefore, increased relative humidity occurs under conditions of decreasing room sensible load and relatively constant room latent load and outdoor air load.

The heavy lines in *Fig. 60* represent the cycle for design conditions. The light lines illustrate the initial cycle of the air when bypass control first begins to function. The new room conditions, mixture conditions and apparatus dewpoint continue to change until the equilibrium point is reached.

Point *(2)* on *Figs. 60 and 61* is the condition of air leaving the dehumidifier. This is a result of a smaller bypass factor and lower apparatus dewpoint caused by less air thru the cooling equipment and a smaller load on the equipment. Line *(2-3-4)* represents the new RSHF line caused by the reduced room sensible load. Point (3) falls on the new RSHF line when bypassing return air only.

Bypassing a mixture of outdoor and return air causes the mixture point *(3)* to fal on the GSHF line, *Fig. 60*. The air is then supplied to the space along the new RSHF line (not shown in *Fig. 60*) at a higher moisture content than the air supplied when bypassing return air only. Thus it can be readily observed that humidity control is further hindered with the introduction of unconditioned outdoor air into the space.

VOLUME CONTROL

Volume control of the supply air quantity provides essentially the same type of control that results from bypassing return air around the heat transfer equipment, Fig. 60. However, this type of control may produce problems in air distribution within the space and, therefore, the required air quantity at partial load should be evaluated for proper air distribution.

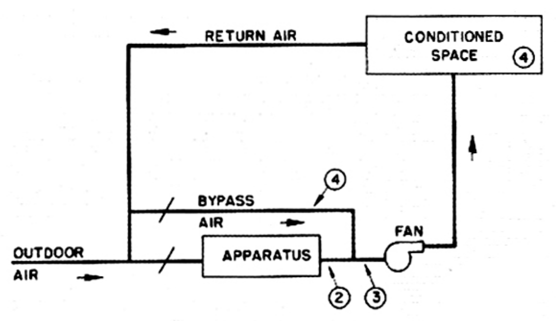


FIG. 61- SCHEMATIC SKETCH OF BYPASS CONTROL WITH BYPASS OF RETURN AIR ONLY

ON-OFF CONTROL OF AIR HANDLING EQUIPMENT

On-off control of air handling equipment (fan-coil units) results in a fluctuating room temperature and space relative humidity. During the "off" operation the ventilation air supply is shut off, but chilled water continues to flow thru the coils. This method of control is not recommended for high latent load applications, as control of humidity may be lost at reduced room sensible loads.

ON-OFF CONTROL OF REFRIGERATION EQUIPMENT

On-off control of refrigeration equipment (large packaged equipment) results in a fluctuating room temperature and space relative humidity. During the "off" operation air is available for ventilation purposes but the coil does not provide cooling. Thus, any outdoor air in the system is introduced into the space unconditioned. Also the condensed moisture that remains on the cooling coil, when the refrigeration equipment is turned off, is re-evaporated in the warm air stream. This is known as re-evaporation. Both of these conditions increase the space latent load, and excessive humidity results. This method of control is not recommended for high latent load applications since control of humidity may be lost at decreased room sensible loads.

REFRIGERATION CAPACITY CONTROL

Refrigeration capacity control may be used on either chilled water or direct expansion refrigeration equipment. Partial load control is accomplished on chilled water equipment by bypassing the chilled water around the air side equipment (fan-coil units). Direct expansion refrigeration equipment is controlled either by unloading the compressor cylinders or by back pressure regulation in the refrigerant suction line.

Refrigeration capacity control is normally used in combination with bypass or reheat control. When used in combination, results are excellent. When used alone, results are not as effective. For example, temperature can be maintained reasonably well, but relative humidity will rise above design at partial load conditions, because the latent load may not reduce in proportion to the sensible load.

PARTIAL LOAD CONTROL

Generally, reheat control is more expensive but provides the best control of conditions in the space. Bypass control, volume control and refrigeration capacity control provide reasonably good humidity control in average or high sensible heat factor applications, and poor humidity control in low sensible heat factor applications. On-off control usually results in the least desirable method of maintaining space conditions. However, this type of control is frequently used for high sensible heat factor applications with reasonably satisfactory results.

TABLE 65- APPARATUS DEWPOINTS

90 - 80 F DB

DB (F)	RH (%)	WB (F)	W (gr/lb)	EFFECTIVE SENSIBLE HEAT FACTOR AND APPARATUS DEWPOINT*									
90	20	62.7	42.0	ESHF	1.00	.96	.92	.90	.88	.86	.84	.82	.81
				ADP	43.5	41	39	37	35	32	29	24	22
	25	65.1	52.7	ESHF	1.00	.96	.92	.88	.84	.82	.80	.78	.75
				ADP	49.6	48	46	44	41	39	36	32	22
	30	67.3	63.6	ESHF	1.00	.92	.87	.83	.80	.76	.74	.72	.70
				ADP	54.5	52	50	48	46	42	38	34	24
	35	69.3	74.2	ESHF	1.00	.92	.85	.81	.76	.73	.71	.69	.66
				ADP	58.8	57	55	53	50	48	45	42	33
	40	71.2	84.8	ESHF	1.00	.92	.83	.78	.74	.69	.66	.63	.62
				ADP	62.4	61	59	57	55	52	48	44	40
	45	73.0	95.5	ESHF	1.00	.92	.82	.76	.70	.66	.62	.60	.58
				ADP	65.8	65	63	61	59	56	52	49	43
	50	74.9	106.4	ESHF	1.00	.92	.78	.68	.64	.60	.58	.56	.54
				ADP	68.9	68	66	63	61	58	56	53	47
	55	76.7	117.5	ESHF	1.00	.92	.76	.68	.64	.57	.54	.52	.50
				ADP	71.6	71	69	67	66	62	59	57	50
	60	78.4	128.4	ESHF	1.00	.86	.68	.60	.56	.52	.50	.48	.46
				ADP	74.2	73	71	69	67	64	62	59	50
	65	80.0	139.6	ESHF	1.00	.75	.68	.62	.55	.50	.47	.45	.43
				ADP	76.8	75	74	73	71	69	66	64	59
	70	81.6	151.0	ESHF	1.00	.78	.66	.60	.52	.47	.43	.41	.39
				ADP	79.0	78	77	76	74	72	69	66	58
85	20	59.6	35.8	ESHF	1.00	.98	.95	.92	.90	.88	.87	.86	.84
				ADP	39.4	38	36	34	32	30	28	26	22
	25	61.7	44.8	ESHF	1.00	.98	.93	.90	.86	.84	.82	.80	.78
				ADP	45.2	44	42	40	37	35	32	28	20
	30	63.7	54.1	ESHF	1.00	.94	.89	.85	.81	.79	.77	.75	.73
				ADP	50.2	48	46	44	40	38	35	31	22
	35	65.5	62.9	ESHF	1.00	.92	.86	.82	.78	.74	.72	.70	.69
				ADP	54.1	52	50	48	45	41	38	32	27
	40	67.4	71.7	ESHF	1.00	.92	.84	.79	.76	.73	.69	.67	.65
				ADP	57.9	56	54	52	50	48	44	40	32
	45	69.1	81.1	ESHF	1.00	.92	.83	.77	.72	.68	.64	.62	.61
				ADP	61.2	60	58	56	54	51	46	41	36
	50	70.8	90.1	ESHF	1.00	.92	.80	.73	.68	.64	.61	.59	.57
				ADP	64.2	63	61	59	57	54	51	48	39
	55	72.3	99.4	ESHF	1.00	.92	.83	.73	.67	.60	.57	.56	.54
				ADP	66.9	66	65	63	61	57	54	52	47
	60	73.9	108.8	ESHF	1.00	.92	.76	.67	.61	.56	.54	.52	.50
				ADP	69.5	69	67	65	63	60	58	55	49
	65	75.5	118.2	ESHF	1.00	.88	.69	.61	.56	.53	.50	.48	.47
				ADP	71.9	71	69	67	65	63	61	58	54
	70	77.0	127.6	ESHF	1.00	.81	.63	.55	.51	.49	.47	.45	.43
				ADP	74.0	73	71	69	67	66	64	62	55
82	35	63.3	57.0	ESHF	1.00	.92	.88	.84	.80	.76	.74	.72	.71
				ADP	51.6	49	48	46	43	39	36	31	27
	40	65.0	65.1	ESHF	1.00	.90	.87	.82	.78	.74	.71	.69	.67
				ADP	55.2	53	52	50	48	45	41	38	31
	45	66.7	73.5	ESHF	1.00	.91	.87	.80	.75	.72	.68	.65	.63
				ADP	58.5	57	56	54	52	50	46	41	53
	50	68.3	81.9	ESHF	1.00	.90	.80	.74	.70	.64	.62	.60	.59
				ADP	61.5	60	58	56	54	50	47	42	37
	55	69.8	90.2	ESHF	1.00	.90	.83	.74	.68	.64	.61	.58	.56
				ADP	64.2	63	62	60	58	56	54	50	44
	60	71.3	98.5	ESHF	1.00	.92	.76	.68	.63	.59	.56	.54	.52
				ADP	66.7	66	64	62	60	58	55	52	44
	65	72.8	107.0	ESHF	1.00	.86	.71	.63	.58	.54	.52	.51	.49
				ADP	69.1	68	66	64	62	60	58	56	51
	70	74.2	115.5	ESHF	1.00	.80	.71	.65	.60	.54	.51	.48	.46
				ADP	71.2	70	69	68	67	65	63	60	56
81	35	62.5	55.2	ESHF	1.00	.94	.89	.84	.81	.77	.75	.73	.71
				ADP	50.8	49	47	45	43	39	36	32	21
	40	64.2	63.2	ESHF	1.00	.94	.87	.82	.78	.75	.72	.69	.67
				ADP	54.4	53	51	49	47	45	41	36	23
	45	65.9	71.2	ESHF	1.00	.96	.91	.83	.78	.74	.70	.67	.64
				ADP	57.6	57	56	54	52	50	47	43	36
	50	67.5	79.0	ESHF	1.00	.90	.84	.80	.74	.70	.66	.62	.60
				ADP	60.5	59	58	57	55	53	50	45	38
	55	69.0	87.4	ESHF	1.00	.90	.77	.71	.66	.62	.60	.58	.56
				ADP	63.2	62	60	58	56	53	51	47	35
	60	70.5	95.4	ESHF	1.00	.92	.77	.68	.63	.59	.56	.54	.53
				ADP	65.8	65	63	61	59	56	53	50	46
	65	71.9	103.7	ESHF	1.00	.85	.76	.71	.66	.60	.56	.52	.50
				ADP	68.2	67	66	65	64	62	60	56	52
	70	73.3	111.9	ESHF	1.00	.80	.71	.61	.55	.52	.48	.47	.46
				ADP	70.3	69	68	66	64	62	58	56	52
80	20	56.4	30.4	ESHF	1.00	.98	.95	.93	.91	.89	.88	.87	.86
				ADP	35.4	34	32	30	28	26	24	22	20
	25	58.3	38.0	ESHF	1.00	.96	.93	.90	.88	.86	.84	.82	.81
				ADP	40.9	39	37	35	33	31	28	24	21
	30	60.0	45.6	ESHF	1.00	.96	.91	.88	.85	.83	.80	.78	.76
				ADP	45.7	44	42	40	38	36	32	28	21
	35	61.8	53.5	ESHF	1.00	.94	.88	.85	.82	.79	.77	.73	.72
				ADP	49.8	48	46	44	42	40	37	29	24
	40	63.5	61.2	ESHF	1.00	.94	.90	.84	.80	.76	.73	.70	.68
				ADP	53.5	52	51	49	47	44	41	36	28
	45	65.1	68.9	ESHF	1.00	.96	.87	.81	.76	.73	.70	.67	.64
				ADP	56.8	56	54	52	50	48	45	41	31
	50	66.7	76.7	ESHF	1.00	.89	.80	.74	.70	.66	.64	.62	.61
				ADP	59.7	58	56	54	52	49	46	42	39
	55	68.2	84.6	ESHF	1.00	.89	.82	.74	.69	.65	.61	.59	.57
				ADP	62.3	61	60	58	56	54	50	47	40
	60	69.6	92.3	ESHF	1.00	.91	.83	.72	.66	.62	.59	.57	.54
				ADP	64.8	64	63	61	59	57	55	53	47
	65	71.1	100.4	ESHF	1.00	.85	.76	.71	.63	.58	.55	.52	.50
				ADP	67.2	66	65	64	62	60	58	54	47
	70	72.4	108.3	ESHF	1.00	.78	.71	.65	.61	.55	.52	.49	.47
				ADP	69.4	68	67	66	65	63	61	58	53

See page 147 for notes.

TABLE 65- APPARATUS DEWPOINTS (Continued)

79 – 72 F DB

ROOM CONDITIONS DB (F)	RH (%)	WB (F)	W (gr/lb)	EFFECTIVE SENSIBLE HEAT FACTOR AND APPARATUS DEWPOINT*									
79	35	61.0	51.5	ESHF	1.00	.96	.91	.89	.85	.82	.78	.75	.73
				ADP	48.9	48	46	45	43	41	37	32	26
	40	62.7	59.2	ESHF	1.00	.97	.90	.84	.80	.76	.74	.71	.69
				ADP	52.7	52	50	48	46	43	41	36	29
	45	64.3	66.7	ESHF	1.00	.91	.83	.78	.75	.72	.70	.67	.65
				ADP	55.9	54	52	50	48	46	44	39	32
	50	65.9	74.2	ESHF	1.00	.89	.80	.75	.71	.68	.66	.63	.61
				ADP	58.9	57	55	53	51	49	47	42	33
	55	67.4	81.9	ESHF	1.00	.96	.82	.74	.69	.66	.63	.60	.58
				ADP	61.4	61	59	57	55	53	51	47	41
	60	68.8	89.3	ESHF	1.00	.90	.76	.69	.64	.61	.57	.55	.54
				ADP	63.9	63	61	59	57	55	51	47	41
	65	70.2	97.0	ESHF	1.00	.84	.71	.64	.59	.56	.54	.52	.51
				ADP	66.3	65	63	61	59	57	55	51	48
	70	71.6	104.8	ESHF	1.00	.81	.71	.65	.58	.54	.52	.50	.48
				ADP	68.5	67	66	65	63	61	59	57	53

DB (F)	RH (%)	WB (F)	W (gr/lb)										
78	35	60.3	50.0	ESHF	1.00	.96	.91	.87	.83	.79	.77	.75	.73
				ADP	48.2	47	45	43	41	37	35	31	22
	40	61.9	57.3	ESHF	1.00	.93	.87	.82	.79	.77	.73	.71	.69
				ADP	51.7	50	48	46	44	42	38	34	25
	45	63.5	64.6	ESHF	1.00	.95	.86	.81	.76	.74	.70	.68	.66
				ADP	55.0	54	52	50	48	46	42	39	34
	50	65.0	71.9	ESHF	1.00	.94	.83	.76	.73	.70	.67	.64	.62
				ADP	57.9	57	55	53	51	49	47	42	36
	55	66.6	79.2	ESHF	1.00	.96	.83	.75	.70	.65	.62	.60	.59
				ADP	60.5	60	58	56	54	51	48	44	41
	60	67.9	86.4	ESHF	1.00	.90	.82	.76	.69	.64	.60	.57	.55
				ADP	63.0	62	61	60	58	56	53	49	42
	65	69.3	93.8	ESHF	1.00	.85	.77	.71	.67	.62	.58	.54	.52
				ADP	65.2	64	63	62	61	59	57	53	48
	70	70.6	101.2	ESHF	1.00	.71	.66	.62	.59	.55	.52	.50	.48
				ADP	67.5	65	64	63	62	60	58	55	48

DB (F)	RH (%)	WB (F)	W (gr/lb)										
77	35	59.6	48.3	ESHF	1.00	.96	.91	.87	.83	.79	.77	.75	.74
				ADP	47.3	46	44	42	40	36	33	28	24
	40	61.2	55.5	ESHF	1.00	.96	.89	.84	.81	.78	.76	.73	.70
				ADP	50.9	50	48	46	44	42	40	36	27
	45	62.7	62.4	ESHF	1.00	.94	.86	.81	.77	.74	.72	.69	.66
				ADP	54.1	53	51	49	47	45	43	39	29
	50	64.2	69.7	ESHF	1.00	.94	.84	.77	.73	.70	.68	.65	.63
				ADP	57.0	56	54	52	50	48	46	42	37
	55	65.6	76.6	ESHF	1.00	.95	.83	.75	.70	.67	.63	.61	.59
				ADP	59.6	59	57	55	53	51	48	44	37
	60	67.1	83.6	ESHF	1.00	.89	.82	.77	.73	.67	.62	.58	.56
				ADP	62.0	61	60	59	58	56	53	48	43
	65	68.5	90.8	ESHF	1.00	.84	.72	.64	.60	.57	.55	.54	.53
				ADP	64.4	63	61	59	57	55	53	51	48
	70	69.8	97.9	ESHF	1.00	.79	.66	.60	.55	.53	.51	.50	.49
				ADP	66.5	65	63	61	59	57	55	53	49

ROOM CONDITIONS DB (F)	RH (%)	WB (F)	W (gr/lb)	EFFECTIVE SENSIBLE HEAT FACTOR AND APPARATUS DEWPOINT*									
76	35	58.9	46.7	ESHF	1.00	.96	.91	.87	.84	.81	.79	.77	.74
				ADP	46.3	45	43	41	39	37	34	31	21
	40	60.4	53.7	ESHF	1.00	.96	.89	.84	.81	.78	.76	.72	.70
				ADP	49.9	49	47	45	43	41	39	32	22
	45	61.9	60.4	ESHF	1.00	.94	.86	.81	.77	.74	.71	.69	.67
				ADP	53.2	52	50	48	46	44	40	37	31
	50	63.4	67.4	ESHF	1.00	.93	.83	.77	.73	.69	.67	.65	.63
				ADP	56.2	55	53	51	49	46	43	40	32
	55	64.9	74.0	ESHF	1.00	.94	.82	.75	.70	.67	.65	.62	.60
				ADP	58.7	58	56	54	52	50	48	44	38
	60	66.2	80.9	ESHF	1.00	.90	.77	.70	.66	.62	.60	.58	.57
				ADP	61.1	60	58	56	54	52	49	46	43
	65	67.6	87.6	ESHF	1.00	.84	.72	.65	.61	.58	.56	.54	.53
				ADP	63.4	62	60	58	56	54	52	48	43
	70	68.9	94.6	ESHF	1.00	.80	.67	.60	.56	.54	.52	.51	.50
				ADP	65.5	64	62	60	58	56	54	52	49

DB (F)	RH (%)	WB (F)	W (gr/lb)										
75	20	53.2	25.7	ESHF	1.00	.98	.96	.94	.92	.90	.89		
				ADP	31.5	30	28	26	24	22	20		
	25	54.8	32.1	ESHF	1.00	.95	.92	.90	.88	.86	.84		
				ADP	36.9	34	32	30	28	25	21		
	30	56.5	38.5	ESHF	1.00	.97	.93	.90	.87	.85	.82	.80	.79
				ADP	41.4	40	38	36	34	32	28	24	20
	35	58.1	45.2	ESHF	1.00	.96	.91	.87	.84	.80	.78	.76	.75
				ADP	45.5	44	42	40	38	34	31	27	22
	40	59.6	51.8	ESHF	1.00	.96	.89	.84	.81	.79	.76	.73	.71
				ADP	49.1	48	46	44	42	40	37	32	24
	45	61.1	58.2	ESHF	1.00	.94	.87	.81	.77	.75	.72	.69	.67
				ADP	52.2	51	49	47	45	43	40	35	21
	50	62.6	65.0	ESHF	1.00	.92	.84	.78	.74	.71	.69	.66	.64
				ADP	55.2	54	52	50	48	46	44	40	34
	55	64.0	71.5	ESHF	1.00	.94	.87	.78	.73	.69	.65	.63	.61
				ADP	57.8	57	56	54	52	50	47	44	39
	60	65.3	77.9	ESHF	1.00	.90	.77	.71	.66	.63	.61	.59	.58
				ADP	60.1	59	57	55	53	51	49	46	43
	65	66.7	84.8	ESHF	1.00	.84	.72	.65	.61	.59	.57	.55	.54
				ADP	62.4	61	59	57	55	53	51	48	44
	70	68.0	91.2	ESHF	1.00	.80	.73	.68	.61	.57	.54	.52	.51
				ADP	64.5	63	62	61	59	57	55	52	49

DB (F)	RH (%)	WB (F)	W (gr/lb)										
72	35	55.9	40.8	ESHF	1.00	.98	.93	.89	.86	.83	.81	.79	.77
				ADP	42.8	42	40	38	36	34	31	28	22
	40	57.3	46.7	ESHF	1.00	.95	.92	.87	.84	.81	.77	.75	.73
				ADP	46.3	45	44	42	40	38	34	30	23
	45	58.7	52.7	ESHF	1.00	.94	.87	.82	.79	.76	.74	.71	.69
				ADP	49.5	48	46	44	42	40	38	32	22
	50	60.1	58.8	ESHF	1.00	.92	.88	.81	.77	.73	.70	.68	.66
				ADP	52.4	51	50	48	46	43	40	37	30
	55	61.4	64.4	ESHF	1.00	.93	.83	.77	.72	.68	.66	.64	.63
				ADP	54.9	54	52	50	48	45	42	39	36
	60	62.7	70.2	ESHF	1.00	.89	.79	.72	.68	.65	.63	.61	.60
				ADP	57.3	56	54	52	50	48	46	42	39

TABLE 65- APPARATUS DEWPOINTS (Continued)

72 – 55 F DB

DB (F)	RH (%)	WB (F)	W (gr/lb)		EFFECTIVE SENSIBLE HEAT FACTOR AND APPARATUS DEWPOINT*								
72	65	64.0	76.3	ESHF	1.00	.84	.73	.67	.63	.61	.59	.58	
				ADP	59.5	58	56	54	52	50	48	47	
	70	65.2	82.3	ESHF	1.00	.80	.69	.62	.59	.56	.54	.53	
				ADP	61.6	60	58	56	54	51	48	44	

DB (F)	RH (%)	WB (F)	W (gr/lb)		EFFECTIVE SENSIBLE HEAT FACTOR AND APPARATUS DEWPOINT*								
70	20	49.9	21.6	ESHF	1.00	.98	.96	.94	.93				
				ADP	27.6	26	24	22	21				
	25	51.5	27.0	ESHF	1.00	.97	.94	.92	.90	.88			
				ADP	33.7	31	29	27	25	22			
	30	53.0	32.8	ESHF	1.00	.98	.94	.91	.88	.86	.84	.82	
				ADP	37.1	36	34	32	30	27	25	20	
	35	54.4	38.0	ESHF	1.00	.97	.93	.89	.86	.84	.82	.80	.78
				ADP	41.1	40	38	36	34	32	30	27	22
	40	55.8	43.5	ESHF	1.00	.95	.90	.86	.83	.80	.78	.76	.74
				ADP	44.5	43	41	39	37	35	32	29	22
	45	57.1	49.1	ESHF	1.00	.93	.87	.82	.79	.77	.75	.73	.71
				ADP	47.7	46	44	42	40	38	36	33	27
	50	58.5	54.8	ESHF	1.00	.92	.84	.80	.76	.74	.71	.69	.67
				ADP	50.5	49	47	45	43	41	38	35	25
	55	59.7	60.1	ESHF	1.00	.93	.83	.77	.73	.71	.68	.66	.64
				ADP	53.1	52	50	48	46	44	42	38	32
	60	60.9	65.5	ESHF	1.00	.89	.79	.73	.69	.66	.64	.62	.61
				ADP	55.4	54	52	50	48	46	43	40	36
	65	62.2	71.1	ESHF	1.00	.93	.78	.71	.66	.63	.61	.59	.58
				ADP	57.7	57	55	53	51	49	47	44	40
	70	63.4	76.9	ESHF	1.00	.90	.74	.66	.61	.59	.57	.56	.55
				ADP	59.8	59	57	55	53	51	49	47	45
	75	64.5	82.5	ESHF	1.00	.88	.70	.62	.57	.55	.53	.52	.51
				ADP	61.7	61	59	57	55	53	51	49	44
	80	65.7	88.0	ESHF	1.00	.87	.73	.65	.60	.54	.51	.49	.48
				ADP	63.5	63	62	61	60	58	56	53	49
	85	66.8	93.7	ESHF	1.00	.71	.56	.52	.50	.48	.47	.46	.45
				ADP	65.3	64	62	61	60	59	58	57	54
	90	67.9	99.3	ESHF	1.00	.66	.56	.50	.47	.45	.43	.42	.41
				ADP	66.9	66	65	64	63	62	61	60	56
	95	69.0	105.0	ESHF	1.00	.60	.47	.42	.39	.38	.37		
				ADP	68.5	68	67	66	65	64	62		

DB (F)	RH (%)	WB (F)	W (gr/lb)		EFFECTIVE SENSIBLE HEAT FACTOR AND APPARATUS DEWPOINT*								
65	60	56.6	55.0	ESHF	1.00	.95	.84	.77	.73	.70	.68	.66	.65
				ADP	50.6	50	48	46	44	42	39	36	34
	65	57.7	59.7	ESHF	1.00	.92	.85	.80	.73	.69	.66	.64	.62
				ADP	52.9	52	51	50	48	46	44	41	37
	70	58.9	64.5	ESHF	1.00	.89	.80	.76	.69	.65	.62	.60	.58
				ADP	55.0	54	53	52	50	48	46	43	37
	75	59.9	69.2	ESHF	1.00	.88	.78	.72	.65	.61	.58	.56	.55
				ADP	56.9	56	55	54	52	50	48	45	41
	80	51.0	73.8	ESHF	1.00	.75	.68	.63	.60	.58	.55	.53	.52
				ADP	58.7	57	56	55	54	53	51	48	46
	85	62.0	78.6	ESHF	1.00	.71	.63	.58	.55	.52	.50	.49	
				ADP	60.3	59	58	57	56	54	52	50	
	90	63.0	83.2	ESHF	1.00	.70	.58	.53	.50	.48	.46	.45	
				ADP	61.9	61	60	59	58	57	55	53	
	95	64.0	88.0	ESHF	1.00	.69	.51	.46	.43	.42	.41		
				ADP	63.5	63	62	61	60	59	58		

DB (F)	RH (%)	WB (F)	W (gr/lb)		EFFECTIVE SENSIBLE HEAT FACTOR AND APPARATUS DEWPOINT*								
60	60	52.3	46.2	ESHF	1.00	.94	.89	.81	.77	.74	.72	.70	.68
				ADP	46.0	45	44	42	40	38	36	34	28
	65	53.3	50.0	ESHF	1.00	.91	.86	.78	.74	.70	.69	.67	.65
				ADP	48.1	47	46	44	42	40	39	36	31
	70	54.3	53.9	ESHF	1.00	.89	.83	.74	.70	.67	.65	.63	.62
				ADP	50.1	49	48	46	44	42	40	37	34
	75	55.3	57.8	ESHF	1.00	.79	.74	.71	.68	.64	.62	.60	.59
				ADP	52.0	50	49	48	47	45	43	40	37
	80	56.3	61.7	ESHF	1.00	.85	.76	.70	.66	.61	.59	.57	.56
				ADP	53.8	53	52	51	50	48	46	44	41
	85	57.2	65.5	ESHF	1.00	.75	.67	.63	.57	.56	.54	.53	
				ADP	55.4	54	53	52	50	49	47	45	
	90	58.2	69.4	ESHF	1.00	.72	.62	.57	.54	.52	.50	.49	
				ADP	57.0	56	55	54	53	52	50	47	
	95	59.1	73.5	ESHF	1.00	.69	.55	.49	.47	.46	.45		
				ADP	58.5	58	57	56	55	54	52		

DB (F)	RH (%)	WB (F)	W (gr/lb)		EFFECTIVE SENSIBLE HEAT FACTOR AND APPARATUS DEWPOINT*								
55	60	47.9	38.4	ESHF	1.00	.93	.89	.85	.80	.77	.75	.73	.71
				ADP	41.3	40	39	38	36	34	32	29	23
	65	48.8	41.4	ESHF	1.00	.91	.86	.83	.78	.74	.72	.70	.68
				ADP	43.3	42	41	40	38	36	34	31	24
	70	49.7	44.6	ESHF	1.00	.90	.84	.80	.74	.71	.69	.67	.66
				ADP	45.2	44	43	42	40	38	36	33	31
	75	50.6	48.0	ESHF	1.00	.89	.82	.74	.69	.66	.65	.64	.63
				ADP	47.1	46	45	43	41	39	37	36	34
	80	51.5	51.2	ESHF	1.00	.88	.79	.74	.67	.64	.62	.61	.60
				ADP	48.8	48	47	46	44	42	40	39	37
	85	52.4	54.5	ESHF	1.00	.77	.70	.66	.63	.60	.58	.57	
				ADP	50.4	49	48	47	46	44	42	40	
	90	53.2	57.7	ESHF	1.00	.76	.67	.61	.58	.55	.54	.53	
				ADP	52.0	51	50	49	48	46	44	41	
	95	54.2	61.2	ESHF	1.00	.69	.58	.54	.51	.49			
				ADP	53.6	53	52	51	50	48			

*The values shown in the gray areas indicate the lowest effective sensible heat factor possible without the use of reheat. This limiting condition is the lowest effective sensible heat factor line that intersects the saturation curve. Note that the room dewpoint is equal to the required apparatus dewpoint for an effective sensible heat factor of 1.0.

NOTES FOR TABLE 65:

1. For Room Conditions Not Given; The apparatus dewpoint may be determined from the scale on the chart, or may be calculated as shown in the following equation:

$$ESHF = \frac{1}{1 + .628\,\dfrac{(W_{rm} - W_{adp})}{(t_{rm} - t_{adp})}}$$

This equation in more familiar form is:

$$ESHF = \frac{0.244\,(t_{rm} - t_{adp})}{0.244\,(t_{rm} - t_{adp}) + \dfrac{1076}{7000}\,(W_{rm} - W_{adp})}$$

(Cont.)

where w_{rm} = room moisture content, gr/lb of dry air
W_{adp} = moisture content at apparatus dewpoint, gr/lb of dry air
t_{rm} = room dry-bulb temperature
t_{adp} = apparatus dewpoint temperature
0.244 = specific heat of moist air at 55 F dewpoint, Btu per deg F per lb of dry air
1076 = average heat removal required to condense one pound of water vapor from the room air
7000 = grains per pound.

2. **For High Elevations**. For effective sensible heat factors at high elevations, see *Table 66*.
3. **For Apparatus Dewpoint Below Freezing**. The latent heat of fusion of the moisture removed is not included in the calculation of apparatus dewpoint below freezing or in the calculation of room load, in order to simplify estimating procedures. Use the same equation as in Note 1. The selection of equipment on a basis of 16 to 18 hour operating time provides a safety factor large enough to cover the omission of this latent heat of fusion, which is a small part of the total load.

TABLE 66- EQUIVALENT EFFECTIVE SENSIBLE HEAT FACTORS FOR VARIOUS ELEVATIONS*

For use with sea level psychrometric chart or tables

Effective Sensible Heat Factor from Air Conditioning Load Estimate	Elevation (Feet) and Barometric Pressure (Inches of Hg) at Installation									
	1000 (28.86)	2000 (27.82)	3000 (26.82)	4000 (25.84)	5000 (24.89)	6000 (23.98)	7000 (23.09)	8000 (22.12)	9000 (21.39)	10000 (20.57)
	Equivalent Effective Sensible Heat Factor Referred to a Sea Level Psychrometric Chart or Tables									
.95	.95	.95	.95	.96	.96	.96	.96	.96	.96	.96
.90	.90	.91	.91	.91	.92	.92	.92	.92	.93	.93
.85	.85	.86	.86	.87	.87	.88	.88	.88	.89	.89
.80	.81	.81	.82	.82	.83	.83	.84	.84	.85	.85
.75	.76	.76	.77	.78	.78	.79	.80	.80	.81	.81
.70	.71	.72	.72	.73	.74	.75	.75	.76	.77	.77
.65	.66	.67	.68	.68	.69	.70	.71	.71	.72	.73
.60	.61	.62	.63	.64	.64	.65	.66	.67	.68	.69
.55	.56	.57	.58	.59	.60	.61	.61	.62	.63	.64
.50	.51	.52	.53	.54	.55	.56	.57	.57	.58	.59

*Values obtained by use of equation

$$ESHF_e = \frac{1}{\dfrac{(p_1)(1 - ESHF)}{(p_o)(ESHF)} + 1}$$

where p_o = barometric pressure at sea level
p_1 = barometric pressure at high elevation
ESHF = ESHF obtained from air conditioning load estimate
$ESHF_e$ = equivalent ESHF referred to a sea level psychrometric chart or *Table 66*

NOTES FOR TABLE 66:

1. The required apparatus dewpoint for the high elevation is determined from the sea level chart or Table 65 by use of the equivalent effective sensible heat factor. The relative humidity and dry-bulb temperature must be used to define the room condition when using this table because the above equation was derived on this basis. The room wet-bulb temperature must not be used because the wet-bulb temperature corresponding to any particular condition, for example, 75 F db, 40% rh, at a high elevation is lower (except for saturation) than that corresponding to the same condition (75 F db, 40% rh) at sea level. For the same value of room relative humidity and dry-bulb temperature, and the same apparatus dew-point, there is a greater difference in moisture content between the two conditions at high elevation than at sea level. Therefore, a higher apparatus dewpoint is required at high elevation for a given effective sensible heat factor.
2. Air conditioning load estimate (See *Fig. 44*). The factors 1.08 and .68 on the air conditioning load estimate should be multiplied by the direct ratio of the barometric pressures $\frac{(p_1)}{(p_o)}$.

   Using this method, it is assumed that the air quantity (cfm) is measured at actual conditions rather than at standard air conditions. The outdoor and room moisture contents, grains per pound, must also be corrected for high elevations.
3. Reheat-Where the equivalent effective sensible heat factor is lower than the shaded values in *Table 65*, reheat is required.

INDEX PART 1 LOAD ESTIMATING

I

L

M

O

P

R

S

FIGURE PART 1 LOAD ESTIMATING

Carrier

TABLE PART 1 LOAD ESTIMATING

CHART PART 1 LOAD ESTIMATING

SYMBOLS

cfm_{ba}	bypassed air quantity around apparatus
cfm_{da}	dehumidified air quantity
cfm_{oa}	outdoor air quantity
cfm_{ra}	return air quantity
cfm_{sa}	supply air quantity
h	specific enthalpy
h_{adp}	apparatus dewpoint enthalpy
h_{es}	effective surface temperature enthalpy
h_{ea}	entering air enthalpy
h_{la}	leaving air enthalpy
h_{m}	mixture of outdoor and return air enthalpy
h_{oa}	outdoor air enthalpy
h_{rm}	room air enthalpy
h_{sa}	supply air enthalpy
t	temperature
t_{adp}	apparatus dewpoint temperature
t_{edb}	entering dry-bulb temperature
t_{es}	effective surface temperature
t_{ew}	entering water temperature
t_{ewb}	entering wet-bulb temperature
t_{ldb}	leaving dry-bulb temperature
t_{lw}	leaving water temperature
t_{lwb}	leaving wet-bulb temperature
t_{m}	mixture of outdoor and return air dry-bulb temperature
t_{oa}	outdoor air dry-bulb temperature
t_{rm}	room dry-bulb temperature
t_{sa}	supply air dry-bulb temperature
W	moisture content or specific humidity
W_{adp}	apparatus dewpoint moisture content
W_{ea}	entering air moisture content
W_{es}	effective surface temperature moisture content
W_{la}	leaving air moisture content
W_{m}	mixture of outdoor and return air moisture content
W_{oa}	outdoor air moisture content
W_{rm}	room moisture content
W_{sa}	supply air moisture content

ABBREVIATIONS

adp	apparatus dewpoint
BF	bypass factor
(BF) (OALH)	bypassed outdoor air latent heat
(BF) (OASH)	bypassed outdoor air sensible heat
(BF) (OATH)	bypassed outdoor air total heat
Btu/hr	British thermal units per hour
cfm	cubic feet per minute
db	dry-bulb
dp	dewpoint
ERLH	effective room latent heat
ERSH	effective room sensible heat
ERTH	effective room total heat
ESHF	effective sensible heat factor
F	Fahrenheit degrees
fpm	feet per minute
gpm	gallons per minute
gr/lb	grains per pound
GSHF	grand sensible heat factor
GTH	grand total heat
GTHS	grand total heat supplement
OALH	outdoor air latent heat
OASH	outdoor air sensible heat
OATH	outdoor air total heat
rh	relative humidity
RLH	room latent heat
RLHS	room latent heat supplement
RSH	room sensible heat
RSHF	room sensible heat factor
RSHS	room sensible heat supplement
RTH	room total heat
Sat Eff	saturation efficiency of sprays
SHF	sensible heat factor
TLH	total latent heat
TSH	total sensible heat
wb	wet-bulb

FORMULAS PART 1 LOAD ESTIMATING

PSYCHROMETRIC FORMULAS

A. AIR MIXING EQUATIONS (Outdoor and Return Air)

$$t_m = \frac{(cfm_{oa} \times t_{oa}) + (cfm_{ra} \times t_{rm})}{cfm_{sa}} \quad (1)$$

$$h_m = \frac{(cfm_{oa} \times h_{oa}) + (cfm_{ra} \times h_{rm})}{cfm_{sa}} \quad (2)$$

$$W_m = \frac{(cfm_{oa} \times W_{oa}) + (cfm_{ra} \times W_{rm})}{cfm_{sa}} \quad (3)$$

B. COOLING LOAD EQUATIONS

$$\text{ERSH} = \text{RSH} + (\text{BF})(\text{OASH}) + \text{RSHS}^* \quad (4)$$
$$\text{ERLH} = \text{RLH} + (\text{BF})(\text{OALH}) + \text{RLHS}^* \quad (5)$$
$$\text{ERTH} = \text{ERLH} + \text{ERSH} \quad (6)$$

$$\text{TSH} = \text{RSH} + \text{OASH} + \text{RSHS}^* \quad (7)$$
$$\text{TLH} = \text{RLH} + \text{OALH} + \text{RLHS}^* \quad (8)$$
$$\text{GTH} = \text{TSH} + \text{TLH} + \text{GTHS}^* \quad (9)$$

$$\text{RSH} = 1.08\dagger \times cfm_{sa} \times (t_{rm} - t_{sa}) \quad (10)$$
$$\text{RLH} = .68\dagger \times cfm_{sa} \times (W_{rm} - W_{sa}) \quad (11)$$
$$\text{RTH} = 4.45\dagger \times cfm_{sa} \times (h_{rm} - h_{sa}) \quad (12)$$
$$\text{RTH} = \text{RSH} + \text{RLH} \quad (13)$$

$$\text{OASH} = 1.08 \times cfm_{oa}(t_{oa} - t_{rm}) \quad (14)$$
$$\text{OALH} = .68 \times cfm_{oa}(W_{oa} - W_{rm}) \quad (15)$$
$$\text{OATH} = 4.45 \times cfm_{oa}(h_{oa} - h_{rm}) \quad (16)$$

$$\text{OATH} = \text{OASH} + \text{OALH} \quad (17)$$
$$(\text{BF})(\text{OATH}) = (\text{BF})(\text{OASH}) + (\text{BF})(\text{OALH}) \quad (18)$$

$$\text{ERSH} = 1.08 \times cfm_{da}\ddagger \times (t_{rm} - t_{adp})(1 - \text{BF}) \quad (19)$$

$$\text{ERLH} = .68 \times cfm_{da}\ddagger \times (W_{rm} - W_{adp})(1 - \text{BF}) \quad (20)$$

$$\text{ERTH} = 4.45 \times cfm_{da}\ddagger \times (h_{rm} - h_{adp})(1 - \text{BF}) \quad (21)$$

*RSHS, RLHS and GTHS are supplementary loads due to duct heat gain, duct leakage loss, fan and pump horsepower gains, etc. To simplify the various examples, these supplementary loads have *not* been used in the calculations. However, in actual practice, these supplementary loads should be used where appropriate. *Chapter 7* gives the values for the various supplementary loads. *Fig. 1, Chapter 1,* illustrates the method of accounting for these supplementary loads on the air conditioning load estimate.

†*Item H, page 151,* gives the derivation of these air constants.

‡When no air is to be physically bypassed around the conditioning apparatus, $cfm_{da} = cfm_{sa}$.

$$\text{TSH} = 1.08 \times cfm_{da}\ddagger \times (t_{edb} - t_{ldb})^{**} \quad (22)$$
$$\text{TLH} = .68 \times cfm_{da}\ddagger \times (W_{ea} - W_{la})^{**} \quad (23)$$
$$\text{GTH} = 4.45 \times cfm_{da}\ddagger \times (h_{ea} - h_{la})^{**} \quad (24)$$

C. SENSIBLE HEAT FACTOR EQUATIONS

$$\text{RSHF} = \frac{\text{RSH}}{\text{RSH} + \text{RLH}} = \frac{\text{RSH}}{\text{RTH}} \quad (25)$$

$$\text{ESHF} = \frac{\text{ERSH}}{\text{ERSH} + \text{ERLH}} = \frac{\text{ERSH}}{\text{ERTH}} \quad (26)$$

$$\text{GSHF} = \frac{\text{TSH}}{\text{TSH} + \text{TLH}} = \frac{\text{TSH}}{\text{GTH}} \quad (27)$$

D. BYPASS FACTOR EQUATIONS

$$\text{BF} = \frac{t_{ldb} - t_{adp}}{t_{edb} - t_{adp}};\ (1 - \text{BF}) = \frac{t_{edb} - t_{ldb}}{t_{edb} - t_{adp}} \quad (28)$$

$$\text{BF} = \frac{W_{la} - W_{adp}}{W_{ea} - W_{adp}};\ (1 - \text{BF}) = \frac{W_{ea} - W_{la}}{W_{ea} - W_{adp}} \quad (29)$$

$$\text{BF} = \frac{h_{la} - h_{adp}}{h_{ea} - h_{adp}};\ (1 - \text{BF}) = \frac{h_{ea} - h_{la}}{h_{ea} - h_{adp}} \quad (30)$$

E. TEMPERATURE EQUATIONS AT APPARATUS

$$t_{edb}^{**} = \frac{(cfm_{oa} \times t_{oa}) + (cfm_{ra} \times t_{rm})}{cfm_{sa}\ddagger} \quad (31)$$

$$t_{ldb} = t_{adp} + \text{BF}(t_{edb} - t_{adp}) \quad (32)$$

t_{ewb} and t_{lwb} correspond to the calculated values of h_{ea} and h_{la} on the psychrometric chart.

$$h_{ea}^{**} = \frac{(cfm_{oa} \times h_{oa}) + (cfm_{ra} \times h_{rm})}{cfm_{sa}\ddagger} \quad (33)$$

$$h_{la} = h_{adp} + \text{BF}(h_{ea} - h_{adp}) \quad (34)$$

F. TEMPERATURE EQUATIONS FOR SUPPLY AIR

$$t_{sa} = t_{rm} - \frac{\text{RSH}}{1.08\,(cfm_{sa}\ddagger)} \quad (35)$$

**When t_m, W_m and h_m are equal to the entering conditions at the cooling apparatus, they may be substituted for t_{edb}, W_{ea} and h_{ea} respectively.

G. AIR QUANTITY EQUATIONS

$$cfm_{da} = \frac{\text{ERSH}}{1.08 \times (1 - \text{BF})\,(t_{rm} - t_{adp})} \tag{36}$$

$$cfm_{da} = \frac{\text{ERLH}}{.68 \times (1 - \text{BF})\,(W_{rm} - W_{adp})} \tag{37}$$

$$cfm_{da} = \frac{\text{ERTH}}{4.45 \times (1 - \text{BF})\,(h_{rm} - h_{adp})} \tag{38}$$

$$cfm_{da}\ddagger = \frac{\text{TSH}}{1.08\,(t_{edb} - t_{ldb})} \tag{39}$$

$$cfm_{da}\ddagger = \frac{\text{TLH}}{.68\,(W_{ea} - W_{la})} \tag{40}$$

$$cfm_{da}\ddagger = \frac{\text{GTH}}{4.45\,(h_{ea} - h_{la})} \tag{41}$$

$$cfm_{sa} = \frac{\text{RSH}}{1.08 \times (t_{rm} - t_{sa})} \tag{42}$$

$$cfm_{sa} = \frac{\text{RLH}}{.68 \times (W_{rm} - W_{sa})} \tag{43}$$

$$cfm_{sa} = \frac{\text{RTH}}{4.45 \times (h_{rm} - h_{sa})} \tag{44}$$

$$cfm_{ba} = cfm_{sa} - cfm_{da} \tag{45}$$

Note: cfm_{da} will be less than cfm_{sa} only when air is physically bypassed around the conditioning apparatus.

$$cfm_{sa} = cfm_{oa} + cfm_{ra} \tag{46}$$

H. DERIVATION OF AIR CONSTANTS

$$1.08 = .244 \times \frac{60}{13.5}$$

where .244 = specific heat of moist air at 70 F db and 50% rh, Btu/(deg F) (lb dry air)
60 = min/hr
13.5 = specific volume of moist air at 70 F db and 50% rh

$$.68 = \frac{60}{13.5} \times \frac{1076}{7000}$$

where 60 = min/hr
13.5 = specific volume of moist air at 70 F db and 50% rh
1076 = average heat removal required to condense one pound of water vapor from the room air
7000 = grains per pound

$$4.45 = \frac{60}{13.5}$$

where 60 = min/hr
13.5 = specific volume of moist air at 70 F db and 50% rh