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Research Article

Optimizing Commercial Building HVAC System through Accurate Load Calculation and Efficient Ducting Design

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Keywords

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Heat Load calculations,
CFM values.

Abstract

This paper introduces an innovative approach to create cost-effective supply air and return air ducting systems, aiming to optimize indoor conditions in commercial buildings. Following ISHRAE standards and employing CAD Plans, theoretical heat loads are compared with practical E20 Sheet results, ensuring comprehensive accuracy. These calculations drive cubic feet per minute (CFM) determinations, guiding ducting design and equipment selection. Our distinctive contribution lies in seamlessly integrating theoretical and practical data, fortifying system robustness. Furthermore, our commitment to sustainability bolsters energy efficiency. The study concludes by aligning the estimated 80.51 TR equipment capacity with the 70-80 TR range, affirming precision and cost-effectiveness. This work furnishes a pragmatic framework and underscores its relevance in contemporary energy-conscious installations.

1. Introduction

The proportion of energy consumption originating from both residential and commercial buildings has been progressively on the rise, reaching approximately 40% in developed nations. This has surpassed the energy usage of other major sectors such as industry and transportation. The combination of population expansion, heightened requirements for building amenities and comfort, along with the increased duration of time spent indoors, all contribute to the ongoing surge in energy demand. Notably, within the realm of building amenities, the expansion of energy consumption in HVAC (Heating, Ventilation, and Air Conditioning) systems holds particular significance. In fact, HVAC systems account for 50% of building energy consumption and 20% of overall energy consumption in the United States [1],[2]. Energy-efficient buildings offer energy, economic, and environmental benefits. They reduce energy expenditures and environmental pollutants. They also create economic opportunities for business and industry by promoting new energy efficient technologies. Owners of commercial buildings pass on energy costs to consumers or tenants, eliminating any incentive for energy-efficient design and construction. Homebuyers often are motivated more by up-front costs than operating costs [3]. The purpose of a building's HVAC design is both high indoor air quality and energy efficiency. These dual considerations require an integrated design approach. A building's heating, ventilation, and air conditioning system (HVAC) creates a climate that allows for maximum comfort by compensating for changing climatic conditions [4],[5]. The humidity level inside the building must be maintained below a critical level for the survivability of electronics devices because excessive humidity leads to many deleterious effects on moisture sensitive electronics devices[6],[7]. Efficiency improvements in HVAC systems can lead to substantial savings. HVAC systems need to be in a good balance with the buildings in general and they need to be of a proper size

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which fits with the actual heating, cooling, and ventilation needs[8]–[11]. Hence all our energy-efficiency solutions must consider HVAC system load calculations. This will help us to avoid installing an undersized or oversized HVAC system. In the realm of HVAC design for commercial structures, adopting a larger approach isn't synonymous with enhanced outcomes. To illustrate, the installation of an air conditioning system that surpasses the necessary size can lead to discomfort. This arises due to the system's frequent cycling on and off. Consequently, it operates insufficiently to eliminate surplus humidity, resulting in a damp environment containing pockets of varying temperatures. The process of determining the load for HVAC systems involves considerations beyond mere square footage. It's imperative to factor in additional elements that impact heating and cooling requirements. These variables encompass aspects such as the arrangement of lighting, utilization patterns within the space, exposure to natural light, and the composition of building materials used [12]–[19].

Some other literatures and articles on designing a proper HVAC system are:

Comparison of Cooling Load Calculations by E20 and HAP Software describes a complete air conditioning system to control ambient conditions such as temperature, relative humidity, air movement, etc. in an economical manner for Al Fahad Mosque located Unaizah in Qassim region K.S.A. Three methods are used for calculation of thermal load i) E20 Method ii) HAP software iii) by giving dimensions of building which, the sellers are estimating tonnage of refrigeration (market method or ALGhaith company method) [20]. It presents the difference in results of the three methods. But a particular method for calculation of thermal load has not been described in this paper.

Design And drafting of HVAC, Central Air Conditioning System for An Office Building calculates the cooling load of a central air conditioning system for an office building in Hyderabad, India [21].

Strategy Guideline: Accurate Heating and Cooling Load Calculations demonstrate the impact on the loads when common inaccurate adjustments (also known in the industry as “safety factors”) are made to the house. It shows that small manipulations such as changing the outdoor/indoor design conditions can result in exaggerated loads. Oversizing not only impacts the heating and cooling equipment costs, but duct sizes and numbers of runs must also be increased to account for the significantly increased system airflow [22].

The Heating and Cooling Load Calculation (Tertiary Non-ITS) computes individual room heating loads through the perimeter heat loss factor equation. Wall and ceiling partition loads are factored in solely when there's a heat loss. Heat gain isn't considered for spaces on the partition-exposed side. The summary outlines heating load components and Total Room Heat Load for each space. A 20% safety margin is included in room heat loads to accommodate uncertainties in this design phase [23].

Energy-saving practices, encompassing both direct technological interventions and behavioral adjustments, as well as the extension of product lifespans, have emerged as crucial strategies to mitigate rising energy demands and environmental impacts. Ensuring energy efficiency entails key elements like insulation, efficient HVAC systems, and optimized natural lighting. Moreover, contemporary building designs underscore the essential role of effective air conditioning in addressing ventilation and pollution concerns[24].

In cooling load calculation, TEDT/TA, HB, TFM, CLTD/SCL/CLF and RTS methods for calculating cooling loads are compared according to their using data, coefficients, and calculation procedure. The elementary school located in Istanbul is selected to be an example for comparison of numerical differences among these methods [25].

2. Methodology and Calculations

2.1. Cubic Feet per Minute (CFM)

A case study was done in a commercial building located at Hyderabad, India (latitude 17. 86°N). Table 1 presents the summary of material used for the analysis purpose. Transmission coefficient (ΔU) for the materials glass, masonry wall, roof, partition wall and the floor used in the civil building were assumed to be 0.2, 0.31, 0.3, 0.2, 0.33 respectively, mentioned in the ISHRAE standard. Additionally, the temperature changes (ΔT) across the dissimilar materials varies according to its geographical orientations (i.e., east, west, north, south). For the wall and glass, ΔT were found to be 38.5, 29, 21, 30 °F, and 13, 14.5, 17, 13 °F, respectively in the east, west, north, and south direction. Furthermore, a constant ‘ΔT’ of 48.5 °F was assumed across the roof. The details for the temperature difference are shown in Table 3.

Figure 1 presents the civil plan of room ‘Room-1’ taken for the demonstration purpose. The building thermal conditions are provided in Table 2, which provides the information on dry bulb temperature (DBT), wet bulb temperature (WBT), relative humidity and specific humidity. The daily range was 14^0F.

Note: Correction Factor is the correlation provided to the materials which are exposed to sun directly. Correction factor is considered taking the maximum of daily range and correction value.

Table 1. Materials used in the building.

Material	Description
Window	Ordinary glass, outside awning, medium color (3’ x 3’)
Door	Wood (7ft. x 3ft)
Masonry wall	Solid brick by falls and commands of the 12 3/18” plastic on wall sepsis’ board.
Partition wall	Concrete, sand, and granites aggregate of 6” with suspended plaster and 0.5" of insulation on top of the deck.
Roof	Concrete blocks made up of light weight aggregate of 0.5" plaster 8” thick
Floor	13/6” wooden block (or) Slab of sand aggregate of 8” thick and 0.5" light weight plastics with no furring

Table 2. Building thermal conditions

Condition	DBT($°F$)	WBT($°F$)	RH (%)	Sp. Humidity (Gr/Lb)	Daily Range($°F$)
Ambient/ Surrounding	106	78	28	98	14
Room Thermal Conditions	75	63	50	65	-19
Difference(Δ)	31	15		33	

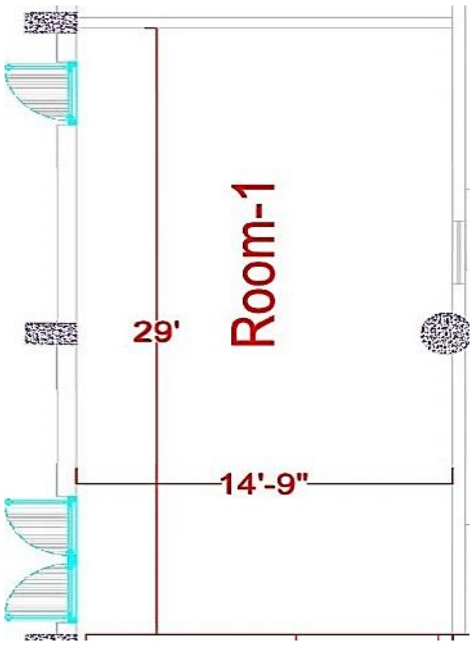

Figure 1. Civil plan for room **1**

Table 3. Equivalent temperature difference for the materials

Material	Equivalent temperature difference & solar heat gain (in case of glass only) (a)	Correction factor (b)	Total temperature difference (a + b)
Wall (123 lbs)	$E - 19.5$ °F $W - 10$ °F $N - 2$ °F $S - 11$ °F	19 °F	$E = 38.5$ °F $W = 29$ °F $N = 21$ °F $S = 30$ °F
Roof (70 $lbs/sq.ft$)	29.5 °F	19 °F	48.5 °F
Partitions (Walls between rooms)	(Non-AC. Temperatures - AC Temperatures) $101 - 75 = 26$ °F (Non-AC Temperature = Atmospheric Temperature - 5°F) = $106 - 5 =$ 101°F	No need to add correction factor because it is not exposed to the sun	26 °F
Glass	$E = 13\ Btu/hr.sq.ft^0F$ $W = 14.5\ Btu/hr.sq.ft^0F$ $N = 17\ Btu/hr.sq.ft^0F$ $S = 13\frac{Btu}{hr}.sq.ft^0F$	Not needed in case of solar heat gain	$E = 13\ Btu/hr.sq.ft^0F$ $W = 14.5\ Btu/hr.sq.ft^0F$ $N = 17\ Btu/hr.sq.ft^0F$ $S = 13\ Btu/hr.sq.ft^0F$

2.2. Building Information

On the first floor, R-01 - Cabin 1 (shown in Figure 1) has an area of 428 sq. ft. (2" x 14'9"). The roof height is 12 ft and has been obtained from the National Building Code (NBC)[26]. The false ceiling height is 9" and the room volume is calculated to be 38,525 cubic ft. (i.e., volume = A x H = 428 x 9). The lightning load for the application can be provided by the client or the electrical department or can be taken from the standard electrical data book [27]. In the context of hot climates, this study presents an innovative window design system that incorporates dynamic shading elements made from colored transparent polymers, a concept considered in our research paper, offering an effective solution for reducing cooling loads and enhancing HVAC efficiency in energy-efficient buildings [28]. Equipment load is categorized based on the wattage of the equipment or the wattage per area. The standard equipment load ranges from 0.5 W/sq. ft. to 0.9 W/sq. ft. depending on the application.

Internal sensible heat load

Q_{people} = 8 $*$ 245 btu/hr = 1960 btu/hr

$Q_{lights} =$ 428 ft^2 x 1.25 $w/s.ft$ =1679.90 btu/hr

Q_{equip} = 428 $ft^2 *$ 0.5 w/ft^2 = 671.96 btu/hr

Room total internal sensible heat load $= Q_{people} + Q_{light} + Q_{equip} = 4310.96\ btu/hr$

External sensible heat load

$Q_{wall\ west} = U_{wall} * A_{west\ wall} * \Delta T_{west\ wall} = 0.31 * 408 * 29 = 3649.94\ btu/hr$

$Q_{floor} = 0.33 * 428 * 26 = 3672.24\ btu/hr$

Room total external sensible heat loads = $Q_{wall} + Q_{floor} + Q_{Roof} + Q_{glass} = 7322.18\ btu/hr$

(To find CFM outside air)

$CFM_{people} = no\ of\ people * CFM\ per\ person = 8 * 20 = 160\ CFM$

$CFM_{(NACPH)} = volume * \left(\frac{NACPH}{60min}\right) = 3849 * \frac{8}{60} = 513.6\ CFM$

[NACPH: no of air changes per hour]

$Q_{air} = 513 * 1.08 * 31 * 0.3 = 5158.60\ btu/hr$

Room total external to internal sensible heat load $= Q_{infiltration} + Q_{air} = 5326 btu/hr$

Room total sensible heat loads $(RTSH) = Q_{internal} + Q_{external} + Q_{external\ to\ internal} = 16960\ btu/hr$

Effective room sensible heat $(ERSH) = RTSH + F.O.S\ \ [F.O.S = 10 - 15\%\ of\ RTSH]$

(F.O.S is considered on sensible and latent heat loads of the room to accommodate this loss caused due to unexpected occupancy over infiltration duct leakages and duct losses)

$$ERSH = 16960 + \left(\frac{12.5}{100} * 16960\right) = 19{,}080\ btu/hr$$

Room latent heat loads:

$$Q_{people} = no.of\ people * latent\ heat\ gain\ per\ person = 8 * 205 = 1640\ btu/hr$$

$Q_{infiltration} = CFM_{infiltration} * 0.68 * \Delta gr = 5 * 0.68 * 33 = 112.2\ btu/hr$

$Q_{outside} = 513.6 * 0.68 * 33 * 0.3 = 3457.56\ btu/hr$

Room total latent heat loads $= Q_{people} + Q_{infiltration} + Q_{outside} = 5209.76\ btu/hr$

Effective room latent heat loads (ERLH):

$$ERLH = RTLH + FOS(FOS = 2.5 - 5\%\ of\ RTLH) = 5209.76 + \left(\frac{2.5}{100}\right) * 5209.76 = 5340\ btu/hr$$

Effective room total heat loads $(ERTH) = ERSH + ERLH = 24{,}420\ btu/hr$

Total heat loads at cooling coil $= Q_{outside\ air\ contact\ air\ sensible} + Q_{outside\ air\ contact\ air\ latent}$

$$Q_{cooling\ coil\ contact} = 20{,}104.36\ btu/hr$$

Grand total heat load $(GTH) = ERTH + Q_{cooling\ coil\ contact} = 44{,}524.40\ btu/hr$

Grand total heat load $= GTH + (F.O.S\ (1 - 3\%) of\ G.T.H) = GTH + (0.03 * GTH)$

$$= 45{,}860.13\ btu/hr$$

Capacity in T. R of A. C machines $= GTH/12000 = 45{,}860.13/12000 = 3.82\ TR$

Effective sensible heat factor (E.S.H.F represents the rate of change of sensible heat)

$ESHF = ERSH/ERTH = 0.78$

Apparatus dew point temperature (ADP represents the temperature at which condensation of atmosphere air water vapor takes place. It is the temperature at which dehumidification starts.)

For room condition ADP = 50^0F.

Dehumidify rise (DR is the temperature of the contact air after passing over the coils. It is the dehumidified air which is enter)

$DR = (R.T - ADP)C.F = (RT - ADP)(1 - BF) = (75 - 50)(1 - 0.3) = 17.5\ °F$

Dehumidified CFM (It is the quantity of dehumidified air i.e., removal to condition the space.) Dehumidified CFM $= ERSH/\ (D.R * 1.08) = 19{,}080/\ (17.5 * 1.08) = 1009.53\ CFM$

3. Calculation by Using Excel-20 Sheet

The following E-20 sheet was used to compare the result obtained from hand calculation. The manually calculated value was 1009.53CFM and E-20 Sheet Calculated value was 1009.53CFM. Both the values are equal, so the remaining all calculations can be done by using E-20 Sheet.

Table 4. E20-Sheet for Room 1

LOCATION	Hyderabad								
CLIENT	XYZ								
CONSULTANT	ABC								
126.00									
Item	Area or Quantity			Sun Gain or Temp. Diff.			Factor (U)	Btu/Hour	Watts
		ROOM HEAT			Q= U*A*ΔT				
ROOM SENSIBLE HEAT									
Solar Gain - Glass	Area			ΔT			U		
Glass - N		SqFt	x			x		0.00	
Glass - NE		SqFt	x			x		0.00	
Glass - E		SqFt	x			x		0.00	
Glass - SE		SqFt	x			x		0.00	
Glass - S		SqFt	x			x		0.00	
Glass - SW		SqFt	x			x		0.00	
Glass - W		SqFt	x			x		0.00	
Glass - NW		SqFt	x			x		0.00	
Skylight		SqFt	x			x		0.00	
Solar & Transmission Gain - Walls & Roof									
Wall - N		SqFt	x		F	x		0.00	
Wall - NE		SqFt	x		F	x		0.00	
Wall - E		SqFt	x		F	x		0.00	
Wall - SE		SqFt	x		F	x		0.00	
Wall - S		SqFt	x		F	x		0.00	
Wall - SW		SqFt	x		F	x		0.00	
Wall - W	406.00	SqFt	x	29.00	F	x	0.31	3,649.94	
Wall - NW		SqFt	x		F	x		0.00	
Roof	428.00	SqFt	x	48.50	F	x		0.00	
Transmission Gain - Except Walls & Roof									
All Glass		SqFt	x	31.00	F	x	1.17	0.00	
Partition		SqFt	x	26.00	F	x	0.28	0.00	
Ceiling		SqFt	x		F	x		0.00	
Floor	428.00	SqFt	x	26.00	F	x	0.33	3,672.24	
INFILTRATION AND BY-PASSED AIR									
Infiltration	5.00	CFM	x	31.00	T. Diff	x	1.08	167.40	
Outside Air	513.60	CFM	x	31.00		x	1.08	5,158.60	
Internal Heat									
People	8.00	Nos.	x	245.00	Btu/Hour Per Person			1,960.00	
Lighting	428.00	SqFt	x	1.25	W/SqFt	x	3.14	1,679.90	
Equipment	428.00	SqFt	x	0.50	Watts	x	3.14	671.96	
Power		kW/Hp	x					0.00	
Sub Total								16,960.04	
Factor							5-15%	2,120.00	
Effective Room Sensible Heat								19,080.04	1.00
ROOM LATENT HEAT									
Infiltration	5.00	CFM	x	33.00	Gr/Lb	x	0.68	112.20	
Outside Air	513.60	CFM	x	33.00	Gr/Lb	x	BFx0.68	3,457.56	
People	8.00	Nos.	x	205.00	Btu/Hour Per Person			1,640.00	
Sub Total								5,209.76	
Factor							2.5 - 5%	130.24	
Effective Room Latent Heat								5,340.00	2.00
EFFECTIVE ROOM TOTAL HEAT								24,420.04	
		OUTSIDE AIR HEAT							
Sensible	513.60	CFM	x	31.00	F(TD)	x	CF x 1.08	12,036.73	3.00
Latent	513.60	CFM	x	33.00	Gr/Lb	x	CF x 0.68	8,067.63	4.00
OUTSIDE AIR TOTAL HEAT								20,104.36	
GRAND SUB-TOTAL HEAT								44,524.40	
Factor							1 - 3%	1,335.73	
GRAND TOTAL HEAT								45,860.13	
TONS=GRAND TOTAL HEAT/12000								3.82	

SPACE REFERENCE	Room-01					
AREA (SqFt) (WxH)	428.00					
Falce Ceiling Height (Ft)	9.00					
Volume (CuFt)	3,852.00					
Estimate for	Summer					
Design Conditions	DB (°F)		WB (°F)	RH (%)	SH (Gr/Lb)	
Ambient (Outside)	106.00		78.00	28.00	98.00	
Room (Indoor)	75.00		63.00	50.00	65.00	
Difference Δ	31.00		15.00	22.00	33.00	
Bypass Factor (BF)					=	0.30
Contact Factor (CF = 1 - BF)					=	0.70
CFM Ventilation						
CFM Per Person	25.00	No	=	8.00	=	200.00
CFM Per SqFt	0.25	Sqft	x	428.00	=	107.00
Air Change Per Hour (CFM)			=	8.00		
CFM Cu. ft	3,852.00	x	8.00	x1/60	=	513.60
CFM Infiltration						
Swinging		x		cfm/door	=	0.00
Revolving Doors (People)		x		cfm/door	=	0.00
Open Doors		x	1.00	cfm/door	=	0.00
Crack (feet)		x		cfm/ft	=	0.00
						0.00
Supply CFM from Machine						
Effective Room Sensible Heat Factor =						
Effective Room Sensible Heat/Eff Room Total Heat					=	0.78
Apparatus Dew Point (ADP)						
Indicated ADP (°F)					=	
Selected ADP (°F)					=	50.00
Dehumidified Rise						
(Room DB - ADP) x CF					=	17.50
DEHUMIDIFIED AIR QUANTITY						
Effective Room Sensible Heat			=	1,009.53	CFM	
Dehumidified Rise x 1.08						
			=	474.48	L/s	
TOTAL HEAT CAPACITY						
Grand Total Heat			=	3.82	TR	
SENSIBLE HEAT CAPACITY						
Grand Sensible Heat			=	3.82	TR	
12,000.00						
Notes:						

4. Results

The CFM value for each room, from room 1 to the hall was found separately by using the 20 steps which are mentioned above in the methodology section. And for the convenience of the data calculation, we have used the excel sheet known as E-20 sheet, to minimize the time for the calculation procedure. Excel-20 sheet was designed in such a way that we have added each process that we have followed for manual data calculations. The CFM and TR for individual rooms vary due to the varying area of each room which are indicated in the table below.

Table 5. Volume and load of each room

Room	Volume (ft^3)	CFM	TR
1	3852	1009.53	3.82
2	5832	1828.19	5.96
3	24417	5549.62	22.96
4	10998	2322.56	10.61
5	11565	3010.8	11. 78
6	1836	370.1	1.8
7	7263	2529.38	8.48
8	7920	1835.86	7.52
Reception	3600	980.28	3.62
Hall	4050	1032.51	3.96
Total		20468.83	80.51

4.1. Duct Design

For approximately 500 CFM, one diffuser is required and the data for the diffusers for each room is given in the table below. And the estimated size for the supply and return duct is also mentioned below. The size for the 500 CFM is 12” x12” for which the CAD plan was designed for supply and return air ducting system using same data.

Table 6. Type of diffuser for each room

Room	CFM	Number of diffusers
1	1009.53	$2 * 500$
2	1828.19	$2 * 1000$
3	5549.62	$6 * 900$
4	2322.56	$4 * 500$
5	3010.8	$6 * 500$
6	370.1	$1 * 500$
7	2529.38	$5 * 500$
8	1835.86	$2 * 1000$
Reception	980.28	$2 * 500$
Hall	1032.51	$2 * 500$

Table 7. Diffuser selection based on CFM.

CFM	Supply and Reuse air diffuser
$0 - 150$	$6" * 6"$
$150 - 300$	$9" * 9"$
$300 - 500$	$12" * 12"$
$500 - 700$	$15" * 15"$
$700 - 1000$	$18" * 18"$
$1000 - 1400$	$24" * 24"$

5. Conclusion

From the above calculations, the estimated value is 20,468.83 CFM air supply, and an 80.51 TR capacity machine is required. For this LPCP 50 Series, according to Trane Air Conditioners, an air handler is used to maintain proper air conditioning. It is suitable for a 20,000-25,000 CFM flow rate and a 70-80 TR capacity. In this work, the calculated CFM values of each room on each floor, using the excel-20 sheets, and TR values of every room, the total capacity of TR was estimated. The capacity of the unit required is approximately 80.51 TR, but a range of 70-80 TR was used to avoid fluctuations in the working. All parameters were taken into consideration for high accuracy and proper estimation of a suitable machine. Also, based on the obtained CFM for each room and for all floors, duct design was done using AutoCAD. All diagrams were shown on the civil plan. From this, we can conclude that our estimated values are enough to establish the air conditioning system in the specified location. By using the HVAC system, energy consumption of the building is reduced as much as possible by avoiding unnecessary losses. This is one of the most well-designed and useful methods in present-day installations.

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