Proceedings of 8th IOE Graduate Conference
Peer Reviewed
ISSN: 2350-8914 (Online), 2350-8906 (Print)
Year: 2020 Month: June Volume: 8

Design, Simulation and Analysis of an Air-Conditioning System: A Case Study of the Proposed Aerospace Building of Pulchwok Engineering Campus in the Context of Nepal

Sudin Bhuju Shrestha [a], Hari Bahadur Dura [b], Nawraj Bhattarai [c]

[a, b, c] *Department of Mechanical and Aerospace Engineering, Pulchowk Campus, IOE, TU, Nepal*
Corresponding Email:
[a]074msmde019@ioe.edu.np, [b]duraharis@pcampus.edu.np, [c]bnawraj@ioe.edu.np

Abstract

Optimization of energy consumption has been on the lime light from the beginning of this 21st century, with more than half of the energy being spent on controlling indoor atmospheric condition. Taking into account that the field of Heating, Ventilation and Air-conditioning (HVAC) is in its preliminary developing stage in Nepal, researches associated to HVAC system optimization is only starting to germinate with detailed HVAC optimization studies being implied only in major HVAC projects. Commercially available software are available for that however, they only do so much in predicting the load not so much on how that load will affect the temperature distribution with respect to time and space. At such Computational fluid dynamics (CFD) can be a very effective tool in simulating the steady and unsteady temperature distributions. The present work focuses on design, simulation and analysis of an air conditioning system best suited for the auditorium hall of the proposed Aerospace building of the Pulchwok Engineering Campus. Considering the fact, that the climate of Kathmandu is of subtropical type, the cooling loads were calculated for summer months from March to October via various methods implying Cooling Load Temperature Differences(CLTD), Hourly Analysis Program (HAP) and Autodesk RevitMEP. Consequently the loads were estimated to be 9.87 Tons, 8.52 Tons and 9.487 Tons respectively. Simulations were then performed in Fluent ANSYS for boundary conditions meeting the requirements of the weather data for 8, 9, 10 and 11 Tons of air-conditioning system setups which led to the conclusion that for desired design temperature of 21.5ºC, 10 Tons had the best performance in terms of average of kWh used per hour for comfortable condition as well as the total energy consumed.

Keywords

CFD, Fluent, RevitMEP, Radiant Time Series, Houly Analysis Program, Conduction Transfer Function, Cooling Load Temperature Difference, Air Conditioning

1. Introduction

With the exponential growth of the world's economy, the problem of insufficient power supply has taken place in many countries in recent years, especially during the peak period where building energy consumption accounted for 40% of the total energy consumption in the world while the air-conditioning systems in buildings consume about 60% to 70% of total electricity consumption in some countries and contribute over 30% of the CO_2 emissions [1, 2]. This increasing demand for heating, cooling and electricity supplies in buildings stimulates the search for higher-more efficient and low-emission energy production, conservation and usage methods [3].

Located at 27°4′N 85°21′E and an elevation of 1338m, Kathmandu valley is located in the warm temperate zone (1,200m-2,300m). The average summer temperature varies from 28°C to 30°C (82°F to 86°F) while the average winter temperature is 10.1°C (50.2°F) in the valley [4, 5]. Similarly, while the highest temperature in summer reaches to about 32.5°C, lowest temperature reaches to about -2°C [6]. Because of this type of climatic conditions Kathmandu valley in general faces about 8 months of summer (March to October) and 4 months of winter (November to February). Moreover, in this type of climatic conditions the cooling load during the summer days are much higher than the heating load during the winter. As a result, the HVAC systems installed in the valley are designed prioritizing the summer cooling load conditions. In Kathmandu valley alone, energy demand of residential sector was found to be about 7,500 TJ in 2013 with increase at the rate of 4% per annum. About 4% of total energy consumed in an urban house in the country is for purpose of space heating and cooling [7]. One of the difficult aspects of estimating the maximum cooling load for a space is determining the time at which this maximum load will occur. The walls and roof that make up a building's envelope have the capacity to store heat energy which causes the time lag for the heat transfer from outdoors to the space. As a result individual components that make up the space cooling load often peak at different times of the day, or even different months of the year. [8]. HVAC is the total study of the systems which regulates, controls and maintain the required atmospheric condition of an indoor or vehicular environment irrespective of external conditions [9]. Application of CFD in the study area of HVAC is not new. Simulations by EXACT3 CFD code were carried out to analyze the performance of split-type air conditioners with respect to the temperature rise of condensing units installed at building re-entrant and it was reported that CFD technique is capable of providing accurate results associated to the stack effect on the performance of the AC units serving at different floors of the building [10]. Analysis of the feasibility and energy saving property of clean air conditioning technology in clean operating rooms in a hospital where air flow distribution was simulated using the CFD software Airpak 3.0 and Fluent and it was found that an increase of the air supply area and return air inlets can increase the area of unidirectional flow regions of the main flow regions and avoid indoor vortexes and turbulivity in the operating area as well as the application of a secondary air return system in summers can reduce energy consumption [11].

Despite the effectiveness of CFD in the field of HVAC, CFD is still very new to the Nepalese science society. It all boils down to the fact that the immense amount of computational resources required for CFD calculations which are not readily accessible to all here. Consequently, the application of CFD in HVAC is only limitedly applied on large commercial HVAC project in which access to computational resources are possible.

1.1 Governing equations

The governing equations for three-dimensional, steady, turbulent and incompressible flow with heat transfer are given by the known continuity equation, the energy equation, N–S equation (momentum equation).

Continuity equation:-

$$\frac{\partial p}{\partial t} + \nabla \cdot (\rho u) = 0 \tag{1.1}$$

Momentum equation (N-S equation):-

$$\frac{\partial u}{\partial t} + u.\nabla u = V\nabla^2 u - \frac{1}{\rho}\nabla p + \rho g \tag{1.2}$$

Energy equation:-

$$\frac{\partial}{\partial t}\left(\rho\left(e+\frac{V^2}{2}\right)+\nabla\cdot\left(\rho\left(\mathrm{e}+\frac{V^2}{2}\right)\cdot V\right)\right.$$
$$+\nabla\cdot(p\vee) - \text{ viscous force } + \nabla\cdot(\dot{q}) = 0 \tag{1.3}$$

where,

u,v, w =Velocity of fluid in X, Y and Z directions

ρ, μ, ∇, p =Density, Viscosity, Divergence and Pressure

1.2 Heat gain and cooling load calculation methods

1.2.1 Cooling Load Temperature Difference (CLTD)

The CLTD method accounts for the thermal response (lag) in the heat transfer through the wall or roof, as well as the response (lag) due to the radiation of part of the energy from the interior surface of the wall to the object within the space which varies with heat gain with time, the massiveness of the structure, and the geographical location [12]. The total classification of 41 walls and 42 roofs in CTF method was simplified and a practical usable version of the CLTD tables was described(10 roofs and 16 walls) in GRP 158 [13] which is based on work done by [14]. The CLTD values corresponded to the heat gain caused by outdoor air temperature and solar radiation under a set of standard conditions, which included a latitude of 40°N, date of July 21, maximum outdoor temperature of 95°F, daily temperature range of 21°F, and an inside design temperature of 78°F and to calculate for other locations latitude and month correction factor were provided to use. Later on however separate tables for latitude 24°N, 36°N and 48°N were devised avoiding the need for latitude and month correction factor. The equations involved in calculation of load using CLTD are as follows.

a) External walls and roofs:-

$$\dot{q}_{\theta 1} = UA(CLTD)_{\mathrm{corr}\ \theta} \tag{1.4}$$

$$(CLTD)_{\text{corr }\theta} = [(CLTD*K) + (78-t_i) + (t_{om}-85)]*f \quad (1.5)$$

$$t_{om} = t_{o-}\frac{DR}{2} \quad (1.6)$$

b) Internal partition walls and floor:-

$$\dot{q}_{\theta 2} = UA\,(t_o - t_r) \quad (1.7)$$

c) Solar gain through glass:-

$$\dot{q}_{\theta 2} = A\,(SC)\,(SCL)_{\boldsymbol{\theta}} \quad (1.8)$$

d) Conductive heat gain through glass:-

$$\dot{q}_{\theta 3} = UA\,(t_o - t_r) \quad (1.9)$$

e) People:-

$$\dot{q}_{\theta_{\mathbf{s4}}} = N(sensibleheatgain)(CLF) \quad (1.10)$$

$$\dot{q}_{\theta_{\mathbf{L4}}} = N(\text{ Latent heat gain}) \quad (1.11)$$

$$\dot{q}_{\theta 4} = \dot{q}_{\theta s4} + \dot{q}_{\theta 14} \quad (1.12)$$

f) Lights:-

$$\dot{q}_{\theta 5} = N(BF)(W)(CLF) \quad (1.13)$$

g) Equipments:-

$$\dot{q}_{\theta \mathbf{6}} = N(UF)(W)(CLF) \quad (1.14)$$

h) Ventilation and infiltration air load estimation:-

$$\dot{q}_{\theta_{s7}} = \dot{m}(cp)\,(t_o - t_r) \quad (1.15)$$

$$\dot{q}_{\theta 17} = \dot{\mathrm{m}}\,(h_g)\,(W_{\mathbf{1}} - W_{\mathbf{2}}) \quad (1.16)$$

$$\dot{q}_{\theta \mathbf{7}} = \dot{q}_{\theta \mathbf{s}7} + \dot{q}_{\theta \mathbf{17}} \quad (1.17)$$

$$\dot{V} = A_{leak}\sqrt{a_s\,(t_o - t_r) + a_w v^2} \quad (1.18)$$

where,

A = Area, ft^2 or m^2

$\mathrm{CLTD}_{\text{cort }\theta}$ = Corrected CLTD which gives the temperature difference equating to the cooling load at temp θ, ^{0}F or ^{0}C

CLTD = Tabular CLTD, ^{o}F or ^{o}C

$\mathrm{t_i}$ and $\mathrm{t_{om}}$ = Actual inside and mean outside design dry bulb temperature, ^{o}F or ^{o}C

DR = Daily range, ^{o}F or ^{o}C

K = Colour adjustment factor, 1 if dark coloured or light in an industrial area,0.83 if permanently medium coloured (rural area), 0.65 if permanently light-coloured (rural area) (ASHRAE, 1980)

U= Overall heat transfer coefficient, Btu/$\left(\mathrm{hrft^{2o}F}\right)$ or W/$\left(m^{2o}\mathrm{C}\right)$

$\mathrm{t_o}$ = Temperature in adjacent space or exterior environment, °F or °C

$\mathrm{t_c}$ = Inside design temperature (constant) in conditioned space, °F or °C

SC = Shading coefficient (internal shade)

SCL_{θ} = Solar cooling load factor, Btu/(hr-ft^2°F) or W/$\left(\mathrm{m^2}°\mathrm{C}\right)$

$\dot{\mathrm{q}}_{\theta s4}$ and $\dot{\mathrm{q}}_{\theta 14}$ = Sensible and latent heat load from people

CLF = cooling load factor

N = number of respective element

BF = Ballast factor, 1.0 for incandescent bulb and 1.2 for fluorescent light

W = Watts input from electrical plans or lighting fixture data, Btu/hr

UF = Usage factor

$\dot{\mathrm{q}}_{\theta s7}$ and $\dot{\mathrm{q}}_{\theta 17}$ = Sensible and latent heat load from infiltration

$\dot{\mathrm{m}}$ = Mass flow rate of ventilation/infiltration air, kg/sec or 1b/hr

$\mathrm{C_p}$ = Specific heat of air at constant pressure, J/kg or Btu/lb

$\mathrm{h_g}$ = latent heat of vaporization, J/kg or Btu/lb

$\mathrm{W_1 - W_2}$ = Difference of specific humidity, g/kg or oz/lb

Aleak = Effective leakage (ELA), cm^2 or in^2

$\mathrm{a_s}$ = Stack coefficient [15], $\left[(\mathrm{L/s})^2/\left(\mathrm{cm}^4\cdot\mathrm{K}\right)\right]$ or $\left[\left(\mathrm{ft}^3/\min\right)^2/\mathrm{in}^4\cdot{}^o\mathrm{F}\right)\right]$

a_{W} = Wind coefficient [15], $\left[(\mathrm{L/s})^2/\left(\mathrm{cm}^4\cdot(\mathrm{m/s})^2\right)\right]$ or $\left[\left(\mathrm{ft}^3/\min\right)^2/\mathrm{in}^4\cdot(\mathrm{mph})^2\right)\right]$

v = Wind speed, m/s or mph

1.2.2 Conduction Transfer Function Model

A method for grouping walls and roofs with similar transient heat transfer characteristics i.e. on the basis of their thermal response characteristics particularly time lag in order to obtain a compact set of conduction transfer function (CTF) coefficients(42 roofs and 21 walls) was devised [16]. The CTF coefficients were then used in the CTF equation to calculate the representative heat gain or loss for any wall or roof in that particular group. The 1-D conductive heat gain (or loss) $\dot{\mathrm{Q}}_{\mathrm{cond,t}}$ at time t hour through the roof and walls is calculated according to the conduction transfer function (CTF) model [16] which is show in the eqn 1.19. The CTF method is used by HAP (Hourly analysis program) which one of the method used in this research to calculate the load.

$$\dot{\mathbf{Q}}_{\text{cond},t} = -\sum_{n\geq 1} d_n \dot{\mathbf{Q}}_{\text{cond},t-n\Delta t} + A\left(\sum_{n\geq 0} b_n T_{os,t-n\Delta t} - T_i \sum_{n\geq 0} C_n\right) \quad (1.19)$$

where,

T_o = Outside air temperature

α = Absorptance of surface

I_t = Total radiation incident on surface

h_o = Outside convective and radiative heat transfer coefficient

ε = Emmitance of the surface

F = Difference between the long-wave length radiation incident on the surface and the radiation emitted by a black body at outdoor air temperature

A = Area of roof or wall, m^2 or ft^2

$\triangle t$ = time step of 1 hour

$T_{os,t}$ = Sol–air temperature of outside surface at time t

b_n, c_n, d_n = CTF coefficients [16]

1.2.3 Radiant Time-Series Method

The RTSM method makes simplifications such as there is no internal or external heat balance rather it is assumed all the surface are effectively at the zone air temperature and thus facilities the use of single convection coefficients, radiation coefficients as well as fixed surface conductance independent of surface temperature, sky temperature etc. [15]. The storage and release of energy by the surfaces are approximated with predetermined zone response values. The RTSM method if heat load calculation is used by Autodesk RevitMEP software which is another method used to calculate the heat load in this research. The basic equations involved in the RTS methods are given below.

$$q''_{\text{convection,ext},j,\theta} = h_c\left(t_e - t_{os,j,\theta}\right) \tag{1.20}$$

$$t_e = t_o + {}^{\alpha G_t}/_{h_o} - {}^{\varepsilon \delta R}/_{h_o} \tag{1.21}$$

$$q''_{\text{conduction},in,j,\theta} = \sum_{n=0}^{23} Y_{pn}\left(t_{e,j,\theta-\mathbf{n}\delta} - t_{rc}\right) \tag{1.22}$$

where,

h_o = Combined exterior convection and radiation coefficients, Btu/hrft^2F or W/m^2K

δR = Difference between thermal radiation incident on the surface from the sky and surroundings and the radiation emitted by a blackbody at outdoor air temperature, Btu/hrft^2 or W/m^2

Y_{pn} = nth response factor, Btu/hrft^2F or W/m^2K

$t_{e,j,\theta-n\delta}$ = Sol-air temperature, n hours ago, F or C

t_{rc} = Presumed constant room air temperature, F or C

2. Methods and Methodology

2.1 Description of the Auditorium Hall

Table 1: Information related to the Auditorium Hall

S.N.	Element	Value
1.	Floor area	3585.0ft^2
2.	Floor size	68.1ft × 49.2ft
3.	Roof area	3596.4ft^2
4.	Roof exposure	NW
5.	Roof slope	5 degrees
6.	Average ceiling height	25.3ft
7.	Location of Partition	SW side
8.	Location of hall	NE side of the building
9.	Longer Side	NE, SW
10.	Shorter side	NW, SE
11.	Floor location of the hall	2nd floor
12.	Total number of floors occupied	2nd and 3rd
13.	Probable location of window placement	NW and NE side.
14.	Door width	5ft

Table 2: Monthly temperature pattern for Kathmandu Valley (°F) (ASHRAE, 1993)

Month	Max DBT	Min DBT	Max WBT	Min WBT
Jan	77.2	52.2	70.6	51.7
Feb	79.2	54.2	71.6	53.7
Mar	82.4	57.4	74.8	56.9
Apr	83.6	58.6	75.0	58.1
May	86.0	61.0	76.0	60.5
Jun	88.0	63.0	78.0	62.5
Jul	89.0	64.0	78.0	63.5
Aug	89.0	64.0	78.0	63.5
Sep	87.0	62.0	77.0	61.5
Oct	84.8	59.8	75.8	59.3
Nov	80.6	55.6	73.8	55.1
Dec	78.2	53.2	71.8	52.7

2.2 Considerations Made

2.2.1 Material Assumptions

Table 3: Load element assumptions

Element	R-value (hrft^{2o}F/Btu)
Wall	5.621 (Type 12)
Roof	14.48 (Type 4)
Floor	2.129
Door	1.41206
Window	1.03891 (SC 0.6 Zone type C)
Number of occupancy	100 (Per person 350 Btu/hr)
Lighting	0.9 W/ft^2
Infiltration	91.9301 CFM (Calculated)
Electrical appliance	20% of 16A/220V supply lost as heat

2.2.2 Inlet and Return Vent Parameters

Table 4: Return-vent parameters

S.N.	Tons	A/C arrangement		CFM	Return vent size (in^2)
		1 Ton	2 ton		
1.	8		4	2966.4	1053.748
3.	9	1	4	3407.8	1209.577
5.	10		5	3708	1315.557
7.	11	1	5	4308.2	1527.448

Table 5: Inlet-vent parameters

Capacity	1.083 Tons	2.041 Tons
Area(mm)	900*100	1000*100
CFM	441.4	741.6
Velocity (m^2/s)	2.314	3.499
Temp. of air exiting (^{o}C)	16.83	15.03
Watts	1095	2010

The sizing of the return vent in based on (Engineering ToolBox, 2010). While the inlet vent parameters are based on the split air-conditioner models found in the market manufactured by LG (VM242H6 and VM122H6).

2.2.3 Boundary Conditions

The wall boundary conditions were set up based on the type of material and the weather conditions. Radiation boundary condition was chosen assuming that the outer surface of the wall is in equal temperature as the ambient. For representing the heat generation through occupants inside the auditorium, heat generation markers of diameter 30cm were placed equidistant to each other (in total of 52 markers) and heat flux of 100 occupants are assumed to be produced from them. Velocity inlets were setup for each vents as mentioned in the table 5. Similarly for the return vent (outlet-vent) boundary condition was chosen in ANSYS with non-existent backflow pressure and backflow temperature of the ambient space.

2.3 Geometry, Mesh and Mesh Independence

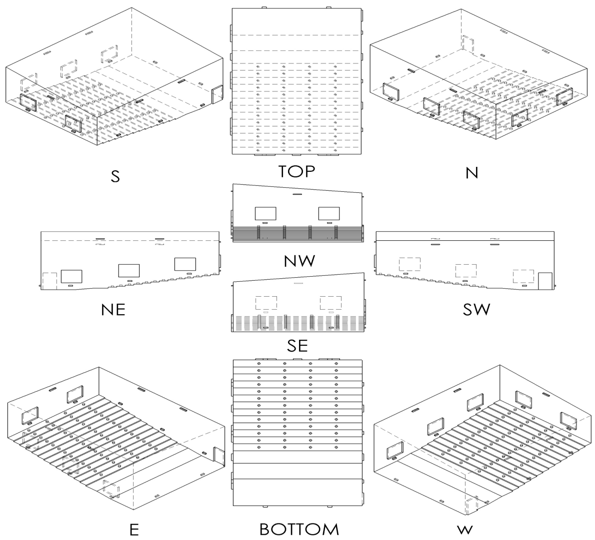

Figure 1: Geometry developed in SolidWorks

The geometry of the auditorium hall was developed using AutoCAD and SolidWork. The podium for the presenter is assumed to be on the SE side and the attendees are assumed to sit on the NW side facing south east side. It is assumed that there is a total of 13 tires of 1m length and 0.15m height steps and a 1.8m length disable friendly tire at the front as well where the attendees will remain sit. the windows are only permitted on the North-East and on the North-West walls. On the North-West walls there are 2 windows, while on the North-East side there are 3 windows all equidistant to each other of size 2.45m×1.82m. The door is located at the South-West side sized (1.6m×2.44m). The total volume of the model was accounted to be 2180.41m^3.

The mesh for the simulation was made using ANSYS Fluent meshing tool. Meshes ranging from cell no. 71,170 to 1,289,265 were used and while comparing the average volume temperature of the space after 40mins of run time for 10 ton A/C configuration following results were found shown in the figure 3 below. Since the deviation of the result (i.e. 0.32ºC) between the largest and smallest mesh used was very small, the mesh having 152,573 no. of cells was chosen for further processing.

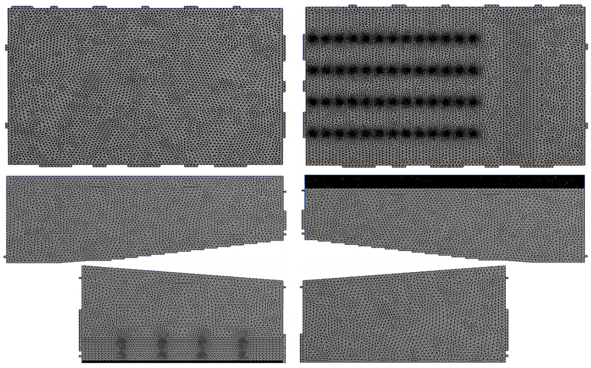

Figure 2: Mesh Developed

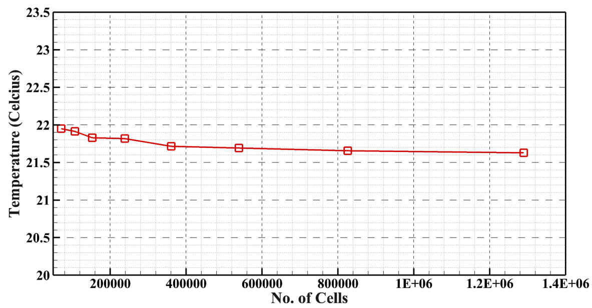

Figure 3: Mesh Independence Test

3. Results and Discussions

The calculation of internal loads were calculated initially to be constant for all methods which is shown in table 6. The external loads however calculated through different methods are explained below.

Table 6: Load through internal sources

Elements	Loads (Btu/hr)
Infiltration load	1816.93
Overhead lighting load	11037.35
Internal appliance load	2402.34
Partition load	1343.83
Occupancy load	35000

3.1 Cooling Load Temperature Difference

By using the CLTD method the maximum load for the July and August month was calcualed to be 9.87 Tons at 5pm in the evening. The breakdown of individual external loads as well as montlhy load calculated are shown below.

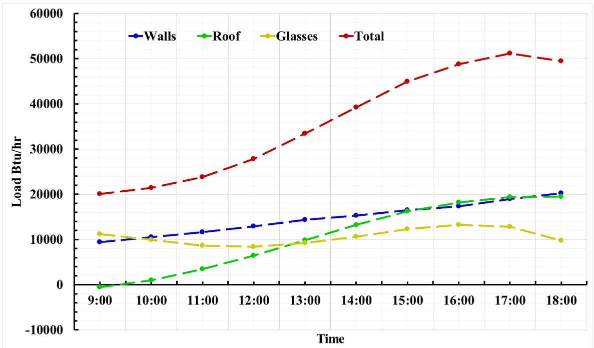

Figure 4: Hourly heat transfer for July (CLTD)

Table 7: External load by CLTD

Element	Load (Btu/hr)
Wall	18938.40
Roof	19372.87
Floor	15155.00
Door	509.89
Window	12868.27
Partition Load	1343.83

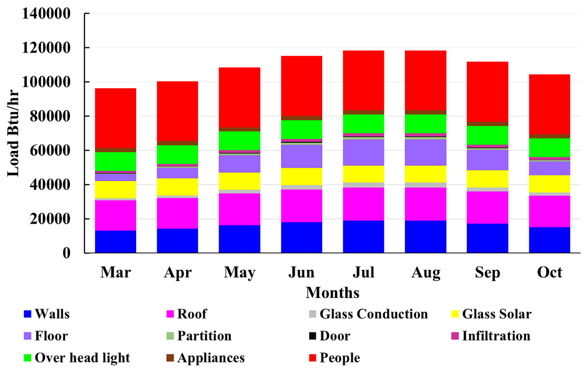

Figure 5: Monthly load pattern (CLTD)

3.2 Hourly Analysis Program

Unlike CLTD which uses only one temperature value, HAP takes into consideration of the hourly varying temperature through the day. By doing so the maximum load was calculated to be 8.52 Tons at 5pm in the month of July/August. To make equivalent comparisons to CLTD, assumptions were made. **Assump 1:** the peak design temp. is reached at the peak load time (5pm) and **Assump 2:** the peak design temp. occurs throughout the day. The maximum load calculated by HAP using Assump:1 and Assump:2 were 8.7 Tons and 9.1 Tons respectively.

Table 8: External load by HAP(CTF) Btu/hr

Element	Hourly temp.	Assum.1	Assum.2
Wall	17277	17325	20043
Roof	13226	13228	14081
Window	10294	10691	10910
Door	350	399	426
Floor	10004	11580	12450
Partition	887	1027	1104

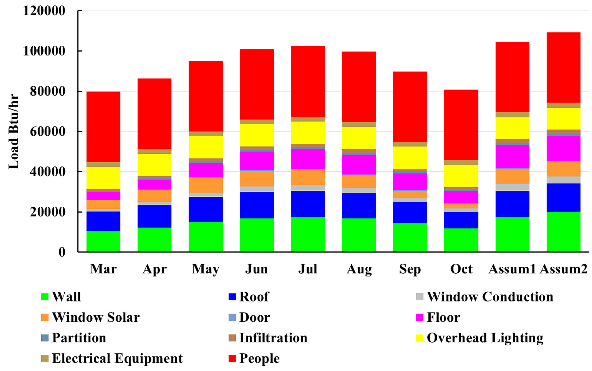

Figure 6: Monthly load pattern (HAP)

3.3 RevitMEP

The total load was calculated to be about 9.487 Tons at 5pm of July respectively. It is to be noted that RevitMEP is only able calculates the design maximum cooling load and hourly or month wise load generation is not possible.

Table 9: External load by RevitMEP(RTSM)

Elements	Loads (Btu/hr)
Walls	18109.40
Windows	13382.10
Door	41.50
Roof	16907.31
Floor	15155
Total	63595.31

3.4 CFD Results

The simulations were performed for various A/C Ton configurations for March to October. For assisting the simulation as well as the analysis, it was assumed that the human comfort conditions to begin below 23.5°C and final desired inner targeted temperature conditions to be multiple one; being 22.5°C, 22°C and 21.5°C.

Considering the response time, in all cases of desired inner temperature, it was found that the performance ranged from the highest A/C configuration i.e. 11 Tons to cool the quickest while the 8 Tons to be the slowest, which was obvious. The 8 Ton A/C configuration was not able to cool down the temperature of the space to 21.5°C. The 8 Ton A/C configuration suffered significant performance loss in the month of July and August being able to drop the temperature to human comfort range only after the 30-40 minute mark, yet never able to attain the 21.5°C. However, in other months overall it is able to drop temperature to human comfort range within 15-25 minutes.

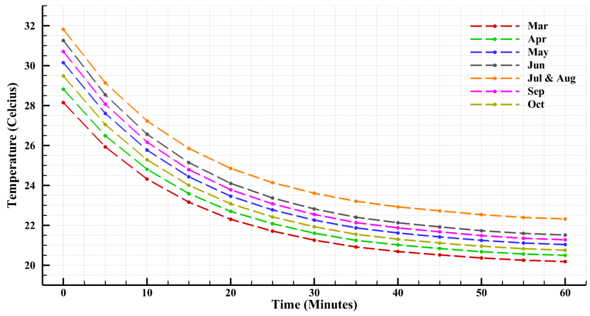

Figure 7: 8 Ton A/C performance

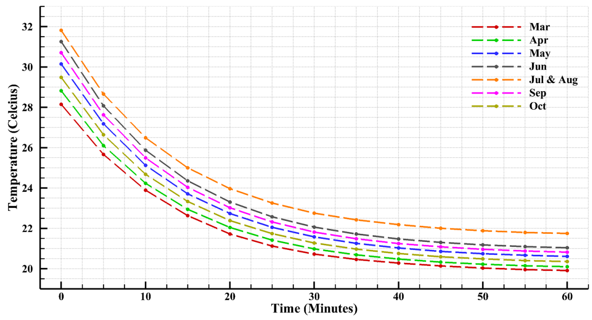

Figure 8: 9 Ton A/C performance

On the contrary, the 11 ton A/C configuration seemed to perform very well in the month of July and August. It was able to drop the temperature to human comfort levels in 15-20 minutes.

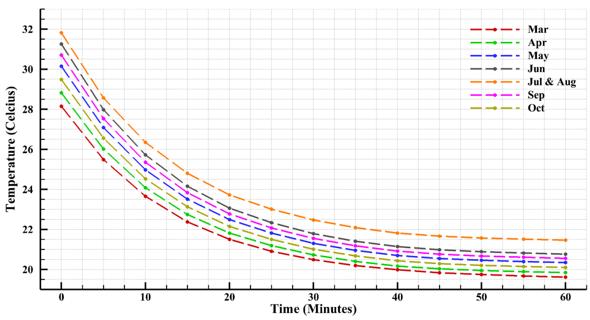

Figure 9: 10 Ton A/C performance

Similarly, simulations were also perform to evaluate how fast did the temperature increased from the

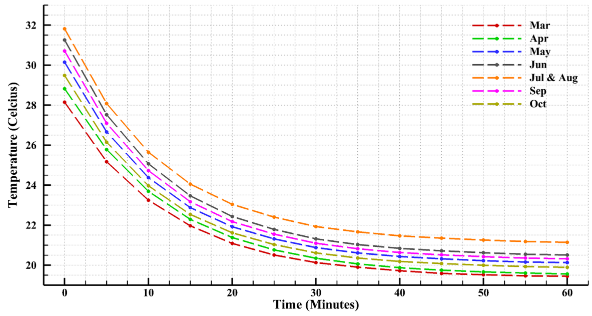

Figure 10: 11 Ton A/C performance

desired temperature to the temperatures beyond human comfort conditions for different months. For this the same geometry, and boundary configurations were used but the boundary conditions of A/C inlet vents were turned off.

Table 10: Time in minutes to reach desired temperature in July/August

Temperature	22.5°C	22°C	21.5°C
8 Tons	51.52	74.00	Cannot reach
9 Tons	35.75	45.89	76.68
10 Tons	31.24	37.98	56.41
11 Tons	24.74	28.43	39.82

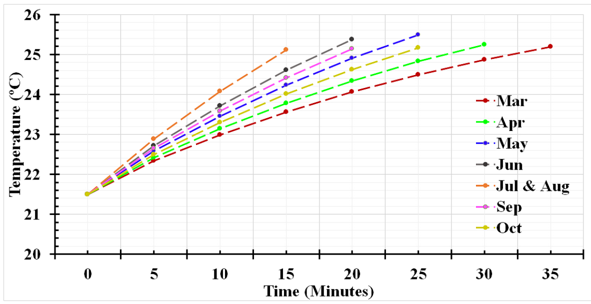

Figure 11: Temperature increase curve

Assuming that the A/C is to be turned on 8 hours a day from March to October and the A/C cut-off thermostat temperature (temperature at which A/C turns back on once the temperature starts to increase again) to be 23.5°C, by using the temperature drop curves of each A/C configurations and the temperature increase curves of each month it was possible to calculate the total time the A/C would be turned on each day and month, consequently to calculate the total energy consumed for various cases of desired inner targeted temperature (22.5°C, 22°C and 21.5°C).

If we consider average kWh used per hour of thermal comfort condition as well as the total energy consumed to be the performance criteria; it was found that in the case of 22.5ºC (desired inside temperature), the performance was in the order of 8, 10, 11 and 9 Tons. In the case of 22ºC (desired inside temperature), the performance was in the order of 8, 11, 10 and 9 Tons. While in the case of 21.ºC (desired inside temperature), the performance was in the order of 10, 11, 9 and 8 Tons.

Table 11: Total kWh used for cooling

Temperature	22.5°C	22°C	21.5°C
8 Tons	9425.33	10060.98	N/A
9 Tons	10160.55	10725.46	11902.49
10 Tons	89726.63	10445.57	11554.31
11 Tons	9823.53	10422.57	11836.80

Table 12: Average kWh used per hour of comfort condition obtained

Temperature	22.5°C	22°C	21.5°C
8 Tons	5.17	5.51	N/A
9 Tons	5.50	5.81	6.45
10 Tons	5.26	5.64	6.24
11 Tons	5.28	5.60	6.36

4. Concluisons

The entire process of design, simulation and analysis of an air-conditioning system was successfully carried out for the proposed aerospace building of Pulchwok Engineering Campus in this research. With respect to the modern construction standards, weather data as well as proposed design plans, necessary considerations were made for the materials, occupancies, possible infiltration, etc. The total design maximum cooling was calculated to be about 9.87, 8.52 and 9.487 Tons using ASHRAE recognized CLTD, TFM(HAP) and RTSM(RevitMEP) methods. By analyzing the sources of cooling load, it was found that even though the area roof is about 75% of the area of the total wall surfaces the amount of heat transfer in both cases is almost similar. Thus, one of the minor conclusions obtained was that it is effective to insulate the roof than to insulate the walls. The simulations performed in ANSYS Fluent for 8, 9, 10 and 11 Tons revealed the primary conclusions of this research. Considering the time required to attain the desired design temperature to be the criteria that defines the performance, in all cases the 11 Tons A/C configuration tends to be superior followed by 10, 9 and 8 Tons. Nevertheless, considering average kWh used per hour for comfortable condition as well as the total energy consumed for cooling to be the criteria that defines performance and choosing the desired design temperature to be 22.5ºC the performance was in the order of 8, 10, 11 and 9 Tons. On the contrary, choosing the desired design temperature to be 22ºC the performance was in the order of 8, 11, 10 and 9 Tons. However, if the desired design temperature was chosen to be 21.5ºC, the performance was in the order of 10, 11 and 9 Tons respectively (since 8 Tons is disqualified as it cannot achieve that particular temperature).

References

[1] Chen-Yi Zhou, Xiao Zhong, Song Liu, Shuai Han, and Peng Liu. Study on the relationship between thermal comfort and air-conditioning energy consumption in different cities. *Journal of Computers*, 28(2):135–143, 2017.

[2] Liu Yang, Haiyan Yan, and Joseph C Lam. Thermal comfort and building energy consumption implications–a review. *Applied energy*, 115:164–173, 2014.

[3] J Deng, RZ Wang, and GY Han. A review of thermally activated cooling technologies for combined cooling, heating and power systems. *Progress in Energy and Combustion Science*, 37(2):172–203, 2011.

[4] Department of of Hydrology and Meteorology. Normals from 1981-2010. 2012.

[5] World Metrological Organizaton. World weather information service-kathmandu. 2013.

[6] ISHRAE. Nepal weather data 2019. 2019.

[7] Utsav Shree Rajbhandari and Amrit Man Nakarmi. Energy consumption and scenario analysis of residential sector using optimization model–a case of kathmandu valley. pages 476–483, 2014.

[8] ILE Schweizerische Kontaktstelle für Angepasste Technik am ILE Gut Paul, Ackerknecht Dieter. *Climate Responsive Building*. SKAT, St. Gallen, Switzerland, 1993.

[9] RS Kurmi and JK Gupta. *A Textbook of Refrigeration and Air Conditioning. Fist Multicolour Revised & Updated Edition*. Eurasia Publishing House (P) Ltd. Ram Nagar, New Delhi–110055, 2006.

[10] TT Chow and Z Lin. Prediction of on-coil temperature of condensers installed at tall building re-entrant. *Applied Thermal Engineering*, 19(2):117–132, 1999.

[11] XR Ding, YY Guo, and YY Chen. Design and simulation of an air conditioning project in a hospital based on computational fluid dynamics. *Archives of civil engineering*, 63(2):23–38, 2017.

[12] Mohd Firdaus Musa. Building energy analysis using cooling load factor/cooling load temperature different (clf/cltd). 2010.

[13] ASHRAE. Ashrae handbook: Fundamentals. 1979.

[14] W Rudoy and F. Duran. Development of an improved cooling load calculation method. 1975.

[15] Faye C McQuiston, Jerald D Parker, and Jeffrey D Spitler. Heating, ventilating, and air conditioning: analysis and design. 2004.

[16] McQuiston F.C Harris, Steven Merrill. Study to categorize walls and roofs on the basis of thermal response. 1988.